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Benefits of Compressor Inlet Air Cooling for Gas Turbine Cogeneration Plants

Compressor inlet air cooling is an effective method for enhancing the performance of gas turbine plants. This paper presents a comparative analysis of different solutions for cooling the compressor inlet air for the LM6000 gas turbine in a cogeneration plant operated in base load. Absorption and evaporative cooling systems are considered and their performance and economic benefits compared for the dry low-NO_x LM6000 version. Reference is made to two sites in Northern and Southern Italy, whose climatic data series for modeling the variations in ambient temperature during the single day were used to account for the effects of climate in the simulation. The results confirmed the advantages of inlet air cooling systems. In particular, evaporative cooling proved to be cost effective, though capable of supplying only moderate cooling, while absorption systems have a higher cost but are also more versatile and powerful in base-load operation. An integration of the two systems proved to be able to give both maximum performance enhancement and net economic benefit.

Introduction

Gas turbine performance is strongly dependent on environmental conditions, especially air temperature. For example, for heavy frame gas turbines, at a temperature of 35°C (95°F), power output decreases as much as 20 percent with respect to ISO conditions (15°C, or 59°F, and 60 percent relative humidity), while heat rate increases by approximately 5 percent, and all these figures are almost independent of machine rating (De Lucia et al., 1994). The same is also true for aeroderivative gas turbines like the GE LM6000, whose performance charts show great sensitivity to ambient conditions, with maximum power output for an inlet air temperature of about 10°C (50°F) (Casper, 1993). The dry low-NO_x version shows similar but smoother behavior, with a nearly constant power output for inlet air temperatures less than 8°C (46°F), decreasing above this value (Fig. 1).

In summer months, when ambient temperature in temperate climates can differ widely from nominal condition values, adding an air cooling system at the compressor inlet can restore nominal air conditions. In this way power generation is increased and specific consumption reduced. Cooling system design must take into account actual environmental conditions for the plant site, mainly air temperature and relative humidity. Then, data series over a period of some years should be collected, in order to use models for climate condition simulation reliably (DeLucia et al., 1994). However, differences in gas turbine plant types and applications, e.g., simple or combined cycle, base or peak-load operation, and electric or thermal load following mode, can impose different air cooling solutions. Moreover, in evaluating and comparing different air cooling plant solutions and their pros and cons, one has to take into account the strong influence of the tariff system, which often prevents energy efficiency and economies from coinciding (Hill, 1992).

The choice of the most suitable cooling system, or combination of systems, will then be based on economic and technical considerations concerning the three main means of air cooling: (1) evaporative cooling, (2) compression refrigeration systems with or without thermal energy storage (TES), and (3) absorption chillers.

Evaporative cooling is the least complicated and least expensive solution for cooling inlet air. However, its cooling power is limited by the wet bulb temperature, so that its performance can be severely reduced for high-humidity sites and periods. Therefore, for high temperatures, which are common in summer months, it cannot cool the inlet air back to nominal conditions, but can only provide moderate cooling, variable with ambient conditions. Kolp et al. (1995) discuss the performance of such a system, in both simple and combined cycles, including supercharging.

Compression refrigeration systems, with or without TES, are much less dependent on ambient conditions. They can cool inlet air to low temperatures (down to 4°C, or 39°F, if necessary), but are more expensive and, without TES, have the disadvantage that their efficiency decreases considerably with decreasing load. Moreover, they use electricity as a primary energy source, and this can be a drawback during peak hours, when electricity demand reaches its maximum. By using an ice-based TES system, the latter disadvantage can be partly overcome. The compression refrigeration unit will produce ice during low-cost off-peak electricity periods, while during the day, ice-cooled water will be used to reduce inlet air temperature (Ebeling et al., 1994, 1992). System complexity and equipment size appear to be a major drawback, especially if the plant has to be operated in base load.

Absorption chillers (DeLucia et al., 1994) have the advantage of using thermal energy instead of electricity. They are more expensive than compression cooling without TES, but they can adapt to variable load, virtually without efficiency decrease, and appear to be suitable for base load service. Their lowest temperature limit is 6–7°C (44–45°F), and therefore they cannot be used with an ice-based TES system. The two-stage absorber version is more efficient and requires less energy and cooling water than its single-stage counterpart.

