MODELING PART-LOAD PERFORMANCE OF A GAS TURBINE

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Abstract. *In this paper a gas turbine producing electricity is modeled while operating in part-load condition. The study presents the hypothesis adopted and the simulation strategies, for full load and part-load conditions. The effect of blade cooling-air consumption was incorporated. Two load control methods were evaluated: the conventional temperature decrease control (with constant air flow and decreasing fuel flow) and the combined effect of air flow control with constant temperature (through variable geometry of the inlet guide vane) followed by the temperature control when the former is no longer effective.*
The full-load performance results were analyzed, and compared with traditional gas turbine behavior. The part-load results show the effects of the load control method on the performance of the gas turbine, on the operating condition of the compressor, on the exhaust gas flow and on the turbine exhaust temperature. The sensitivity of the gas turbine operation to environmental conditions (temperature and atmospheric pressure) are also explored.
This study of a gas turbine in simple cycle is the first part of a broader study [8], dealing also with combined cycles at part-load (subject of a second paper submitted to COBEM 2003) and exergoeconomic analysis.

Keywords: *Gas turbine, Part-load, Simulation, Efficiency*

1. Introduction

The huge Brazilian dependence on hydropower to generate electric energy is noteworthy. Although a renewable source, hydropower is subject to climatic changes and this impacts the electricity offer. The need for other sources to produce electricity became apparent in 2001, when the Brazilian society was compelled to reduce its consumption of electricity.

The use of natural gas with gas turbines to generate electricity, both in simple cycle or combined cycle, is a possibility that has some advantages, such as short implementation period, low investment cost and a positive impact on natural gas pipeline investments. In the past, gas turbines were used to generate electricity only in a few places in Brazil, with light distillates (diesel oil) as fuel and only in simple cycle. However, many gas turbine-based thermoelectric power plants were proposed in the last years and some of these are under construction or are already in operation, developing thus the participation of natural gas in the brazilian energetic matrix.

Gas turbines are widely used in combined cycles and cogeneration plants due to its high exhaust temperatures and high efficiency (about 35% in simple cycle and 55% in combined cycle plants). Gas turbines can still be employed in repowering plants where fuels like oil or coal are replaced by natural gas, with advantages both for the environment and plant efficiency.

The sharp growth in the use of gas turbines requires simulation tools that take into account the effects of site conditions and part load operation, to obtain dependable technical and economic evaluations in the early stages of a project. This paper presents a model to simulate heavy-duty gas turbines, not only in design condition but also under part-load, the latter being the main objective.

A parametric analysis for the design condition is carried out to evaluate the influence of combustion gas temperature and compressor pressure ratio on efficiency and specific work.

Two methods are considered for the simulation under part-load: IGV (inlet guide vane control) and TIT (turbine inlet temperature control). Equations found in technical papers are used to describe the behavior of the main gas turbine components under part-load. Finally, the sensitivity of gas turbine performance to environmental conditions is also assessed.

2. Modeling of gas turbine

Gas turbine was modeled considering variable specific heat for air and the combustion/exhaust gases. The composition of the combustion gas results from a reaction of the compressed air with the natural gas in the combustion chamber. A model for blade cooling was also considered in the simulation.

The gas turbine GE PG7241FA is chosen for the study in the modeling and simulation for part-load conditions. Table (1) presents the main parameters at ISO conditions for this turbine.

Table 1: Parameters for the gas turbine PG7241FA.

Pressure ratio[a]	15,5
flow (kg/s)[b]	432
Power (kW)[c]	171700
Exhaust gas temperature(°C)[d]	602
efficiency (%)[e]	36,5
Combustion gas temperature (K)[f]	1588

References (a,c,d): Gas Turbine World 1998-1999 Handbook [6]
References (b,e,f): GE Power Systems [7]

