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A NEW GAS TURBINE CYCLE FOR ECONOMICAL POWER BOOSTING

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ABSTRACT

A Moisture Air Turbine (MAT) cycle is proposed for improving the characteristics of land based gas turbine by injecting atomized water at inlet to compressor.

The power boosting mechanism of MAT is understood as composites of those of following existing systems: inlet air cooling system, inter-cooling and steam injection. Experiments using a 15MW class axial flow load compressor have been carried out to reveal that water evaporation in compressor could reduce compressor work in an efficient manner. Moreover, this technology has been demonstrated by means of 130MW class simple cycle gas turbine power plant to show that a small amount of water consumption is sufficient to increase power output. Very efficient evaporation could be achieved provided the size of water droplet is controlled properly. The amount of water consumption is much less than that of conventional inlet air cooling system with cooling tower for heat rejection.

Incorporating water droplet evaporation profile into consideration, realistic cycle calculation model has been developed to predict power output with water injection.

It has been shown that this technology is economically achievable. It should be stressed that contrary to well known evaporative cooler, MAT cycle could provide power output at a desired value within its capability regardless of ambient humidity condition.

NOMENCLATURE

- Cp: Isobaric specific heat (J/(kgK))
- f: Specific fuel heat (J/kg)
- g: Water content to dry air ratio (-)
- Hi: Specific enthalpy at location i (J/kg)
- h': Specific enthalpy of saturated water (J/kg)
- h'': Specific enthalpy of saturated steam (J/kg)
- Lij: Specific compressor work from the state i to j (J/kg)
- m: 1-1/κ (-)

- Ps: Water vapour pressure (Pa)
- Pcd: Compressor discharge pressure (Pa)
- p: Specific power output (W/(kg/s))
- Q: Gas turbine power output (W)
- R: Compressor work ratio (-)
- s: Specific entropy (J/(kgK))
- T: Temperature (K)
- Tcd: Compressor discharge temperature (K)
- v: Specific volume (m³/kg)
- W: Compressor intake air flow (kg/s)
- x: Dryness (≡ (h-h')/p) (-)
- α: Portion of mechanism (2) to a total power increment (-)
- κ: Specific heat ratio (-)
- η: Gas turbine thermal efficiency (-)
- Compressor adiabatic efficiency (-)
- p: Latent heat of evaporation (= h'' - h') (J/kg)
- Δi: Increment of property i
- φij: Pressure ratio of state j to state i (-)

INTRODUCTION

Needs are on the rise for technologies for recovering the output of gas turbines in summer, especially in cheaper way. One typical system is one that increases output by cooling the incoming air into the gas turbine, thus increasing air density and the mass flow of working fluids. Several of these systems are in use at power plants (Ebeling et al., 1992), as well as research efforts under way for putting such systems into practical use (Saito, K., 1996) (Tanaka, M. et al., 1996). With these systems, the most common method is to accumulate heat using nighttime electric power, discharge the heat during peak daytime hours, and increase output. This makes the running time with increased output depend on the amount of heat accumulated, and furthermore considerable space is required for a heat storage tank on a practical level.

The present study aims to propose, and examine the possibilities of, a Moisture Air Turbine (MAT) cycle, a new technology for increasing the output of a gas turbine by entering a fine normal temperature water spray into the incoming air in the compressor.

This technology is characterized by the fact that it is subject to few practical constraints in that it requires no heat accumulation and that it can increase output on demand.

Analogous to the MAT cycle, water injection into compressor is known to get extra thrust in aircraft engine on taking off at high altitude (Rolls-Royce plc, 1986).

Mechanical or chemical interaction between blades and particles injected into turbomachines has been extensively investigated so far. Blade erosions by the solid particles were studied (Hamed, A. and Tabakoff, W., 1994).

If the droplet size were of the order of $5\mu\text{m}$, collisions with the compressor blades would be minimized; such droplets would follow the flow streamlines (Bannister R.L. et al., 1995).

It is also reported that the absence of erosion was confirmed by long-term operating experience when the water is properly injected directly into compressor and no corrosion of the metal or deposits in the compressor flow path were observed (Arsen'ev L.V. and Berkovich A.L., 1996).

The MAT cycle assumes generation of very fine water droplets to minimize mechanical and chemical interaction with compressor blades. This paper reports the findings in a theoretical study of the heat cycle and in a verification of its principles in experiments.

THE PRINCIPLES OF INCREASED OUTPUT WITH A WATER SPRAY

The Compression Process of Wet Air with Phase Changes

The MAT cycle aims to enter fine water droplets into the incoming air in the compressor, thus increasing the output of the gas turbine. Figure 1 is a psychrometric chart representing how a working fluid becomes compressed under ambient air conditions. Suppose, for example, that the ambient air is at 20°C and 70%RH, and the state can be defined by point A in Fig. 1. Supposing that the ambient air is humidified and cooled along the equivalent wet-bulb temperature line, reaching saturation before entering the compressor, the incoming air moves to state B at the compressor inlet. Liquid droplets that have not evaporated completely outside the compressor evaporate continuously in later compression. Assuming that the evaporation process remains saturated and that the total entropy of the two-component two-phase mixture is conserved, the liquid finishes boiling and the C-to-D process enters the single-phase compression process, resulting in a temperature rise. If evaporation is accompanied with an entropic rise, the mixture becomes unsaturated. In reality, since the rate of evaporation from liquid droplets is limited, the state do not reach thermal equilibrium. The result is that the actual process presumably deviates from the saturation line, following the locus of the broken line. On the other hand, the normal compression process follows the A-to-D'' locus.