Description of the Plant

The plant considered in this paper produces both electricity and steam in cogeneration for an industrial user, in base-load operation. This plant is an interesting example of integrated energy production. The realization of several such plants is

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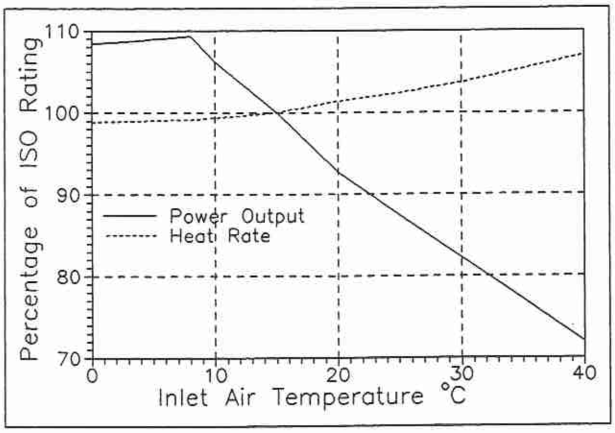

Fig. 1 Performance chart for GE LM6000 dry low-NO_x

scheduled in the next few years in Italy on various industrial sites, where there are industrial users capable of using the thermal energy produced. In the case of gas turbine combined plants, cogeneration operation for both electricity and heat production adds further constraints on heat production and enhances the gas turbine section with respect to the lower steam plant. These constraints can make the use of an inlet air cooling section for a cogeneration plant even more economical than for a normal combined cycle plant.

The plant layout is shown in Fig. 2. The topper is a GE LM6000 dry low-NO_x in combined cycle, which is equipped with a two-pressure level unfired HRSG, which produces steam at 40 bar (580 psia) and 8 bar (116 psia). The low-pressure steam is purchased by an industrial user, who needs it on a three eight-hour shift basis, i.e., round-the-clock operation (e.g., paper and pulp, food processing, or building materials sectors). Steam delivery to the industrial user is a priority and must be constantly guaranteed, i.e., the plant is operated in a thermal load following mode. The high-pressure steam expands in a turbine where two steam bleedings take place at 8 and 2.5 bar (36.3 psia), to integrate the low-pressure steam flow rate, in case the steam demand of the industrial user exceeds low-pressure boiler production. The condenser, at a pressure of 0.05 bar (0.73 psia) with cooling towers, is capable of condensing the whole steam production of the HRSG. For cogeneration plant simulation, the performance chart for inlet air temperature variation for the dry low-NO_x LM6000 gas turbine was used (Fig. 1).

Cooling Section Design

The installation of an inlet air cooling section was planned to enhance plant performance during summer months. The compression refrigeration option turned out to be hardly viable in the present application. Since the plant is designed for base-load operation, a compression refrigeration system, without TES, would consume electricity during the day, when it is most expensive, and its efficiency is strongly affected by part-load operating conditions. On the other hand, an ice-based TES for cooling inlet air back to 10°C, during the whole day in summer's hottest months, would require large equipment size, increasing plant complexity, and investment costs.

As for the absorption system, we considered both single- and two-stage configurations. Two-stage absorbers have a COP (coefficient of performance) of 1.2, almost double that of single-stage ones, typically 0.7 (ASHRAE, 1989). On the other hand, two-stage units use steam at 8 bar, whereas their single-stage equivalents need a larger steam flow rate, but at a lower pressure (2.5 bar). Since in both cases the steam is bled from the steam turbine, the cost of the electricity not produced must be taken into account for a correct comparison. Considering condenser pressure level and an average steam turbine efficiency of 0.85, the electricity not produced is 58.3 kWh per GJ of cooling thermal load in the case of the two-stage absorber (61.5 kWh/MBtu), and 77.8 kWh per GJ of cooling thermal load in the single-stage one (82.0 kWh/MBtu). The latter's incremental electricity loss amounts to 32 percent. A further element of comparison is the cooling water required by the unit: The single-stage absorber utilizes a cooling water flow rate equal to one and a half times that of two-stage units of equal power.

