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Power Increase of Gas Turbines by Inlet Air Pre-Cooling with Absorption Refrigeration Utilizing Exhaust Waste Heat

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ABSTRACT

Using heat energy from the tail-end of gas turbine exhaust, an ammonia absorption refrigeration system can precool the inlet air to a temperature slightly above the freezing point of the air humidity.

The concept is described and shows how it indicates a significant increase of gas turbine power output, depending on ambient conditions.

INTRODUCTION

It is well-known that the power output of gas turbines depends, among other factors, on the inlet air temperature. The trend to improve efficiency and power output leads to higher combustion and gas turbine inlet temperatures, but this also increases NO_x emission which is now limited by environmental legislation.

By precooling the inlet air, either a significant power output increase, or a reduction of NO_x emission due to decreased combustion temperatures is possible.

Precooling can be performed by a separate refrigeration plant either with a mechanical vapor compressor using part of the additional power, or by an absorption refrigeration plant utilizing exhaust waste heat.

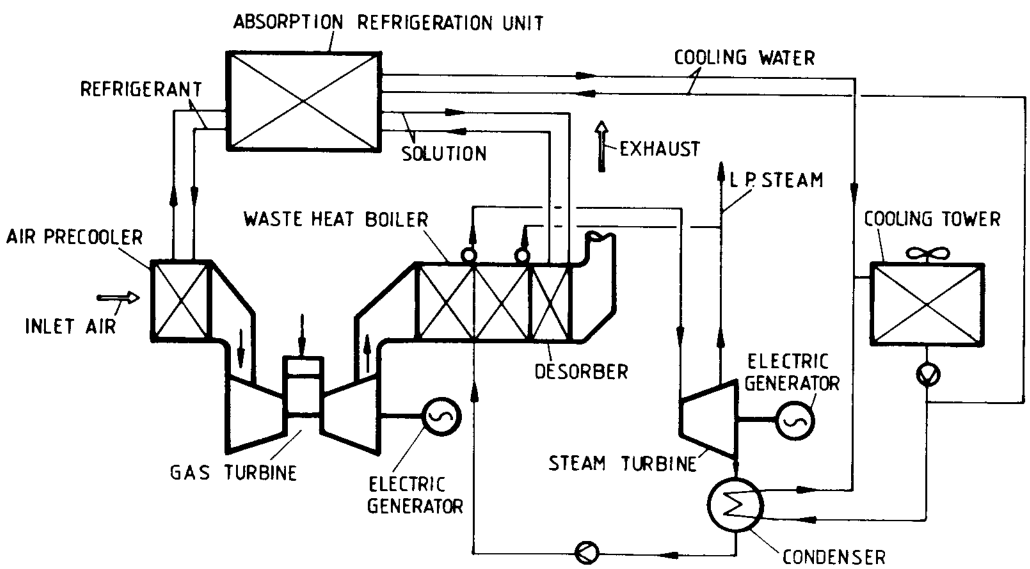

FIG. 1 Power generation system with absorption refrigeration for precooling gas turbine inlet air

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TOTAL CONCEPT

The inlet air for the gas turbine should be cooled down as low as possible without freezing the air humidity. The required power demand for mechanical refrigeration working under these conditions has a magnitude of 200 kW per MW of refrigeration. This loss of power can be avoided with absorption refrigeration operating with heat source temperatures down to 120 °C.

Packaged refrigeration units with working pair water-lithiumbromide are only available for water chilling and hot water or steam as heat source. Ammonia-water absorption refrigeration systems, however, permit lower evaporation temperatures and direct heating by gas turbine exhaust gas.

Such a combination is shown in the simplified flow-sheet of figure 1.

Air from the ambient is precooled in an evaporator which is supplied with the refrigerant ammonia from the ammonia absorption system. The cooled air enters the compressor part of the gas turbine. The exhaust gas leaves the power turbine and its energy is recovered in waste heat exchangers producing steam. The remaining tail-end exhaust heat energy is transferred to the desorption part of the absorption refrigeration.

The absorption refrigeration system is a closed cycle using ammonia as refrigerant and water as solvent. Figure 2 shows a simplified flow-sheet of this cycle:

Ammonia vapour from the evaporator (air precooler) is absorbed in a shell-and-tube-absorber of falling-film type by a weak solution of ammonia-water. Solution enriched with ammonia is collected and pumped to a rectification column pre-heated in a solution heat exchanger by hot desorbed solution. The rectified ammonia vapour leaving the column is condensed by heat transfer to the ambient. The liquified ammonia is fed to the evaporator again where a small sidestream has to be retransferred to the column top necessary for rectification. Desorption heat for rectification is taken from the exhaust gas using a reboiler with forced circulation.