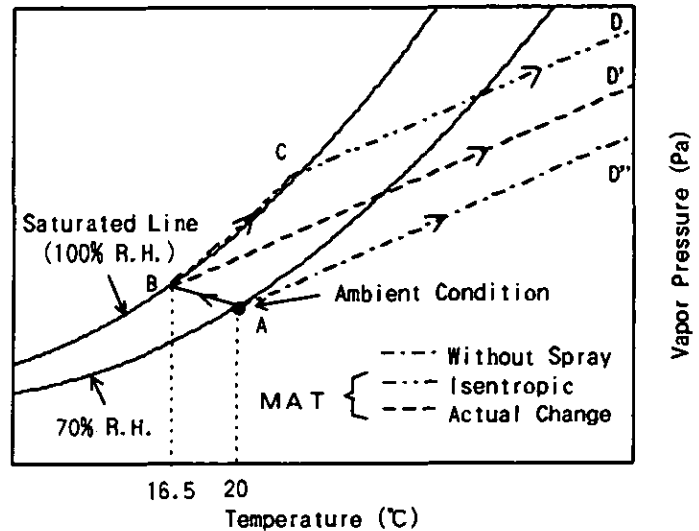


FIGURE 1 CHANGE IN STATE OF WORKING FLUID DURING COMPRESSION

Output Increment Mechanism

The output increment mechanism can be divided into the following sections in qualitative terms:

- (1) Cooling of the incoming air on the equivalent wet-bulb temperature line with regard to the front flow in the compressor.
- (2) Cooling of the internal gas due to the evaporation of liquid droplets sucked in by the compressor.
- (3) The difference between the amounts of working fluids passing through the turbine and compressor. This difference corresponds to the amount of evaporation in the compressor.
- (4) An increase in the isobaric specific heat of the mixture due to the ingress of steam having a high isobaric specific heat.

Figure 2 expresses the above mechanism as a combination of conventional techniques. Since the output of the gas turbine is the product of specific output multiplied by the incoming flowrate of the compressor, the output increase can be expressed by the sum of ① contribution due to an increase in the incoming flowrate and ② contribution due to a rise in specific output. The mechanism (1) is classified as ①, while the mechanisms (2), (3), and (4) stemming from the behavior of liquid droplets introduced into the compressor are classified as ②. The MAT cycle can be interpreted as a combination of the effects of existing techniques.

$\Delta Q = P \cdot \Delta W + W \cdot \Delta P$
 Q : GT Power Output W : Inlet Air Flow, P : Specific Power Output

Mechanism of MAT Cycle	ΔQ	Existing System	Concept
(1) Compressor External Cooling (Incremental Inlet Air Flow)	$P \cdot \Delta W$	Inlet Air Cooling (Evaporative Cooler)	Inlet Air Cooling Panel
(2) Compressor Internal Cooling (Decrease in Work)		Intercooler (Evaporative Latent Heat, No Heat Loss)	
(3) Mass Flow Increase (4) Larger Specific Heat	$W \cdot \Delta P$	Steam Injection	

FIGURE 2. INCREMENTAL POWER MECHANISM OF MAT CYCLE

The Mechanism for Reducing Compressor Work by Evaporation

Figure 3 represents how the evaporation of liquid droplets reduces work consumed by the compressor. For simplification, this process is described schematically with the reversible adiabatic compression process ($ds = 0$) with a piston. Suppose that the piston is at a certain compression pressure P_1 and a temperature T . Since the state change in a particular stage in the compressor is in the isobaric process ($dP = 0$), we make a simulation where pressure P_1 is kept, with the spray liquid droplets evaporating and the temperature going down by ΔT . At that time, we see $dh = 0$ from $dh = Tds + vdp$, resulting in this process becoming an isenthalpic process. When air is compressed to P_2 , the subsequent temperature is $(T - \Delta T) \phi^m$ where ϕ is the pressure ratio P_2 / P_1 .

