Elsevier Editorial System(tm) for Applied Thermal Engineering
Manuscript Draft

Manuscript Number: ATE-2009-124R1

Title: Families of Optimal Thermodynamic Solutions for Combined Cycle Gas Turbine (CCGT) Power Plants

Article Type: Research Paper

Keywords: CCGT power plant; thermodynamic optimization; families of optimal solutions

Corresponding Author: Dra. Sonia Judith Benz, Ph.D.

Corresponding Author's Institution: Universidad Tecnológica Nacional

First Author: Ezequiel Godoy, Chemical Engineer

Order of Authors: Ezequiel Godoy, Chemical Engineer; Nicolás J Scenna, Ph.D.; Sonia Judith Benz, Ph.D.

Abstract: Optimal designs of a CCGT power plant characterized by maximum second law efficiency values are determined for a wide range of power demands and different values of the available heat transfer area. These thermodynamic optimal solutions are found within a feasible operation region by means of a non linear mathematical programming (NLP) model, where decision variables (i.e. transfer areas, power production, mass flow rates, temperatures and pressures) can vary freely. Technical relationships among them are used to systematize optimal values of design and operative variables of a CCGT power plant into optimal solution sets, named here as optimal solution families. From an operative and design point of view, the families of optimal solutions let knowing in advance optimal values of the CCGT variables when facing changes of power demand or adjusting the design to an available heat transfer area.

To: Applied Thermal Engineering Journal Editor

Subject: Submission of manuscript of the paper “Families of Optimal Thermodynamic Solutions for Combined Cycle Gas Turbine (CCGT) Power Plants” for revision and publication

Dear Sir

Please find annexed with this letter the manuscript of the paper “Families of Optimal Thermodynamic Solutions for Combined Cycle Gas Turbine (CCGT) Power Plants” for revision and publication in the Applied Thermal Engineering Journal.

This paper has not been published previously, it is not under consideration for publication elsewhere, and if accepted it will not be published elsewhere in substantially the same form, in English or in any other language, without the written consent of the Publisher.

I look forward to hearing from you.

Yours faithfully

Sonia Benz

Manuscript

Families of Optimal Thermodynamic Solutions for Combined Cycle Gas Turbine (CCGT) Power Plants

E. Godoy[a], N. J. Scenna[a,b] and S. J. Benz[a,*]

[a] UTN, FRRo – Universidad Tecnológica Nacional, Facultad Regional Rosario – Zeballos 1341 – (2000) Rosario – Argentina
[b] INGAR / CONICET – Instituto de Desarrollo y Diseño – Avellaneda 3657 – (3000) Santa Fe – Argentina

---

Abstract

Optimal designs of a CCGT power plant characterized by maximum second law efficiency values are determined for a wide range of power demands and different values of the available heat transfer area. These thermodynamic optimal solutions are found within a feasible operation region by means of a non linear mathematical programming (NLP) model, where decision variables (i.e. transfer areas, power production, mass flow rates, temperatures and pressures) can vary freely. Technical relationships among them are used to systematize optimal values of design and operative variables of a CCGT power plant into optimal solution sets, named here as optimal solution families. From an operative and design point of view, the families of optimal solutions let knowing in advance optimal values of the CCGT variables when facing changes of power demand or adjusting the design to an available heat transfer area.

Keywords: CCGT power plant; thermodynamic optimization; families of optimal solutions

---

Nomenclature

Symbols

AAR	=	area allocation ratio
X	=	area fraction
CR	=	compression ratio
$\dot{E}$	=	exergy flow rate (kW)
$\dot{Q}$	=	heat flow rate (kW)
A	=	heat transfer area (m^2)
THRSG	=	HRSG temperature relation

[*] Corresponding Author. Tel.-Fax.: +54-341-4484909
E- mail address: ezgodoy@frro.utn.edu.ar (E. Godoy), nscenna@ceride.gov.ar (N. J. Scenna) , sbenz@frro.utn.edu.ar (S. J. Benz)

LHV	=	lower heating value (kJ/kg)
$\dot{m}$	=	mass flow rate (kg/s)
MR	=	mass flow ratio
$\dot{W}$	=	power (kW)
PR	=	power ratio
p	=	pressure (atm)
TSAT	=	saturation temperature relation
h	=	specific enthalpy (kJ/kg)
s	=	specific entropy (kJ/kg ºK)
e	=	specific exergy (kJ/kg)
TSS	=	superheated steam temperature relation
t	=	temperature (ºK)

Greek Symbols

η_E	=	exergetic efficiency
η_T	=	thermal efficiency

Subscripts

0	=	ambient condition
a	=	air
bs	=	bled steam
cond	=	condensate
cw	=	cooling water
fw	=	feed water
f	=	fuel
g	=	gas
Net	=	net or total
sat	=	saturated steam
ss	=	superheated steam

Acronyms

AC	=	air compressor
CC	=	combustion chamber
Co	=	condenser
De	=	deaerator
Ec	=	economizer

Ev	=	evaporator
FP	=	feed pump
GT	=	gas turbine
HRSG	=	heat recovery steam generator
RP	=	recirculation pump
ST	=	steam turbine
Su	=	superheater
TGT	=	turbine of gas cycle

1. Introduction

The increasing demand of electricity, the limited sources of fossil fuel and the urgent necessity of reducing environmental pollution have induced the developing of new technical means for energy production. In this context and in comparison with other energy conversion systems, the approval of combined cycle gas turbine (CCGT) power plants continues increasing due to its reliability and its high efficiency in the use of available resources.

Traditionally, the design of power generation systems is performed following a sequential strategy. Generally, as first step, the thermodynamic efficiency of the plant is maximized. In this context, numerous approaches exist in the literature, including maximum first and second law efficiencies, maximum power production, and minimum entropy generation, among others (Valdés and Rapún [1]; Franco and Giannini [2]; Consonni and Silva [3]; Franco and Casarosa, [4]).

Then, as second step, the economic optimization of the system is addressed. For this purpose, pure economic (Biezma and San Cristóbal [5]; Rodrigues et al. [6]; Söderman and Pettersson [7]), thermoeconomic (Franco and Casarosa [8]; Valdés et al. [9]), and exergetic- economics (Kwak et al. [10]; Rosen and Dincer [11]; Sarraf Borelli and Oliveira Junior [12]) methods are commonly applied. Even ecological efficiency may be used to quantify the performance of the system (Corrado et al. [13]; Villela and Silveira [14]).

On the other hand, efforts dedicated to the simultaneous multi- objective optimization of power generation systems are not as numerous (Giannantoni et al. [15]; Li et al. [16]; Mussati et al. [17]), due to the complexity of such techniques.

As Vargas and Bejan [18] claimed, thermodynamic optimization may be used by itself (without cost minimization) in the preliminary stages of design, in order to identify trends and the existence of optimization opportunities. The optima and structural characteristics identified based on thermodynamic optimization can be made more realistic through subsequent refinements based on global cost minimization. It is beyond dispute that in the end any practical engineering design will be selected and refined based on the minimization of total cost. Thermodynamic optimization may be used as a first-cut approach for those applications where the total cost is dominated by the costs associated with the destruction of exergy, as it is the design of CCGT power plants.

The design of CCGT power plants is inherently complex due to the presence of two different power cycles which are intimately coupled through the heat recovery steam generator (HRSG). The performance of the whole system strongly depends on the optimal integration between the power units. As common practice, gas and steam turbines are selected from within a set of commercially available ones, while the HRSG is the only component of a combined cycle which is tailored specifically for each power plant. In the present study, each unit is sized as a result of considering continuous variable values within practical limits. Therefore, this approach enables to design them to meet specifically the expected optimal process conditions of the plant.

Valdés and Rapún [1] presented a method for the optimization of the HRSG with the aim of improving the performance of the system without increasing the total heat transfer area. The HRSG design method is based on the application of influence coefficients, and is supported by a CCGT simulation tool.

Exhaust gas and steam temperature profiles at the HRSG are restricted by both the minimum temperature difference at the pinch point and the approach point, previously fixed by the designer. Adequate values for these parameters are derived from experience, according to thermodynamic and economic considerations (Valdés and Rapún [1], Franco and Giannini [2]).

Since the HRSG is a complex piece of equipment, Franco and Giannini [2] approached its design performing in a first level an operational optimization by minimizing the exergy losses, and then, progressed towards its detail design in a second level by minimizing a compactness index. However, this sequential approach may exclude regions of feasible solutions from the space explored by the optimization algorithm.

Aguirre and Scenna ([19] and [20]) presented a thermodynamic approach for the synthesis of heat and power integrated systems, determining the matching policy among external streams and operating cycle fluids by minimizing the entropy generation, given the values of the total heat exchange area and the isentropic efficiencies of turbines and compressors. Mussati et al [21] extended the synthesis procedure to dual purpose desalination plants including gas turbines, using the exhausted gas as heat source for steam turbine cycle.

