Contribution to the Numerical simulation of the effect of Considering Blades Cooling Flow Rate and Working Fluid Properties on the Thermodynamic Performances of real Gas Turbine's Plants

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Abstract

This work examines the effect of considering blades cooling flow rates in the thermodynamic working fluid model on the estimation of GTs' thermodynamic performances. So, three models were tested: the first takes into account blades cooling flow rates and uses relationship of the compression and the expansion under their differential forms to follow the progressive variation of the C_p *according to the temperature. The second one the* C_p *averaged value variation is a function of the temperature and without considering the turbine blades' cooling. The last, differs from the previous model only by supposing the* C_p *as a constant value. Data of a real operating GT plant were used for validation. Results highlighted that the first model provided a realistic results, principally in terms of the exhaust gas temperature (EGT) and efficiency of the GT plant, while the two others models were not. In fact, the second model overestimates considerably the EGT and the efficiency, while the third one underestimates those last parameters.* **.** *Taking into account the blade cooling effect corrects the prediction of the studied parameters during the predesign step of GT's plants, so NOx productions can be reduced for better efficiencies.*

Key words: Gas turbines 'plants efficiency, air cooling flow rates, C_p, thermodynamic performances, TOT.

1. Introduction

Gas turbines (GT) are known to be a widely used energy conversion system to generate electricity; and in combined cycle (CC) that allows them getting a better efficiency around 60% compared to internal combustion systems. Improving and optimizing GT cycles aims to increase their performances, reduce their fuel consumption and consequently decrease the emissions and/or costs.

Many authors took interest in thermodynamic analysis of GT and CC. The first studies have investigated the effect of GT design parameters on CC global performance and concluded that the most important ones are the turbine inlet temperature (TIT) and the pressure ratio [1-5].

It is well known that optimizing CCs is a way to improve them; Horlock [6] has done it to the combined cycle pressure ratio by considering one pressure level steam cycle. Bassily [7-9] used optimized CC with two and three-pressure levels with reheat adapted to a GT with simple cycle and reheated expansion. Gody et al. [10] used NLP to maximize the CC exegetic efficiency determining HRSG optimal surface for a wide range of power. Koch et al. [11] used an evolutionary algorithm in order to minimize CC cost production. Kotowicz and Bartela [12] and Ahmadi and Dincer [13] used GA for thermo-economic optimization to show the influence of fuel price on optimal design parameters.

GT realization procedure is mainly composed of four steps: Market research, thermodynamic optimization, aero-thermodynamic design (1D design), CFD and experimental analysis. After following the analysis of the above-mentioned four steps, we notice the unavailability of the thermodynamic optimization in spite of its simplicity comparatively to the aero-thermodynamic and CFD steps. This is due to the fact that its results are the input data of the other following steps and the detailed turbo-components design is based on the results of the thermodynamic optimization. A wrong determination of the thermodynamic parameters at design point may prevent designers from taking fully advantage of the potential of the cycle. Thus, the credibility and the validity of the thermodynamic model are of a paramount importance.

The analysis of the previous studies in relation to their used thermodynamic models reveals that the variation of working fluid thermodynamic properties particularly the C_p(Heat capacity at constant pressure) during the compression and the expansion, which is generally evaluated as an averaged value. However, few studies supposed the progressive variation of the $C_p$as properties of the working fluid evolve. Furthermore, turbine blade cooling is mostly taken into account during the aero-thermodynamic, CFD and experimental designs steps but in thermodynamic analysis. In a convection-air-cooled gas-turbine engine the cooling air discharges from the tips of the rotor blades. This cooling air mixes with the combustion gases to increase the mass flow and decrease the gas temperature for the following turbine stages [14].In many studies the cooling bleeding air fraction is neglected during the thermodynamic modeling knowing that it is mixed with burned gases with about 15% to 20% of total mass flow rate [15]. This may not only lead to significantly different results from the reality but it can also falsify steam cycle calculations (when the GT is a part of a combined cycle) because the increase of coolant fraction leads to decrease of exhaust gas temperature (TOT), and this impacts directly steam cycle performances prediction.