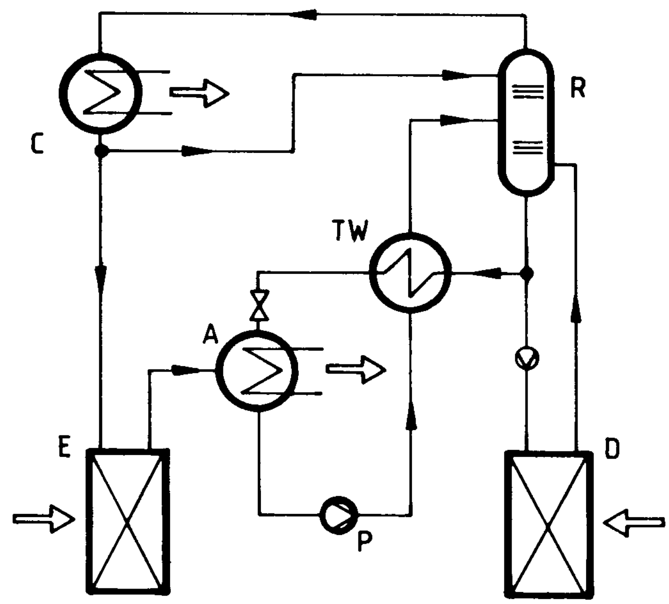

FIG. 2 Flow-sheet of absorption refrigeration system

E - Evaporator
A - Absorber
C - Condenser
R - Rectification column
D - Desorber
P - Solution pump
TW - Solution heat exchanger

TECHNICAL DESIGN

A gas turbine power generation plant with a combined steam generation cycle has enough heat energy available in the tail-end exhaust to cover the demand of the absorption refrigeration process. This can be seen from the following example case:

A power generation unit using a General Electric LM 2500 which is designed for continuous load rating at sea level altitude and power engine speed of 3 600 rpm operating with natural gas fuel has performance data as listed in table I for different inlet air temperatures based on zero humidity and no duct losses.

TABLE I

Performance data of LM 2500 [1)]

Ambient air temperature	°C	32	27	21	4
Inlet air mass flow	kg/s	57.0	59.2	62.2	68.1
Exhaust gas mass flow	kg/s	57.6	59.8	62.8	68.9
Power generation	MW	16.45	17.53	18.94	21.86
Corrected power generation for duct losses and air wet bulb temperature of 21 °C	MW	16.07 [2)]	17.12 [2)]	18.60 [2)]	20.76 [3)]

1) Source: GE gas-turbine performance data MID-TD-2500-5/Febr. 1979
2) Duct losses: inlet 100 mm, outlet 200 mm water column
3) Duct losses: inlet 100 mm, outlet 300 mm water column

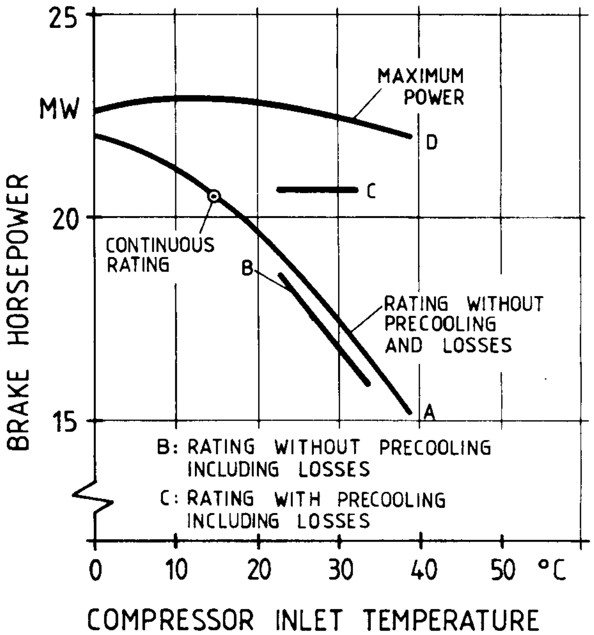

FIG. 3 Power generation with General Electric LM 2500

These values are indicated in figure 3 where line A shows the power generation without humidity and duct losses, line B the reduced values relating to actual conditions.

The refrigeration demand for air cooling from ambient to a lower temperature of 4 °C depends on the dry-air temperature and its humidity at compressor entrance. For given inlet-air flow to the gas turbine compressor, the calculated refrigeration capacity is shown in figure 4.