Vargas and Bejan [18] studied the thermodynamic behaviour of a power plant associated solely with the stream-to-stream interaction (meaning the interaction between a cold sink and a hot source), while operating at maximum power. They demonstrated that thermodynamic optima exist with respect to both the mass flow ratio and the area allocation ratio for this simple system. Moreover, the authors mentioned that it may be useful to extend these relationships to more complex systems.

The objective of this paper is to examine in a fundamental way the thermodynamic optimization of CCGT power generation systems, where every component is driven by a limited amount of available exergy. In order to accomplish this task, a NLP optimization model is coded for the analyzed CCGT plant using GAMS software. A parametric study of the thermodynamic optima is systematically carried out by varying several operative and design parameters in order to discover trends in the system behaviour, and identify families of thermodynamic solutions and its fundamental characteristics.

2. NLP Thermodynamic Optimization Model of the CCGT Power Plant

The CCGT power plant here analyzed consists of a gas turbine and a single pressure steam cycle. The gas turbine produces part of the required power. Natural gas is used as fuel. The steam turbine, in which expansion of the superheated steam produced in the HRSG occurs, generates the rest of the required power.

The mathematical model of the CCGT single pressure power plant is structured as a NLP optimization problem, since all the variables are continuous and most of the equations describing the CCGT model are non-linear. Then, the NLP model formulation is generically represented by Eq. (1), Eq. (2) and Eq. (3), denoting the objective function (function to be optimized, η_E) and a set of equality and inequality constraints which define the feasible operation region, respectively.

$$\max_{\bar{x}} \eta_E(\bar{x}) \text{ : Objective Function} \tag{1}$$

$$s.t. \quad \bar{f}(\bar{x}) = 0 \text{ : Equality Constraints} \tag{2}$$

$$\bar{g}(\bar{x}) \leq 0 \text{ : Inequality Constraints} \tag{3}$$

The modelling scope of this work is oriented towards finding thermodynamic optimal CCGT designs. Solving the NLP optimization problem implies finding the optimum value of the selected thermodynamic objective function, within a feasible operation region defined by mass and energy balances, thermodynamic properties correlations, design equations, and operative restrictions according to practical experience. The objective function is briefly introduced further ahead in this section; while the equality and inequality constraints are summarized in Appendix I, along with the assumptions considered to build the CCGT model.

Solving the equation system is inherently complex due to convergence and variables initialization difficulties. The non-linear mathematical programming model is here solved using the reduced gradient algorithm CONOPT included in the software GAMS.

2.1. Description of the Objective Function

The thermal efficiency of the cycle (net power production per unit of combustion heat) is usually used as first law efficiency. This definition of the CCGT efficiency is widely found in the literature (Valdés and Rapún [1], Aguirre and Scenna [19] and [20], Rapún [22], Sue and Chuang [23]).

$$\eta_T = \frac{\dot{W}_{Net}}{\dot{Q}_{Net}} = \frac{\left(\dot{W}_{GT} - \dot{W}_{AC}\right) + \left(\dot{W}_{ST} - \dot{W}_{FP} - \dot{W}_{RP}\right)}{\dot{m}_f \ . LHV} \tag{4}$$

The first law efficiency, however, does not quantify the irreversible losses of the power plant since it makes no distinction between power and heat. Instead, the second law efficiency expressly accounts the irreversibilities of the power production process (Vargas and Bejan [18]). As Sue and Chuang [23] stated, by using exergy to evaluate the power plant cycle, a more accurate performance of the system can be obtained. If less exergy is consumed, a cycle can produce more efficiently. Therefore, the exergetic efficiency is defined as the net power production per unit of exergy consumed by the CCGT, and is used here as the second law efficiency, according to the next equation.

$$\eta_E = \frac{\dot{W}_{Net}}{\dot{E}_{Net}} = \frac{\left(\dot{W}_{GT} - \dot{W}_{AC}\right) + \left(\dot{W}_{ST} - \dot{W}_{FP} - \dot{W}_{RP}\right)}{\dot{m}_a \ . e_a + \dot{m}_f \ . e_f - \dot{m}_g \ . e_g} \tag{5}$$

2.2. Definition of a Feasible Operation Region

The CCGT power plant is modelled by means of thermodynamic properties correlations, mass and energy balances and design equations (see Appendix I).

In order to set useful boundaries to circumscribe a feasible operation region, inequality constraints according to practical experience are considered in our model. For example, as was shown by Franco and Giannini [2], operative constraints related to the HRSG are necessary for the optimization of the steam generator.

Moreover, only a few hypotheses (i.e. no make-up stream, constant overall heat transfer coefficients) are considered to build the combined cycle model, which are widely accepted for most applications and usually employed for modelling purposes (see for example Valdés and Rapún [1], Franco and Giannini [2], Mussati et al [21]).

2.3. NLP Model Performance

The NLP here proposed is a deterministic type model, since no uncertainty in the parameters is considered. In addition, errors linked to the correlations introduced in the model for the prediction of fluid properties are widely discussed in the source literature ([23], [24] and [25]).

For results comparison purposes, the NLP model here proposed is tested by searching the CCGT thermodynamic optima for a HRSG area of 60400 m^2 and a power requirement of 227.8 MW as obtained by Valdés and Rapún [1]. The simulation model used there, and developed by Rapún [22], includes continuous variables and non-linear equations. It demands to take in advance fixed values for the gas turbine parameters (i.e. the gas inlet temperature, the compression ratio, the air mass flow rate), according to manufacturer data. On the other hand, in the present NLP optimization model, those parameters are treated as operative variables which values are bounded by inequality constraints in order to consider the whole range of commercially available options. In addition, the Pinch Point, the Approach Point, the temperature difference between gas and steam at the superheater entry, and the boiler and the condenser operative pressures need to be fixed in Rapún's model [22]; while here they can vary freely within feasible ranges (see Appendix I) to obtain an optimal HRSG design.

Table 1 presents the optimal values of the main practical interest variables obtained with the NLP thermodynamic optimization model here proposed, paired with the ones obtained by Valdés and Rapún [1] with the simulation model. The optimal design redistributes the power production between both turbines resulting in an increase of the computed thermal efficiency and a diminution of the gas consumption. The optimal HRSG area distribution results are similar to the ones obtained by Valdés and Rapún [1] with their optimization procedure of the HRSG design based on influence coefficients.

Finally, a sensitivity analysis is performed to investigate the influence of the input parameters in the CCGT thermodynamic optima. Table 2 presents the percentage modification of the exergetic efficiency with respect to the value presented in Table 1, caused by the variation of the isentropic efficiencies and the ambient temperature from - 5 % to + 5 % with respect to the values adopted as input data. As it is well known, the sensitivity analysis proves that any slight increment of the isentropic efficiencies is accompanied by a reduction in the system irreversibilities and an increase of the exergetic efficiency. On the other hand, minor increases of the ambient temperature produce a diminution of the exergetic efficiency, as can be seen in Table 2.

3. Results and Discussion

3.1. Space of Optimal Values. Optimal Thermodynamic Families

Runs of the CCGT model are performed maximizing the exergetic efficiency for given values of the power production demand and the available total heat transfer area, in order to obtain parametrical solutions. As these thermodynamic optimal solutions are found within a feasible operation region, decision variables can vary freely; so can operative and design variables.

The values of the optimal exergetic efficiency versus the power requirement, for four different values of the total heat transfer area, are presented in Fig. 1. Each parametrical solution covers usual intervals for the design and operation of the CCGT power plant. For example, for a design value of the total heat transfer area of 50000 m^2, it is possible to address power requirement demands between 210 MW and 265 MW, operating with a decrease in the exergetic efficiency from 0.5068 to 0.5045. Then, these sets of parametrical solutions are hereafter identified as families of optimal solutions. The corresponding computed values of the thermal efficiency (from 0.5209 to 0.5155) result higher than the ones published by other authors (see for example Valdés and Rapún [1]), since the variables of the CCGT model are allowed to adjust their values within wide ranges, which enables to reach further improvements of the system performance.

In order to reduce the number of graphics and improve their presentation, in Fig. 1 and Fig. 3 to Fig. 6, cuts in the ordinates axis are used, a differentiated line type is used to identify curves corresponding to a particular ratio, and different marker types are used to categorize the results parameterized on different total heat transfer area values.

At this point, the study of the optimal thermodynamic behaviour of the CCGT system, designed for a given total heat transfer area, reveals that there is a unique power production requirement which secures obtaining a maximum value of the exergetic efficiency among the range of optimal ones. Fig. 2 summarizes the optimal values of the total heat transfer area which correspond to the maximum optimal CCGT exergetic efficiency against different power production requirements. As expected, it is verified that an increase of the power production requires a CCGT design based on a bigger total heat transfer area in order to achieve the maximum optimal performance for the operation of the cycle.

Linked to the optimal efficiency values presented in Fig. 1, optimal values of the design and operative variables of the CCGT power plant (as transfer areas, power production, mass flow rates, temperatures and pressures) are also systematized into optimal solution families according to interest technical relationships among such variables. Dimensionless ratios proposed for this study are presented in Table 3.

