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PERFORMANCE IMPROVEMENTS OF A NATURAL GAS INJECTION STATION USING GAS TURBINE INLET AIR COOLING

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ABSTRACT

Gas Turbine (GT) performance seriously deteriorates at increased ambient temperature. This study analyses the possibility of improving GT power output and efficiency by installing a gas turbine inlet air cooling system.

Different cooling systems were analyzed and preliminary cost evaluations for each system were carried out.

The following three cooling systems were considered in detail:

a) Traditional compression cooling system;
b) Absorption single-acting cooling system using a solution of lithium bromide;
c) Absorption double-acting cooling system using a solution of lithium bromide.

Results clearly indicate that there is a great potential for GT performance enhancement by application of an Inlet Air Cooling (IAC). Technical and economical analyses lead to selection of a particular type of IAC for significant savings in capital outlay, operational and maintenance costs and other additional advantages.

INLET AIR COOLING EFFECTS

A gas turbine IAC system may be used to increase machine power as well as efficiency, under the same ambient conditions.

Performance improvements are as follows:

1) net available power increases (due to decreasing inlet air temperature, increasing density and thus increase in machine mass flow rate)
2) increase in machine efficiency (due operation closer to design conditions).

Fig. 1 shows the main operating trends of a GE gas turbine model MS5002-C vs ambient temperature. A 20°C inlet air temperature increase causes a of 15-18% power loss.

Various authors have demonstrated the competitiveness of IAC both in the production of electric power peaks (Ebeling et al, 1992, Ebeling et al, 1994), combined plant for base load production (Vand Der Linder et al, 1996) and cogeneration plants (De Lucia et al, 1993, De Lucia et al, 1995, Utamura et al, 1996). The possibility of using these systems for natural gas compression stations is still to be investigated.

AIR COOLING SYSTEMS

The air cooling process is greatly affected by two factors: the initial cooling temperature and relative humidity. The latter parameter, even though not particularly affecting gas turbine behavior, does affect cooling capacity considerably. To better understand the importance of this parameter it should be noted that the power required to cool air from 30°C to 15°C practically doubles (increasing by 110%) if the relative air humidity is 90% instead of 50% (passing from 24 kJ/kg to 51 kJ/kg). All this must be carefully considered when selecting the most appropriate IAC.

Let us examine the various IAC systems available. The commonest IACs applicable to gas turbines may be classified in two different categories:

Evaporative Cooling Systems

They are systems exploiting the latent heat of water vaporization in a process of adiabatic air saturation permitting temperature reduction from a dry to a wet bulb value. Therefore, their success in realizing higher ΔT depends on the relative air humidity contents (the nearer one gets to 100% in terms of relative humidity, the lower is the obtainable ΔT). These systems are particularly simple and economical and are suitable for hot, dry climates rather than hot, humid ones, where they are unable to reach satisfactory ΔT.

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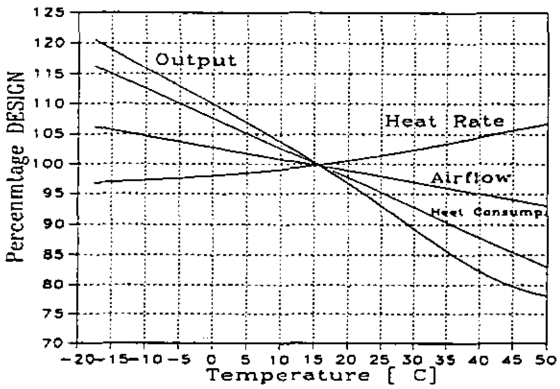

Figure 1 Effect Of Compressor Inlet Temperature (MS5002 C)

Cooling systems

They are more complex systems where, by using thermodynamic cycles, the heat is transferred from a low to a higher temperature source, exploiting the fluid property of vaporization or condensation at a different temperature as a function of pressure.

These systems are the most suitable for application in a hot, humid climate. The Coefficient Of Performance (COP)[1], which indicates the efficiency of cooling cycles, is defined as the ratio between the heat removed at the cooling stage Q_{refr} and the energy used to perform this operation L_u. The thermodynamic cycles usually used for industrial cooling systems are of two types: compression system and absorption system.