2.1. Modeling for the design condition

The model proposed by Gallo [5] is used to predict gas turbine blade cooling. By this model, the necessary air for cooling is deviated from the compressor exit and mixed with the combustion gases that leave the combustion chamber. Korakianitis and Wilson [12] had used a similar model to predict the effect of blade cooling, also considering the addition of all the air for blade cooling to the combustion gas. The necessary air flow for blade cooling ($\dot{m}_c$) is given by Eq. (1):

$$\frac{\dot{m}_c}{\dot{m}_g} = \left(\frac{cp_g}{cp_c}\right) \cdot St_g \cdot \left(\frac{\Omega_b}{\Omega_g}\right) \cdot \left(\frac{1}{\varepsilon_h}\right) \cdot \frac{(T_g - T_b)}{(T_b - T_c)} \tag{1}$$

where $\dot{m}_g$ is the combustion gas flow, cp_g and cp_c the specific heat of gas and air, St_g is the gas Stanton number, Ω_b the blade surface area, Ω_g the hot gas flow area, ε_h the heat transfer effectiveness, T_g the hot gas temperature at combustion chamber exit, T_c the temperature of the cooling air flow at compressor exit and T_b the maximum blade temperature. The values for internal cooling with air indicated by Gallo [5] are presented in Tab. (2).

Table 2: Parameters for blade cooling.

maximum blade temperature (K)	1073
heat transfer effectiveness (ε_h)	0,3
gas Stanton number (St_g)	0,005
surface area ratio $\left(\frac{\Omega_b}{\Omega_g}\right)$	4
mixing pressure drop (%)	0,5

For heavy-duty gas turbine, some simulation parameters employed are indicated in Tab. (3).

Table 3: Parameters used in the simulation.

compressor isentropic efficiency	0,87[1]
turbine isentropic efficiency	0,89[2]
combustion chamber pressure drop (%)	2,7
generator efficiency	0,98

Reference: Dechamps [1]

A transmission efficiency of 98% is considered between the shaft that transmits the power from the turbine to the compressor. The fuel utilized in the simulation is the natural gas from Bolivia. The pressure loss in the inlet air filter is set equal to 996,3 Pa (4 inH_2O) and an inlet temperature of 185°C is used for the natural gas at the combustion chamber [14].

2.2. Modeling under part-load

It is important to predict the performance of power plants in off-design conditions, mainly of gas turbines in combined cycles plants, because, as observed by Facchini [4] "the part-load operation of the entire combined cycle is controlled by the gas turbine, while the steam cycle adapts itself to the variation of exhaust gas flow and temperature".

[1]Original value presented as 0,876, modified to 0,87 to better fit the results from the proposed model with the respective values presented in Tab. (1).
[2]Original value presented as 0,899, modified to 0,89.

Facchini [4] compared five off-design methods for power control in gas turbines:

1) throttling of air flow at the compressor inlet
2) variation of air flow at the compressor inlet through the use of variable inlet guide vane (IGV)
3) bleeding of air flow down the compressor
4) recirculation of part of air flow from compressor exit to its inlet
5) variation of fuel flow in the combustion chamber in order to control the turbine inlet temperature (TIT).

Comparing the five alternatives above, it was observed that the principal part-load control method is the fuel flow control in the combustion chamber, as the other methods basically act on the compressor. Methods 1, 3 and 4 are not usually employed due to the dissipation of energy with an associated detrimental effect to gas turbine efficiency at part-load.

The part-load methods generally used in heavy-duty gas turbines are the reduction of the air mass flow rate at compressor inlet through inlet guide vane (IGV) and the reduction of turbine inlet temperature (TIT) through the control of the fuel mass flow rate. According to Dechamps et al. [2], Facchini [4] and Kim and Ro [11], it is necessary to combine the two part-load methods (IGV and TIT) to cover an wide range of part-load conditions with high efficiency.

The equations proposed by El-Sayed [3] are used to characterize the behavior of gas turbine main components at part-load condition.

A pressure drop described by Eq. (2) is considered for the combustion chamber:

$$\Delta \mathrm{P} = \Delta \mathrm{P_d} \cdot \left(\frac{\dot{\mathrm{m}}_\mathrm{g}}{\dot{\mathrm{m}}_\mathrm{gd}}\right)^{1,75} \tag{2}$$

where $\Delta \mathrm{P_d}$ and $\Delta \mathrm{P}$ are, respectively, the pressure losses in design and off-design conditions, $\dot{\mathrm{m}}_\mathrm{gd}$ and $\dot{\mathrm{m}}_\mathrm{g}$ are the combustion gas mass flow rate in design and off-design conditions, respectively.