As a result, the following solutions were considered:

- Case A—Two-stage absorption chiller.
- Case B—Single-stage absorption chiller.
- Case C—Evaporative cooling.

Subsequently, a further case (Case D) was also considered as an integrated solution with evaporative cooling and a two-stage absorption unit.

The single- and two-stage absorption units considered had a cooling power of 3300 kW (939 tons): enough to reduce the temperature of air flow rate at LM6000 compressor inlet from 25°C (77°F) (at 60 percent relative humidity) to 10°C. For higher air temperatures and similar humidity conditions, air cooling diminishes, because the amount of water condensing during cooling increases. So as to maintain plant load at a level not lower than 20 percent of design conditions, the absorber should work only when ambient air temperature is higher than 15°C. Total inlet pressure loss in all cases can be calculated, at a conservative estimate, as not exceeding a 25 mm (1 in.) water-head.

Climate Effect Modeling

A crucial factor in the size and performance of an air cooling section is the climate of the chosen site. Climatic variations were simulated, using the model described by De Lucia et al. (1994), for the purpose of making a comparison between the three solutions. This model assumes that the temperature depends on solar radiation and imposes maximum temperature when solar radiation is at its highest, and vice versa. Thus the ambient temperature follows a sinusoidal curve between maximum and minimum measured temperature values. The amplitude of this curve depends also on the length of the period of light, the latitude of the site, and the meteorological day considered. The model was modified in order to take into account variations in nighttime temperature, which was assumed to be constant by De Lucia et al. (1994). A further modification was introduced with the aim of a more accurate simulation of

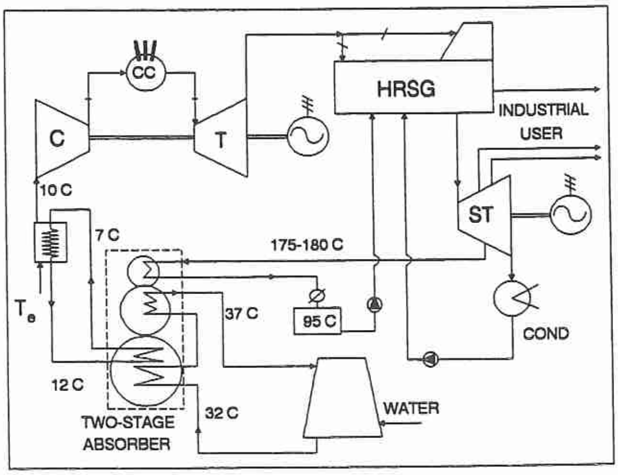

Fig. 2 Plant layout (two-stage absorber)