Optimal values of the area allocation ratio (*AAR*) and the power ratio (*PR*) are plotted in Fig 3, while the optimal area distribution of the heat exchange sections of the HRSG (X_{Ec}, X_{Ev}, X_{Su}) can be observed in Fig. 4. Evolution of the optimal mass flow ratio (*MR*) and the optimal compression ratio (*CR*) are shown in Fig. 5. Optimal trends of temperatures ratios recognized from a practical point of view (*THRSG*, *TSAT*, *TSS*) are documented in Fig. 6.

It can be observed from Fig. 3 to Fig. 6 that the optimal parametrical curves plotted for four different total transfer area values are characterized by a linear functionality within the range of power production requirements for each design. Each optimal design so conceived, is then characterized by optimal values of the HRSG heat transfer area distribution, the power production distribution (power ratio), and operative temperatures and pressures.

From an operative and design point of view, the families of optimal solutions allow to estimate in advance optimal values of the CCGT variables when facing changes of the power demand or adjusting the design to an available heat transfer area.

3.2. Influence of Ratios Values on the Thermodynamic Optima

In order to identify the impact of the ratio values in the thermodynamic optima behaviour, parametrical solutions are determined for different fixed values of such technical relationships.

Study cases are driven considering net power production requirements in the range of 170 MW to 370 MW and total heat transfer areas between 40000 m^2 and 70000 m^2, as in the previous section. The NLP model is then solved by maximizing the exergetic efficiency not only for fixed values of the power demand and the total heat transfer area, but also for the ratio whose influence wants to be studied. Then, the percentage variation of the exergetic efficiency is computed fixing different ratios values, which represent percentage variations of +/- 5 % to +/- 20 % with respect to the corresponding optimal ones. For sake of simplicity, results presented in this section are obtained for a net power production of 270 MW and a total heat transfer area fixed at 55000 m^2, though similar behaviour is verified over the whole range of power demand and heat transfer area.

The CCGT thermodynamic optima exhibit a maximum with respect to the area allocation ratio (*AAR*), as can be seen in Fig. 7. A variation of the *AAR* value with respect to the reference one causes a pronounced diminution of the combined cycle exergetic efficiency. For example, a 5 % diminution produces a 10 % decrease in the cycle efficiency. The slightest change in the distribution of HRSG area to condenser area drives to an increment in the *LMTDs* and eventually, to an increase in the exergy losses. Similarly to the results reported by Vargas and Bejan [18], here it is found that thermodynamic optima with respect to the matching stream and the overall allocation of heat transfer area exist.

Fig. 8 shows that a mass flow ratio (*MR*) deviation from the optimum value (±20 %) causes a moderate variation of the exergetic efficiency (lower than 1.4 %). Also, as shown in Fig. 8, the optimized CCGT efficiency is weakly influenced by percentage variations till ±20 % of the compression ratio (*CR*) and the power ratio (*PR*).

Similar results are obtained when parameterizing on the temperature ratios. Any variation of the optimal HRSG inlet temperature or the HRSG outlet gas temperature cause minor diminutions of the optimal exergetic efficiency, due to the limitation imposed to the recovery of the heat contained in the gas stream. These trends are traduced in a diminution of the optimum efficiency, as the value of the total heat transfer area is fixed. Besides, in the steam side at the HRSG, the optimized performance exhibits a maximum with respect to the evaporator operative temperature and the superheated steam temperature.

4. Conclusions

In order to accomplish the thermodynamic optimization of a CCGT power plant, a NLP model for the combined cycle is generated, and afterwards used to obtain optimal values for operative and design variables when the CCGT operates at its maximal exergetic efficiency.

Thermodynamic optimization (without cost minimization) results useful in the preliminary stages of design of a process plant, allowing to identify trends in the system behaviour and revealing optimization opportunities. As well, such optimal values can be used as feasible initialization points to encourage more complex optimization problems in the design and operation of CCGT power plants.

The families of optimal thermodynamic solutions, for different power demands and available heat transfer area values, summarize the information regarding the optimal values of the exergetic efficiency and the operative and design variables (as transfer areas, power production, mass flow rates, temperatures and pressures). Thus, results are systematized according to interest technical relationships among such variables.

The families of optimal solutions (Fig. 1, Fig. 3, Fig. 4, Fig. 5, and Fig. 6) allow to estimate in advance optimal values of the CCGT variables when facing changes of the power demand or adjusting the design to an available total heat transfer area.

Also, it is established that associated to each given power production requirement, a maximum optimal exergetic efficiency design exists (Fig. 2), determining a bound to the increment of the total heat transfer area. Such bound is produced by the activation of at least one of the technological restrictions considered in the model. This way, an increase of the power production requires a CCGT design based on a bigger total heat transfer area to achieve an optimal performance for the operation of the cycle, although this is not always possible due to economic restrictions.

In addition, the impact of the ratio values on the thermodynamic optima behaviour is identified through parametrical solutions determined for different fixed values of such technical relationships, providing complementary information (Fig. 7 and Fig. 8) about the expected behaviour of the exergetic efficiency of the CCGT power plant when facing changes in the values of operative and design variables.

Acknowledgements

The authors gratefully acknowledge the financial support of the Agencia Nacional de Promoción Cientifica y Tecnológica (ANPCyT) and the Universidad Tecnológica Nacional- Facultad Regional Rosario (UTN- FRRo) from Argentina.

Appendix I

As mentioned in Section 2, the model based on mathematical programming, consists of an objective function (presented in Section 2.1) and a feasible region (see Section 2.2). In the present Appendix, the description of the model of the CCGT power plant is completed, introducing the set of equality and inequality constraints that define the feasible operation region.

AI.1. Thermodynamic Properties Correlations

Enthalpy, entropy and exergy of each fluid line included in the model are calculated by considering their dependence on temperature, pressure and compositions.

$$h = \mathrm{f}\left(t , p , x_{N_2} , x_{O_2} , x_{H_2O} , x_{CO_2}\right) \tag{6}$$

$$s = \mathrm{f}\left(t , p , x_{N_2} , x_{O_2} , x_{H_2O} , x_{CO_2}\right) \tag{7}$$

$$e = h + t_0 \,.\, s \tag{8}$$

In addition, a standard correlation is used to estimate the steam saturation pressure.

$$p_{sat} = \mathrm{f}\left(t_{sat}\right) \tag{9}$$

The thermodynamic properties correlations can be obtained from the standard literature (i.e. the correlations produced by IAPWS ([24] and [25]) are used to predict liquid water and steam properties; and the correlations given in Perry [26] are used to predict gas properties).

AI.2. Energy Balances and Design Equations at HRSG, Condenser and Deaerator

Energy balances are applied to every HRSG component, as well as the condenser. Design equations, which include the heat transfer area and the logarithmic mean temperature difference calculation, for each heat transfer section, are also considered.

$$\dot{Q}_k = U_k \,.\, A_k \,.\, LMTD_k = \dot{m}_{cf} \,.\, \Delta h_{cf} = \dot{m}_{hf} \,.\, \Delta h_{hf} \qquad , \qquad k = Ec\,, Ev\,, Su\,, Co \tag{10}$$

In the deaerator, bled steam from the steam turbine is used for incondensable gases elimination.

$$\dot{m}_{cond} \,.\, h_{cond} + \dot{m}_{bs} \,.\, h_{bs} = \dot{m}_{fw} \,.\, h_{fw} \tag{11}$$

AI.3. Design Equations at Turbines and Compressors

Expressions of the isentropic efficiency are used to account the irreversibilities of the expansion processes in the turbine of the gas cycle and the steam turbine.

$$\eta_i^k = \frac{\Delta h}{\Delta h\big|_i} \qquad , \qquad k = TGT\,, ST \tag{12}$$

A similar expression is considered regarding the compression process in the air compressor.

$$\eta_i^{AC} = \frac{\Delta h\big|_i}{\Delta h} \tag{13}$$

AI.4. Mass and Energy Balances at Combustion Chamber

A mass balance for each chemical compound and a global energy balance are considered at the combustion chamber.

$$x_{j,g} \,.\, \dot{m}_g = x_{j,a} \,.\, \dot{m}_a + x_{j,f} \,.\, \dot{m}_f + \sum_z \dot{r}_{j,z} \qquad , \qquad j = N_2\,, O_2\,, H_2O\,, CO_2 \tag{14}$$

$$\dot{m}_g \,.\, h_g = \dot{m}_a \,.\, h_a + \dot{m}_f \,.\left(h_f + LHV \,.\, \eta_{adiab}^{CC}\right) \tag{15}$$

AI.5. Power Production and Consumption

The turbine of the gas cycle and the steam turbine fulfil the electricity production requirement. Turbines power production is calculated from the corresponding mass and energy balances.