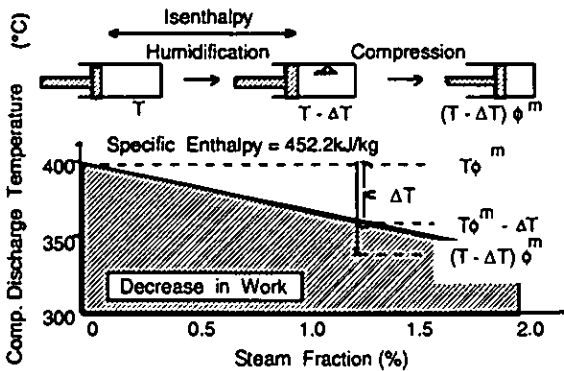


FIGURE 3 MECHANISM OF DECREASE IN COMPRESSOR WORK

On the other hand, supposing that the gas temperature has a constant evaporation latent heat when an equal amount of liquid droplets evaporate after the gas is compressed from pressure P_1 to P_2 , the result is $T \phi^m - \Delta T$. In that case, it is clear that piston

work remains constant before and after spraying. In the spray amount-enthalpy diagram in Fig. 3, the envelope in the shaded portion represents an isenthalpic line of the mixture, corresponding to $(T \phi^m - \Delta T)$ above. The difference between this and the discharge temperature in evaporation in the above intermediate stage is

$$(T \phi^m - \Delta T) - (T - \Delta T) \phi^m = \Delta T(\phi^m - 1) > 0.$$

This indicates that, since the latter temperature is lower than the former, the outlet enthalpy of the latter is smaller, so that the required work of the piston decreases.

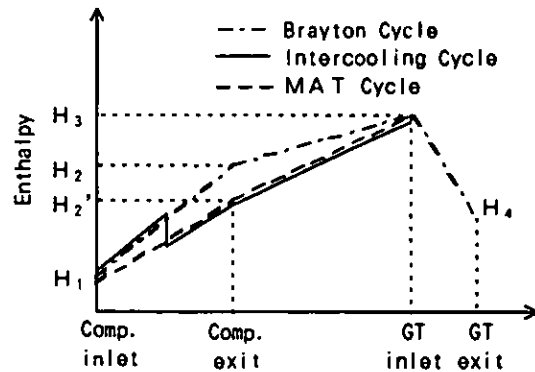
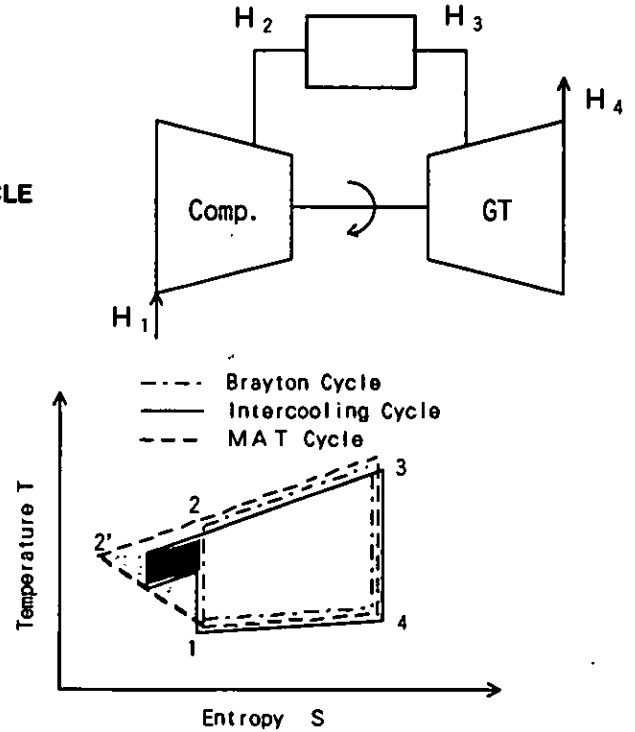


FIGURE 4 ENTHALPY PROFILE WITHIN GT

THERMAL EFFICIENCY OF THE THEORETICAL CYCLE

Thermal Efficiency of the Increased Output

Of all output increment mechanisms described in the preceding section, (1) is well known as a rise in the weight flow of the incoming air due to cooling. We are therefore going to study how the cooling of the gas in the compressor affects the specific output and thermal efficiency. Figure 4 is a thermal cycle diagram. For simplification, let us ignore the effects of the increase in working fluids due to evaporation, or mechanisms (3) and (4), and examine output increment mechanism (2) alone. The output per unit weight of incoming air, or specific output p , is

$$p = (\text{turbine shaft output}) - (\text{compressor work}) \\ = (H_3 - H_4) - (H_2 - H_1) \quad (1)$$

On the basis of the above, suppose that the turbine inlet enthalpy H_3 is constant. We will then find that output increment Δp is caused by a reduction in the enthalpy at the compressor outlet to H_2' , due to the use of the MAT cycle.

$$\Delta p = H_2 - H_2' \quad (2)$$

On the other hand, fuel consumption f is $f = H_3 - H_2$. At that time, fuel increment Δf is

$$\Delta f = H_2 - H_2' \quad (3)$$

From formulae (2) and (3), we find that $\Delta p = \Delta f$ ($\equiv \Delta$) and thus the thermal efficiency of the mechanism (2) of the MAT cycle η_1 exceeds that of the Brayton cycle η_0 because the inequality

$$\eta_1 = \frac{p + \Delta}{f + \Delta} > \frac{p}{f} = \eta_0 \text{ holds provided } f > p \text{ which is always satisfied.}$$

It should be noted the heat rate to gain incremental output Δp i.e. $\Delta f/\Delta p$ is found to be η_0 times smaller than that of the Brayton cycle as far as additional fuel consumption is concerned. As compared to that, the energy efficiencies achieved with traditional output increment techniques are lower than the thermal efficiency of the gas turbine by an amount equivalent to the energy consumption required for cold heat production. Water injection in combustor does not cause the reduction of compressor work and hence results in the reduction of thermal efficiency.