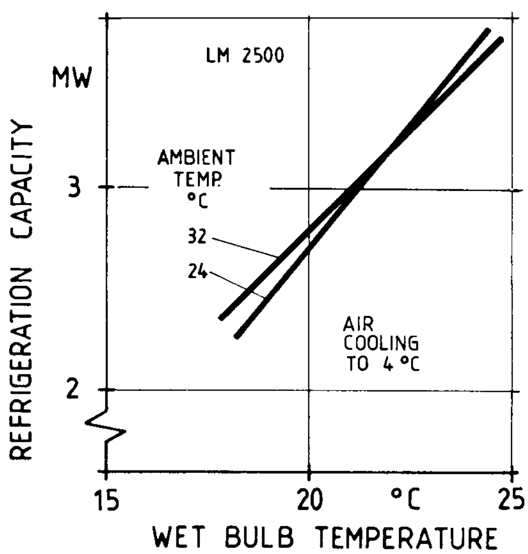

FIG. 4 Refrigeration demand for air precooling of General Electric LM 2500

An ammonia absorption refrigeration system designed as single-stage desorption and two-stage absorption cycle (two-stage air cooling to 4 °C) with an evaporative condenser, water cooled absorbers (water supply temperature 27 °C) and desorber direct-heated by exhaust gas has specific performance data per 1 MW refrigeration as listed in table II. These values are proportional to the refrigeration capacity.

The desorber design depends on the available temperature difference between exhaust discharge from the steam generating system and the lowest necessary desorber entrance temperature. That is fixed by the minimum exhaust gas temperature at desorber discharge plus the exhaust gas temperature decrease.

The absorption system will be a separate and independent working unit, with evaporator (air precooler) and desorber arranged in the inlet system and exhaust gas duct respectively and with a forced circulation of refrigerant and solution. Surfaces and dimensions for air precooler and desorber have to be optimized with regard to the pressure drops in the ducts. Figures 5 and 6 show the approximate cross section and depth in relation to the pressure loss, based on a total refrigeration capacity of 3.5 MW distributed with 60 % at 7.5 °C and 40 % at about 0 °C evaporation temperature and an exhaust gas temperature of 250 °C.

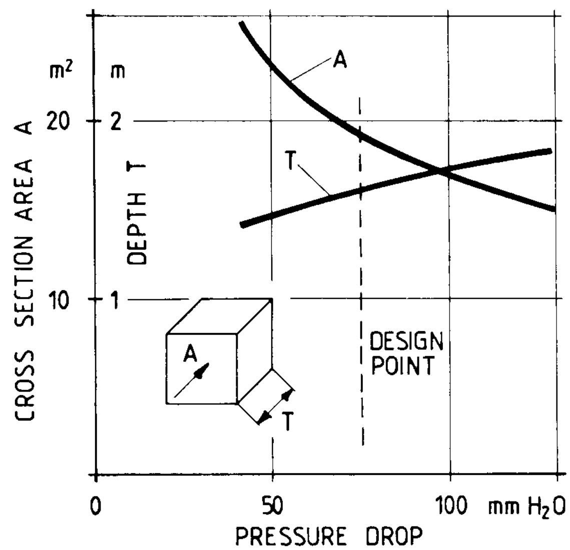

FIG. 5 Dimensions of air precooler for 3.5 MW refrigeration plant

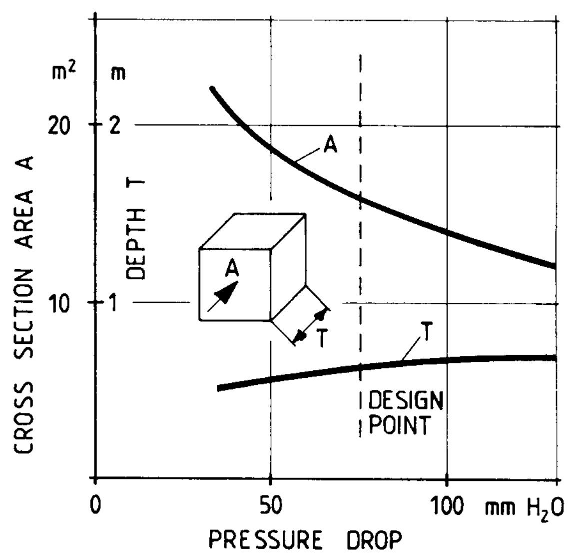

FIG. 6 Dimensions for desorber direct exhaust gas heated for 3.5 MW refrigeration plant

All other equipment of the absorption plant can be arranged in an open-air structure in some distance from the power generation plant. The ground space requirement is about 9 x 18 m.