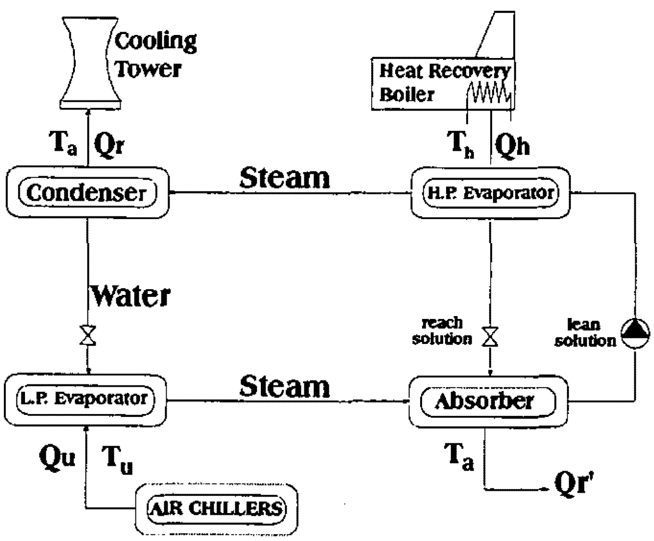

Figure 2 Absorption Machine Scheme

1 $COP = \frac{Q_{Refr}}{L_u}$

Compression cooling system This is the most conventional system where the circulation of a suitable fluid between different pressure areas is carried out mainly by using the mechanical power supplied by a compressor. It transfers the fluid from a lower pressure area (evaporator) to a higher pressure one (condenser).

Absorption cooling system. The absorption cooling system uses heat instead of the mechanical power of compression system as its energy source. This allows the recovery and use of waste heat coming from the gas turbine exhaust gases, by simply installing a heat recovery boiler.

A very small quantity of mechanical energy is required in any case to drive the cooling fluid and cooling water circulation pumps.

The COP of these systems is lower with respect to traditional compression cycles, but this parameter is not a valid comparison element, since in its evaluation the difference between mechanical power and thermal power is not taken into account. Moreover, in our case, considering that the required heat is available and, if not used, would be somehow scattered in the atmosphere, it is a zero cost source.

Absorption cycles are to be considered technologically very mature with an increasing market. In 1992 40% of large type chillers were absorption systems, and they have reached excellent reliability levels in relation to compression systems. Usually, they employ a solution of water and lithium bromide that is used to absorb steam acting as a cooling fluid. Because of the type of fluid used, temperatures lower than 4°C cannot be reached, in order to avoid problems of lithium bromide crystallization in the solution. They are characterized by very favorable behavior in part load operating conditions (De Lucia et al, 1993), and this makes them particularly suitable for applications where the required power is extremely variable.

COP can be expressed as follows (for symbol details see Fig 2):

$$COP = \frac{Q_u}{Q_h + W_p} \cong \frac{Q_u}{Q_h} = \left(\frac{1}{T_a} - \frac{1}{T_h}\right) / \left(\frac{1}{T_u} - \frac{1}{T_a}\right)$$

Both single-stage and double-stage machines are available on the market. Single-stage machines are older, very simple and use low pressure (about 1.5bar) and relatively low temperature saturated steam or even overheated water. They reach COP values of about 0.7. Double-stage machines are more sophisticated and may reach a COP value, of 1.1-1.2, according to the highest value of the evaporator temperature (T_h) of the heat source that requires a higher supply steam pressure (about 8 bar) from the heat recovery boiler

COMPRESSION STATION DESCRIPTION

The high pressure compression station considered is designed to recover the natural gas coming from the oil treatment system and re-inject it into the oil reservoir, to maintain pressure and enhance oil recovery.

The required duty is to compress 373 Nm^3/s (1200 MMSCFD) starting from a suction pressure of 75 bar (110 psi) up to a discharge pressure of 620 bar (9000 psi) (to keep the reservoir pressure of 540 bar). The natural gas [2] flow at station inlet has a temperature of 49°C (120°F). The overall service is achieved by gas turbine driven

2 (molecular weight = 21 and composition of 80% C1, 8% C2, 6% CO_2, 3% C3 and 3% others)

centrifugal compressor trains operating in parallel, each one consisting of three compression phases, with intermediate gas cooling. The total power required by the gas compressors is: 134 MW.