Equation (3) presents the compressor efficiency at off-design condition:

$$\eta_\mathrm{comp} = \eta_\mathrm{comp,d} \cdot \left(\frac{0,3337 + 1,0917 \cdot \mathrm{M_r} - 0,5254 \cdot \mathrm{M_r}^2}{0,9}\right) \tag{3}$$

with $\eta_\mathrm{comp,d}$ and η_comp as the compressor isentropic efficiency at design and off-design conditions, respectively. The parameter $\mathrm{M_r}$ expresses the ratio between the compressor inlet air mass flow rate at off-design ($\dot{\mathrm{m}}_\mathrm{air,comp}$) and design conditions ($\dot{\mathrm{m}}_\mathrm{air,comp,d}$):

$$\mathrm{M_r} = \frac{\dot{\mathrm{m}}_\mathrm{air,comp}}{\dot{\mathrm{m}}_\mathrm{air,comp,d}} \tag{4}$$

The turbine efficiency is corrected to off-design condition in accordance with Eq. (5):

$$\eta_\mathrm{turb} = \eta_\mathrm{turb,d} \cdot \left(\frac{0,6164 + 0,6179 \cdot \mathrm{P_r} - 0,3343 \cdot \mathrm{P_r}^2}{0,9}\right) \tag{5}$$

with $\eta_\mathrm{turb,d}$ and η_turb as the turbine isentropic efficiency at design and off-design conditions, respectively. The parameter $\mathrm{P_r}$ expresses the ratio of turbine pressure ratio in off-design ($\mathrm{pr_{turb}}$) to turbine pressure ratio in design condition ($\mathrm{pr_{turb,d}}$):

$$\mathrm{P_r} = \frac{\mathrm{pr_{turb}}}{\mathrm{pr_{turb,d}}} \tag{6}$$

The flow function $\left(\dot{\mathrm{m}}\sqrt{\mathrm{T}}/\mathrm{P}\right)$ is used for the turbine inlet in accordance with Eq. (7):

$$\frac{\dot{\mathrm{m}}_\mathrm{turb} \cdot \sqrt{\mathrm{T}}}{\mathrm{P}} = \mathrm{f} \cdot \left(\frac{\dot{\mathrm{m}}_\mathrm{turb,d} \cdot \sqrt{\mathrm{T_d}}}{\mathrm{P_d}}\right) \tag{7}$$

where $\dot{\mathrm{m}}_\mathrm{turb}$, P, T refer respectively to mass flow rate (including blade cooling air), pressure and temperature of the combustion gas at the turbine inlet for off-design condition, having been used the sub-index ‘d’ to assign the same variables to the design condition case. The following equations proposed by El-Sayed [3] were used for the parameter f:

- for $\mathrm{P_r} \geq 0,53$ the parameter f is described by Eq. (8):
  $$\mathrm{f} = 1 \tag{8}$$
- for $\mathrm{P_r} < 0,53$ the parameter f is given by Eq. (9):
  $$\mathrm{f} = 0,1228 + 2,8283 \cdot \mathrm{P_r} - 2,2145 \cdot \mathrm{P_r}^2 \tag{9}$$

Although both Eqs. (8) and (9) have been used in the simulation, according to Kim and Ro [10] only the use of Eq. (8) is a reasonable assumption even for low pressure ratios.

3. Gas turbine simulation

In accordance with the modeling presented in sections 2.1 and 2.2 for the gas turbine at ISO and part-load conditions, respectively, a simulation for the gas turbine was performed in order to analyze its main parameters.

3.1. Parametric analysis

A parametric analysis is carried out to evaluate the results obtained with those from technical literature which employed similar gas turbine simulation methods.

Figure (1) shows the fraction of air mass flow rate at the compressor exit that is diverted to blade-cooling, suggesting the following conclusions:

1) for a certain pressure ratio, the fraction of air for blade cooling increases as the gas turbine maximum temperature is higher.
2) for a specified maximum temperature, as the pressure ratio increases the air fraction for blade cooling also increases due to the higher temperatures for the air at the compressor exit, therefore demanding a greater cooling air mass flow.
3) for high maximum temperatures (1800 K) and high pressure ratios the method of blade cooling with air is impracticable, since the necessary fraction of air for cooling becomes very high.