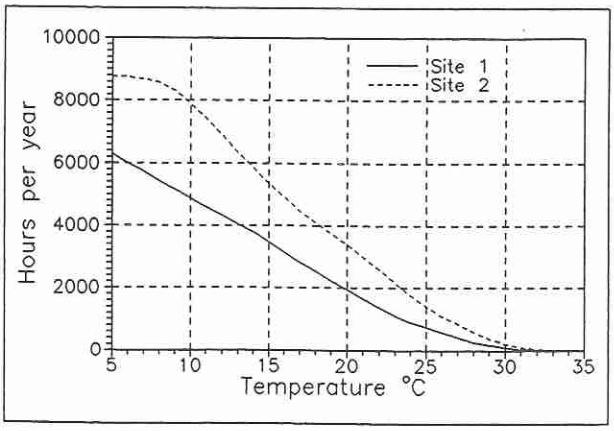

Fig. 3 Cumulative annual hours for ambient temperature

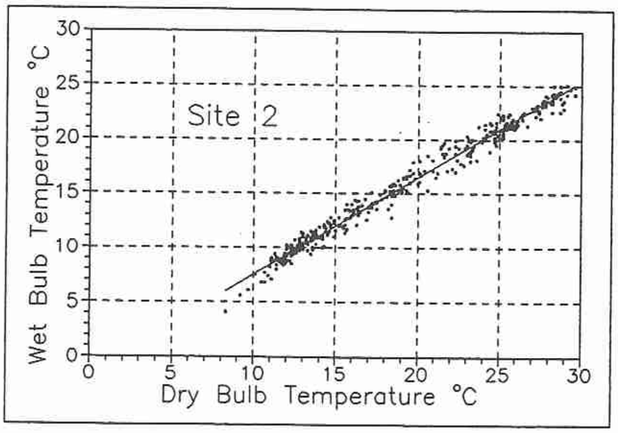

Fig. 5 Wet and dry bulb temperature correlation (site 2)

daytime temperature variations, considering the time shift always present between ambient temperature and solar radiation. It was verified that by imposing the maximum temperature one hour later than maximum solar radiation, the measured values available could be fitted more correctly.

In order to evaluate climatic influence on plant performance, two sites were considered, which are representative of two different types of climate. At the first site considered (Site 1, Northern Italy, latitude 45°08′ N) the ambient temperature is higher than 15°C for approximately 3500 hours per year (40 percent of total working hours). Most of these hours are obviously in summer (96 percent of total summer hours) and spring (47 percent of total spring hours), while in fall and winter the air cooling system would only seldom be activated, if it operates for external temperatures higher than 15°C. The second site (Site 2, Southern Italy, latitude 38°13′ N) has a higher average temperature all over the year (ambient temperature > 15°C for 61 percent of total working hours). In these conditions an absorption unit would always be in operation in summer months, while in spring and fall the percentage would also remain considerably high (respectively, 66 and 62 percent). Figure 3 shows the cumulative annual hours in which the ambient temperature is higher than the value on the x axis. In comparison with site 1, we can see that site 2 ambient conditions appear more favorable for the installation of an air cooling system, which would be much more intensively exploited.

Wet bulb temperature evaluation is of the utmost importance for correct design of the evaporative cooling section, and is also a very important parameter for absorption systems. The wetter the inlet air, the higher the amount of condensation in the cooling coils for the same final temperature. The result is an increase in thermal load on the chiller, without air temperature reduction, and with decrease in cooling section performance. In order to calculate wet bulb temperature, average daily wet and dry bulb temperature values were correlated. The graphs, reported in Figs. 4 and 5 for each site, show a marked linear distribution. Least-square linear relations obtained from the data shown are represented by continuous curves in the figures. These relations allowed correlation of average dry and wet bulb temperatures in our simulation. The method is admittedly approximate, since it supplies a deterministic value of wet bulb temperature for each dry bulb temperature, consequently neglecting statistical variations of wet bulb temperature. This is a compromise between the need for avoiding excessive complication of the model and correlation of the two values in such a way as to reflect the measured data. With the cumulative annual curves for both wet and dry bulb temperatures, it was possible to determine when and for how long the evaporative cooling coils can operate.

Simulation Results

The influence of ambient condition variations on plant performance was calculated on the basis of the analysis given above. Figure 6 shows the power output for three of the solutions considered (Cases A, B, and C) for site 2; the case of site 1 is similar, and there are no significant variations in performance. Evidently both absorption solutions are able to maintain nominal conditions for inlet air temperatures below 25°C, but, of the

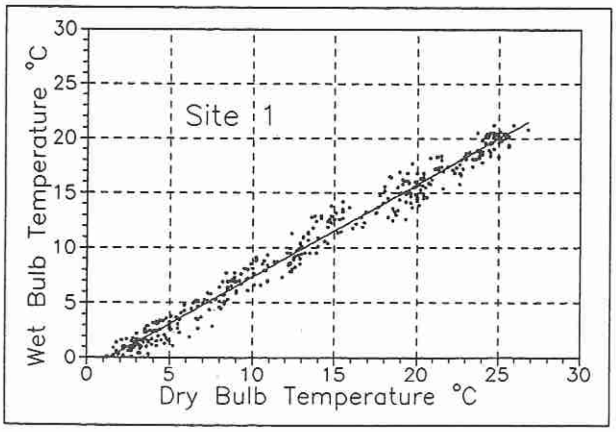

Fig. 4 Wet and dry bulb temperature correlation (site 1)