Table 1 is based on the assumption that the temperature ratio ($\equiv T_3 / T_1$) and pressure ratio ($\equiv P_2 / P_1$) are under identical conditions and shows the rankings of the gas turbine characteristics in each cycle. As shown in Fig. 4, the MAT cycle is the best thermal cycle in terms of both specific output and thermal efficiency. The inter-cooling cycle achieves a high specific output but has a low cycle thermal efficiency because of heat loss in the compressor. Why the MAT cycle achieves a high thermal efficiency can be attributed to being that the compression process with this cycle approaches isothermal changes, or follows an asymptotic curve with regard to the Ericsson cycle having a theoretical thermal efficiency equal to that of the Carnot cycle.

The section given below gives a theoretical study of how much reduction in work is expected.

TABLE 1 CHARACTERISTICS RANKING OF VARIOUS CYCLES

Cycle	Thermal Efficiency	Specific Output
Brayton Cycle	2	3
Intercooling Cycle	3	2
MAT Cycle	1	1

Theoretical Work Due to the Compression of a Two-Component Two-Phase Mixture

The isentropic compression work L_{12} of dry air is

$$L_{12} = C_p T (\phi_{AD}^m - 1) \quad (4)$$

On the other hand, compression work L_{12}' in the two-component two-phase compression process including the evaporation of liquid droplets can be expressed with the formula given below as the sum of work in the liquid-evaporation process (B to C in Fig. 1) and the work of the two-component single-phase mixture.

$$L_{12}' = g [h_c'' - (h_b' + x_b \rho_b)] + C_p (T_c - T_b) \\ + C_p T_c (\phi_{CD}^m - 1) (1 + g) \quad (5)$$

Total pressure P at evaporation completion point C on the psychrometric chart is expressed with the formula given below, using steam pressure P_s when the two-phase saturated wet air in state B is compressed isentropically and when dryness x reaches 1 in state C.

$$P = \frac{0.622 + g}{g} \cdot P_s \quad (6)$$

Ratio R of spray work to non-spray work is

$$R = \frac{L_{12}'}{L_{12}} \quad (7)$$

Figure 5 is a representation of R with regard to the spray amount. The compressor discharge pressure is used as a parameter.

External air conditions were assumed to be equal to the experimental conditions using axial compressor mentioned later in this report. Since $R < 1$ at the time of spraying, we found that there theoretically existed a decrease in motive power due to the evaporation of liquid droplets in the compression process. This indicates that the dependence on discharge pressure is low. The decline in motive power requirements in the compressor is almost proportionate to the amount of spray, while the rate of decrease of work due to spraying with an air ratio of 1% is 6.8%.

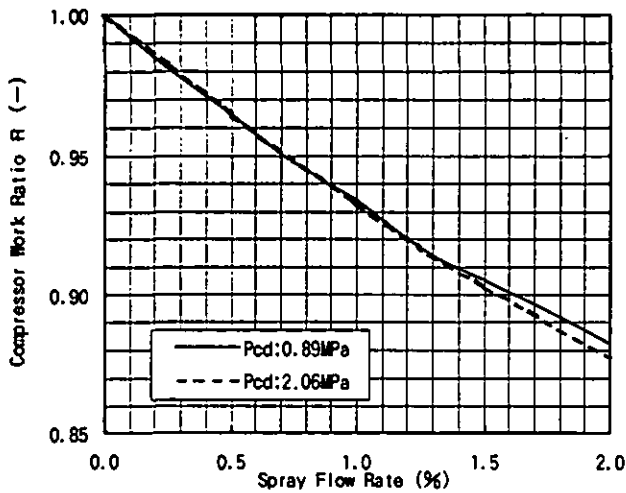


FIGURE 5 INFLUENCE OF SPRAY FLOW RATE ON COMPRESSOR WORK RATIO, R

PREDICTION OF THE CHARACTERISTICS OF AN ACTUAL CYCLE

This section makes an attempt to quantify the characteristics of the MAT cycle, using a calculation model similar to the characteristics of an actual gas turbine, such as adiabatic efficiency, air extraction of the compressor, and changes in gas composition due to water evaporation. In particular, this section evaluates the effects of the spray amount and compares the performance of this model with that of a traditional output increasing solution that injects water into a combustion device.

One of the calculation conditions is that parameters have been selected that assume an axial flow gas turbine of the order of 150 MW. The ambient temperature and humidity are assumed to be 35°C and 53%RH. We did not include liquid droplets in the flowrate of the compressor, and performed calculations based on the assumption that all would evaporate at a time during the intermediate stage. Figure 6 shows how the spray amount depends on the output increment ratio and thermal efficiency. The output increment sensitivity with regard to the unit spray amount is higher in the cooling outside the compressor (mechanism ①) than in the cooling (mechanisms ②, ③, and ④) in the compressor. The water consumption required to restore the gas turbine output at an external air temperature of 35°C to the output during a baseload running at 5°C is about 2.3% of the incoming air. The thermal efficiency rise at 2.3% spray is 2.8% in relative terms.