Table II

Specific Performance Data for an Ammonia Absorption Refrigeration System °)

Refrigeration capacity	1	MW
Heat ratio (Refrigeration capacity per desorption heat)	0.64	-
Electric energy for solution pumps and evaporative condenser	30	kW
Heat input to desorber	1.563	kW
Cooling water demand at 10 °C temperature rise	125	m^3/h
Exhaust gas temperature decrease	80	°C
Minimum exhaust gas temperature at desorber discharge	120	°C

°) Air precooling to 4 °C in two cooling steps
Cooling water supply at 27 °C;
Air wet bulb temperature at 21 °C

CONCLUSIONS

The air precooling improves the power output in absolute terms according to figure 7. Depending upon the ambient air temperature, the power output can be increased by 17 to 29 %. Line C in figure 3 illustrates this situation. The necessary desorber heat can be recovered from the exhaust as long as the exhaust leaving the steam generation section has temperatures of more than 200 °C. All results are based on a refrigeration demand for air precooling of 3.5 MW. For other values the temperatures will vary.

The efficiency of the gas turbine is slightly improved with lower air inlet temperatures due to better compressor performance of the gas turbine unit. Furthermore, the exhaust gas flow increases by 2 ... 4 % with pre-cooled air which results in a possible higher output of steam production at the same exhaust discharge temperatures from the steam generation section. Figure 8 indicates the relevant gas turbine efficiencies with and without air precooling.

The NO_x-emission will be reduced due to decreased air and thus combustion temperatures. Absolute values depend on the amount of water injection, combustion chamber design and combustion velocity. NO_x reductions may be achieved in cases where the possible power increase is not fully required for design or steady operation. If NO_x reduction by air precooling or other procedures is not sufficient for environmental legislation limits, the higher power output can compensate the power sacrifice caused by Selective Catalytic Reduction (SCR) systems which would then be necessary to meet the requirements.

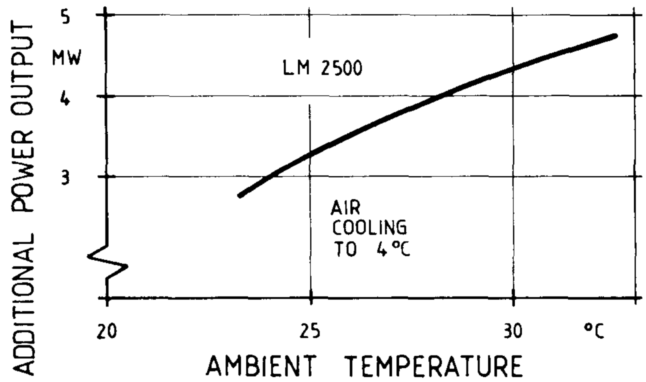

FIG. 7 Additional power output of General Electric LM 2500 with air precooling down to 4 °C

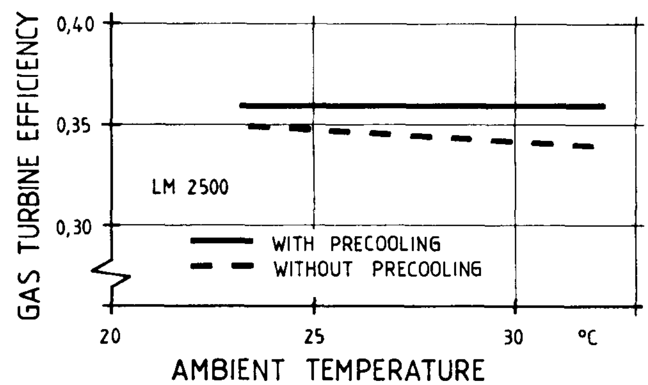

FIG. 8 Efficiency of General Electric LM 2500 without and with air precooling

Normally, maintenance intervals for gas turbine units depend on the maximum allowable turbine inlet temperature. Operation with pre-cooled air at reduced power loads can increase the time between overhauls.

LITERATURE

/1/ Malewski, W. Selected criteria for absorption refrigeration plants and examples of their design and construction

Chemiker Zeitung Vol. 95, 1971, 186/192

/2/ Holldorff, G. Revisions up absorption refrigeration efficiency

Hydrocarbon Processing july 1979, 149/155

/3/ Holldorff, G. Methods for lowering the heat medium quality in absorption refrigeration plants

Klima-Kälte-Heizung 9/1984, 345/349