Compressors operate between 56-68 Nm^3/s (180-218 MMSCFD) by passing from the normal operating condition to that with maximum load achieved at 105% of nominal speed.

AMBIENT CONDITIONS

Ambient condition data are very important for a correct selection and evaluation of the cooling system. This compression station is located in an area particularly unfavorable climatewise; it is a typically equatorial climate (humid and hot). Average main characteristics, over the last 25 years (temperature and rainfall), are shown in Figs 3 and 4.

Though limit values of 37.8°C (100°F) and 97% relative humidity were recorded, for the plant design the following values were assumed as design ambient parameters:

Air temperature	32.2°C (90 F)
Relative humidity	82%

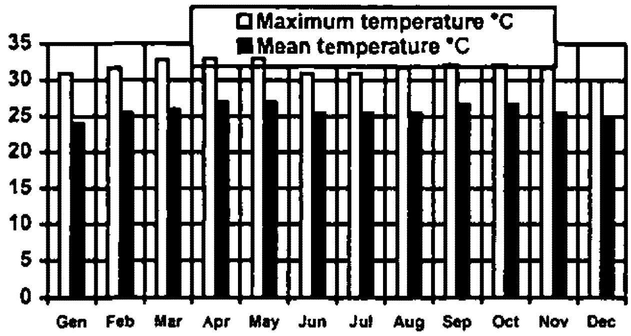

Figure 3 Maximum And Monthly Site Temperature

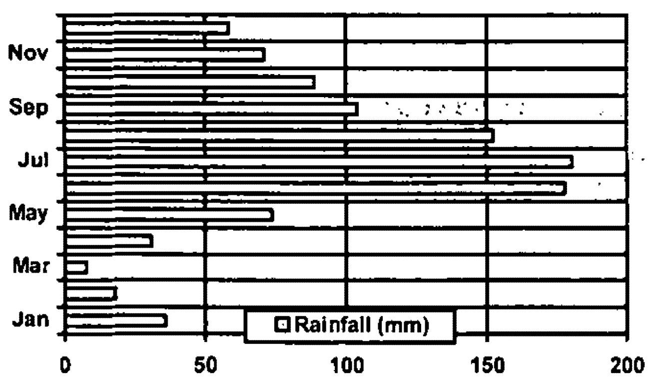

Figure 4 Average Monthly Rainfall

Fig. 5 shows the average power available form each GT as a function of the monthly average temperature, compared with that available at 10°C. Power losses, up to 3-4 MW, can be noted (referring to 10°C air temperature), not only in the worst periods (i.e. Summer), but also throughout the year.

Fig. 6 shows the performance increase in term of both power and efficiency achieved by cooling the air at turbine inlet at 10°C.

