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Performance Assessment of Wet-Compression Gas Turbine Cycle with Turbine Blade Cooling

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***Abstract*—**Turbine blade cooling is considered as the most effective way of maintaining high operating temperature making use of the available materials, and turbine systems with wet compression have a potential for future power generation because of high efficiency and high specific power with a relatively low cost. In this paper performance analysis of wet-compression gas turbine cycle with turbine blade cooling is carried out. The wet compression process is analytically modeled based on non-equilibrium droplet evaporation. Special attention is paid for the effects of pressure ratio and water injection ratio on the important system variables such as ratio of coolant fluid flow, fuel consumption, thermal efficiency and specific power. Parametric studies show that wet compression leads to insignificant improvement in thermal efficiency but significant enhancement of specific power in gas turbine systems with turbine blade cooling.

***Keywords*—**Water injection, wet compression, gas turbine, turbine blade cooling.

I. Introduction

THE efficient use of energy resources and applications is of great importance and has been attracting much attention in recent years[1-2]. The gas turbine is unquestionably one of the most inventions of the 20th century and gas turbine power plants are widely used all over the world for electricity generation [3-4]. One of the solutions can be using the humidified gas turbines which mean the systems in which water or steam is injected at various positions to enhance their power output. In these systems evaporative cooling is a key process which can be classified as inlet cooling, exit cooling, and wet compression [5-8].

Wet compression is a process whereby liquid water droplets are injected at the entrance of a compressor, and the applied evaporative cooling reduces compression work and increases the mass flow rate in the turbines, resulting in additional power production [9]. White and Meacock [10] simulated the effect of small water droplets in a compressor with two conceptions of equilibrium and non-equilibrium evaporation inside a compressor by assuming mean-line design. Zheng et al. [11] considered a thermodynamic model of the wet compression process and Sa et al. [12] investigated the effects of varying ambient temperature. Kim et al. [13-15] carried out energy and exergy analyses of the performance of gas turbine cycles with wet compression. They developed a modeling for the transport operations for the non-equilibrium wet compression process based on droplet evaporation and obtaining the analytic expressions with algebraic equations as solutions. Perez-Blanco et al. [16] investigated the general case of evaporatively-cooled compression with high-pressure refrigerants.

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The turbine blade cooling has been a challenging area for increasing the power efficiency and improving the performance of gas turbine systems. The gas turbine blade surface is cooled by using the compressed air from the compressor. In film cooling the cooling air flowing out of the many small holes forms a thin film over the blade surfaces, and reduces the heat transfer from the hot gases to the blade metal besides cooling the blade surface [17]. Various kinds of models for turbine blade cooling in many studies have been reported in the literature.

Najjar et al. [18] and Albeirutty et al. [19] investigated three different cooling techniques such as air-cooling, open-circuit steam cooling and closed-loop steam cooling for gas turbine systems. Cleeton et al. [20] carried out the cooling optimization of humid air turbine cycle and steam injected gas turbine cycle. Nowak et al. [21] discussed the optimization of internal cooling passages within a turbine blade with use of conjugate heat transfer analysis. Sanjay et al. [22-24] conducted parametric energy and exergy analysis of reheat gas-turbine cycle using closed-loop steam cooling. They carried out a comparative study of different turbine blade cooling methods on the thermo-dynamic performance of combined cycle power plant and compared the thermodynamic performance of gas turbines.

In this study, effects of wet compression on the performance of gas turbine systems with turbine blade cooling are parametrically investigated. Important system variables such as ratio of coolant fluid flow, fuel consumption, thermal efficiency, and specific power are estimated for various water injection ratios as well pressure ratios.

II. System Analysis

The schematic diagram of the system is shown in Fig. 1. Air enters the compressor at T_1, P_1 and RH_1 and at the same time liquid droplets are injected into the air with initial droplet diameter of D_1 at a rate of f_1, mass of liquid per unit mass of dry air.

Important assumptions used in this work are as follows [13]:

1) Gases are ideal gases whose thermodynamic properties are varied with temperature.
2) The dry air at compressor inlet consists of 0.2095 moles of O_2, 0.7902 moles of N_2 and 0.0003 moles of CO_2.
3) The fuel is methane (CH_4) and the combustion is complete

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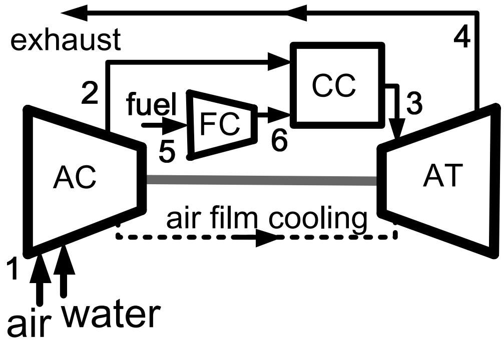

Fig. 1 Schematic diagram of the system

and adiabatic.

