PART LOAD CONDITIONS OF COMPLEX CYCLE POWER PLANTS
WITH INTERCOOLED GAS TURBINE

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ABSTRACT

The present paper evaluates the behavior, in design and part load working conditions, of a complex gas turbine cycle with multiple intercooled compression, and the optional preheating of the air at the high pressure compressor outlet by means of the gas turbine outlet hot gas.

The results are then compared with those obtained by a Brayton cycle gas turbine, with or without preheating of the air at the high pressure compressor outlet.

Subsequently, the performance of complex combined cycles, with intercooled gas turbine as topper and one, two or three pressure level steam cycle as bottomer, in design and part load working conditions is also evaluated.

The performance of these complex combined plants is then compared with that obtained by a Brayton cycle gas turbine as topper and one, two or three pressure level steam cycle as bottomer.

Part load working conditions are realized by varying either the inlet guide vane angle of the first compressor nozzles or the maximum temperature at the combustor outlet.

The study shows that in part load working conditions obtained by varying IGV, the complex cycles, in the examined gas turbine or in the combined cycle power plants, give conversion efficiencies decidedly greater than those obtainable by varying combustor exit temperature.

Furthermore it is found that these complex power plant efficiencies, in part load working conditions, are far greater than those obtained by the Brayton cycle gas turbine, or by combined cycle with Brayton cycle gas turbine as topper, if IGV adjustment is adopted.

If power variation is obtained with combustor outlet temperature adjustment, the efficiencies of the combined power plants with complex or Brayton cycle gas turbines, are substantially the same, for the same relative power variation.

INTRODUCTION

The most recent industrial gas turbine power plants utilize rather complex thermodynamic cycles, such as gas turbines with multiple intercooled compression, with and without regeneration, rather high overall pressure ratio and maximum temperature values.

In design working conditions, multiple intercooled compression cycle gas turbines supply greater specific works than Brayton cycle gas turbines at the same conditions. The intercooled gas turbine may also have greater conversion efficiencies for suitable low pressure compressor pressure ratio values. The greater specific work is due both to the intrinsic features of the thermodynamic cycle and to the benefits of disposing of cooling mass flow rates bled by the compressor at lower temperatures.

Moreover, the present technology of the combined cycle power plants involves increasingly
complex thermodynamic cycles. This is true not only with regard to the gas turbine, but also to the bottomer, whose cycle frequently comprises two or three pressure levels.

The use of two or three pressure level steam cycle gives lower heat recovery boiler exhaust gas temperature than in the one level steam cycle, for assigned gas turbine outlet temperature, leading to higher overall conversion efficiencies.

In part load working conditions all the main parameters are modified, essentially pressures, temperatures, mass flow rates and component efficiencies, with rather different relevance according to the typology and layout of the power plant.

Therefore, in part load working conditions, the performance of these complex cycles, especially the overall conversion efficiency, may also be subject to considerable variations, which are of greater or lesser relevance according to the percentage time in part load working conditions.

The present study deals with the gas turbine cycle with multiple intercooled compression and combined cycle with intercooled gas turbine as topper and one, two and three pressure level steam cycle as bottomer, both in design and part load working conditions.

**NOMENCLATURE**

- \( P \) : power plant shaft power
- \( P_{Ds} \): degasator pressure
- \( P_k \): condenser pressure
- \( T_4 \): GT outlet temperature (or downstream PRH when present)
- \( T_{IT} \): turbine inlet temperature
- \( T_{sh} \): steam outlet superheater temperature
- \( T_{st} \): HRSG outlet temperature
- \( \beta \): overall GT pressure ratio
- \( \beta_{lc} \): LPC pressure ratio
- \( \Delta \eta/\eta \): percentage power plant shaft efficiency variation referring to its value in the design condition
- \( \Delta P/P \): percentage power plant shaft power variation referring to its value in the design condition
- \( \Delta P_{b} \): combustor pressure losses
- \( \Delta P_{ec} \): economizer pressure losses (water side)
- \( \Delta P_{v} \): vaporizer pressure losses (water side)
- \( \Delta P_{h} \): preheater pressure losses
- \( \Delta P_{lc} \): IC pressure losses
- \( \Delta P_{ln} \): LPC inlet pressure losses
- \( \Delta P_{sh} \): superheater and re heater pressure losses (water side)
- \( \Delta P_{st} \): stack pressure losses
- \( \Delta T_{ap} \): approach point temperature difference
- \( \Delta T_{pp} \): pinch point temperature difference
- \( \eta \): power plant shaft efficiency
- \( \eta_{p} \): GT compression polytropic efficiency
- \( \eta_{pc} \): GT expansion polytropic efficiency
- \( \eta_{sh} \): mechanical efficiency
- \( \eta_{ST} \): steam turbine isentropic efficiency