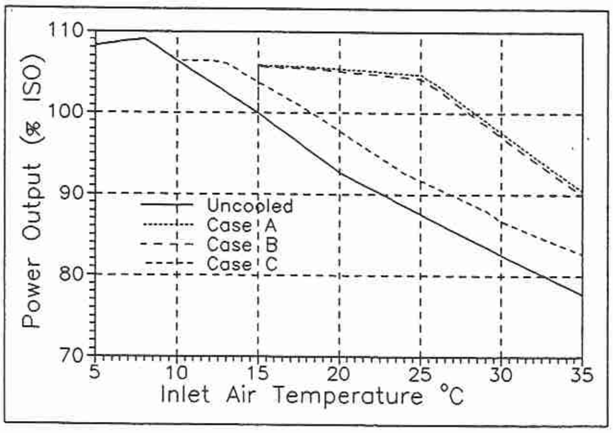

Fig. 6 Plant power output (percentage of ISO rating)

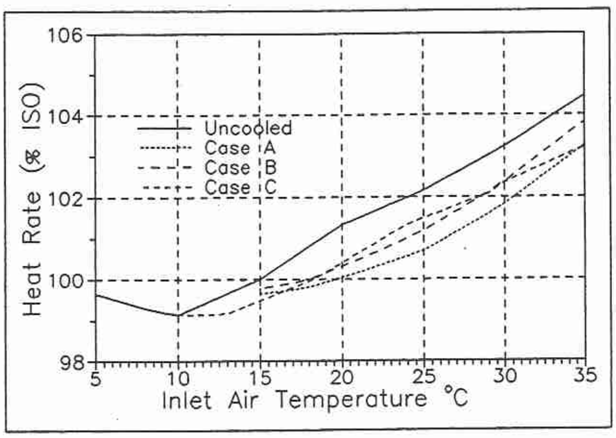

Fig. 7 Plant heat rate (percentage of ISO rating)

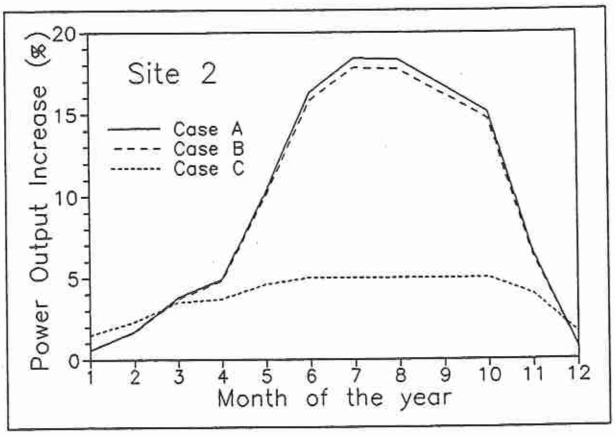

Fig. 9 Power output increase (site 2, percentage of uncooled case)

two, the single-stage unit, with higher accessory consumption, has a slightly lower performance. Figure 7 shows that the use of cooling reduces plant heat rate, which has comparable values for the three solutions.

Performance Evaluation. Simulation was carried out on a yearly basis. Increases in power output are reported in Figs. 8 and 9 for sites 1 and 2, respectively, on a month by month basis, as a percentage of the uncooled case. It can be noted that the two-stage absorber (Case A) gives the maximum power increase in the warmest month (18 percent in July), whereas its single-stage counterpart (Case B) is only slightly less effective. Table 1 shows annual totals calculated from the previous data. While maximum power increase is of the same amount (17–18 percent) for both sites considered, annual total increase for site 2 is much higher than for site 1. In summer months on both sites absorption units work at full load, but ambient conditions on site 2 are more uniform and also give considerable gains in spring and fall. Ebeling et al. (1992) report a 21–25 percent increase in power output on a hot summer day. We also experienced similar high values, but they were limited to a few hours on the hottest days in summer. It should be underlined that results in Figs. 8 and 9 are monthly averages, which are more relevant in the case of a base-load application. Power increase benefits for evaporative cooling are approximately half those of absorption units, but are more evenly distributed over the whole year.

