Exergy Analysis of Wet-Compression Gas Turbine Cycle with Recuperator and Turbine Blade Cooling

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***Abstract*—Turbine blade cooling has been considered as the most effective way for maintaining high operating temperatures while making use of the available component material and has been a challenging area for improving the performance of gas turbine systems. In this work, exergy analysis of the wet-compression gas turbine cycle with recuperator and turbine blade cooling is performed. The exergy destructions in the system components and cycle exergy efficiency are estimated for varying pressure ratios and water injection ratios. Exergy destruction in the combustor and exergy loss due to exhaust gas are dominant and larger for higher pressure ratios and lower water injection ratios. Consequently the exergy efficiency decreases with pressure ratio but can be improved by injecting more water.**

***Index Terms*—water injection, wet compression, gas turbine, turbine blade cooling, exergy**

I. Introduction

The humidified gas turbines in which water or steam is injected at various positions have been attracted much attention, since they have the potential to enhance the power output with low cost. In these systems, evaporative cooling is a key process which can be classified as inlet fogging, after fogging and wet compression [1]-[3]. Wet compression is a process in which water droplets are injected into the air at the compressor inlet and allowed to be carried into the compressor. Since the droplet evaporation in the front stages of the compressor reduces the air temperature, the amount of compression work is reduced [4], [5]. Kim *et al*. [6], [7] studied on the performance of gas turbine cycles with wet compression and developed a model analyzing the transport operations for the non-equilibrium wet compression process based on droplet evaporation.

It is important in a gas turbine design to increase the power output and to reduce the fuel consumption. Improvement in the thermal efficiency of a power plant can be obtained by operating with higher turbine inlet temperatures. The maximum temperature of a gas turbine plant occurs at the entry of the first stage of the turbine. The employment of high temperature for given pressure ratio leads to a better system performance. However, employment of high temperature gas in gas turbines requires materials which can withstand the effects of high temperature operation [8]. One of the ways to overcome the problems resulting from high temperature is to maintain the temperature of the blade at a level low enough to preserve the desired material properties by blade cooling. For a variety of method of turbine blade cooling, there have been many studies modeling the turbine blade cooling process [9]-[13].

The exergy analysis is well suited for furthering the goal of more effective energy resource use, since it enables the location, cause, and true magnitude of waste and loss to be determined [14]. Kim et al. [15] carried out exergy analysis of simple and regenerative gas turbine cycles with wet compression. In this study exergy analysis of the wet-compression gas turbine system with recuperator and turbine blade cooling is carried out. Effects of pressure ratio and water injection ratio are investigated parametrically on the exergy destructions and the exergy efficiency of the cycle.

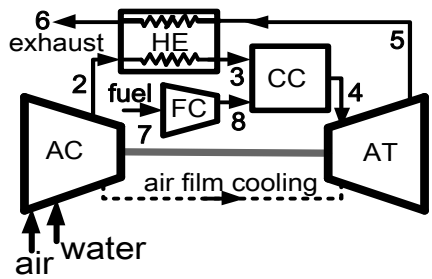

Figure 1. Schematic diagram of the system.

II. System Analysis

The schematic diagram of the system is shown in Fig. 1. Air enters the compressor at temperature T_1, pressure P_1 and relative humidity RH_1. At the same time liquid water droplets are injected into the air with initial diameter of D_1 at a rate such that the ratio of mass of liquid water to dry air is equal to f_1. For the convenience of analysis the injected water droplets are assumed to have uniform size. Fuel is assumed to be pure methane and enters the combustion chamber with air. The

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combustion occurs at constant pressure until the flame adiabatic temperature is reached. The hot gas is then expanded to state 5 and exhausts the turbine. The fuel mass is determined such that the turbine inlet temperature reaches a predetermined value. The recuperation of exhaust heat and the cooling of turbine blade also take place.