$$\dot{W}_{TGT} = \dot{m}_g \,.\, \Delta h_g \tag{16}$$

$$\dot{W}_{ST} = \left(\dot{m}_{ss} - \dot{m}_{bs}\right).\, \Delta h_{ss} + \dot{m}_{bs} \,.\, \Delta h_{bs} \tag{17}$$

The air compressor is directly driven by the turbine of the gas cycle. Power consumption of the air compressor depends upon its discharge pressure and the air mass flow rate.

$$\dot{W}_{AC} = \dot{m}_a \,.\, \Delta h_a \tag{18}$$

In addition, pumps consume power for condensate recirculation and to force feed water into the HRSG.

$$\dot{W}_{RP} = \frac{\dot{m}_{cond} \,.\, \Delta P_{cond}}{\eta_m^{RP}} \tag{19}$$

$$\dot{W}_{FP} = \frac{\dot{m}_{fw} \,.\, \Delta P_{fw}}{\eta_m^{FP}} \tag{20}$$

The net power produced by the gas turbine equals the difference between the power generated by the turbine of the gas cycle and the power consumed by the air compressor.

$$\dot{W}_{GT,Net} = \dot{W}_{TGT} - \dot{W}_{AC} \tag{21}$$

The net power produced by the steam turbine equals the difference between the power generated by the steam turbine and the power consumed by the recirculation and feed pumps.

$$\dot{W}_{ST,Net} = \dot{W}_{ST} - \dot{W}_{RP} - \dot{W}_{FP} \tag{22}$$

AI.7. Introduction of Practical Values Ranges for Operative Parameters

Operative constraints related to the HRSG are necessary for the optimization of the steam generator. Therefore, in order to set useful boundaries to circumscribe a feasible operation region according to practical experience, the following additional inequality constraints are considered in the model:

- Minimum and maximum steam pressure in the HRSG to assure normal operation,

$$10\ atm \le p_{sat} \le 60\ atm \tag{23}$$

- Minimum bled steam pressure is needed to perform elimination of incondensable gases in the deaerator,

$$p_{bs} \ge 20 \,.\, p_{cond} \tag{24}$$

- Minimum operative pressure of the condenser is restricted by temperature of available cooling water,

$$p_{cond} \geq 0.042\ atm \tag{25}$$

- Minimum gas temperature at HRSG discharge is required to prevent corrosion due to water condensation,

$$t_g \geq 420\ ^{o}K \tag{26}$$

- Maximum gas temperature into the turbine of the gas cycle is determined by the resistance of the material the turbine is constructed with,

$$t_g \leq 1350\ ^{o}K \tag{27}$$

- Maximum steam temperature into the steam turbine is determined by the resistance of the material the turbine is constructed with,

$$t_{ss} \leq 850\ ^{o}K \tag{28}$$

- Minimum steam quality at steam turbine discharge is necessary to achieve normal operation of the turbine,

$$y_{cond} \geq 0.88 \tag{29}$$

- Minimum and Maximum Approach Point value (difference between the temperature of the water leaving the economizer and the saturation temperature) is selected to guarantee no water evaporation in the HRSG economizer and to avoid thermal shock at evaporator entry, respectively,

$$2\ ^{o}K \leq t_{sat} - t_{fw} \leq 5\ ^{o}K \tag{30}$$

- Minimum temperature differences between gas and steam across the HRSG must be selected to secure reasonable practical values of heat transfer areas of economizer, evaporator and superheater:

  Minimum and Maximum Pinch Point value (difference between the saturation temperature and the temperature of the gas leaving the evaporator),

$$10\ ^{o}K \leq t_g - t_{sat} \leq 20\ ^{o}K \tag{31}$$

  Minimum temperature difference at economizer entry,

$$t_g - t_{fw} \geq 30\ ^{o}K \tag{32}$$

  Minimum temperature difference al superheater exit,

$$t_g - t_{ss} \geq 30\ ^{o}K \tag{33}$$

AI.8. Input Data

Most of the parameters values are taken from references [1] and [2].

References

[1] Valdés M., Rapún J. L. Optimization of Heat Recovery Steam Generators for Combined Cycle Gas Turbine Power Plants. Applied Thermal Engineering 2000;21:1149-1159.

[2] Franco A., Giannini N. A General Method for the Optimum Design of Heat Recovery Steam Generators. Energy 2006;31:3342-3361.

[3] Consonni S., Silva P. Off-design performance of integrated waste-to-energy, combined cycle plants. Applied Thermal Engineering 2007;27:712-721.

[4] Franco A., Casarosa C. On some perspectives for increasing the efficiency of combined cycle power plants. Applied Thermal Engineering 2002;22:1501-1518.

[5] Biezma M. V., San Cristóbal J. R. Investment criteria for the selection of cogeneration plants- a state of the art review. Applied Thermal Engineering 2006;26:583-588.

[6] Rodrigues M., Faaij A. P. C. Walter A. Techno-economic analysis of co-fired biomass integrated gasification/combined cycle systems with inclusion of economies of scale. Energy 2003;28:1229-1258.

[7] Söderman J. Pettersson F. Structural and operational optimisation of distributed energy systems. Applied Thermal Engineering 2006;26:1400–1408.

[8] Franco A. Casarosa C. Thermoeconomic evaluation of the feasibility of highly efficient combined cycle power plants. Energy 2004;29:1963–1982.

[9] Valdés M., Durán M. D., Rovira A. Thermoeconomic optimization of combined cycle gas turbine power plants using genetic algorithms. Applied Thermal Engineering 2003;23:2169-2182.

[10] Kwak H. Y., Kim D. J., Jeon J. S. Exergetic and thermoeconomic analyses of power plants. Energy 2003;28:343-360.

[11] Rosen M. A., Dincer I. Exergoeconomic analysis of power plants operating on various fuels. Applied Thermal Engineering 2003;23:643-658.

[12] Sarraf Borelli S. J., Oliveira Junior S. Exergy-based method for analyzing the composition of the electricity cost generated in gas-fired combined cycle plants. Energy 2008;33:153-162.

[13] Corrado A., Fiorini P.; Sciubba E. Environmental assessment and extended exergy analysis of a ‘‘zero CO2 emission’’, high-efficiency steam power plant. Energy 2006;31:3186-3198.

[14] Villela I. A. C., Silveira J. Ecological efficiency in thermoelectric power plants. Applied Thermal Engineering 2007;27:840-847.

[15] Giannantoni C., Lazzaretto A., Macor A., Mirandola A., Stoppato A., Tonon S., Ulgiati S. Multicriteria approach for the improvement of energy systems design. Energy 2005;30:1989-2016.

[16] Li H., Maréchal F., Burer M., Favrat D., Multi-objective optimization of an advanced combined cycle power plant including CO2 separation options. Energy 2006;31:3117-3134.

[17] Mussati S. F., Aguirre P. A., Scenna N. Thermodynamic approach for optimal design of heat and power plants. Relationship between thermodynamic and economic solutions. Latin American Applied Research 2006;36:329-335.

[18] Vargas J. V. C., Bejan A. Thermodynamic Optimization of the Match Between Two Streams with Phase Change. Energy 2000;25:15-33.

[19] Scenna N., Aguirre P. Optimal Thermodynamic Synthesis of Dual-Purpose Desalination Plants. Extraction Vs. Back Pressure Turbines. Desalination Journal, 1991;81:419-436.

[20] Aguirre P., Scenna N. J. Optimal Thermodynamic Synthesis of Dual-Purpose Desalination Plants. Chemical Engineering Science 1989;44:283-296.

[21] Mussati S. F., Aguirre P., Scenna N. J. Thermodynamic Synthesis of Cogeneration Systems in Thermal Desalination Plants including Gas Turbines. In: INGAR, CONICET, Santa Fe, Argentina, editor. International Desalination Association IDA World Congress on Desalination and Water Reuse, Madrid, Spain, 1997. p. 6-9.

[22] Rapún J. L. Mathematical Model of the Behaviour of Gas and Steam Turbines Combined Cycles (Modelo Matemático del Comportamiento de Ciclos Combinados de Turbinas de Gas y Vapor), PhD Thesis. Universidad Politécnica de Madrid, Escuela Técnica Superior de Ingenieros Industriales, 1999 (in Spanish). See also: http://oa.upm.es/170/.

[23] Sue D.-C., Chuang C.-C. Engineering design and exergy analyses for combustion gas turbine based power generation system. Energy 2004;29:1183-1205.

[24] The International Association for the Properties of Water and Steam. Supplementary Release on Saturation Properties of Ordinary Water Substance. Saint Petersburg, Russia: IAPWS, 1992. See also: http://www.iapws.org.

[25] The International Association for the Properties of Water and Steam. Release on IAPWS Industrial Formulation 1997 for the Thermodynamic Properties of Water and Steam. Erlangen, Germany: IAPWS, 1997. See also: http://www.iapws.org.

[26] Perry, J. Chemical Engineers Handbook. Mexico: McGraw-Hill, 1995.

