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Optimization of inlet air cooling systems for steam injected gas turbines

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Abstract

The cooling of the inlet air in gas turbines is a practice used to improve power performances. In the past, the authors suggested extending this practice to STIG turbines, using some of the heat generated in the back boiler to supply the absorption systems. Positive results have prompted further developments. For the purpose of verifying the positive effects of compression air cooling in STIG turbines, a calculation model has been developed and studied specifically for gas turbines and STIG turbines. The model has been applied to the Allison 501 KH turbine.

In addition to focusing on the two traditional techniques used to cool the air of the compression system through an absorption unit and an intercooling unit, the present work emphasizes the advantages of using an ejection cooling system. The availability of exhaust heat from the STIG turbines prompted the idea of cooling through a double ejection system. Water cannot be used as a primary refrigerant in compression cooling systems because of the very low pressures that have to be reached, whereas the ejection system used for steam compression allows for the use of water. Moreover, the ejection system is relatively easy to design and construct. The use of this cooling system, enables good results to be obtained in terms of the ejection system's coefficient of performance and consequently of the STIG turbine's performance.

1. Introduction

The performances of gas/steam mixed turbines depend not only on the quantity of steam or water injected, but also on variations in the operating conditions in relation to ISO conditions. The performances of new, clean machines supplied with natural gas are considerably influenced by suction pressure drops and by the ambient temperature and pressure of the inlet air.

A variation in the ambient pressure implies a variation in the cycle mass flow and power. A variation in the ambient temperature influences the exhaust temperature of the compressor, the internal temperature of the turbine, the external temperature of the turbine, the mass flow, the specific work, the specific consumption and the power. When the ambient temperature drops, the power supplied by the machine increases. That is why it is useful in many cases to cool the compressor inlet air with a view to obtaining a greater production of electric power.

The mixed gas and steam cycles can be divided into three categories: the category of water injection cycles, in which water is injected into one or more points of the gas cycle; the category of humid air cycles (HAT cycles), in which water is injected through a saturator; and the category of steam injection cycles, called STIG cycles. The present work deals with the optimization of gas cycles with steam injection through cooling of the compressor intake air. The performance of the machines was assessed considering three different techniques for cooling the intake air:

- with a single or double-stage absorption-chiller, using the exhaust gases of the turbine as a heat source;
- with intercooled compression achieved by using two compressors and an intercooler;
- with a steam ejection system using the exhaust gases of the turbine as a heat source.

Though the practical applications are still limited and concern the medium and small powers, the increasing number of publications in the field of STIG turbines has shown the great interest of the scientific and industrial world in these cycles. The lack of experimental data, due to difficulties in obtaining the necessary information from turbine constructors (bound by business secrets) and the chance to collect data on the cogeneration plant of the city-owned public utility in Osimo Italy, which uses the Allison

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Nomenclature

COP	coefficient of performance	
EHV	equivalent heat value	$kJ \cdot kg^{-1}$
ISTIG	intercooled steam injected gas turbine	
h	heat content	$kJ \cdot kg^{-1}$
m	mass flow	$kg \cdot s^{-1}$
P	power	kW
s	entropy	$kJ \cdot kg^{-1} \cdot K^{-1}$
STIG	steam injected gas turbine	
T	temperature	K
TIT	turbine inlet temperature	K
TOT	turbine outlet temperature	K
χ	mole fraction	
β	compression ratio	
η	efficiency	

Subscripts

a	air
b	compressor discharged
comp	compressor
cool	cooling
dif	diffuser
el	electrical
f	fuel
g	gas
ic	intercooling
id	ideal
l	liquid
mix	mixing
noz	nozzle
refr	refrigeration
st	steam
ther	thermal
turb	turbine
wa	waste
0	total

501 KH turbine [1] prompted the plan to develop a model to assess the performance of this type of turbine [2,3]. A further calculation model was created and applied to verify the optimization of the cooled STIG cycle. This model enables an estimation of the performance of certain selected, small and large power gas turbines, of certain gas turbines with steam injection, and of others cooled through a single or a double-stage absorption chiller, an ejection system or an intercooled compression system.