By considering the combustion process, total five kinds of gases, namely O_2, N_2, CO_2, H_2O and CH_4 are included in the analysis. All calculations are based on unit mass of dry air at the inlet of air compressor. The isobaric specific heat of gas of component i, specific enthalpy and the function s^0 used to calculate the entropy can be calculated as:

$$c_{p,i} = \sum c_{i,j} T^j \tag{1}$$

$$h(T,m) = \sum m_i \int_{T_0}^{T} c_{p,i}(T)dT \tag{2}$$

$$s^0(T,m) = \sum m_i \int_{T_0}^{T} \frac{c_{p,i}(T)}{T} dT \tag{3}$$

where T_0 is the reference temperature of 298.15 K, and m_i and m are the mass of component i and mass of gas mixture per unit mass of dry air, respectively.

The compression process can be modeled as follows [6]. The changing rate of mass and energy of the water droplets suspended in air are written with the quasi-steady relations as:

$$\frac{df}{dt} = -A \cdot I \tag{4}$$

$$f \cdot c_{pw} \cdot \frac{dT_s}{dt} = A \cdot (q_S - q_L) = A \cdot (q_S - h_{fg} \cdot I) \tag{5}$$

From mass and energy balance in the air and fuel compressors, coolant mass and works of air and fuel compressors are calculated as:

$$\sum m_{c,i} = m_{ac,in} - m_{ac,out} \tag{6}$$

$$w_{ac} = h_{ac,out} + \sum h_{c,i} - h_{ac,in} \tag{7}$$

$$w_{fc} = h_{fc,out} - h_{fc,in} \tag{8}$$

Here subscripts *ac*, *fc*, and *c* denote air compressor, fuel compressor, and coolant, respectively.

In the turbine, hot gas of mass m_g passes over the blade surface, while the coolant of mass m_c passing internally through the blade channels is ejected out from the leading edge which forms a film over the blade surface and finally mixes with hot gas at the trailing edge. The cooling factor R_c is defined as:

$$R_c = \frac{\left(T_{g,in} - T_b\right) c_{p,g} \left(1 - \eta_f\right)}{\varepsilon_f \left(T_b - T_{c,in}\right) c_{p,c}} \tag{9}$$

Here subscripts *g* and *b* denote the main gas and the turbine blade, respectively, and the term (1-η_f) takes into account of the reduction in heat transfer due to the coolant film formed over the blade surface. In addition, ε_f is the heat transfer effectiveness defined as

$$\varepsilon_f = \frac{\left(T_{c,out} - T_{c,in}\right)}{\left(T_b - T_{c,in}\right)} \tag{10}$$

This factor is introduced to treat the gas turbine blades cooled by internal convection as heat exchangers operating at uniform temperature and the coolant exit.

The expansion and mixing processes of main air and cooling air in each row are modeled to occur through the following four steps:

1) Main air flow is polytropically expanded.
2) Main air flow is cooled down by heat transfer.
3) Main air and cooling air are mixed at a constant pressure.
4) Pressure drops due to mixing process.

The details of modeling process and the related coefficients used for analyzing the turbine blade cooling can be found in [13].

The exergy which is a property of a stream is defined as the maximum useful work available when the stream evolves reversibly to reach equilibrium with the environment which is called to be in dead state. The reference environment used in this analysis has the same properties as the intake air to the compressor. The contributions of the kinetic and potential exergy are usually neglected and the rate of exergy flow can be written by the sum of contributions of physical exergy and chemical exergy as:

$$\dot{E} = \dot{E}^{PH} + \dot{E}^{CH} \tag{11}$$

In a gas turbine system the exergy can be supplied by the inlet air and fuel. Since the inlet dry air is in equilibrium with the environment (dead state) and the exergy of the dead state is defined zero, the inlet air has only the exergy of liquid water and vapor. A portion of the exergy supplied by the fuel is used to produce net power, while the hot exhaust gas stream still carries some exergy. The ratio of useful exergy production rate to the rate of fuel input exergy, that is, the ratio