**Acronyms**

- 1p : 1 pressure level steam bottomer cycle
- 2p : 2 pressure level steam bottomer cycle
- 3p : 3 pressure level reheat steam bottomer cycle
- BGT : Brayton cycle gas turbine
- BCC : combined cycle with Brayton cycle GT as topper
- CC : combined cycle
- CT : gas turbine
- HP : steam cycle high pressure
- HPC : high pressure compressor
- HRSG : heat recovery steam generator
- IC : intercooler
- IC1CC : combined cycle with ICGT1 as topper
- ICGT1 : intercooled gas turbine (\( \beta = 24 \))
- ICGT2 : intercooled gas turbine (\( \beta = 30 \))
- IGV : LPC inlet guide vane
- Ip : steam cycle intermediate pressure
- Lp : steam cycle low pressure
- LPC : low pressure compressor
- PRH : HPC exit air preheater

![Fig.1 ICGT schematic layout (case with PRH).](image)

**TOPPER ANALYSIS**

The study was performed on single spool intercooled gas turbine power plants, with and without the preheater exchanger (PRH) located downstream of the expander. The schematic layout with PRH is represented in Fig.1.

Once the performance of the intercooled gas
turbine power plant, with PRH, in design condition was determined, performance variations were evaluated for different power outputs. Furthermore, the design and part load behavior was studied also for an intercooled gas turbine cycle, but without the air preheating at the high pressure compressor (HPC) outlet.

The part load working condition is realized by adjusting either the LPC inlet mass flow rate, obtained by IGV rotation, or the combustor outlet gas temperature.

The part load performance of the two complex gas turbine power plants (with and without PRH) were then compared with those of a common Brayton cycle gas turbine (with and without PRH). Cycle analysis was carried out in all cases taking into account of specific heat variations with temperature and the compounds of the gas mixture working in the plant components. It also considered the pressure losses at the LPC inlet, in the stack, intercooler, combustor and PRH when present.

In the design working condition, evaluation was performed on the cooling mass flow rate of the more thermally stressed parts of the gas turbine power plant machinery under study, employing a calculation code developed in collaboration with the Author, previously presented in Benvenuti et al., 1993, and Benvenuti et al., 1994.

As far as the stage expansion pressure ratio is concerned, it was assumed, as is quite widespread, that the pressure turbine pressure ratio is equally distributed over the turbine stages.

The gas properties (pressure, temperature, etc.) at the exit of each rotor stage were evaluated with a polytropic expansion efficiency that takes into account the cooling mass flow rates and fluidodynamic losses. Cooling mass flow pressures at the compressor bleedings are adequate for the cooled nozzle and rotor pressures.

In the intercooled compression, the HPC inlet pressure was determined by maximizing the conversion efficiency of the complex gas turbine plant, and was therefore different according to whether the preheating of the HPC outlet air was considered or not.

The polytropic compression efficiency was assumed to be the same for all the compressor stages, but different for the two compressors, and in particular, higher for the HPC.

As far as the part load working conditions obtained by varying IGV are concerned, the variation of the LPC inlet guide vane angle leads to a variation in the mass flow rate. It was evaluated by adopting a linear correlation between the off-design/design mass flow rate ratio and IGV angle, considering that the first three nozzle rows of the LPC are adjustable simultaneously.

The mass flow rate variations give rise to changes in the turbine pressure flow inlet, if the same turbine inlet temperature is assumed to hold. More in detail, the inlet/outlet pressure ratio for the turbine changes to a varying extent according to whether the outlet mass flow of the turbine first nozzle is choked or not.

In the former case, the mass flow function at the turbine inlet does not change with the passage from the design to part load cases, and therefore the inlet pressure is directly proportional to the inlet mass flow rate.

In the latter, the mass flow function is also a function of the first nozzle expansion ratio. Thus, the evaluation of the pressure at the first nozzle inlet must be performed iteratively with the constraints that the turbine back pressure is assigned. Moreover also in part load working conditions, the turbine pressure ratio is equally distributed among the stages. This procedure was previously described in Negri di Montenegro and Peretto, 1996.

Once the pressure at the inlet of the turbine stages was estimated, the corresponding temperatures could also be determined, taking into account that the cooling mass flow rates were choked with constant mass flow functions.

It was also assumed that the stage polytropic expansion efficiency changes from design to part load, depending on the isentropic enthalpy drop between turbine inlet/outlet (Benvenuti et al., 1991) due to the variation of the expansion pressure ratio.

Taking into account the pressure losses in the combustor, the above-estimated turbine inlet pressure defines the HPC outlet pressure.

For the evaluation of the HPC inlet pressure, in part load conditions, a non-dimensional map together with energy balance and heat exchange equations in the intercooler were used, according to a procedure previously indicated in Negri di Montenegro and Peretto, 1996.