Calculation Conditions	
Compressor Inlet Air Mass Flow Rate	417kg/s
Compressor Polytropic Efficiency	0.915
Turbine Adiabatic Efficiency	0.89
Combustion Temperature	1290°C
Compressor Bleed Ratio	20%
Pressure at Compressor Discharge	1.48MPa
Pressure at Evaporation Stage	0.25MPa

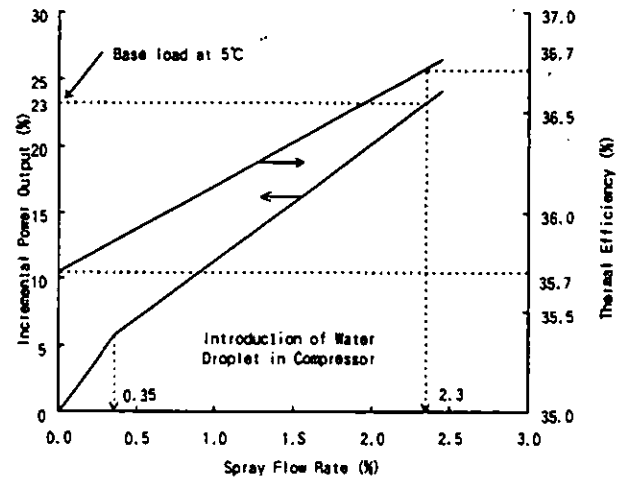
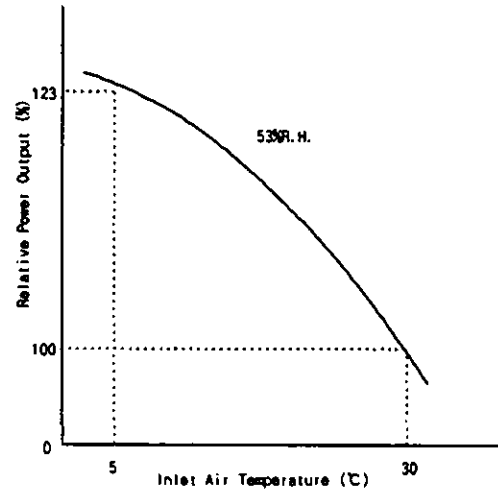


FIGURE 6 GT CHARACTERISTICS VS. WATER INJECTION FLOW RATE

Table 2 gives the details of the output increment for each mechanism in Fig. 2. The figure gives calculated values at 2.3% water spray. The portions of mechanisms ③ and ④ correspond to the injection of steam into an existing type of combustion device, accounting for a third of the total. This means that the MAT cycle approximately triples the output at the same water amount as compared with the steam injection method. Table 2 and Fig. 6 prompt us to expect achieving an output increment of 17.9% from the present calculation case with a 2% spray amount, even though the external air is in a state of saturated wet air. Thus, the MAT cycle achieves a required output within the range specified by its capacity without being affected by the moisture of external air, due to the contribution of mechanisms ②, ③, and ④. This is an essential difference in terms of functionality from

existing evaporative coolers.

Table 3 shows the effect of the location of the water injection with the water amount of 2.3% common to all cases. Case 1 represents the conventional method that injects water in the expander. Case 2 is a direct injection in compressor. Case 3 is MAT where fine water droplets is sprayed upstream of compressor inlet. It is evident that MAT is most effective in both power increment and thermal efficiency.

Table 4 compares the water consumption of this model with a conventional incoming-air cooling system. Although conventional methods also assume that a cooling tower consumes water to remove discharged heat during heat accumulation, the water consumption of the MAT cycle is found to be smaller than that (Utamura, M. et al., 1996).

TABLE 2 FRACTION OF INCREMENTAL POWER OUTPUT

Mechanism of MAT cycle	Air Condition: 35°C 53%R.H. Injected Water: 2.3%
Compressor External Cooling (1)	24%
Compressor Internal Cooling (2)	48%
Mass Flow Increase Larger Specific Heat (3, 4)	28%

TABLE 3 EFFECT OF LOCATION OF WATER INJECTION

Item	Without injection	Water Injection		
		case 1*	case 2	case 3 (MAT)
Turbine Output (MW)	280	287.7	287.7	300
Comp. Work (MW)	156	156	142.1	147.5
Gen. Output (MW)	124 (Base)	131.7 (+6.2%)	145.6 (+17.4%)	152.5 (+23%)
Thermal Efficiency (-)	35.7 (Base)	34.2 (-4.2%)	36.4 (+2.0%)	36.7 (+2.8%)

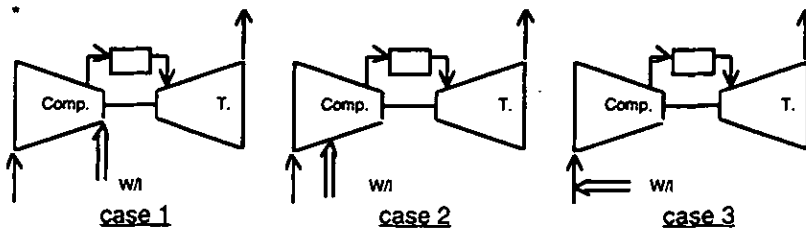


TABLE 4 WATER CONSUMPTION

MAT Cycle		Consumed Water
Inlet Air Cooling with Thermal Energy Storage	Ice Storage (Turbo-Refrigerator) (COP = 2.5)	1.8
	Liquid Air Storage (COP = 0.6)	2.5