Figures Captions

Fig. 1 Optimal Exergetic Efficiency vs. Power Production Requirement

Fig. 2 Optimal Total Heat Transfer Area vs. Power Production Requirement, for Achieving Maximum Optimal Exergetic Efficiency

Fig. 3 Optimal Power Ratio and Area Allocation Ratio vs. Power Production Requirement

Fig. 4 Optimal Area Allocation Distribution vs. Power Production Requirement

Fig. 5 Optimal Mass Flow Ratio and Compression Ratio vs. Power Production Requirement

Fig. 6 Optimal Temperature Relations vs. Power Production Requirement

Fig. 7 Thermodynamic Optima Sensitivity Analysis for the Area Allocation Ratio

Fig. 8 Thermodynamic Optima Sensitivity Analysis for the Mass Flow Ratio, the Power Ratio and the Compression Ratio

Figure 1
Click here to download high resolution image

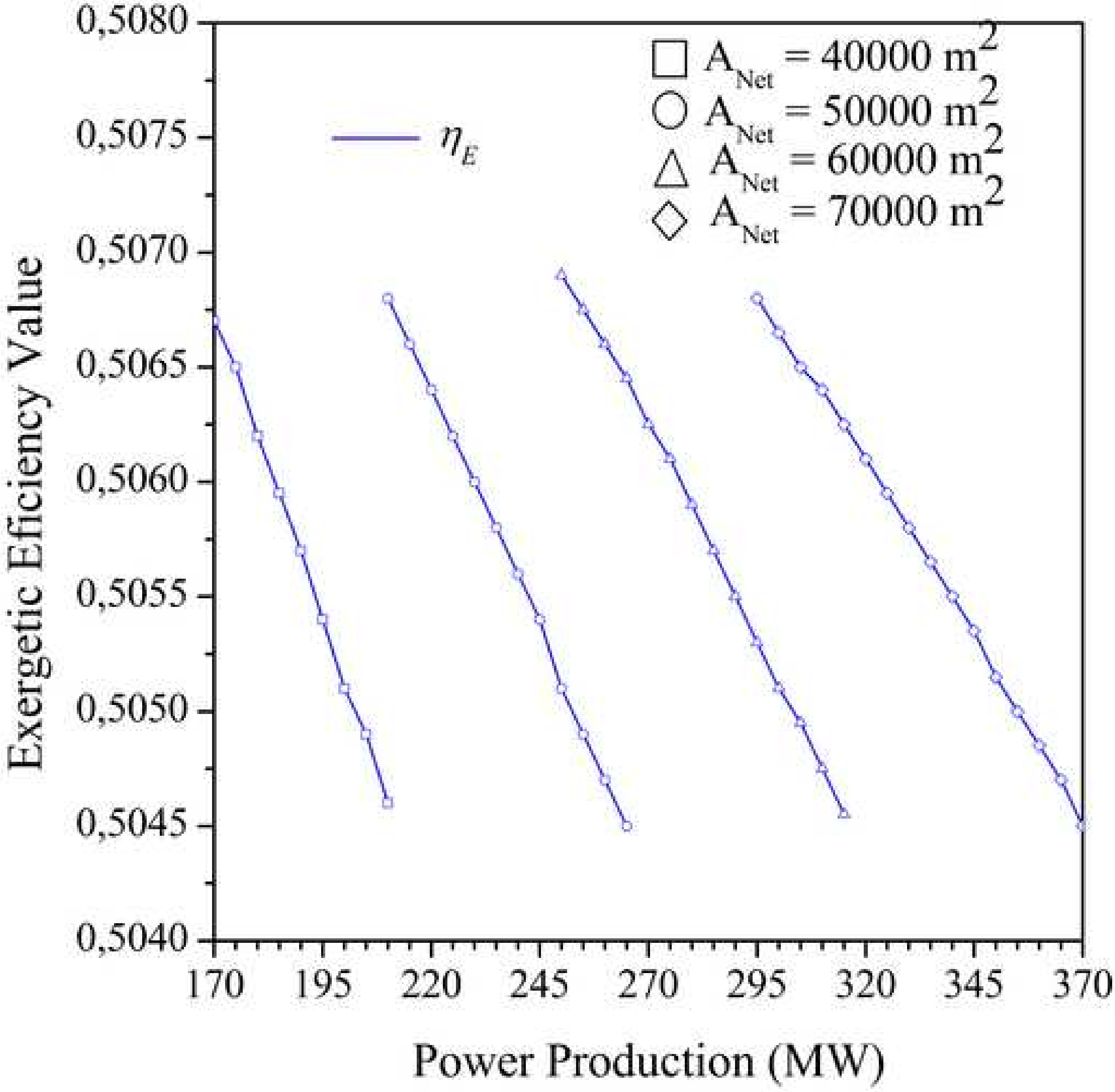

Figure 2
Click here to download high resolution image

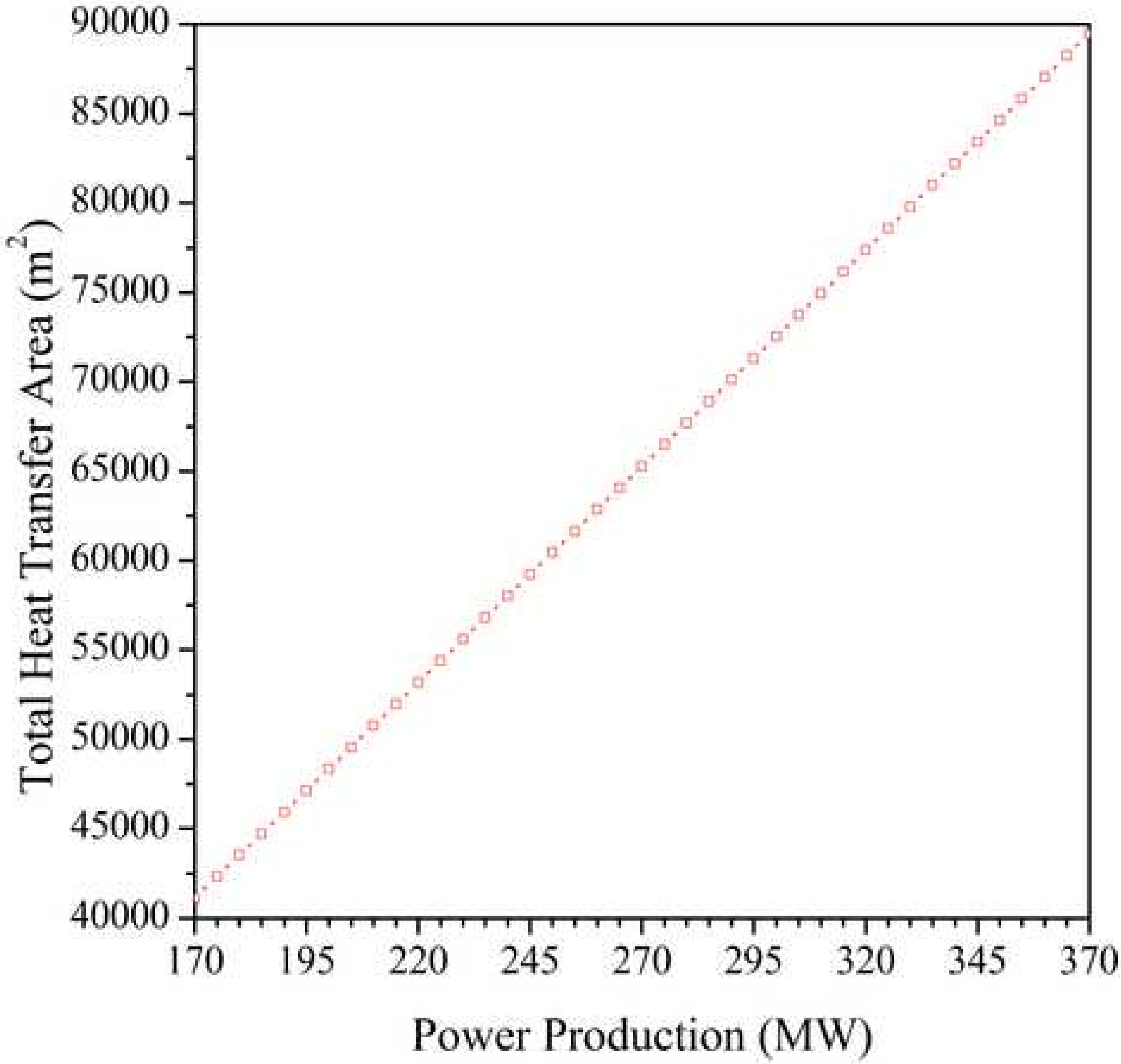

Figure 3
Click here to download high resolution image