4) Compressors and turbine are characterized with polytropic efficiencies.
5) The injected water droplets evaporate completely inside a compressor.

The gases considered in this study are O_2, N_2, CO_2, H_2O and CH_4, which are numbered from 1 to 5 in order. All of the computations in this study are based on unit mass of dry air at compressor inlet. The isobaric specific heat of gas of component i can be expressed as

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

Then specific enthalpy and entropy function can be calculated as [2]:

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

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

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

The compression process can be characterized by the parameter of compression rate, C defined by [13]

$$C = \frac{1}{P}\frac{dP}{dt} \tag{4}$$

$$\Delta t_c = \frac{\ln(R_p)}{C} \tag{5}$$

Here Δt_c is the compression time, R_p the compression ratio, the pressure at compressor discharge, and it is assumed that C has a constant value. Irrespective of whether phase change is occurring, aerodynamic performance may be characterized by polytropic compressor efficiency as [11].

$$dh = \frac{vdP}{\eta_c} \tag{6}$$

The changing rate of mass and energy of the droplets can be written with the quasi-steady relations as [11]

$$\frac{df}{dt} = -A \cdot I \tag{7}$$

$$f \cdot c_{pw} \cdot \frac{dT_s}{dt} = A \cdot (q_S - q_L) = A \cdot (q_S - h_{fg} \cdot I) \tag{8}$$

Here I is the vapor mass flux away from the droplets, q_L the latent heat flux due to droplet evaporation, and q_S the sensitive heat flux due to convection. Here the initial temperature of the droplet is assumed to be same as that of air of T_1.

In this work heat and mass fluxes are expressed with Stokes model as follows [13]

$$I = \frac{2\rho D_v}{D} \ln\left(\frac{1+\omega_s}{1+\omega}\right) \tag{9}$$

$$q_S = \frac{I c_{pv}(T - T_s)}{\left(\frac{1+\omega_s}{1+\omega}\right)^c - 1}, \quad c = \frac{\rho D_v c_{pv}}{k} \tag{10}$$

where k is the thermal conductivity of air, D_v the mass diffusion coefficient of water vapor in air, R_v the gas constant of water vapor, and P_s is the saturated pressure at T_s.

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

$$\sum m_{c,i} = m_{ac,in} - m_{ac,out} \tag{11}$$

$$w_{ac} = h_{ac,out} + \sum h_{c,i} - h_{ac,in} \tag{12}$$

$$w_{fc} = h_{fc,out} - h_{fc,in} \tag{13}$$

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

The combustion process can be expressed as:

$$w_{fc} = h_{fc,out} - h_{fc,in} \tag{14}$$

The fuel consumption per unit mass of dry air, mf can be determined from the following equation:

$$CH_4 + 2O_2 \rightarrow CO_2 + 2\,H_2O \tag{15}$$

Here subscripts of r and p denote reactant and production, respectively, and h_i is the enthalpy of formation of component i.

$$\sum m_i^r \left[h_i(T_i^r) - h_i(T_{ref}) + h_{f,i}^0\right] = \sum m_i^p \left[h_i(T_i^p) - h_i(T_{ref}) + h_{f,i}^0\right] \quad (16)$$

A model of film cooling is as follows [22-24]: The hot gas of m_g passes over the blade surface, while the coolant of 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 film so formed reduces heat transfer from hot gas to the blades. Gas turbine blades cooled by internal convection are treated as heat exchangers operating at uniform temperature and the coolant exit temperature is expressed as a function of heat exchanger effectiveness. Further, isothermal effectiveness, η_f, is introduced to account the reduced heat transfer rate from hot gas to blades. It is defined as the ratio of difference between heat transfer fluxes with and without cooling to an isothermal wall. Heat transfer effectiveness ε_f is defined as $(T_{c,out} - T_{c,in})$ / $(T_b - T_{c,in})$.