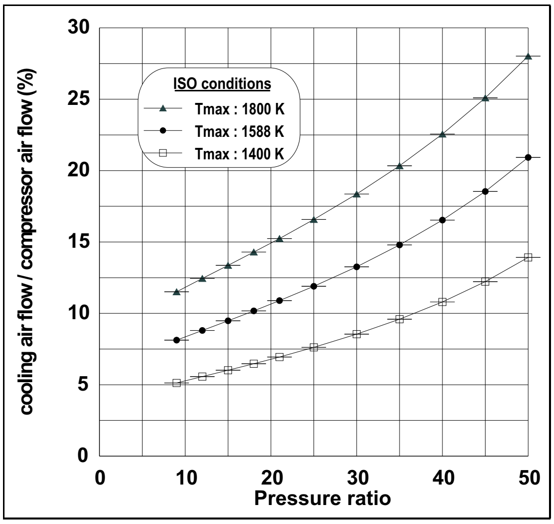

Figure 1: Compressor air fraction diverted for blade-cooling.

Figure (2) presents the effect of blade cooling on gas turbine efficiency, showing that the blade cooling causes a reduction on the efficiency. This effect becomes considerably detrimental to gas turbine efficiency for high pressure ratios due to the losses that results from the mixing of the high mass flow rate of cooling air and the combustion gas.

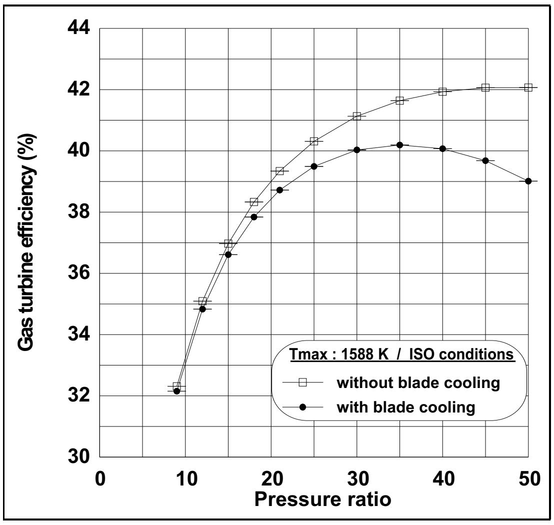

Figure 2: Effect of blade cooling on gas turbine efficiency.

Another important conclusion from Fig. (2) refers to the optimum pressure ratio for maximum efficiency, which is lower for the case with blade cooling than without blade cooling.

3.2. Simulation at ISO conditions

To verify the accuracy of the proposed model, the gas turbine was simulated at ISO conditions and the results obtained from the simulation were compared with those from standard handbooks.

Table 4: Accuracy of the proposed model for gas turbine PG7241.

parameter	standard information	proposed model
efficiency (%)	36,50	36,84
power (kW)	171.700	171.689
exhaust gas temperature (°C)	602	606,7

As it can be noted in Tab. (4) the model yields good results when compared to standard information.

3.3. Part-load simulation

Two load control methods are used to evaluate gas turbine performance at part-load:

- control of the gas temperature at the inlet of the turbine (TIT)
- control of air through inlet guide vane variable geometry (IGV) associated with turbine inlet temperature control (TIT). According to Facchini [4] there is no way to reduce gas turbine power to low load conditions using solely the IGV control because of its narrow operating range.

In accordance with Sehra et al. [13], power is reduced in a heavy-duty gas turbine through the reduction of air mass flow rate at compressor inlet. According to Kim and Ro [11] and Facchini [4], the operating range for the IGV control method is limited to a decrease of 20% of the air mass flow rate, while fuel mass flow rate is controlled to maintain the turbine inlet temperature at its design value. This procedure has also been adopted for the present gas turbine simulation under part-load conditions.

Figure (3) shows the gas turbine exhaust flow at part-load. Reducing the power through TIT control provokes only a small decrease of the exhaust gas flow, since the air flow at compressor inlet remains unchanged and the fuel mass flow rate has a little reduction due to the decreasing of the turbine inlet temperature. Controlling the power with the IGV method causes a mass flow reduction of 20% compared with the design flow, being a proper characteristic of this control mode as already mentioned. It can be perceived in Fig. (3) that the operating range of the IGV control method is limited to a power reduction of only 23%, therefore it is necessary that IGV control be used in association with the TIT control method for extra load reductions.