AN EXPERIMENT FOR CHECKING THE COOLING CHARACTERISTICS

Method of Experiment

Table 5 gives the main specifications and experimental range of a sample axial flow compressor, while Fig. 7 outlines the experimental apparatus. The sample compressor is connected via a drive gas turbine and a fluid joint and its transmission torque can be controlled by adjusting the position of the squeeze tube. During the test, we adjusted the compressor rotation speed to 11,000 rpm and the discharge pressure to a specified value. The spray amount was controlled with a water supply valve, while the grain size was adjusted by controlling the air supply. The amount of flow entering the compressor was measured with a Pitot tube. We also measured the temperatures of various portions and the humidity at the inlet and outlet of the compressor. Spray water temperature was room temperature. The average air flow speed in the suction duct was 20m/s.

TABLE 5 EXPERIMENTAL CONDITIONS

Items	Values
Compressor Specifications	
Rotation Speed (rpm)	11,000
Pressure Ratio	9.07
Mass Flow (kg/s)	36.3
Work (kW)	15,000
Number of Stages	17
Axial Length	1,850mm
Spray Nozzle Specifications	Two Fluid Nozzle
Flow Rate (vs. Inlet Air Ratio)	0 ~ 0.5%

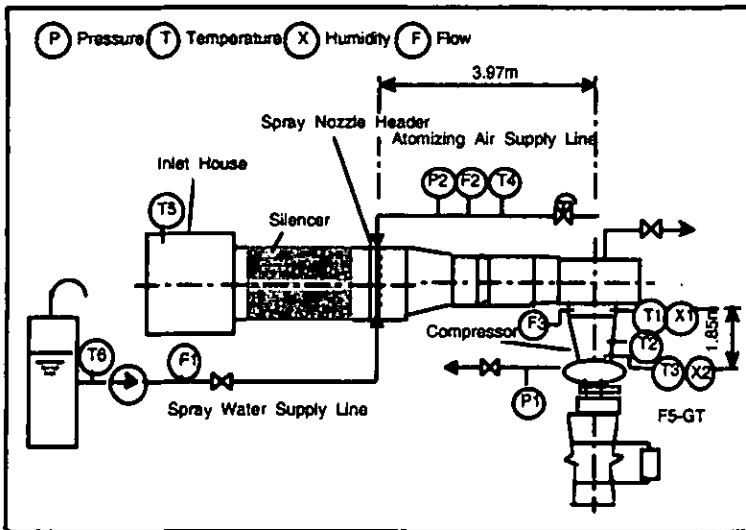


FIGURE 7 COMPONENTS OF EXPERIMENTAL APPARATUS

Experimental Results and Discussion

Figure 8 shows the correlation between the amount of droplets introduced (air ratio) into the compressor based on the spray amount, air flow, and external air conditions and the directly measured absolute humidities at the EGV position at the compressor outlet. Liquid droplets that flowed into the compressor are found to have evaporated at least 95% before reaching the final stage. Figure 9 shows the relationship between the spray amount and the temperature difference at the compressor outlet before and after spraying. For one thing, we find that evaporation and cooling took place efficiently at a low flowrate before the gas flowed into the compressor. The humidity reached was close to 95%. The solid line indicates calculation for the difference between the temperature at the compressor outlet and the temperature before spraying. The former (that is, the temperature at the compressor outlet) was based on two conditions: (1) The absolute humidity of the outlet gas was obtained based on the assumption that all liquid droplets that flowed into the compressor evaporated. (2) The outlet gas enthalpy of the compressor was equal to the value before spraying. This means that this line represents a state where the compressor shows no decline in motive power. The fact that the experimental values exceed them is indicative of the fact that the cooling effect of the gas is higher than the simple evaporation of liquid droplets. These findings led us to determine that there really was a decline in motive power. This can be interpreted as being due to the fact that the temperature decrease due to the evaporation of liquid droplets in the intermediate stage of the compressor is amplified in the compressor process in stages following the evaporation point. This leads us to believe that evaporation in the former stage is advantageous in reducing motive power. The broken line shows the calculation results based on the assumption of isentropic compression using the theory mentioned in the foregoing chapter.

For convenience's sake, we here define the adiabatic efficiency

at the time of spraying as the ratio of isentropic compression work without spray and the motive power at the time of spraying. Figure 10 indicates the ratio S ($\equiv \eta_m / \eta_p$) of the adiabatic efficiency (η_m) at the time of spraying to the adiabatic efficiency (η_p) without spraying, with regard to the spray amount. The solid line is equal to the inverse ($= 1/R$) of the theoretical value of two-phase isentropic compression (formula 7) mentioned in section 3.2. At the same time, the measured values indicate the motive powers calculated with thermal dynamic values measured at the inlet and outlet of the compressor. Within the experimental range, the adiabatic efficiency with regard to unit spray amount (1%) was 4%. S is almost proportionate to the spray amount, and we will examine its gradient.