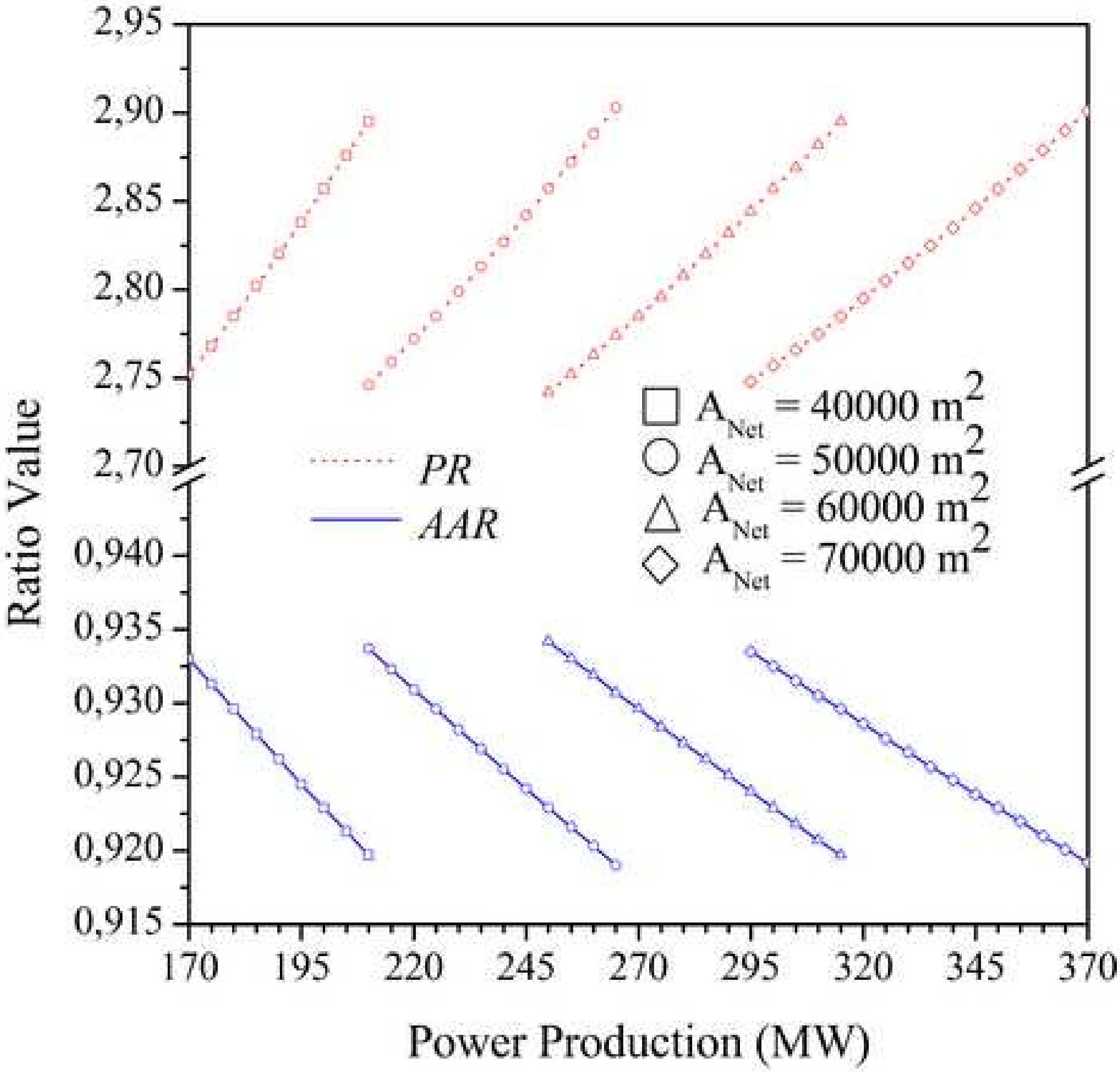

Figure 4
Click here to download high resolution image

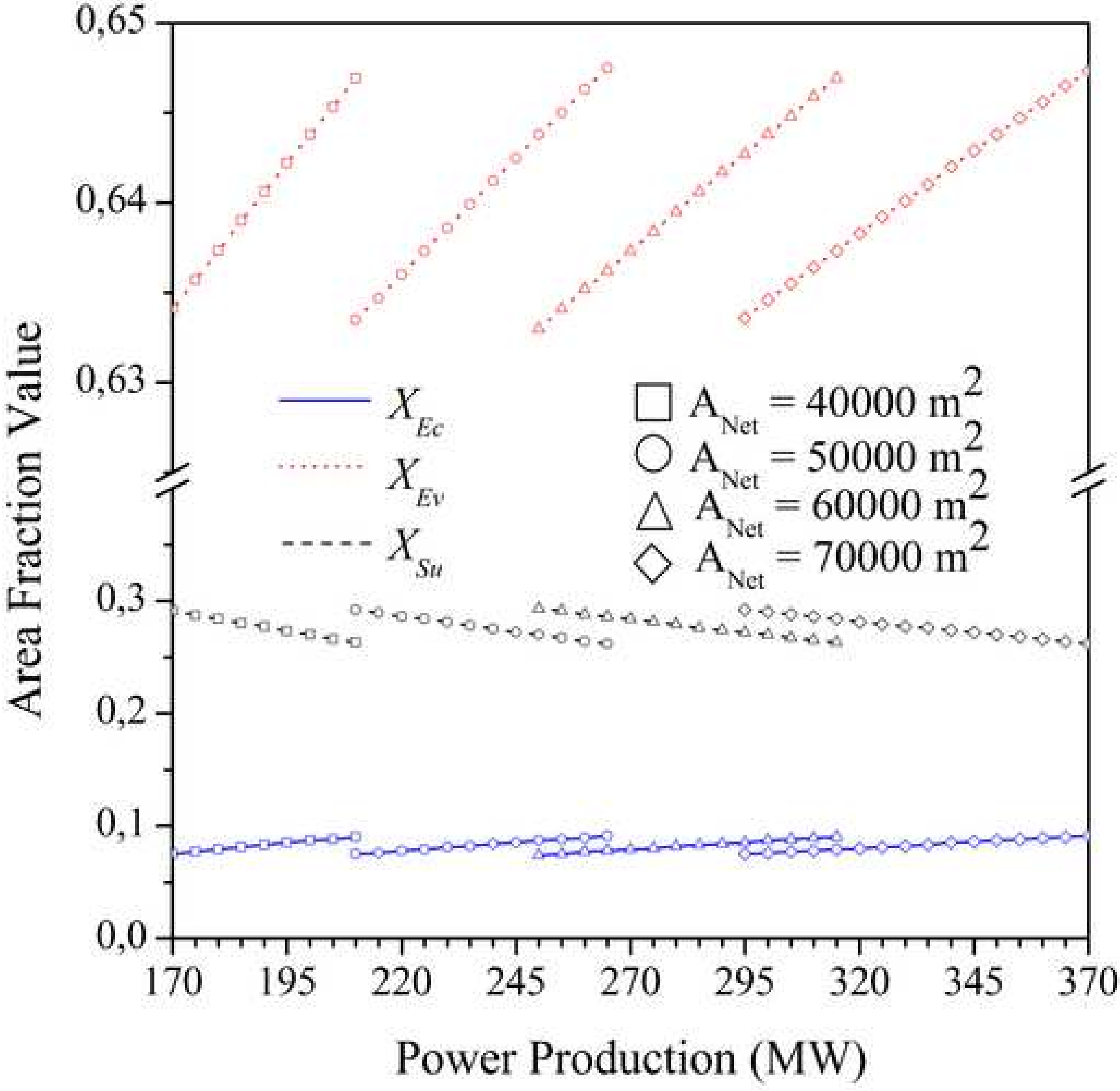

Figure 5
Click here to download high resolution image

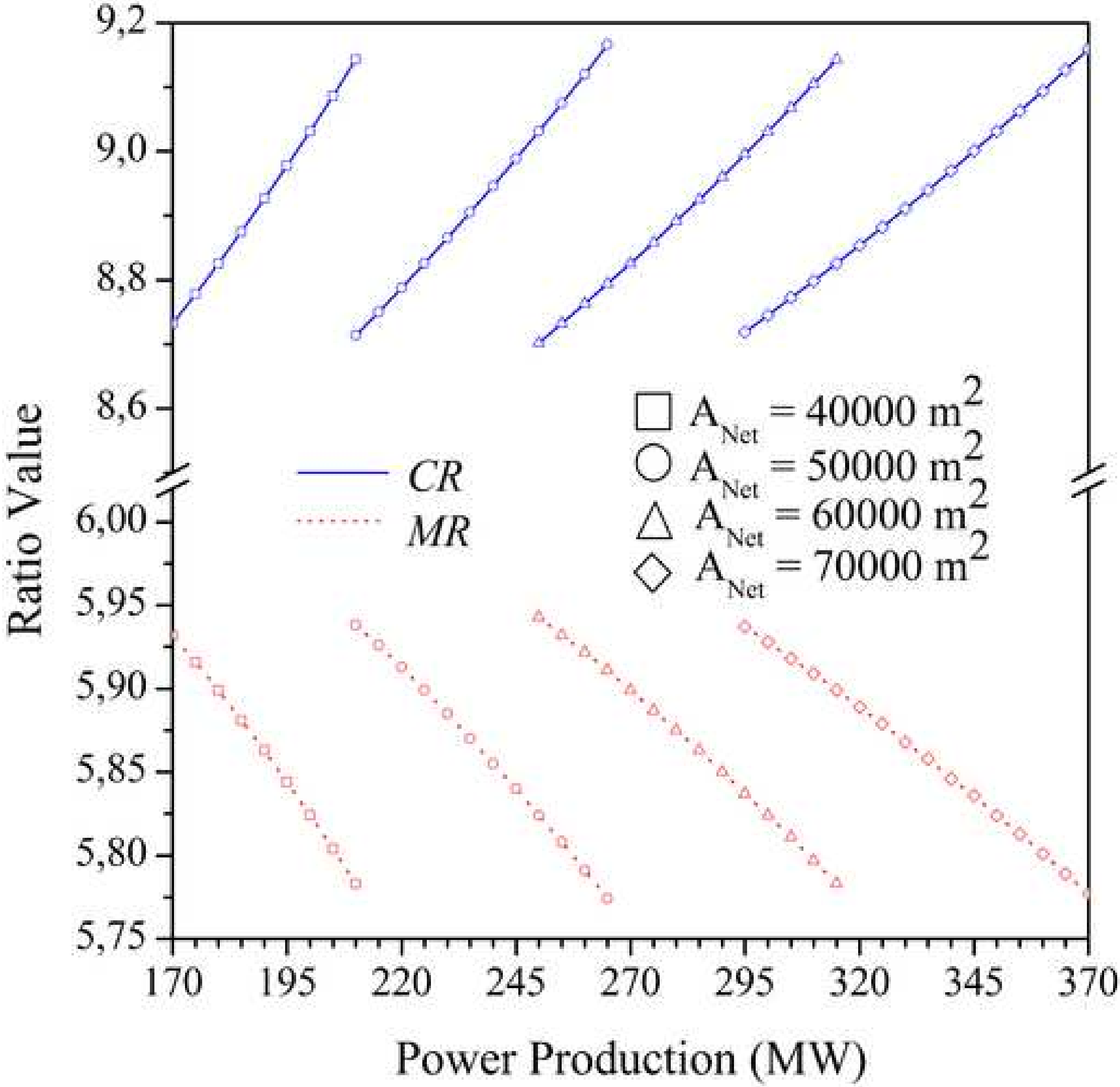

Figure 6
Click here to download high resolution image

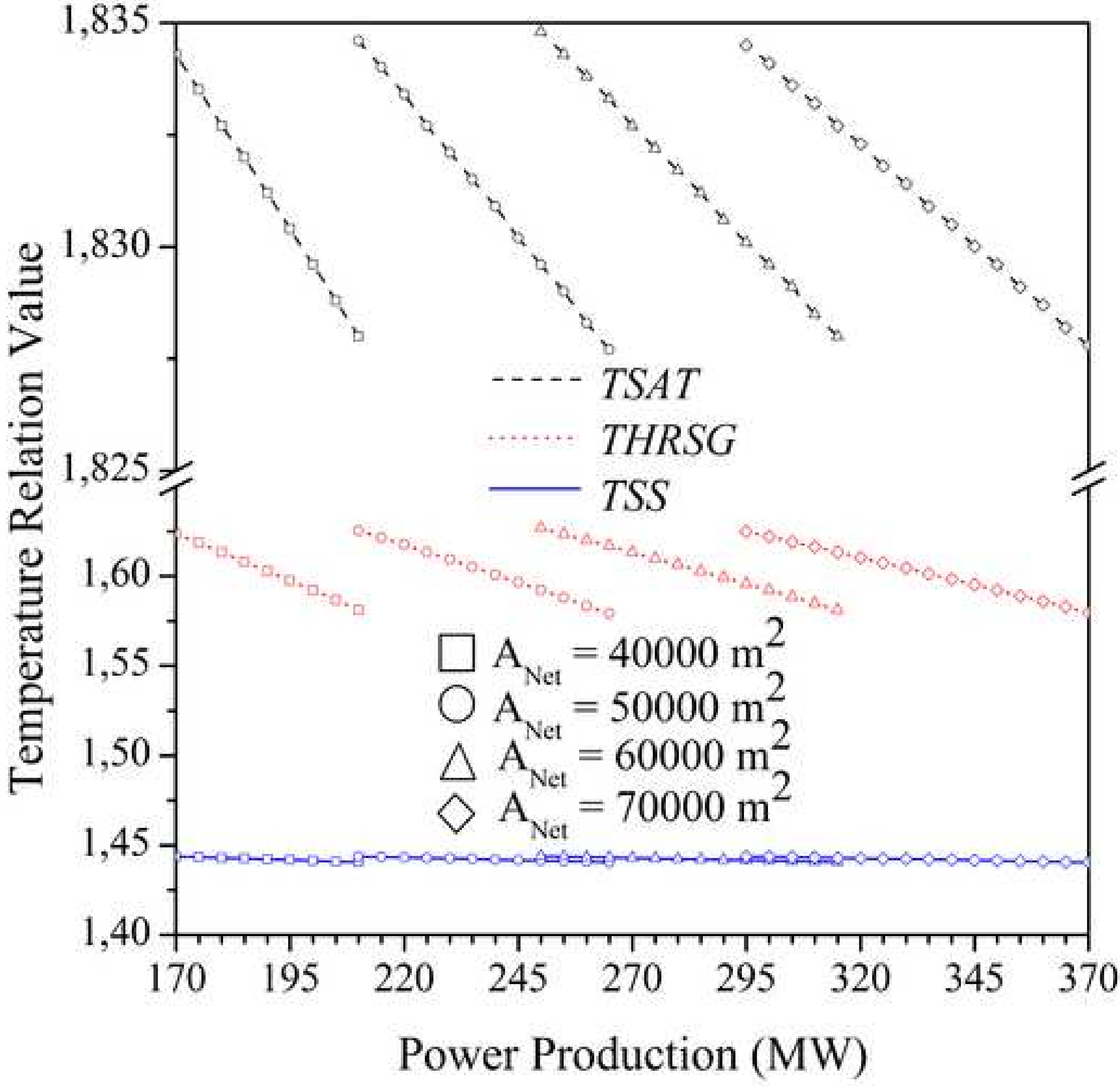

Figure 7
Click here to download high resolution image

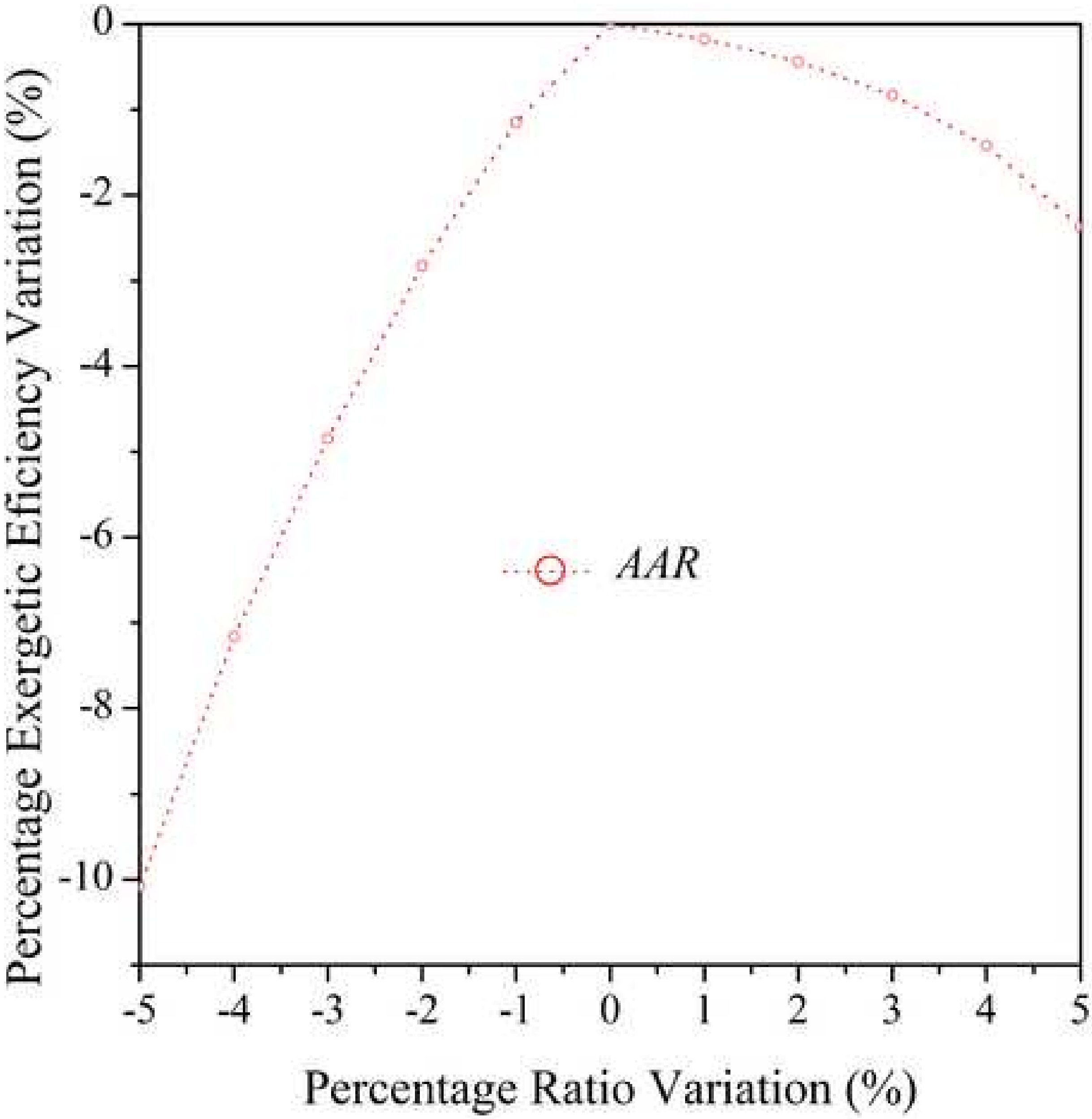

Figure 8
Click here to download high resolution image

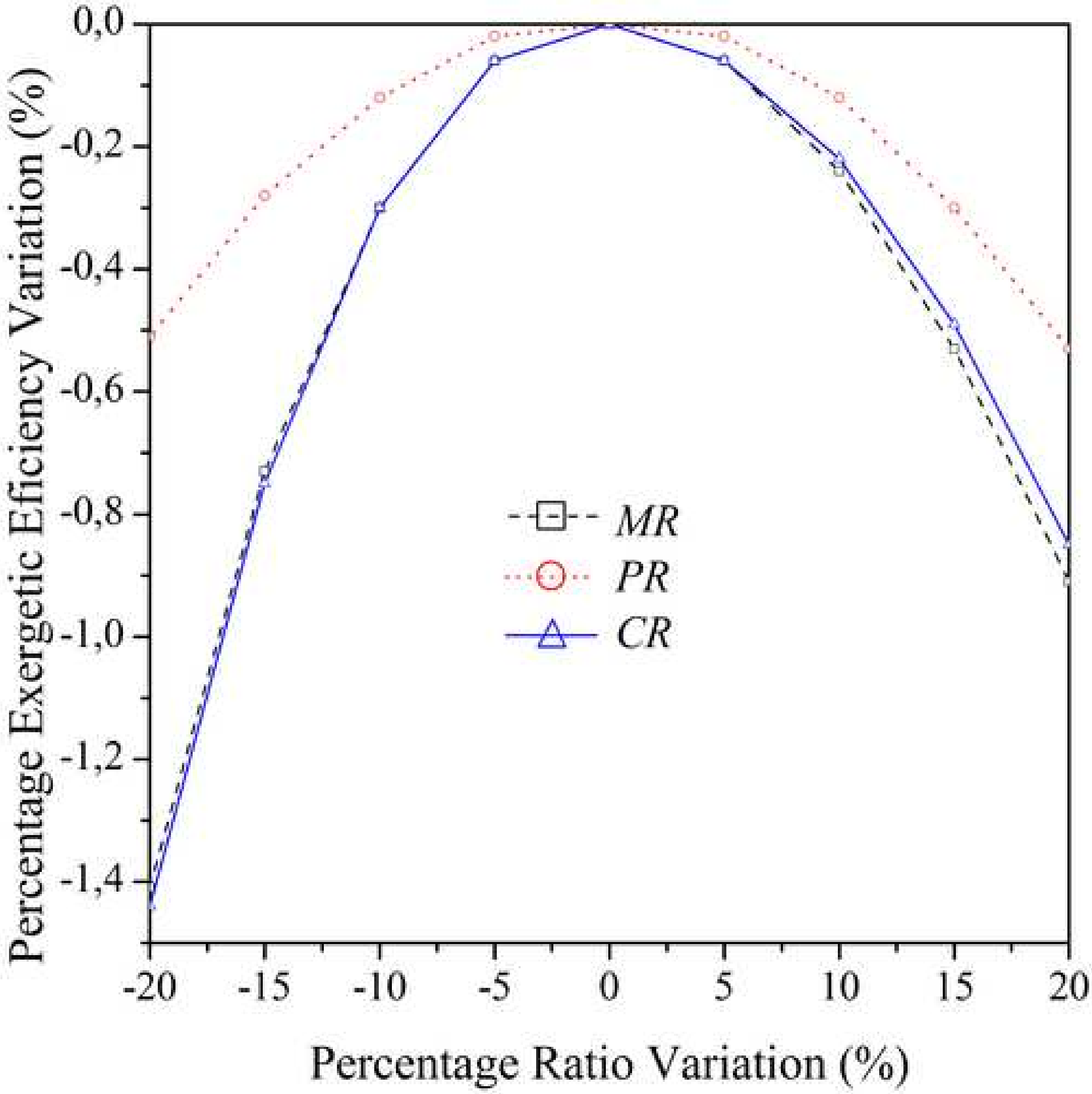

Table 1

Table 1 – Comparison between Rapún's Model and NLP Optimization Model

Description	Rapún's Model	NLP Model
Air mass flow rate (kg/ s)	506.9 [c]	452
Gas turbine inlet temperature (ºK)	1368.1 [c]	1350 [b]
Gas Turbine Gross Power (MW)	151.950	144.7
Gas Exit Temperature (ºK)	818.0	866
Drum Pressure (atm)	49.35 [c]	60 [b]
Temperature Difference al Superheater Exit (ºK)	32 [c]	30 [b]
Pinch Point (ºK)	10 [c]	10 [b]