Integrated Solution. Evaporative cooling proved to be useful for obtaining small temperature variations, while absorption units are more powerful and effective in inlet air cooling, but their use is possible only when ambient air temperature is higher than 15°C. Thus the two methods have complementary operating fields, and the integrated solution using both of them was studied. Evaporative cooling works at below 15°C, whereas the absorption chiller is used at higher temperatures and the evaporative unit is turned off. In this way it is also possible to cover the 10–15°C temperature field, which otherwise would remain unchilled, with a small additional investment. This integrated solution was considered in simulation as case D and supplied the data in Table 1. It can be noted that the power increases supplied by the integrated solution are better than those of the other three solutions. However, incremental benefits diminish, i.e., the benefits from the integrated solution are less than the sum of the benefits resulting from the two separate solutions.

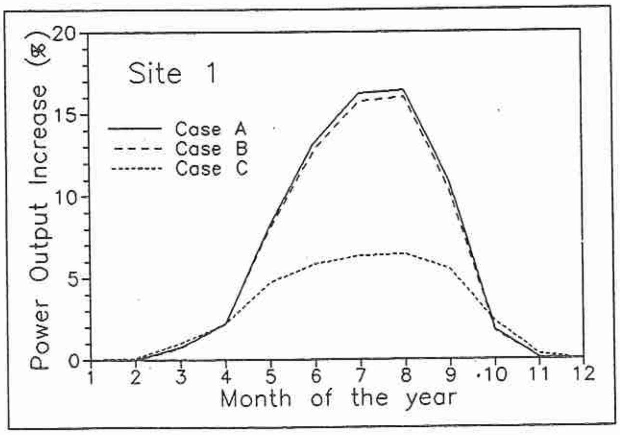

Fig. 8 Power output increase (site 1, percentage of uncooled case)

Economic Results. Annual economic benefits of each solution considered were calculated with respect to the uncooled case. The incremental cost of kWh and investment payback were calculated on the basis of sale of electricity at present Italian prices, which favor day over night production. From 6 a.m. to 10 p.m. the price for private utilities is 9.2 ¢/kWh, whereas at night, weekends, and in the vacation month of August the price sinks to 2.5 ¢/kWh. Natural gas cost is 12.8 ¢/Nm^3 (8.1 ¢/lb). As for capital cost evaluation, for the evaporative cooling system an investment of $130,000 was estimated, $1,000,000 for the single-stage absorber, and $1,200,000 for the two-stage one. The cost analysis was based on a 15-year period and an interest rate of 10 percent. Variable O&M expenses were estimated at a rate of 0.4 ¢/kWh. The vacation month of August was not considered for the economic balance, since in Italy during this period electricity cost is very low. For the economic analysis an exchange rate of 1550 Italian Lire = $1 was assumed (costs referred to 1994).

Incremental kWh cost is the unit cost of electricity produced in excess of the uncooled plant, and is defined as the increase

Table 1 Energy simulation results (annual total)

% of uncooled	Site 1		Site 2	
	Δ Power	Δ Fuel	Δ Power	Δ Fuel
Case A	+5.4	+4.9	+9.0	+8.2
Case B	+5.2	+4.9	+8.7	+8.2
Case C	+2.7	+2.3	+3.8	+3.2
Case D	+5.7	+5.2	+9.7	+8.8

Table 2 Annual economic results (site 1)

Site 1	Net Benefit [$/Year]	Incremental kWh Cost [¢/kWh]	Simple Payback [Years]
Case A	358,000	4.1	2.3
Case B	356,000	4.2	2.1
Case C	251,000	3.1	0.5
Case D	377,000	4.1	2.4

Table 4 Economic results (only daytime)

	Site 1		Site 2	
	Net Benefit [$/Year]	Simple Payback [Years]	Net Benefit [$/Year]	Simple Payback [Years]
Case A	400,000	2.1	732,000	1.3
Case B	402,000	1.9	720,000	1.2
Case D	422,000	2.2	780,000	1.4

in total annual cost due to the installation of air cooling section, divided by the surplus in electricity production. The actual year-averaged cost of kWh for the uncooled plant can be estimated at 4.5 ¢/kWh.