Figure 11 shows an increase in adiabatic efficiency expected when a unit spray amount is given, with regard to the adiabatic efficiency of the compressor in the true sense of the word. Our finding is that the higher the adiabatic efficiency, the higher the rise in adiabatic efficiency, that is, machines with higher adiabatic efficiencies produce higher MAT effect. We have observed no evidence of erosion on compressor blades after 100hr continual operation of water injection.

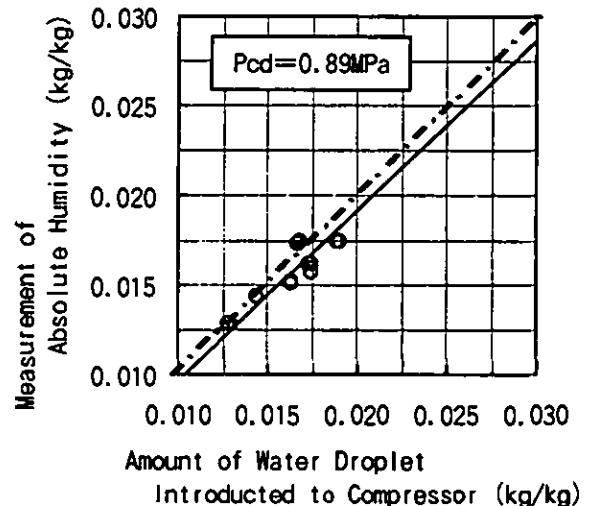


FIGURE 8 FRACTION OF EVAPORATION OF WATER DROPLET

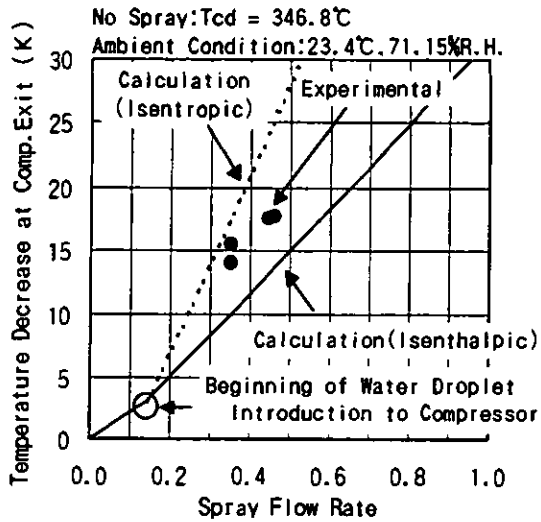


FIGURE 9 COMPRESSOR EXIT TEMPERATURE DIFFERENCE

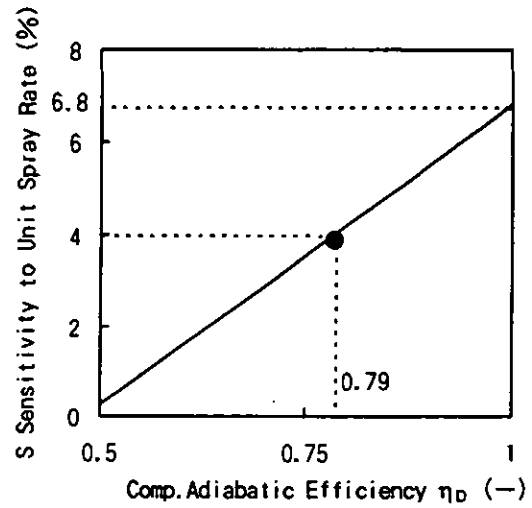


FIGURE 11 S SENSITIVITY TO COMPRESSOR ADIABATIC EFFICIENCY

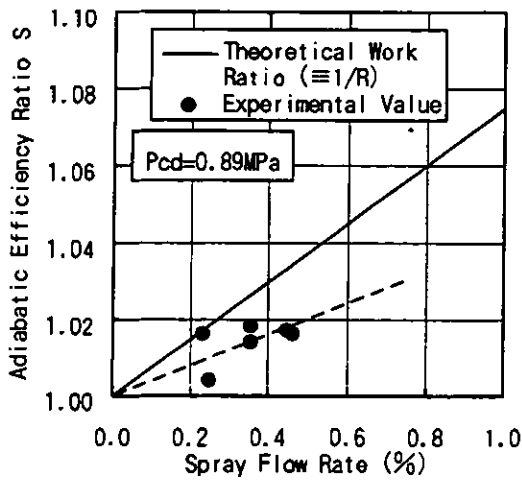


FIGURE 10 S DEPENDENCE ON FLOW RATE

DEMONSTRATIONS OF MAT EFFECT

Demonstration test was conducted using 130MW class simple cycle gas turbine power plant under the ambient condition of 15.3°C , $20\% \text{R.H.}$ with water injection up to 0.4% to air ratio. Plant specifications are shown in Table 6.