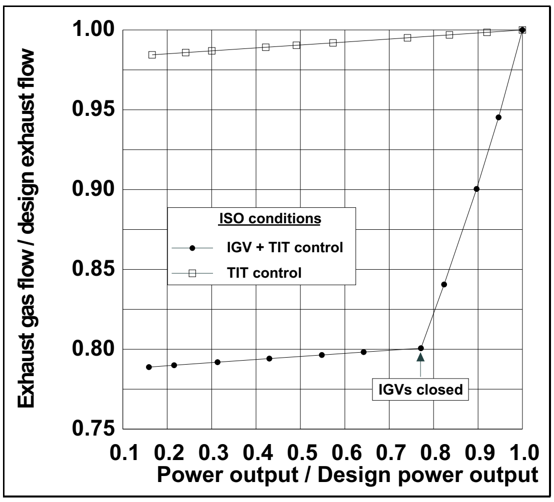

Figure 3: Gas turbine exhaust mass flow rate.

Figure (4) presents the gas turbine efficiency for the part-load control methods considered, where it can be noted that the efficiency at part-load is greater for the TIT control method than the combined method (IGV associated with TIT control). It can still be concluded that, whichever the control mode considered, the gas turbine efficiency decreases considerably for part-loads lower than 50% of the design load.

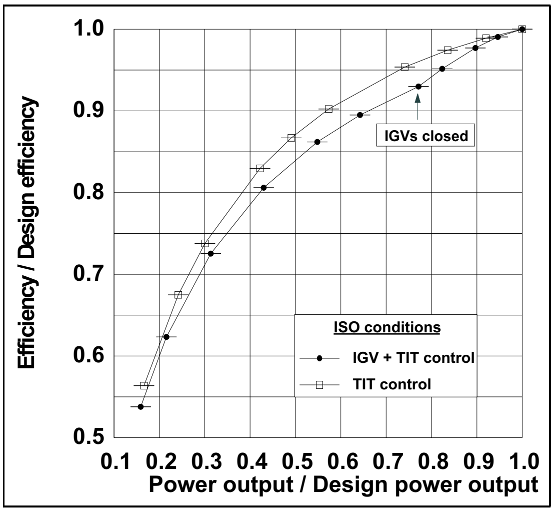

Figure 4: Gas turbine efficiency at part-load.

Figure (5) shows the turbine inlet temperature and compressor pressure ratio for the two part-load methods as load is reduced. It is noteworthy that the TIT method operates with higher pressure ratios than the associated method (IGV/TIT). Moreover, it is interesting to note the differences between the control modes: in IGV control the turbine inlet temperature is kept at its design value (1588 K) and the compressor pressure ratio decreases to match with the turbine owing to the reduction of the compressor air mass flow rate, while in the other method, the pressure ratio diminishes to match the turbine inlet temperature reduction. The two methods analyzed here decreases the load with an associated reduction of the compressor pressure ratio, which has already been remarked by Sehra et al. [13] and Kim and Ro [11].

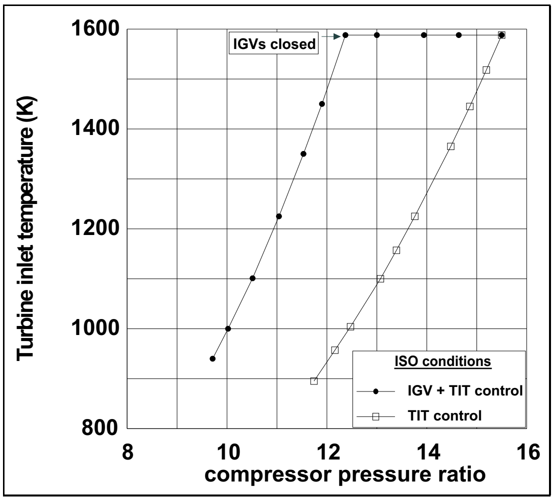

Figure 5: Turbine inlet temperature and compressor pressure ratio at part-load.