Tables 2 and 3 report the net annual benefit, the incremental kWh cost, and simple payback for the four solutions considered for sites 1 and 2. The net benefit reported in the tables is electric savings minus O&M, incremental fuel expenses, and annual capital cost, and is then the annual equivalent of the NPV (net present value) of the investment. Note that incremental kWh costs for the four solutions considered are lower than the value for uncooled plant. Evaporative cooling (case C) is an extremely cost-effective solution, and payback is less than one year, even for a continental climate and cold sites (site 1). Its installation is advisable for all gas turbine plants, if great electricity increases are not required. Both the single- and two-stage systems have a payback of circa 2 years, and their power gain is approximately double that of evaporative cooling. It must be added that payback-oriented analysis can be misleading, since it gives priority to short-term investment. Since we are dealing with paybacks considerably smaller than the lifespan of the components (from 15 to 20 years), the greater yearly saving of the absorption unit leads, in the long run, to higher benefits than those of evaporative cooling. We chose to report our data in terms of simple payback instead of IRR, since use of the former is more widespread, though it is less accurate. The integrated solution (case D), which proved to be the most effective from the power increase standpoint, is also more economical, since it guarantees payback and incremental kWh costs, which are similar to the solution using absorbers, but it has a 6 percent net benefit over the two-stage (case A) and 9 percent over single-stage absorbers (case B).

Incremental cost per installed kW is similar for cases A, B, and D with absorbers, and is approximately $280 per installed kW, while for evaporative cooling (case C) it equals $240 per installed kW. These costs refer to actual annual power increase, calculated by using the curves in Figs. 8 and 9.

The extremely low cost of electricity in Italy during the night and on weekends suggested that we also consider the case in which absorbers are turned off during the night, even in cases where the temperature is higher than 15°C. Economic results are given for the three cases with absorbers (A, B, and D) in Table 4. All three cases, though not saving all possible electricity, turned out to be more economical than the earlier cases of continuous operation. This happens because the Italian tariff system favors daytime production, and at night the cost of producing electricity is higher than the benefit derived from selling it.

Table 3 Annual economic results (site 2)

Site 2	Net Benefit [$/Year]	Incremental kWh Cost [¢/kWh]	Simple Payback [Years]
Case A	651,000	3.7	1.5
Case B	631,000	3.8	1.3
Case C	346,000	3.1	0.4
Case D	691,000	3.7	1.5

In countries where August can also be considered in the economic balance, the net benefit of cooling, as compared with the Italian situation, increases by 18 percent for site 2 and by 26 percent for site 1, with respect to the cases reported above.

Moreover, savings with inlet air cooling improve, if the gas turbine plant is operated in an electric load following mode, i.e., matching the electricity demand curve, because generally the average electricity purchase price is higher than sale price. In this case net benefits are expected to be even higher than those found here.

Conclusions

The simulation results lead to the following conclusions:

- Evaporative cooling is able to supply valid power increases (2–4 percent on a yearly basis) with rather low investment.
- Absorption cooling can enhance power production by 5–10 percent on a yearly basis, depending on site climate, and up to 18 percent in the warmest months. These benefits are double those supplied by evaporative cooling, and absorption cooling depends much less on humidity conditions. However, investments are higher and plants more complicated. This solution is advisable for base-load production plants.
- The air cooling section with two-stage absorber is more effective than a single-stage one, from the point of view of power increase. Besides, the single-stage unit, which requires about 50 percent more cooling water, has higher running costs.
- The integrated absorber plus evaporative cooling solution assures highest plant performance and yearly net benefit increase.
- In Italy the best solution is to use air cooling plants only during daytime. This is due to the extremely low cost of electricity during nighttime and on weekends.

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