The expansion and mixing processes of main and cooling air in each row are modeled as following four steps: 1) main air flow is polytropically expanded, 2) main air flow is cooled by heat transfer, 3) main and cooling air are mixed at a constant pressure, 3) the pressure is dropped due to mixing process. Each row is treated as expander whose walls continuously extract work. Then the temperature of main air flow after polytropically expanded can be computed as:

$$dh = \eta_t v dP \quad (17)$$

$$s_{out}^0 - s_{in}^0 = -\eta_t m_m R_m \ln(P_{in} / P_{out}) \quad (18)$$

Temperature of main air flow after cooled by heat transfer can be obtained from:

$$m_g(h_{g,in} - h_{g,out}) = m_c(h_{c,out} - h_{c,in}) \quad (19)$$

Enthalpy loss from mixing between hot main air and cold cooling air can be obtained from:

$$m_g h_{g,out} + m_c h_{c,out} = (m_g + m_c) h_{mix,out} \quad (20)$$

Turbine work from all stages is given as:

$$w_t = \sum_{row} m_g(h_{g,in} - h_{g,out})_{cooled} + \sum_{row} m_g(h_{g,in} - h_{g,out})_{uncooled} \quad (21)$$

III. Results and Discussions

In this study the basic data for analyses are initial pressure P_1 = 1 atm, inlet temperature T_1 = 15 °C, relative humidity at inlet, RH_1 = 60%, compression rate C = 200 s^{-1}, polytropic compressor efficiency η_c= 88%, initial droplet diameter D_1= 10 μm, polytropic turbine efficiency η_t= 94%, turbine blade temperature T_b= 1123 K, isothermal efficiency η_f = 0.4, and heat transfer effectiveness ε_f = 0.3.

The ratio of coolant flow is plotted against pressure ratio in Fig. 2 for various water injection ratios. Here, the case of f_1 = 0% denotes no injection of water into a compressor, so dry compression occurs inside the compressor. The ratio of coolant flow is defined as the ratio of mass flow rate of coolant of m_c passing internally through the blade channel to that of the hot gas of m_g passing over the blade surface. The ratio increases with increasing pressure ratio, since temperature of coolant increases with pressure ratio and consequently more coolant is required to maintain the blade surface temperature at a constant value. The ratio decreases with increasing water injection ratio, since higher water injection ratio results in lower temperature of coolant flow.

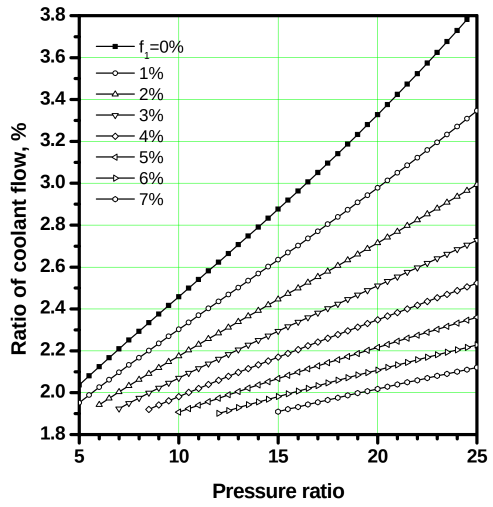

Fig. 2 Variations of ratio of coolant flow with respect to pressure ratio for various water injection ratios

On the other hand it can be seen from the figure that there exist lower limit values of workable pressure ratio for a given water injection ratio and an initial droplet diameter, since the analysis in this study is restricted to the cases of complete evaporation on injected droplets inside a compressor to avoid the liquid water flow outside the compressor. Under the lower limit of pressure for a given water injection ratio, it is

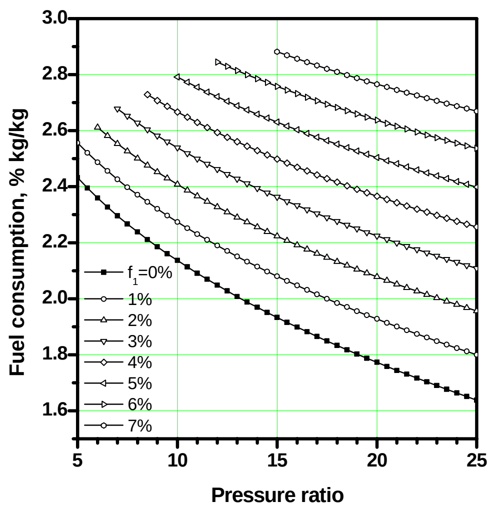

Fig. 3 Effects of pressure ratio on the fuel consumption for various water injection ratios

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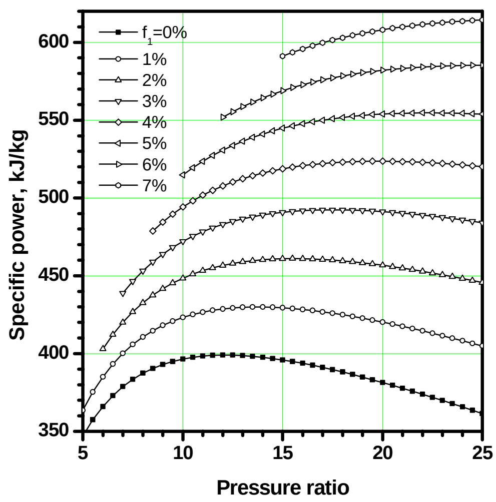