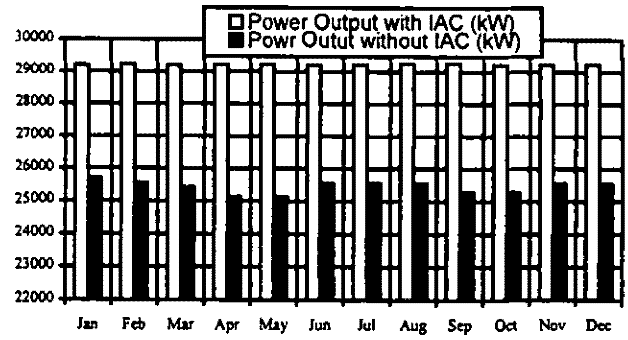

Figure 5 Monthly On-Site Gas Turbine Power

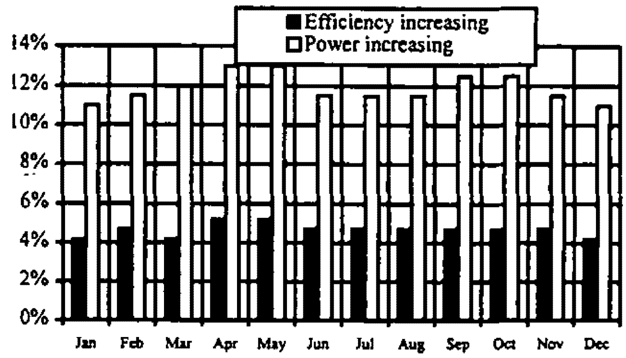

Figure 6 Monthly Efficiency And Power Increase

INJECTION STATION CONFIGURATION

The project (without IAC) is based on 6 operating turbo-compressor units plus a standby one, all driven by a gas turbine model MS5002-C. Each of them treats 17% of total station capacity and each has a power margin of 10% (on design conditions 32.2°C and 82% relative humidity), whilst, under the worst conditions, characterized by an ambient temperature of 37.8°C, they are hardly sufficient to provide service (22970 kW_{TG} against 22400 $kW_{COMPRESSOR}$).

Should only 5 operating units be available, the total capacity that the station can treat would decrease to 83% of the total design value. It can be increased to 91%, by exploiting the available power margin completely (if temperature does not exceed 32.2°C).

Compression station data may be summarized as shown in table 1, where the behavior of single GT units, the compression station and the power margin are shown in relation to the ambient conditions and the number of operating GTs.

For those solutions adopting 5 compressor trains, the value of the total power margin, evaluated on the injection compressors, is higher, due to the increased efficiency achievable by larger machines (by 1-2%).

It is possible to achieve the required compression duty only with 5 machines by keeping a power margin of 10-11% cooling air at 10°C. Air cooling down to 15°C would also allow the required service to be provided, but with a slight power margin, to take into account the unavoidable performance losses caused by machine wear, compressor fouling etc.

Temp [°C].	Relative Humidity %	GT power [kW]	No. of operating GT	Power Station [MW]	Total Power Margin
15	60	28350	6	170.1	26.6%
25	100	26100	6	156.6	16.5%
37.8	82	23000	6	138	2.7%
32.2	82	24660	6	147.96	10%
10	100	29300	6	175.8	31%
10	100	29300	5	146.5	11%(*)
15	100	28350	5	141.7	7.6%(*)
5	100	30330	6	181.98	35.4%
5	100	30330	5	151.65	15.1%(*)

(*) Due to the compressor increased efficiency the compressor power margin will be higher

Table 1

PROPOSED STATION WITH IAC

The station consists of 5 operating turbo-compressor units plus a standby one, driven by the same gas turbines and equipped with an IAC system. In this case the injection compressors are designed for a unitary mass flow rate of 74.6 Nm^3/s, which will be kept constant thanks to the IAC systems.

TYPES OF COOLING SYSTEM ANALYZED

For comparisons between the different systems an identical heat power was considered for all solutions taken into account. This value is required to cool the air at the turbine inlet from reference ambient conditions (32.2°C and 82% relative humidity) to a temperature of 10°C. The heat power required for IAC is about 8 MW for each unit. The heat exchange batteries to be installed in the gas turbine air suction system are identical for the various cases taken into account.

Evaporative cooling system

It is particularly simple and economical, but suitable for hot, dry climates. Its success in realizing higher ΔT depends on the relative air humidity contents at the system inlet. Considering the unfavorable ambient condition described above, this kind of cooling system is not suitable and therefore not considered here.

Propane cooling system

In this case, a propane cycle operating at pressures ranging from 4 to 20 bar, corresponding to -5°C and 81°C respectively, was chosen. A solution of water and glycol whose temperature is to be kept above 0°C, in order to avoid icing on the heat exchange battery surfaces on air side, circulates at the evaporator. The cooling plant is characterized by a COP=2.283 and requires a mechanical power of 17.5 MW. The project also provides for the splitting of the propane compression service into two compression units. For this purpose, it is necessary to provide for two units with centrifugal compressors driven by Nuovo Pignone gas turbines model PGT10 that, under design ambient conditions, have a slight power margin of 5-6% (without providing for IAC).

Fig. 7 shows a detail of station layout with a compressor cooling system.