In addition, the rotational speed was kept constant and the mechanical efficiency linearly reduced with power.

The process described above permits the estimation of the gas turbine efficiency at every power output value.

With regard to the off-design working
conditions due to varying TIT, the part load behaviour is obtained by varying the combustor outlet temperature, while the LPC mass rate remains unchanged. The temperature variations, brought about by varying the fuel mass flow rate in the combustor, mainly influence the pressure and temperature of the fluid mixtures in the expansion through the turbine.

For the evaluation of pressures and temperatures, all the considerations made for the design case hold true. However, it should be noted here that variations of enthalpy drop (and polytropic efficiency) in the turbine are due not only to the variation of its expansion pressure ratio, but in particular to the variation of TIT.

**Topper analysis results**

Employing the above methodology, the complex gas turbine power plant part load performance was evaluated and compared to that of an industrial Brayton cycle power plant. As far as the complex gas turbine is concerned, two industrial machines were considered with same TIT and different pressure ratio. The main design features of the gas turbines are presented in Table 1.

<table>
<thead>
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<th>Table I Gas turbines reference parameters.</th>
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<tr>
<td>( \beta )</td>
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<tr>
<td>PRH</td>
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<tr>
<td>P [MW]</td>
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<td>PRH</td>
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</table>

In addition, the following were assumed:
- ambient air temperature: 15 \(^\circ\)C;
- relative humidity: 60%;
- LHV (natural gas): 49500 \(kJ/kg\).

Figs.2 and 3 represent the curves of efficiency versus power for the intercooled and Brayton cycle gas turbines, for the cases without and with preheating of the HPC outlet mass flow rate, respectively.

It should be observed that, in the design working conditions both with and without PRH, the intercooled gas turbine has a higher efficiency than the Brayton cycle gas turbine. Moreover, the ICGT2 efficiency is greater than that of the ICGT1, if PRH is not considered (Fig.2), while the opposite is true if PRH is introduced (Fig.3).

Fig.3 shows that, if IGV is adjusted, in all three cases, the employment of PRH allows the maintenance of conversion efficiencies not far from the relative design value, also in the case of high power variations.
Fig. 3 $\eta$ versus $P$, obtained by varying TiT or IGV, for the three GT considered (case with PRH).

**BOTTOMER ANALYSIS**

Figs. 4 and 5 show the schematic layout of common two and three pressure level steam bottomer cycles.

Fig. 4 Two pressure level steam bottomer schematic layout.

Fig. 5 Three pressure level reheat steam bottomer schematic layout.

For the complex and Brayton cycle gas turbines considered, the steam pressure level values and corresponding distribution of the steam mass flow rates in the heat recovery steam generator were determined so as to optimize the overall combined cycle conversion efficiency. For this optimization it is necessary to set the minimum temperature difference values in the exchangers and maximum steam temperature.

For each heat exchanger inside the HRSG, the gas and water side pressure drops have been taken into account.

In the design working conditions the steam mass flow rate and the heat exchange surface for each heat exchanger inside the HRSG is determined, once the minimum temperature difference value and the overall heat exchange coefficient are set (Stecco and Desideri, 1991).

The bottomer cycles analyzed here, with one two and three pressure levels, were obtained by determining for each of them the pressures which optimize the efficiency of the combined cycle having a Brayton cycle gas turbine or an intercooled gas turbine (with $\beta=24$) as topper.

The main steam cycle parameters for the two toppers considered are listed below in Tables II and III.

The part load working condition, due to the IGV or TiT adjustment, substantially determines a variation in the thermal power introduced in the HRSG. This gives rise to a variation in the water
mass flow rate through the heat exchangers that is iteratively calculated using the heat exchange surface determined in the design analysis and the overall heat exchange coefficient varied as a function of the gas mass flow rate through the exchanger considered (Dechamps, et al., 1994).

The steam turbines work in sliding pressure, also in the case of the low pressure turbine thus accepting that the gas stack temperature decreases to 95°C, for the maximum load variation considered. The low pressure steam turbine efficiency is varied, taking into account the exhaust losses at the discharge, as expressed in Bartlett, 1958.

A water mass flow rate recirculation in the economizers is adopted to assure that the fluid is liquid at the inlet of the evaporators for all the working conditions considered.

Table II Bottomer combined cycle reference parameters with Brayton cycle GT as topper.

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<td>( H_p ) [MPa]</td>
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Table III Bottomer combined cycle reference parameters with ICGT1 as topper.

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**COMPLEX COMBINED CYCLE ANALYSIS**

The performance of the above-mentioned combined cycles in part load working conditions has been analyzed. The results obtained with the above methodology are reported in the figures below.