2. The absorption refrigeration plant

The characteristic feature of an absorption cooling system, by comparison with the more traditional cooling systems such as those with steam compression or gas, is the need for a supply of heat energy, instead of mechanical energy. Its use is certainly advantageous when waste heat from the heat engines working in cogeneration conditions is available. The absorption cooling system may be a single or double stage system. The single stage machine has only one concentrator, whereas the double-stage machine has two different concentrators. The present study considered two absorption plants with lithium bromide.

If we analyze the STIG cycle applied to the Allison 501 KH turbine, which we suggest implementing through the application of a commercially-available single-stage absorption chiller, the best configuration achievable is the one presented in Fig. 1. For this configuration, we calculated the temperatures at the end of the compression, the turbine exhaust temperatures, the powers absorbed by the compressor and the electric powers generated, with changes in ambient temperature and relative ambient humidity. The results of applying the model to the Allison 501 KH are shown in Figs. 2 and 3.

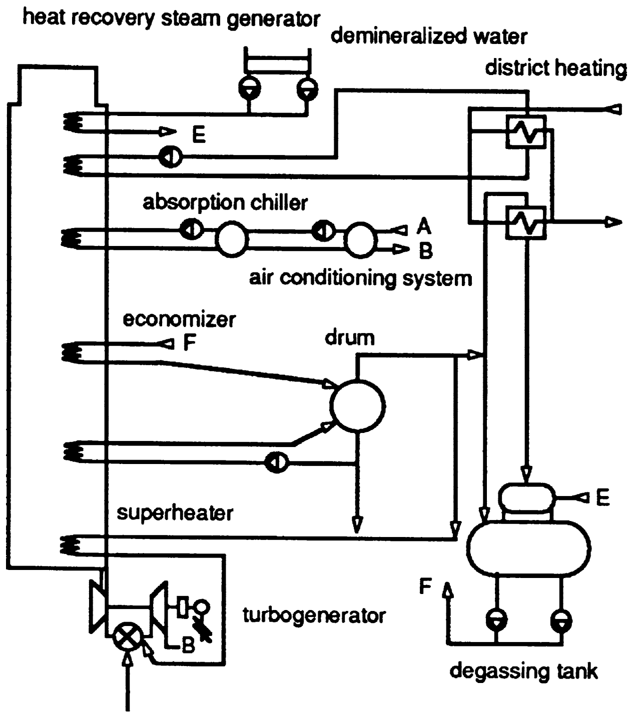

Fig. 1. Lay out of the STIG system provided with an absorption chiller.

If we define the dimensionless quantity $X^{@}$ as the ratio between the thermodynamic quantity when the temperature of the inlet air changes and the thermodynamic quantity when the inlet air temperature equates to 15 °C, illustration 2 shows the trends of the temperature coming from the dimensionless compressor, $T_b^{@}$ and of the temperature emerging from the dimensionless turbine TOT$^{@}$. Fig. 3 shows a simi-

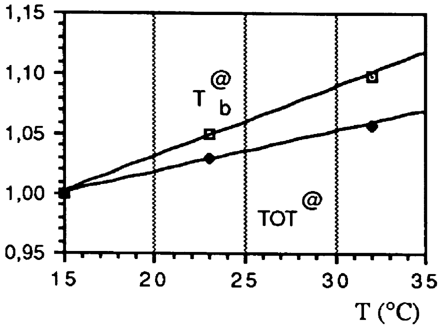

Fig. 2. Dimensionless typical temperatures of the STIG cycle cooled with an absorption chiller.

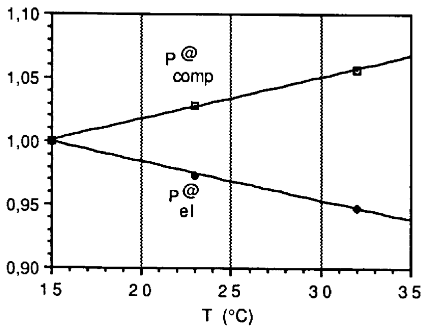

Fig. 3. Dimensionless powers of the STIG cycle cooled with an absorption chiller.