$$\eta_{II} = \frac{\dot{E}_{prod}}{\dot{E}_{fuel}} \tag{12}$$

is called the exergy efficiency. This efficiency measures the fraction of original exergy converted to useful work. The difference between the rate of input and output exergy equals the rate of exergy destruction or loss. The following equation holds as an exergy balance:

$$\dot{E}_{fuel} = \dot{E}_{prod} + \dot{E}_{dest} + \dot{E}_{loss} \tag{13}$$

where the terms with subscripts *prod*, *dest*, and *loss* denote the exergy rate of work production, destruction, and loss, respectively. Identifying the components which have larger exergy destruction or loss is important for the

improvement of system performance. For comparison purposes, it is convenient to use the ratio of exergy destruction rate or exergy loss rate in each component such as combustor, recuperator and exhaust to the rate of exergy input which is supplied by fuel.

III. Results and Discussions

The basic data for the analysis are summarized as follows. Inlet air pressure: P_1= 1 atm, inlet air temperature: T_1= 15°C, relative humidity of inlet air: RH_1= 60%, compression rate: C= 200 s^{-1}, polytropic efficiency of compressor: η_c= 88%, initial diameter of water droplet: D_1= 10 μm, polytropic efficiency of turbine: η_t= 94%, effectiveness of recuperator: ε_r= 0.83, turbine blade temperature: T_b= 1123 K, isothermal efficiency: η_f = 0.4, heat transfer effectiveness: ε_f = 0.3.

The exergy destruction ratio of combustor is plotted with respect to pressure ratio in Fig. 2 for various water injection ratios. The case of f_1= 0% represents a dry compression, while the other cases are for wet compression in the compressor. The ratio of exergy destruction in the combustor is the largest among the components of the gas turbine system. It increases with increasing pressure ratio and decreasing water injection ratio, although the effect of water injection is less important.

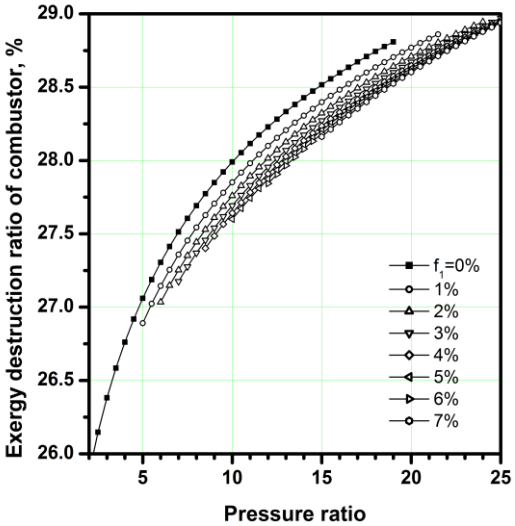

Figure 2. Variations of exergy destruction ratio of combustor with respect to pressure ratio for various water injection ratios.

Fig. 3 shows the exergy destruction ratio of recuperator as a function of pressure ratio for various water injection ratios. Contrary to the combustor the exergy destruction ratio of recuperator decreases with increasing pressure ratio for each water injection ratio and approaches zero as the pressure ratio becomes large. This is because the increased compressor exit temperature matching with high pressure ratio makes heat transfer in the recuperator decrease. It is to be noted that the change of turbine exit temperature with respect to pressure ratio is not significant. The exergy destruction ratio of recuperator increases with the increase of water injection ratio which lowers the compressor exit temperature due to larger cooling effect of droplet evaporation.