Figure (6) introduces the ratio of the exhaust gas temperature at part-load to the exhaust gas temperature at design condition, since this is an important parameter in combined cycle and cogeneration plants.

The exhaust gas temperature is higher with the combined method (IGV/TIT) than with TIT control, which makes IGV method more suitable for combined cycle plants according to the results presented by Gomes and Gallo [9]. Such characteristic is due to the performance of the IGV control that keeps a high efficiency as indicated in Fig. (4), without causing a reduction of the exhaust gas temperature as much as in TIT control. According to Facchini [4] IGV control contributes for a high combined cycle efficiency owing to the delay in the exhaust gas temperature decrease, as may be noted in Fig. (6).

4. Gas turbine sensitivity to environmental conditions

As gas turbines will not operate exclusively at ISO conditions, it is important to predict the sensitivity of its main parameters to environmental conditions (temperature and atmospheric pressure). The correction between design mass flow rate at ISO conditions and other conditions is given by Eq. (10):

$$\frac{\dot{\mathrm{m}}_{\mathrm{iso}} \cdot \sqrt{\mathrm{T}_{\mathrm{iso}}}}{\mathrm{P}_{\mathrm{iso}}} = \frac{\dot{\mathrm{m}}_{\mathrm{corr}} \cdot \sqrt{\mathrm{T}_{\mathrm{amb}}}}{\mathrm{P}_{\mathrm{amb}}} \tag{10}$$

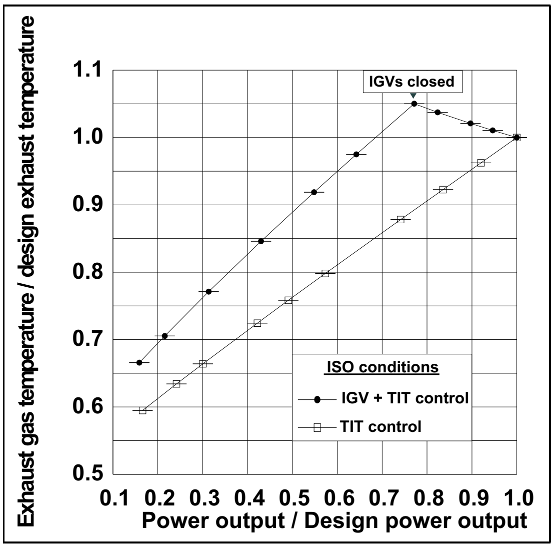

Figure 6: Gas turbine exhaust temperature at part-load.

where $\dot{m}_{corr}$ is the corrected air mass flow rate at compressor inlet for the environmental conditions (T_{amb}, P_{amb}).

The simulation of the gas turbine sensitivity to environmental conditions was carried out considering the same hypothesis for the components described in section 2.2.

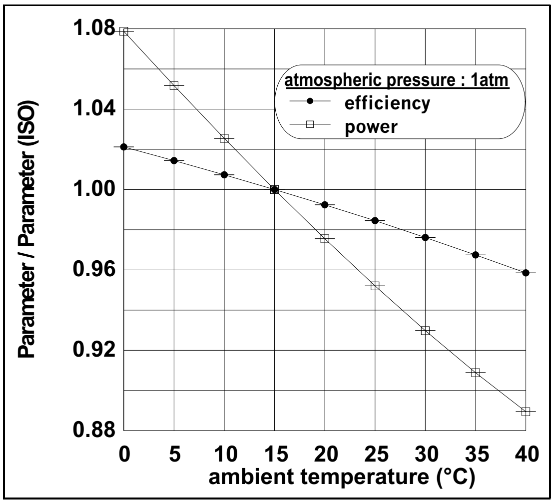

Figure 7: Sensitivity of efficiency and power to environmental temperature.