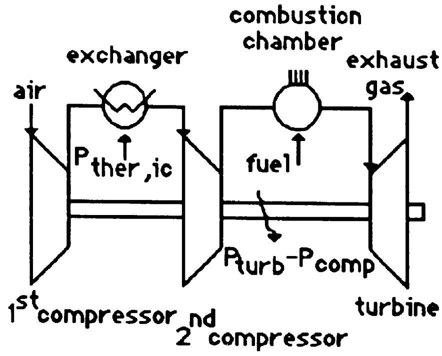

Fig. 4. Scheme of an ISTIG system.

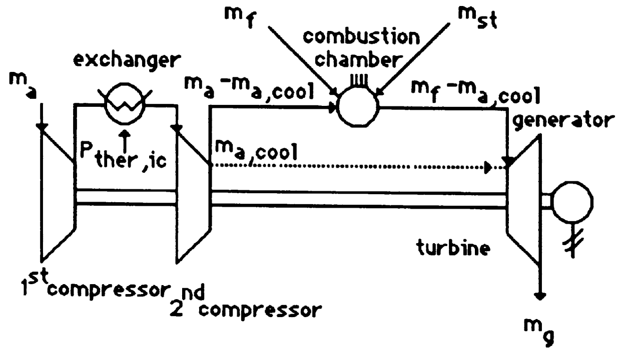

Fig. 5. ISTIG system equipped with two compressors and an intercooler.

lar graph, describing the trend of the dimensionless power of compression $P^{@}_{\mathrm{comp}}$ and of the dimensionless electric power $P^{@}_{\mathrm{el}}$. The points plotted on the graph of Figs. 2 and 3 refer to the inlet air cooling hypotheses adopted in the present study, i.e., from 32 °C (relative humidity 50%) to 23 °C (relative humidity 64%); from 23 °C (relative humidity 64%) to 15 °C (relative humidity 100%).

Fig. 2 shows the rising trend of the dimensionless temperatures (@) when the ambient temperature rises. Fig. 3 describes the rising trend of the dimensionless compression power (@) and the falling trend of the electric dimensionless power (@) when the inlet air temperature rises. According to the hypothesis adopted, if the inlet air temperature is dropped from 32 °C to 15 °C, the compression power decreases by 5.6% and the electric power increases in approximately the same way.

3. Intercooled compression

With a view to enhancing the performance of gas cycles, a common practice adopted in compression systems is intercooled compression, achieved through an intercooler that reduces the temperature of the inlet air for the second compressor (Fig. 4).

The purpose of this intercooling is to reduce the compression work, which has a positive effect on the net work and on the cycle's efficiency, though this latter effect is not so obvious. In fact, as the cycle work increases, there will also be an increase in the inlet heat. If we add intercooling to a simple ideal cycle, its efficiency will suffer. In a real cycle, however, the efficiency of the cycle could be superior to that of the simple cycle. Whether or not this is the case will depend on how much the cycle departs from the ideal conditions. Intercooling is more likely to have beneficial effects on the efficiency of small power machines too, which are characterized by normal yields. Moreover, intercooling reduces the final air compression temperature, which offers some very important design advantages, e.g.,

– a drop in the temperature of the air flow used to cool the turbine blades;
– a reduction in the cooling flows with a consequent reduction in the irreversibility;
– a rise in the internal temperature of the turbine with the same cooling flow, with a consequent improvement in the specific work;
– an increase in the machine's total compression ratio.

Since the practice of intercooled compression in gas cycles is used successfully to improve the cycle's performance, it is logical to study its particular features for the ISTIG cycles too. Fig. 5 shows how the system with a STIG cycle is modified by the two compressors and the intercooler.