Stage by stage calculations were also carried out to predict the increment of the power by water injection assuming the evaporation profile of water droplet inside compressor as is depicted in Fig. 12.

TABLE 6 PLANT SPECIFICATIONS

Item	Value
Gross Power Output (MW)	130 (10°C)
Rotation Speed (rpm)	3000
Mass Flow (kg/s)	411
Compressor Pressure Ratio (-)	12.4
Compressor Adiabatic Efficiency (%)	89.9
Compressor No. of Stages	17
Turbine Inlet Temperature ($^{\circ}\text{C}$)	1155
Turbine Exhaust Temperature ($^{\circ}\text{C}$)	560

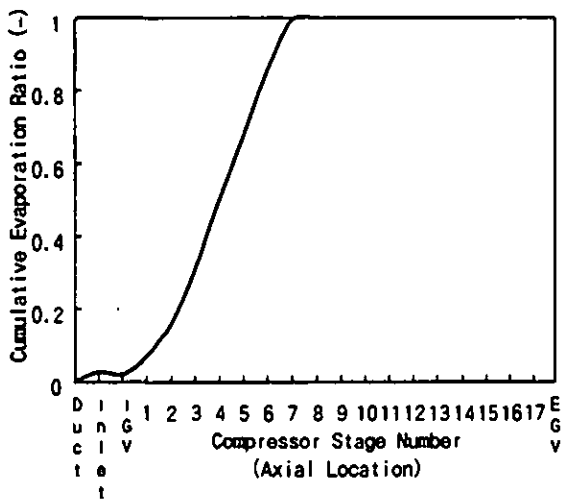


FIGURE 12 EVAPORATION PROFILE INSIDE COMPRESSOR

The evaporation profile was obtained by calculating the change in diameter of a water droplet which moves through gas path with given temperature distribution of working fluid.

Figure 13 shows comparison of experiment with calculation. Significant power increment was observed with very little water consumption. Moreover, according to Fig. 14 thermal efficiency was found to be increased by 1.1% (relative) to 0.4% of water injection due to reduced compressor work as was confirmed in the decrease of the temperature at compressor discharge.

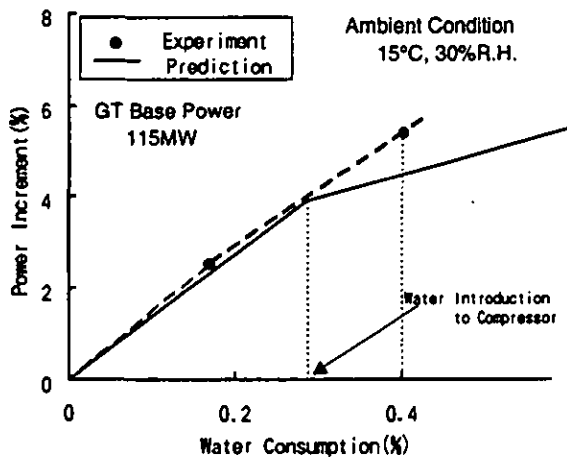


FIGURE 13 COMPARISON OF EXPERIMENT WITH CALCULATION

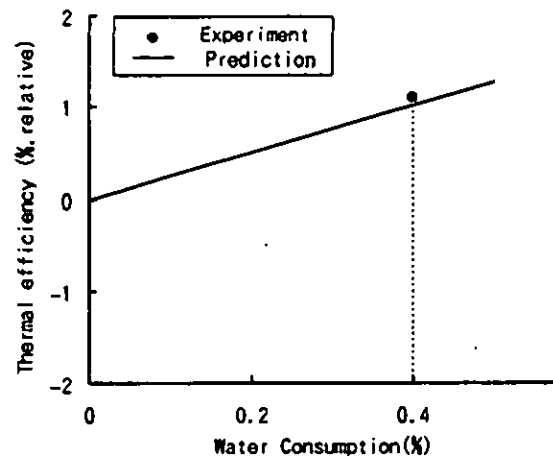


FIGURE 14 GAS TURBINE PERFORMANCE WITH WATER INJECTION

CONCLUSION

In the present study, we proposed a MAT cycle technology, a technique for improving the specific output and thermal efficiency of a gas turbine by entering water spray into the suction side of an axial flow compressor, and demonstrated how the characteristics of a compressor can be improved with relatively little water, in theory and by experiment.

1. This technique is expected to achieve an output increment of about 10% by applying water at an incoming air ratio of 1% with the external air at 35°C and 53%RH.
2. The theoretical heat rate for additional fuel consumption due to an output increment mechanism stemming from a decline in the motive power of a compressor was η_0 times lower than that of the Brayton cycle where η_0 is the thermal efficiency of the Brayton cycle.
3. The theoretical work decrease was 6.8% with regard to 1% water spray, while a compressor with an adiabatic efficiency of 0.79 produced a measurement of 4%.
4. Demonstration test using 130MW class simple cycle power station showed generator output shift from 115MW to 121MW with water injection of 0.4% to air ratio.
5. Thermal efficiency was improved by 1.1% (relative) with water injection of 0.4% to air ratio.

Our future challenges are to determine the top limit spray amount in linkage with the characteristics of a compressor.

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