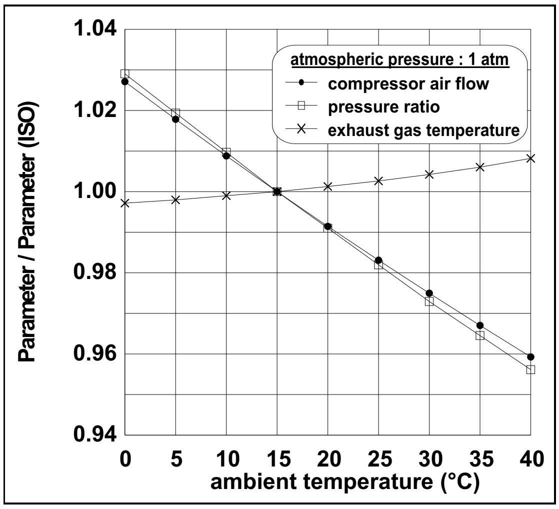

Figure 8: Sensitivity of pressure ratio, air flow and exhaust temperature to environmental temperature.

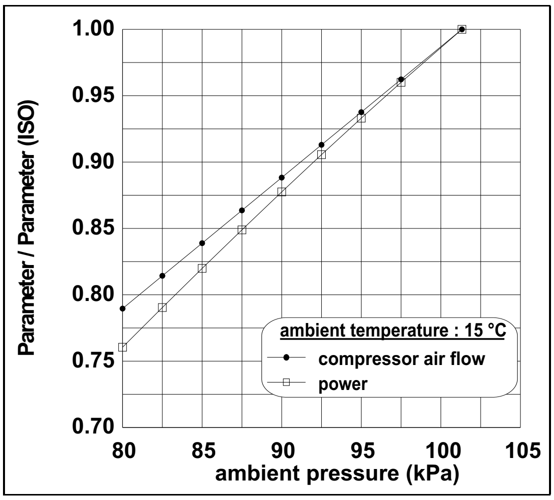

Figure 9: Sensitivity of power and air flow to atmospheric pressure.

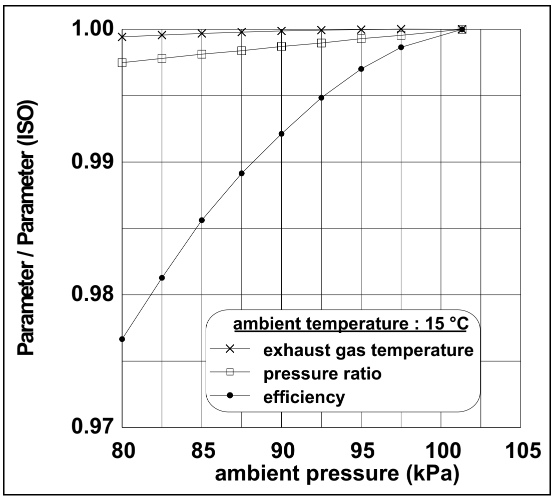

Figure 10: Sensitivity of efficiency, pressure ratio and exhaust gas temperature to atmospheric pressure.

Figure (7) shows the influence of environmental temperature on gas turbine efficiency and power, concluding that power is more sensitive to environmental temperature than efficiency.

Analyzing Fig. (8) it can be noticed that the environmental temperature influences the compressor pressure ratio similarly as the air mass flow rate, with almost no influence on the exhaust gas temperature.

Comparing Figs. (9) and (10) it can be perceived that atmospheric pressure has a remarkable influence on power but almost none on efficiency. As opposed to environmental temperature, atmospheric pressure influences differently the compressor air mass flow rate and pressure ratio, which is practically not sensitive to atmospheric pressure. The exhaust gas temperature is also not influenced by atmospheric pressure.

5. Conclusion

This paper presented a model to predict the behavior of gas turbines at part-load conditions. The model included blade cooling and equations to describe the gas turbine components at part-load.

With regard to the blade cooling method used, the results showed that the greater the gas turbine pressure ratio, the larger will be the air mass flow rate for blade cooling. The gas turbine with blade cooling has an optimum pressure ratio for maximum efficiency lower than that for the case without blade cooling.

The sensitivity analysis to environmental conditions showed that environmental temperature has a greater influence on power than on efficiency. Atmospheric pressure has a marked influence on power and almost none on efficiency.

As regards the two part-load control modes considered for the gas turbine operating in simple cycle, the results pointed out that gas turbine efficiency is higher for turbine inlet temperature control method (TIT) than for the combined method (IGV/TIT). Whichever the part-load control mode, both methods decrease gas turbine efficiency and compressor pressure ratio as load is reduced.

6. Acknowledgments

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