In the present work it is assumed that the two compressors being used have a similar compression ratio, that the temperature of the air leaving the intercooler is 25 °C for small power systems, that there will be an increase in the

Table 1
Features of the compressor of the Allison 501 KH, at the maximum steam injection, processed on the basis of the application of the calculation model

Compressor			
β (dimensionless)	m_a ($kg \cdot s^{-1}$)	P_{comp} (kW)	T_{0b} (°C)
11.7	15.05	5.490	373

Table 2
Features of the combustion chamber of the Allison 501 KH, at the maximum steam injection, processed on the basis of the application of the calculation model

Combustion chamber			
m_{st} ($kg \cdot s^{-1}$)	t_{st} (°C)	m_f ($kg \cdot s^{-1}$)	EHV ($kJ \cdot kg^{-1}$)
2.349	437	0.301	44.870

Table 3
Features of the Allison 501 KH turbine, at the maximum steam injection, processed on the basis of the application of the calculation model

Turbine			
m_g ($kg \cdot s^{-1}$)	T_{st} (°C)	TOT (°C)	P_{el} (kW)
17.7	977	494	5330

quantity of fuel in the combustion chamber and consequently of the heat power introduced as a consequence of the lower temperature of the chamber's inlet air.

In order to assess how intercooling influences the features of the STIG turbine cycle, data regarding the application of the calculation model to the Allison 501 KH turbine are given below. The calculation model has to be supplied with the input data of the compressor, combustion chamber and turbine. For the compressor these are the inlet temperature, the compression ratio and the adiabatic efficiency. For the combustion chamber these are the injected steam flow, the steam injection temperature, the stoichiometric ratio, the fuel flow, the lower heat power of the fuel. For the turbine they are the adiabatic efficiency, the maximum possible temperature on the blades, the wasted powers.

Tables 1, 2 and 3 show some of the characteristic dimensions of the Allison 501 KH turbine. A maximum steam injection was hypothesized for this machine, as recommended by the relevant technique. Table 1 concerns the compressor, Table 2 the combustion chamber, Table 3 the turbine.

If we define the dimensionless quantity $X^{\#}$ as the ratio between the quantity when the steam flow injected in the combustion chamber varies and the quantity for the non-intercooled STIG cycle, when the injected steam flow equates to the maximum one, Figs. 6–11 show the features of the Allison 501 KH, to which the calculation model was applied.

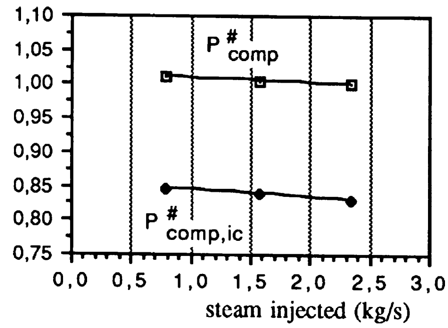

Fig. 6. Comparison of the power of compression between the STIG and the ISTIG cycle.

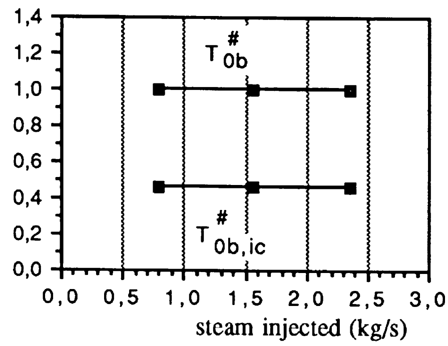

Fig. 7. Comparison of the exhaust compression temperatures between the STIG and the ISTIG cycle.

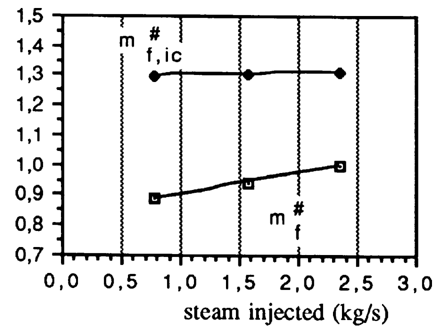

Fig. 8. Comparison of the supply fuel between the STIG and the ISTIG cycle.