Then, in order to study the effect of considering blades cooling and $C_p$progressive variation during the compression and the expansion on the validity of the results of GT thermodynamic modeling, there will be a comparison between three models: the first takes into account C_pVariation according to the temperature keeping the equation that models the compression and the expansion under their differential forms, and this model also considers turbine blades cooling. The second represents $C_p$variation by its average value and doesn't study blades cooling. The last model also neglects blades cooling and considers a constant C_p. The obtained results are confronted to the real ones of the GT Siemens V94.3A (Table 1).

2. Thermodynamic Models

In this part, the three models are presented. GT basic diagram is illustrated by Fig. 1. Figure 1.a corresponds to the first model and Figure 1.b corresponds to the second and the third model.

2.1 The first model

It takes into consideration C_p progressive variation throughout the volution of working fluid keeping the equations that represent compression and expansion processes in their differential form. Turbine blades' cooling is also considered. The equations modeling compression, expansion are respectively:

$$\frac{dP}{P} = \left(\frac{C_{pa}(T).\eta_{pc}}{R}\right)\frac{dT}{T} \quad (1)$$

$$\frac{dP}{P} = \frac{C_{pg}(T)}{R\eta_{pT}}\frac{dT}{T} \quad (2)$$

Equations (1) and (2) are numerically sufficiently resolved by Runge-Kutta fourth-order method known for the accuracy of its results. The required work to compress a unit of air mass is:

$$W_c = \left(c_{pa2}T_2 - c_{Pa1}T_1\right) + (1-e_3)\left(c_{pa3}T_3 - c_{Pa2}T_2\right) + (1-e_3-e_2)(c_{Pa4}T_4 - c_{pa3}T_3) \quad (3)$$

The combustion is not perfectly completed because the exhaust gas is not totally burned. This is represented by combustion efficiency, «□cc ».The ratio of the fuel mass flow rate to the inlet air mass flow rate, in the compressor, is named $f = \dot{m}_f/\dot{m}_a$and expressed as it follows:

$$f = \frac{(1-e_1-e_2-e_3)\ (h_g(TIT)-h_g\ (T_5)]}{PCI.\eta_{cc}-\left(h_g(TIT)-h_g(T_5)\right)} \tag{4}$$

The calculation of proportions of the turbine coolant mass flow rate to gas mass flow rate (e_1, e_2, e_3)is as it follows. To cool a row of a turbine by a film cooling technique, the fraction of the needed cooling air$e = \dot{m}_c/\dot{m}_g$ is expressed by the following equation:

$$e = \frac{0.045(\varepsilon_0-0.12-0.28\varepsilon_0)}{0.7(1-\varepsilon_0)} \tag{5}$$

ε_0 is the overall cooling effectiveness defined as:

$$\varepsilon_0 = \frac{T_g-T_{met}}{T_{met}-T_c} \tag{6}$$

In fact, we have just given the final relationship used in the present study. Concerning the deduction, please refer to [15] for more details.

2.2 Second model

This model takes into account C_{pg}variation by average values. The relationships concerning the combustion chamber remain identical to the precedent model. So, only the modeling of the expansion and the compression is presented here.

If equation (1) is integrated on both sides assuming a constant average $C_p$and replacing it by its expression($C_{pm} = {}^{\gamma_m r}/_{\gamma_m} - 1$), the following well-known relationship is obtained:

$$T_2 = T_1(\pi_c)^{\frac{\gamma_m-1}{\eta_{pc}\gamma_m}} \tag{7}$$

The average$C_{pm}$is expressed as follows:

$$c_{pm} = \frac{1}{T_2-T_1}\int_{T_1}^{T_2} c_{pa}dT \tag{8}$$

The required work to compress a unit mass of air is written as follows:

$$W_c = c_{pa2}T_2 - c_{Pa1}T_1 \tag{9}$$

Following the same line of reasoning as the one of the compression, the expansion is expressed as:

$$TOT = TIT(\pi_T)^{\frac{\gamma_{mg}-1}{\eta_{pc}\gamma_{mg}}} \tag{10}$$

The work produced by mass unit is given by:

$$W_c = (1+f)(c_{pg3}TIT - c_{Pg4}TOT) \tag{11}$$

2.3 The third model

It is perfectly identical to the second one and only the considered $C_p$valueis fixed to be 1.005 $kJ/kg.K$. Thus, it is not necessary to expand on this part.
Finally, the specific work and the efficiency are expressed for the three models as follows:

$$W_{GT} = W_T - W_C/\eta_{mec} \quad (12)$$

$$\eta_{GT}=\frac{W_{GT}}{fPCI} \quad (13)$$

3. Results and discussions

Table 2 compares the obtained TOTs of the three models to the experimental data of the selected GT. It is clear that the first model is giving the best results. A difference of less than 1°C can be noticed between the real case and the first model's data. The two other approaches show significantly different results from the experimental ones. It is noted that, the second /the third ones, respectively, overestimates the TOT and the third one underestimates it. For the V94.3A, the second model provides a higher result of about 37°C and the third model leads to an underestimation of 40°C compared to the experimental case.
Figure 2 compares the results of the three models according to the pressure ratio variations. The two following notes are taken from the analysis of this figure:

- For low pressure ratios (lower than 7), the third model gives higher results than those of the first. From this pressure ratio (i,e 7) the third model indicates to lower values.
- As much as the pressure ratio increases, results of the second model approach those of the first one.

Table 3 compares the efficiency of the three models applied to the GT Siemens V 94.3A. It is clear that only the first model is satisfying results.
On Figure 3, the efficiency according to the pressure ratio is illustrated, for TIT equal to1350°C, whatever pressure ratio value; the second model gives overemphasized results. Furthermore, as much as the pressure ratio increases the difference compared to the first model becomes more significant.
The third model that leads to low values can give relatively acceptable results for the low pressure ratios. This model gives efficiencies similar to those of the first one for lower pressure ratios of 6.
The failure of the second and the third models is more confirmed for the power parameter as it is illustrated in Table 4. The second model provides significantly higher results than those of the real cycle up to 50%. However, results of the third model are lower of about 24%. The first model is satisfactory and the difference between its results and those of the real case is less than 1.5%.
The significantly high (low) power values of the second model (of the third model) are due to the overestimation (the underestimation) of the specific work values. This is justified by Figure 4 which illustrates the power values according

to the pressure ratio. It allows justifying the significant power gap between the two last models and the real case. The gap becomes important for relatively high pressure ratios and decreases for the low ratios.

The importance of considering the $C_p$variation during the compression is then proved to be important, and considering an average C_p is not sufficient to simulate correctly the thermodynamic behavior of the compressor and more specifically for the high pressure ratios. We can summarize the results previously presented in the following:

- It is important to consider C_p progressive variation during the compression and the expansion, because considering average/constant values of C_p will aver/under-estimate predictions of TOT and performances and more specifically the power.

4. Conclusion

The main goal of this work is to examine the effect of the choice of the thermodynamic working fluid model and the consideration of blades cooling on the estimation of GTs thermodynamic performances. For this purpose, a comparison between three models is done: the first takes into account blades cooling and keeps the relationship that models the compression and the expansion under their differential forms to follow the progressive variation of the C_p according to the temperature. The second represents C_p variation as a function to the temperature by its average value and doesn't take turbine blades cooling into consideration. The last one also doesn't consider blades cooling and $C_p$is regarded as a constant. A comparison is done with real operating GT data for validation. The first model shows results close to those of the real plant while the two others couldn't resume the real thermodynamic behavior of the GT. In fact, the second model overestimates considerably the TOT and the performances, while the third one underestimates the TOT as well as the performances**.** Considering $C_p$variation according to the temperature by its average value is not sufficient to obtain valid results, the progressive variation of c_p according to the temperature should then be considered. Furthermore, to correctly estimate the performances and the TOT values, blades cooling flow rates must be taken into account.