Approach Point (ºK)	2 [c]	2 [b]
Total Heat Exchange Area (m^2)		64493
HRSG Heat Exchange Area (m^2)	60400	60400 [a]
Economizer Area Fraction	0.403	0.414
Evaporator Area Fraction	0.499	0.490
Superheater Area Fraction	0.098	0.096
Steam Mass Flow (kg/ s)	65.4	69.1
Steam Title	0.87 [c]	0.884 [b]
Steam Turbine Gross Power (MW)	78.140	83.1
CCGT Gross Power (MW)	227.8	227.8 [a]
Computed Thermal Efficiency	0.499	0.517
Exergetic Efficiency		0.500
Iteration Number		13
Resolution Time (s)		0.141

[a] Values fixed during the resolution of the NLP proposed in this paper

[b] Variables restricted by inequality constraints

[c] Values fixed by Rapún [18] in order to design the CCGT power plant

Table 2

Table 2 – Effects of Parameters Values in the Thermodynamic Optima

Description	Percentage Modification of the Exergetic Efficiency caused by a Given Percentage Variation of the Parameter Value			
	- 5 %	- 1 %	+ 1 %	+ 5 %
Air Compressor Isentropic Efficiency	-2.37	-0.47	0.49	2.39
Turbine of the Gas Cycle Isentropic Efficiency	-5.34	-0.95	1.03	5.64
Steam Turbine Isentropic Efficiency	-0.81	-0.14	0.16	0.63
Ambient Temperature	2.99	0.59	-0.57	-2.81

Table 3

Table 3 – Definition of Operative and Design Ratios		
Symbol	Name	Formula
PR	Power Ratio	$\frac{\dot{W}_{GT,Net}}{\dot{W}_{ST,Net}}$
AAR	Area Allocation Ratio	$\frac{A_{Ec}+A_{Ev}+A_{Su}}{A_{Ec}+A_{Ev}+A_{Su}+A_{Co}}$
X_{Ec}	Economizer Area Fraction	$\frac{A_{Ec}}{A_{Ec}+A_{Ev}+A_{Su}}$
X_{Ev}	Evaporator Area Fraction	$\frac{A_{Ev}}{A_{Ec}+A_{Ev}+A_{Su}}$
X_{Su}	Superheater Area Fraction	$\frac{A_{Su}}{A_{Ec}+A_{Ev}+A_{Su}}$
CR	Compression Ratio	$\frac{p_g / p_0}{\dot{W}_{GT,Net} / \dot{W}_{ST,Net}}$
MR	Mass Flow Ratio	$\frac{\dot{m}_{sat}}{\dot{m}_g}$
THRSG	HRSG Temperature Relation	$\frac{t_{g,in}}{t_{g,out}}$
TSAT	Saturation Temperature in the Evaporator Relation	$\frac{t_{sat}}{t_0}$
TSS	Superheated Steam Temperature Relation	$\frac{t_{ss}}{t_{sat}}$