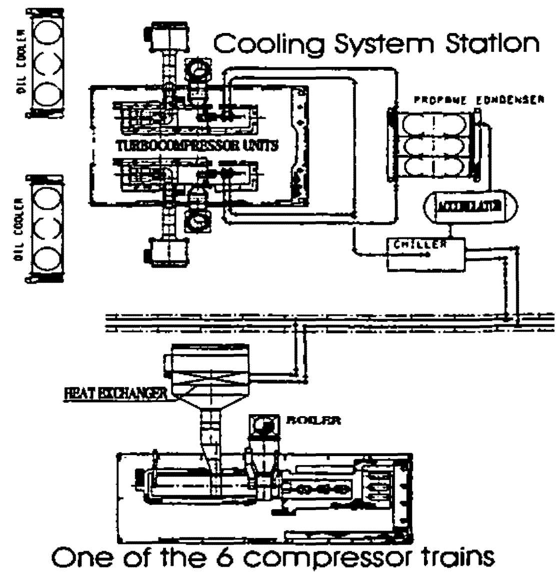

Figure 7 Plant Layout With Compressor Cooling System

Advantages and disadvantages

Advantages

- High temperature at the condenser that permits reduction of its dimensions and increases in its efficiency, even in the case of high ambient temperature and humidity.
- Suitable for installation in various climates

Disadvantages

- High installation cost
- High operating cost mainly due to fuel consumption
- High maintenance cost of the propane turbo-compressor
- Poor performance at part load conditions.

ITEM	SIZE
Turbo-compressors	2 x 10,6 MW
Propane condenser	65 MW
Propane accumulator	100 m^3
Propane evaporator	40 MW
Air chillers	6 x 8 MW
Piping e auxiliaries	--
Total cost (USD*10^6)	**11.5-12**

Table 2 Costs of Compression cooling System

Cost analysis. Table 2 shows an estimate of the costs of the different main components of the compression cooling system.

This type of plant has the disadvantage of high running costs caused by fuel consumption. Supposing a "fuel gas" with Lower Heat Value (LHV) of 35000 kJ/kg, at the price of 0.02 US$/kg were used, an average daily consumption of 140000 kg/day can be estimated. This

	SINGLE STAGE		DOUBLE STAGE	
ITEM	*SIZE*	*COST (USD)*	*SIZE*	*COST (USD)*
Heat recovery boilers	6 x 11.5 MW	1 170 000	6 x 7.5 MW	680 000
Absorbers	12 x 4 MW	4 000 000	12 x 4 MW	5 800 000
Air chillers	6 x 8 MW	1 000 000	6 x 8 MW	1 000 000
Cooling towers	10 x 9 MW	900 000	8 x 9 MW	720 000
Pumps	3 x5000m^3/h	470 000	3x4000m^3/h	400.000
Piping + Auxiliaries	--	750 000	--	650.000
Total cost	--	8 290 000	--	9 250 000

Table 3 Absorption Cooling System Costs

corresponds to an average yearly expenditure of about $1,000,000 i.e about 9-10% of the total investment cost of the cooling plant. The maintenance costs of the other two gas turbines required to drive the cooling unit and of the propane compressors are to be added to the above costs for an exhaustive cost evaluation.

Absorption cooling system

Considering the size of the absorption machines available on the market, we deemed it suitable to use 2 lithium bromide units, 4,000 kW each, to cover all the cooling power required for each turbine. The air cooling plant (see Fig. 8) consists, therefore, of the following main equipment:

- 12 absorbers;
- 6 small low pressure heat recovery boilers (8 bar), using about 15% of heat power recoverable at the gas turbine exhaust,
- water cooling towers and circulation pumps

In this solution mechanical energy is only required to drive the water circulating pumps.

There is a clear difference between single-stage and double-stage systems. The double-stage absorber has a higher efficiency (almost double the single stage) and requires less energy supply, thus lowering the costs of all the auxiliary equipment. The single-stage absorber at present is much more economical (by about 30%) than the double-stage one.

Plant efficiency may be further increased by using cold water (at about 10°C) returning from the heat exchangers (whose capacity may reach 10 m^3/h) for the replenishment of the cooling towers. Thus, there is a reduction in the thermal jump of the water in the cooling towers, that can reach 7% under the worst conditions.