In Fig. 6 the dimensionless power of compression is plotted for the non-intercooled STIG cycle and the ISTIG cycle. The figure shows a dramatic reduction in the compression power needed by the two compressors, by comparison with the single compressor used in the operation of the non-intercooled STIG cycle, and consequently a reduction in the specific absorbed work.

Fig. 7 shows the significant drop in the temperature of the air coming out of the second compressor of the ISTIG system, by comparison to the temperature of the non-intercooled STIG cycle.

Since the temperature of the compressed air is lower in the ISTIG cycle than in the STIG cycle, the combustion chamber clearly demands a larger quantity of fuel in the

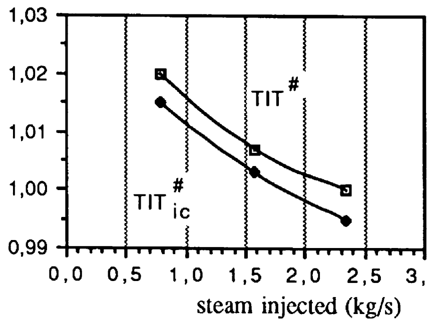

Fig. 9. Comparison between the internal temperature of turbine between the STIG and the ISTIG cycle.

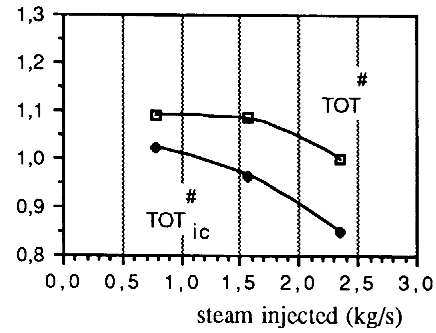

Fig. 10. Comparison between the exhaust temperatures of the turbine between the STIG and the ISTIG cycle.

ISTIG cycle with variations in the quantity of injected steam (see Fig. 8).

By comparison with the non-intercooled STIG cycle, the ISTIG turbine has several advantages when the quantity of steam injected in the combustion chamber varies. In spite of the increase in the quantity of fuel there is an increase in the efficiency. By keeping the cooling flow for the turbine blades constant, the drop in the temperature of the cooling flows will cause a drop in the temperature of the inlet gases of the turbine. This effect is described in Fig. 9. If we hypothesize reducing the cooling flows, because of the considerable reductions in the blade temperatures in the ISTIG system, it will be possible to obtain an increase in the TIT with considerable advantages in terms of the electric power generated. If we observe Fig. 9 we can also deduce that, as the steam flow increases, the TIT of the turbine decreases. In fact, because of steam injection, the flows, heat capacities and Reynolds numbers of the fluid passing through the turbine vary. They increase the capacity for heat exchange as the injected steam increases and consequently foster the rise in the temperature of the blades.

The drop in the temperature of the cooling flows of the turbine implies a consequent decrease in the TOT (Fig. 10). The lower demand for global power absorbed by the compression system implies an increase in the electric power delivered by the turbine. The calculations, using the model for assessing the performance of the machine during the STIG and ISTIG cycles, show a remarkable increase

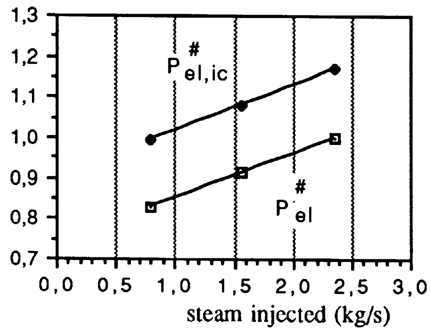

Fig. 11. Comparison between the electric powers of the STIG and the ISTIG cycle.

in the electric power, amounting to approximately 17% (Fig. 11).