Fig. 4 Effects of pressure ratio on the specific work for various water injection ratios

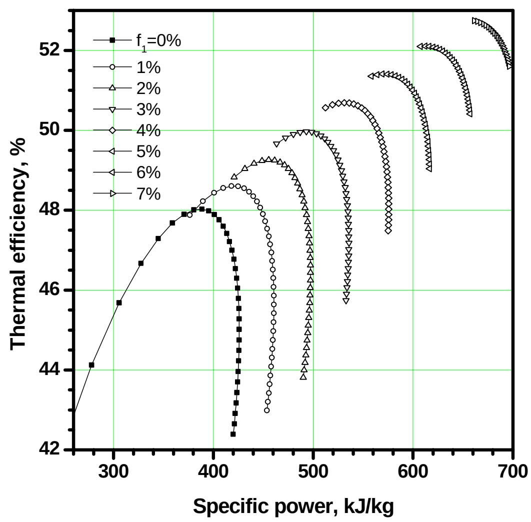

Fig. 6 Variations of thermal efficiency with respect to specific power for various water injection ratios

impossible for water droplets to evaporate completely within the compressor. It is worthy to note that the lower limit of pressure ratio increases with water injection ratio.

Fig. 3 shows variations of fuel consumption per unit mass of dry air with respect to pressure ratio for various water injection ratios. The specific fuel consumption decreases as pressure ratio increases, since rise of pressure ratio causes increase in compressor outlet temperature and consequently results in reduction of fuel consumption. The decrease in mass flow rate of air flowing into the combustor due to coolant flow for turbine blade cooling is also a reason for the reduction of the fuel consumption. For a given pressure ratio, the specific fuel consumption increases with water injection ratio due to cooling by droplet evaporation.

Specific power per unit mass of air is plotted in Fig. 4 with respect to pressure ratio for various water injection ratios. As pressure ratio increases for a specified water injection ratio, the specific power increases and reaches a maximum value and then decreases, so it has a peak value with respect to pressure ratio. This is because specific work is the difference between work production of turbine and work consumption of compressor, and the compression work as well as turbine work increases with increasing pressure ratio. It can be seen from the figure that the optimum value of pressure ratio increases with water injection ratio. For a given pressure ratio, specific power increases significantly with increasing water injection ratio, mainly due to increase in mass flow rate of working fluid.

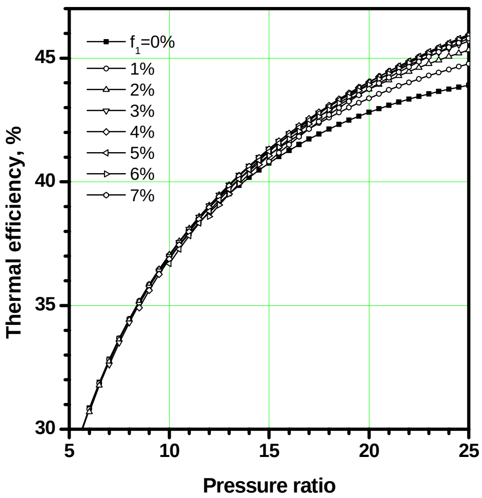

Fig. 5 Effects of pressure ratio on thermal efficiency for various water injection ratios

Thermal efficiency is plotted against pressure ratio in Fig. 5 for various water injection ratios. The thermal efficiency, which is defined as the ratio of specific power to heat addition, increases with pressure ratio, since heat addition decreases with increasing pressure ratio. For a given pressure ratio, thermal efficiency increases with water injection ratio, too. However, the increase of thermal efficiency is insignificant, since fuel consumption as well as specific power increases as water injection ratio increases.

Fig. 6 shows characteristic diagram of thermal efficiency with respect to specific power for various water injection ratios. There exists a peak value of thermal efficiency with respect to specific power, and there exists also a peak value of specific power with respect to specific power. It can be seen from the figure that the optimum pressure ratio for the maximum thermal efficiency is lower than that for the maximum specific power.

IV. Conclusion

In this study effects of wet compression on thermodynamic performance of gas turbine cycle with turbine blade cooling are investigated by using a modeling based on the evaporation of injected liquid droplets and a modeling for turbine blade film cooling. To avoid modeling the liquid water downstream of the compressor, the analysis in this work is restricted to complete evaporation case only. Most significant system parameters of the system are pressure ratio and water injection ratio. Effects of system parameters are thoroughly investigated on the important performance variables such as the ratio of coolant flow, fuel consumption, specific power and thermal efficiency. Numerical results show that wet compression leads to insignificant improvement in thermal efficiency but significant enhancement of specific power in gas turbine systems with turbine blade cooling.

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