The condensate water coming from the GT inlet heat exchangers (about 10 m^3/h at 10°C) can also be used for different purposes within the plant due to its favorable characteristics (low dissolved mineral content etc.).

The performance of the absorption cooling cycle has a remarkable capacity for improvement, thanks to the great interest of industry in this type of technology. Considering that there is still much room for improvement (maximum declared COP on double-stage 1.2 against a theoretical one of 2.9 with $T_h = 180°C$), in the future one can expect great things from these systems.

In the future, the use of ammonia absorption systems, at present not available in the size required by these plants, may allow IAC systems down to temperatures of c.4-5°C, as is normally done with ICE-STORAGE systems, to further reduce the number of compression trains.

Cost analysis. Table 3 shows an estimate of the costs of the main elements of the absorption cooling plant, in the two alternatives considered:

Advantages and disadvantages

Advantages

- installation costs 30-40% lower than compression cooling system
- Complete absence of fuel consumption.
- A negligible electric power consumption
- High flexibility and good performance at part load conditions.
- Low maintenance costs

Disadvantages

- Large cooling towers due to low ΔT available
- Inlet air temperature obtainable cannot be lower than 10°C, due to lithium bromide crystallization

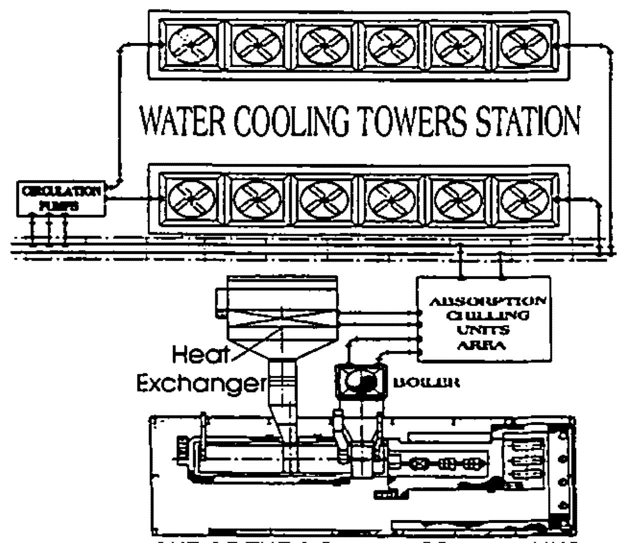

Figure 8 Plant Layout With Absorption Cooling System

COMPARATIVE CONSIDERATIONS

Table 4 shows an estimate of all costs involved in relation to the conventional compression station (without IAC). Thanks to the saving of a complete natural gas injection unit, the capital cost will be reduced by 10-11% of the total station cost. Both the fuel and the maintenance cost for the compression cooling system are higher, due to propane compression trains.

In the case of the absorption cooling system, the fuel costs are expected to be 13% lower than the conventional compression station fuel cost

ITEMS	Compressor cooling system	Absorption cooling system
Capital costs	-10%	-11%
Fuel costs	+2%	-13%
Operating & maintenance	10%	-6%

Table 4 Cost comparison refereed to the conventional station

CONCLUSIONS

The use of gas turbine inlet air cooling turned out to be very advantageous in terms of performance in particularly hot climates. The different IAC systems available at present were investigated:

a) Evaporative cooling system: it is particularly simple and economical, but suitable only for hot, dry climates.
b) Compression cooling system: it is suitable for many different climatic conditions. However, it requires higher O&M costs.
c) Absorption cooling system: its installing cost is slightly lower than the previous one. The O&M cost is markedly reduced but requires a large area to be installed.

All things considered, choosing the best IAC technique allows one:

- to save a complete natural gas compression unit for about US$40÷50,000,000
- to reduce O&M costs of the compression station by at least 10-15%
- to operate all the machines under design conditions independent of climatic conditions.

We can conclude that for our application the absorption cooling system is much more advantageous than other systems.

Even though double stage absorbers are much more efficient and offer several advantages with respect to single stage ones, at present their higher costs cancel any advantage related to the reduction in size and cost of heat recovery boiler, cooling towers, pumps, piping and auxiliary.

ACKNOWLEDGMENTS

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