To sum up, the practice of intercooled compression applied to the STIG cycle examined offers great advantages, such as the decrease in the compression power, the increase in the generated electric power, the increase in the electric efficiency of the system, a greater efficiency of the cooling flows due to the low temperature of the compressor outlet air. However, alongside these advantages, the disadvantages are: the system becomes more complicated due to the presence of two compressors and an intercooler; the flow of the supply fuel is dramatically increased; and a cold source is needed.

4. The ejection refrigerating system

In refrigerating systems with mechanical steam compression, because of the very low pressure that has to be reached in the evaporator (and the consequent presence of very large specific volumes), water cannot be used as a primary refrigerant when temperatures are lower or equal to 0 °C, though water has the best features from the point of view of safety in use and costs.

The ejector used for compressing water vapor enables the use of water as a primary refrigerant in cooling units that use compression systems other than the direct mechanical one.

Fig. 12 shows that the ejector is essentially composed of a convergent-divergent nozzle, a mixing chamber and a diffuser with a divergent profile.

In the nozzle of ejection refrigerating systems, water vapor is expanded (usually at a pressure of 2 bar); it increases its speed as the pressure decreases, thus entraining the steam coming out of the evaporator. In the mixing chamber there is an exchange of momentum between the two steam flows, which combine and retain a high residual speed at the diffuser inlet. Here the kinetic energy is converted into pressure energy. The compression effect allows the consequent condensation, which takes places thanks to the use of a refrigerating source. In the evaporator, water, which is thinly dispersed by special spray nozzles, evaporates at the saturation temperature corresponding to the low pressure

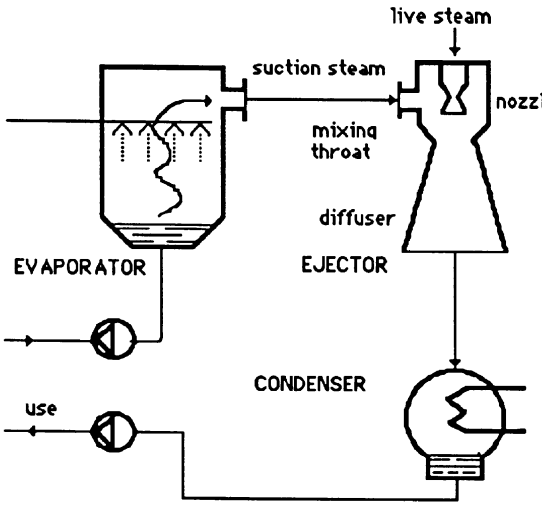

Fig. 12. Scheme of an ejection cooling unit.

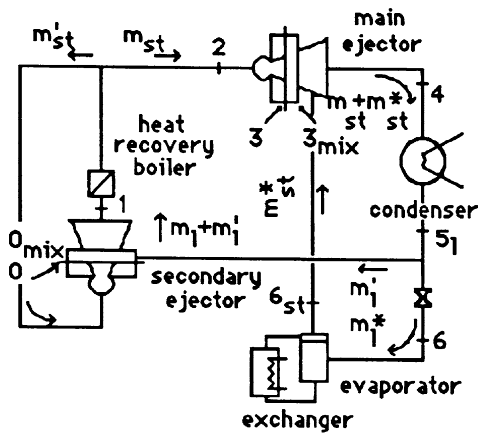

Fig. 13. Scheme of the steam ejection refrigerating system made up by two ejectors.

maintained by the ejector, and produces the refrigerating effect.

Fig. 13 shows the layout of the steam ejection refrigerating system comprising two ejectors. The steam produced inside a cylindrical body, installed on a STIG system, acts as a power fluid for the main ejector. Here the steam flow m_{st} is mixed with the flow m^*_{st}, coming from the evaporator, attracted by the ejector. As a consequence of this, the flow coming out of the main ejector has the thermodynamic features described in point 4 of diagram T–s in Fig. 14. This steam flow reaches the condenser. After being condensed, it splits into two flows: m'_l, and is sucked into the secondary ejector, where it is mixed with the steam flow m_{st}: after being laminated, m^*_l is subsequently sucked into the main ejector. As illustrated in diagram T–s in Fig. 14, there are two circuits in which evaporation takes place and a circuit in which condensation takes place. $m_l + m_{l^*}$ evaporate due to cooling of the back boiler fumes. $m_{st} + m_{st^*}$ condense at the expense of an external fluid which increases its temperature. m_{l^*} evaporates and thus generates the cooling effect.