Fig.7 relates to the combined cycle configurations with one pressure level steam cycle as bottomer and ICGT1 or BGT as topper. The curves represent the overall efficiency versus power, for the two types of adjustment analyzed. It can be observed that in design working conditions the IC1CC has a slightly greater overall efficiency than BCC, and that in part load the IGV adjustment gives higher efficiencies for IC1CC than BCC.

Fig.8 differs from Fig.7 only in relation to the bottomer, in this case a two pressure level steam cycle. The efficiencies of IC1CC and BCC, in the design working conditions, are higher than those found for the previous cases, respectively. As in the previous figure, here too, IGV adjustment permits higher efficiencies for IC1CC than for BCC, in part load working conditions.

In Fig.9 the bottomers of the two plants consist of three pressure level reheat steam cycles. The efficiencies in the design working conditions are higher than in the previous cases. In this figure, as in the previous ones, IGV adjustment allows higher efficiencies for IC1CC than for BCC, in part load working conditions.

To compare the part load percentage efficiency penalization for the two plants analyzed (IC1CC and BCC) and for the two types of adjustment considered, Fig.10 shows the percentage efficiency variations versus percentage power variations for the plant configurations in Fig.9. It can be observed that the intercooled combined cycle power plant maintains its efficiency very close to the design value, until reaching a power variation of about 20%, if IGV adjustment is adopted. In the same power variation range BCC significantly varies its efficiency. On the contrary TIT adjustment gives no benefit, compared to the same regulation in the combined plant with the Brayton cycle as topper.
Fig. 7 $\eta$ versus $P$, obtained by varying TiT or IGV, for IC1CC and BCC (case of 1p).

Fig. 8 $\eta$ versus $P$, obtained by varying TiT or IGV, for IC1CC and BCC (case of 2p).

Fig. 9 $\eta$ versus $P$, obtained by varying TiT or IGV, for IC1CC and BCC (case of 3p).

Fig. 10 $\Delta \eta/\eta$ versus $\Delta P/P$, obtained by varying TiT or IGV, for IC1CC and BCC (case of 3p).
CONCLUSIONS

The analysis carried out in this paper on the performance of complex cycle power plants with an intercooled gas turbine has shown the advantages of this kind of gas turbine.

In the design conditions, the efficiencies of the intercooled gas turbine plants considered in this study reaches about 43% against about 36% for the Brayton cycle gas turbine. Moreover, the adoption of the preheating of the high pressure compressor outlet air mass flow rate with the gas turbine exhaust gas, increases the efficiencies to about 47% for the intercooled gas turbine versus about 42% for the Brayton cycle gas turbine.

In part load working conditions, the intercooled gas turbine gives higher efficiency than the Brayton cycle gas turbine throughout the range of power variation examined, when preheating is absent. It can be observed that the efficiency variation due to one or the other type of adjustment is substantially the same if a Brayton cycle gas turbine is considered. Conversely, in the case of the intercooled gas turbine, the part load efficiency values are very different according to the type of adjustment employed. In fact, if a power variation of about 50% is considered, for intercooled gas turbine with β=30, efficiency passes from about 42% to about 37%, and from about 42% to about 32% if IGV or TiT adjustment is adopted, respectively.

If preheating is considered, adjustment with IGV becomes significantly favorable also for the Brayton cycle gas turbine and permits the intercooled gas turbine to maintain efficiency values very close to design one over a wide power range (until about the 50% of design power).

As regards the use of the intercooled gas turbine as topper in combined cycle applications, it can be seen that in the design working condition this solution gives overall conversion efficiencies very close to that of the combined plant with a Brayton cycle gas turbine as topper. Such efficiency values increase when passing from one to three pressure level steam bottomer cycle.

In part load working conditions, if TiT adjustment is adopted, efficiency variations are not substantially affected by the kind of gas turbine considered. In this case, taking as example the three pressure level reheat steam bottomer cycle, there are significant efficiency variations that reach about 25% for power variations of about 50%.

In part load working conditions, if the IGV adjustment is considered, the combined cycle with intercooled gas turbine as topper gives higher efficiency than that with a Brayton cycle gas turbine as topper, throughout almost all the range of power variation examined. For example, if a combined cycle with three pressure level reheat steam bottomer cycle is considered, with a power variation of 50%, the combined cycle with intercooled gas turbine penalizes its efficiency by only about 5%, while the combined cycle with a Brayton cycle gas turbine reaches about 10%.

Finally it can be observed that the use of the intercooled gas turbine alone, with or without preheating, or in combined cycle applications, especially with a three pressure level reheat steam bottomer cycle, gives high conversion efficiencies in design working conditions. These efficiency values remain very close to the design value also in part load, if IGV adjustment is adopted, rendering the employment of these complex power plants very attractive.

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