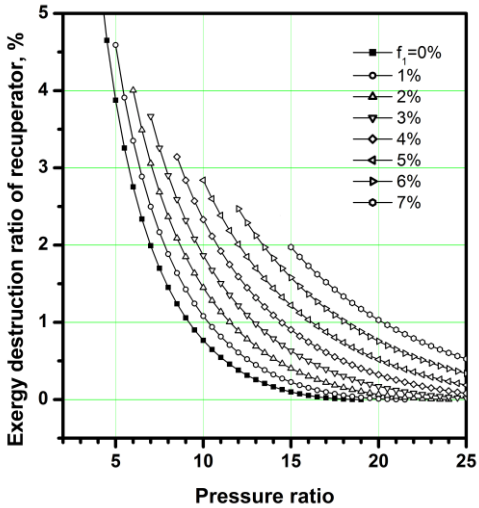

Figure 3. Variations of exergy destruction ratio of recuperator with respect to pressure ratio for various water injection ratios.

Fig. 4 shows the effect of pressure ratio and water injection on the exergy loss ratio due to exhaust gas. The exergy loss ratio increases with pressure ratio for each water injection ratio. However, it decreases significantly as more water is injected, since the compressor exit temperature and consequently the exhaust gas temperature decreases with increasing water injection ratio.

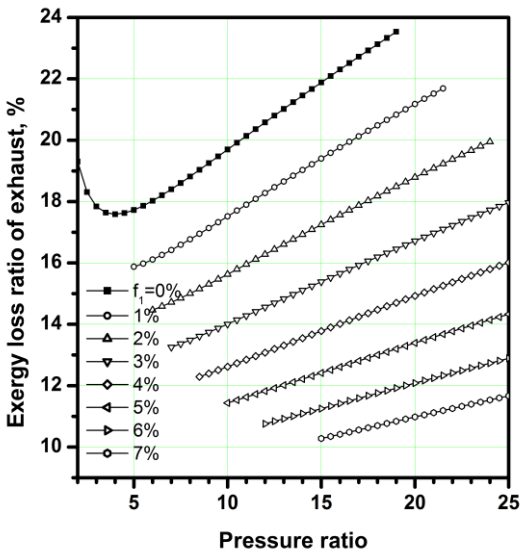

Figure 4. Variations of exergy loss ratio due to exhaust with respect to pressure ratio for various water injection ratios.

Exergy efficiency of cycle is plotted with respect to pressure ratio in Fig. 5 for various water injection ratios. It can have a maximum with respect to pressure ratio for the given values of water injection ratio, if the allowable pressure range is wide enough. For any pressure ratio remarkable improvement of exergy efficiency is expected by injecting water at the inlet of air compressor. This result implies that the effect of increased production of

net work is superior to the effect of increased exergy supply by fuel.

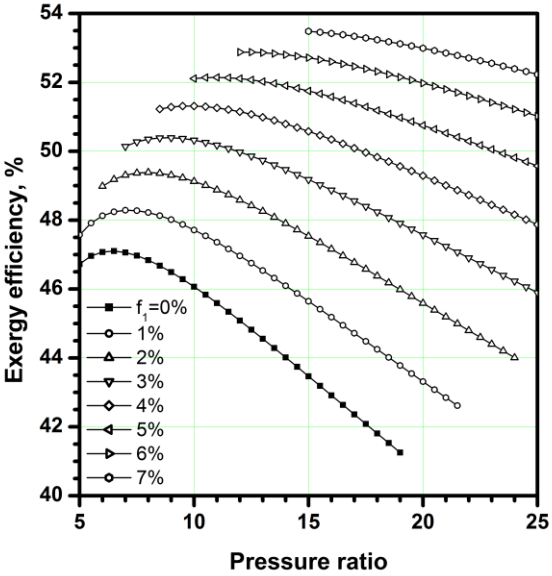

Figure 5. Variations of exergy efficiency of cycle with respect to pressure ratio for various water injection ratios.

IV. Conclusions

In this study exergy analysis of wet-compression gas turbine cycle with recuperator and turbine blade cooling is carried out. Effects of pressure ratio and water injection ratio are investigated on the exergy destruction or loss ratio in each component and the exergy efficiency of cycle. Calculation results show that wet compression reduces exergy destruction of combustor and exergy loss due to exhaust gas and significantly improves the exergy efficiency of gas turbine system with recuperator and turbine blade cooling.

Acknowledgment

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