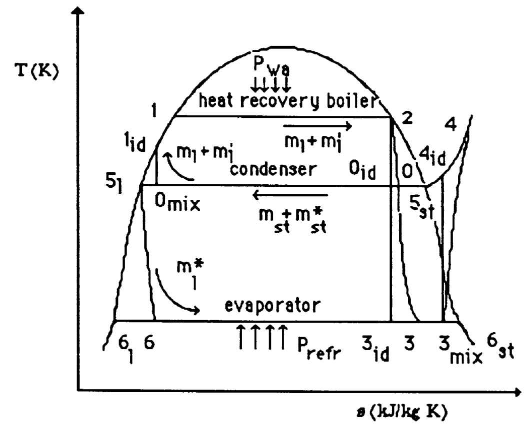

Fig. 14. Cooling cycle with steam ejection.

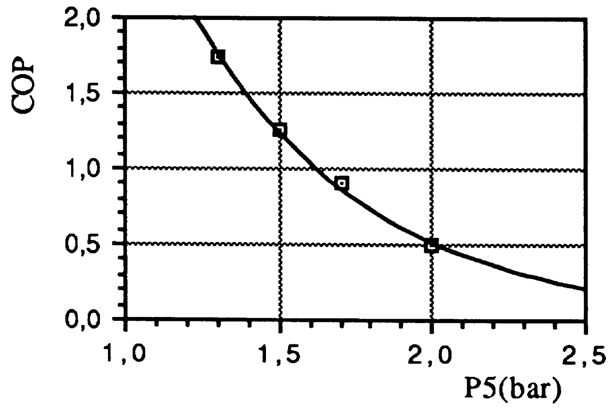

Fig. 15. Variation of the COP concerning a fixed pressure value to the cylindrical body equal to 3 bars, related to the pressure of the condenser.

In the same operational conditions of the STIG systems described above, where the absorption chillers, using the exhaust heat of the gas machine, cool the air from 32 °C to 15 °C, we analyzed the features of the system comprising the Allison 501 KH, cooled by an ejection system.

The following hypotheses were made:

– water evaporation pressure $p_6 = 0.007$ bar;
– nozzle efficiency $\eta_{noz} = 0.85$;
– diffuser efficiency $\eta_{dif} = 0.80$;
– pressure of the steam produced inside the back boiler $p_2 = 3$ bar;
– pressure to the condenser $p_5 =$ various values.

Under these hypotheses we studied the variation in the COP, i.e., the variation in the ratio between the refrigerating power obtained and the heat power absorbed in the back boiler. The results are presented in Fig. 15.

When the pressure to the condenser p_5 increases, the thermodynamic quantities of the cycle act in such a way as to cause a decrease in the coefficient of performance. By applying the calculation model with a view to changing the pressure in the cylindrical body and keeping the ratio between the pressure in the cylindrical body and the one in

Table 4
Values of pressure and temperature in points number 2, 5, 6 of the ejection cycle

p_2 (bar)	p_5 (bar)	p_6 (bar)	T_2 (°C)	T_5 (°C)	T_6 (°C)
3.0	1.3	7×10	133	107	1.84

Table 5a
Values of the enthalpy in the points of ejection cycle

h_0 (kJ·kg^{-1})	h_1 (kJ·kg^{-1})	h_2 (kJ·kg^{-1})	h_3 (kJ·kg^{-1})	h_4 (kJ·kg^{-1})	h_{5st} (kJ·kg^{-1})
2601	1587	2726	2042	2578	2687