5. References

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6. Nomenclature

c_p	specific heat capacity
e	proportion of turbine coolant mass flow rate to gas mass flow rate
f	fraction fuel / air
h_g	gas specific enthalpy
P	pressure
r	specific gas constant
T	temperature
TIT	turbine inlet temperature
TOT	turbine outlet temperature
$W_{s(GT)}$	specific gas turbine work
ζ_{cc}	pressure loss in combustion chamber
π_T	turbine pressure ratio
π_c	compressor pressure ratio
$\pi_{c(eff\_opt)}$	optimum pressure ratio for maximum efficiency
$\pi_{c(sw\_opt)}$	optimum pressure ratio for maximum specific work
η_{cc}	combustion efficiency
η_{mec}	mechanical efficiency
η_{pc}	compressor polytropic efficiency
η_{pT}	turbine polytropic efficiency

7. Subscript

a	air
c	coolant
g	gas
m	mean
met	blade
ex	exit
in	inlet

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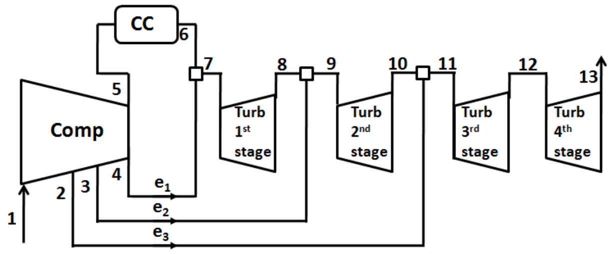
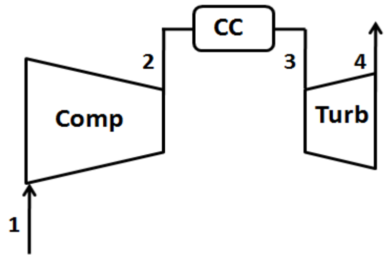


(a)



(b)

Fig. 1: GT schematic diagram: a) first model, b) second and third model

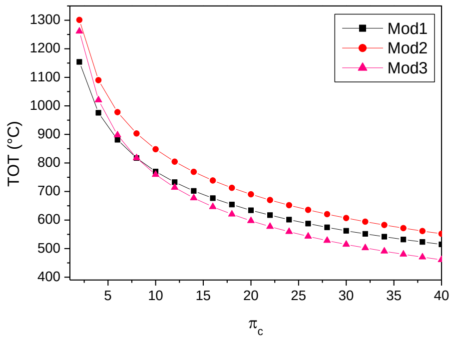


Fig. 2: Turbine outlet temperature (TIT=1350°C)

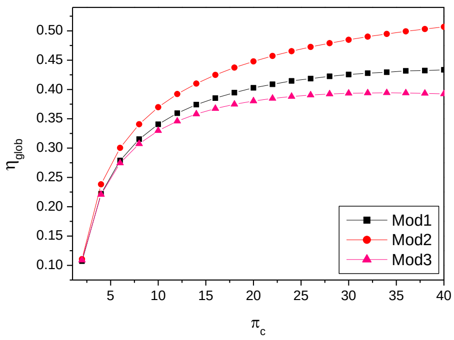


Fig. 3: Thermal efficiency (TIT=1350°C)

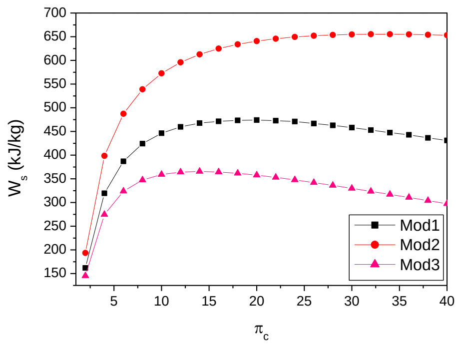


Fig. 4 : Specific work output (TIT=1350°C)

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Table 1: Principal characteristics of selected gas turbines *Siemens V94.3A*

Characteristic	Value
Net efficiency (%)	39.5
Net power output (MW)	288
Turbine inlet temperature (°C)	1350
Pressure ratio	18.2
Turbine outlet temperature (°C)	580
Exhaust masse flow rate (kg/s)	692

Table 2: Models validation – TOT (°C)

	Model 1	Model 2	Model 3	Real Value
Siemens V94.3A	579.32	617.50	542.39	580

Table 3: Models validation – Efficiency (%)

	Model 1	Model 2	Model 3	Real Value
Siemens V94.3A	39.58	43.84	37.53	39.5

Table 4: Models validation – Net power output (MW)

	Model 1	Model 2	Model 3	Real Value
Siemens V94.3A	289.41	433.18	220.63	288