Table 5b
Values of the enthalpy in the points of ejection cycle

h_{6st} (kJ·kg^{-1})	h_{0id} (kJ·kg^{-1})	h_{3id} (kJ·kg^{-1})	h_{4id} (kJ·kg^{-1})	h_{5l} (kJ·kg^{-1})	h_{6l} (kJ·kg^{-1})
2504	2580	1921	2563	449	8

Table 6
Values of the entropy in the point of ejection cycle

S_1 (J·g^{-1} ·K^{-1})	S_2 (J·g^{-1} ·K^{-1})	S_{5l} (J·g^{-1} ·K^{-1})	S_{5st} (J·g^{-1} ·K^{-1})	S_{6l} (J·g^{-1} ·K^{-1})	S_{6st} (J·g^{-1} ·K^{-1})
1.67	6.98	1.38	7.27	0.028	9.10

Table 7
Values of fluid speed at the exit of the respective secondary and main nozzles of the ejection cycle and values of the title

c_0 (m·s^{-1})	c_{0mix} (m·s^{-1})	c_3 (m·s^{-1})	c_{3mix} (m·s^{-1})	χ_{0id}	χ_{3id}
498	249	1169	398	0.95	0.76

Table 8
Values of flows, powers and COP of the ejection cycle

m (kg·s^{-1})	m' (kg·s^{-1})	m^* (kg·s^{-1})	P_{wa} (kW)	P_{refr} (kW)	COP
0.10	0.10	0.20	244	428	1.74

the condenser constant, there will be an increase in the COP as the pressure of the cylindrical body increases.

In order to verify the behavior of the ejection cooling system, Tables 5–8 show the values of the thermodynamic quantities calculated for the Allison 501 KH turbine under the conditions shown in Table 4. The equations employed for the calculation have been solved by using iterative methods.

5. Conclusions

In the past several studies have focused on the installation of a cooling system in order to enhance the performance of gas turbines [4–12]. Taking into account the abundant technical and scientific literature on thermodynamics and on the design of STIG plants, in this work the performances of these cycles were analyzed under the hypothesis of compressor inlet air cooling. The optimization of the gas cycles with steam injection was studied in detail to verify the reduction in the power absorbed by the compression system and in order to evaluate the corresponding increase in the electric power that this practice brings to the STIG cycles. For this purpose we considered:

– the fitting of an absorption-chiller with lithium bromide in the back boiler;
– the fitting of an intercooled compression system comprising two compressors and an intercooler;
– the fitting of a steam ejection system in the heat recovery steam generator.

To verify the behavior of these systems, it was necessary to create a program for calculating the performances of gas turbines with simple cycles and with steam injection. In particular, the program enables the optimization of the system to be analyzed in detail. The calculation program that we developed was applied to a small size turbine: the Allison 501 KH.

Optimization of the steam injection Allison 501 KH turbine with inlet air cooling, using the above-mentioned techniques, produced positive results and a remarkable enhancement of performance was observed in the cases studied.

The fitting of a commercially-available single-stage absorption chiller makes it possible to increase the electric power by approximately 5.6%. This is because of a similar decrease in compression power when the supply air temperature of the machine falls from 32 to 15 °C. Under these conditions, there is a drop in the temperature at the end of the compression by 9.9% and a drop in the exhaust temperature of the turbine by 5.8%.

In order to evaluate the effects of the technique for compression intercooling, we considered setting the value of the inlet air temperature and injecting a given variable quantity of steam in the machine. Under these conditions, we calculated the decrease in compression power, the increase in inlet fuel quantity, the drop in temperature at the end of the compression, the corresponding drop in the turbine's internal temperature and exhaust temperature and the increase, by approximately 17%, of the electric power generated.

Finally, we proposed the technique of compression inlet air cooling through an ejection system supplied by the exhaust heat of the gas turbine. This offers the opportunity to obtain good results using water as a primary refrigerant. An economic feasibility study should be made on the new configuration, which is straightforward from the design and construction point of view. Probably the costs will prove limited, thus offering the opportunity to enhance the performances of STIG turbines through the use of a very simple system for compressor inlet air cooling.

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