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Influence of BladeCooling on the Efficiency of Humid Air Turbine*

Jan SZARGUT, Janusz SKOREK, Ireneusz SZCZYGIEL
Technical University of Silesia
Institute of Thermal Technology
ul. Konarskiego 22, 44 100 Gliwice - Poland
Tel.: 48 (0)32 237-22-12
Fax.: +48 (0)32 237-28-72

Abstract

The influence of an open system blade cooling on the efficiency of the humid air turbine has been analyzed. It has been assumed that three stages of the turbine are cooled by means of compressed air taken from the outlet of the compressor. The thermodynamic analysis showed, that the deleterious impact of blade cooling results mainly from the irreversibility of mixing of cooling air with the working fluid. The influence of the blade cooling on the optimum operational parameters of HAT turbine (compression ratio, excess air, air humidity and preheated air temperature) has been analyzed too.

Key words: gas turbine, cooling of blades, humid air turbine

1. Introduction

The analyzed humid air turbine (HAT) power plant is equipped with a regenerative preheater of the compressed air. The preheater is fed with the turbine exit gases (*Figure 1*). The liquid water is injected into the compressed air before the inlet to the preheater. The humidification of the air allows to decrease the air excess ratio in the combustion process and so to decrease the ratio of mass flow rates in the compressor and in the turbine. This decreases the proportion of the compressor driving power to the turbine power. The interstage cooling of the compressed air ensures an additional decrease of the compressor driving power. As the result, the temperature of the turbine exit gases flowing from the regenerative preheater decreases. The mentioned features provide the high efficiency of the HAT power plant. Therefore many authors have investigated this plant (Bram and De Ruyck 1996, Cohn and Waters 1982, El-Masri 1986, Lazaretto and Segato 1999, Rufli 1990, Stecco et.al. 1993, Szargut 1997), but without any analysis of the influence of blade cooling on the efficiency.

In current gas turbines the temperature of combustion gases is very high (up to 1400 ^{o}C), and therefore the turbine blades are cooled. Usually the compressed air taken from the outlet of the compressor is used for this purpose in the open cooling system (*Figure 1*). After flowing through the inner channels of the blades, the cooling air is mixed with the working gases. This evokes exergy losses and decreases the temperature of working fluid before the next step of expansion. Eventually the efficiency of the power plant decreases. In order to reduce the compressor driving power, the air which cools the further stages of turbine could be taken from prior stages of the compressor, but this would complicate its construction. The losses appearing in the open cooling system could be avoided by the application of a closed system (with air or steam as a cooling medium), but the construction of the turbine would be much more complicated in this case and therefore is not applied in current gas turbines.

2. Influence of Cooling on the Operation of the Turbine Stage

A thermodynamic analysis of the influence of cooling on the work of a gas turbine has been made by El-Masri (Eckert 1984). He assumed a continuous injection of water and continuous mixing with working fluid. This assumption is far from the real course of processes. Similar solutions are presented by Traupel (1988) and

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Rufli (1990). Eckert (1984) analysed the conditions of heat transfer on the blade surface cooled with air film. Cohn and Waters (1982) compared various cooling systems.

In present investigation the mathematical model of the HAT-plant has been developed, taking into account particular stages of the turbine. It has been assumed, that the humidification takes place not in a scrubber (as assumed e.g. Gallo (1996) and Xiao et al. (1996)) but by means of the injection of water, as proposed by Bram and De Ruyck (1996), De Ruyck et al. (1997), and assumed also by Szargut and Cholewa (1997) and Szargut (1997). In this case the chain of processes is shorter, but the air with mist droplets flows to the preheater. The formation of mist can be avoided by means of the staggered injection, Szargut and Szczygiel (1999).

The influence of cooling on the amount of work performed in one stage is small. This conclusion results from the analysis of the energy balance. To simplify analysis, the expansion without friction is taken into account (*Figure 2*).

The specific work performed in a cooled and in an adiabatic stage can be expressed respectively as follows:

$$l_c = i_1 - i_2 - q_c \qquad l_0 = i_1 - i_{2s} \tag{1}$$

where,

i_1, i_2, i_{2s} : specific enthalpy of the working fluid at the inlet (1) and outlet of the cooled (2) and adiabatic (2s) stage,

q_c : heat of cooling per unit of the working fluid.

The decrease of the performed work caused by cooling is presented in *Figure 2* by means of the area A(1 – 2 – 2s – 1)

$$-\Delta l = l_0 - l_c = A(1-2-2s-1) \tag{2}$$

The ratio of the decrease of work to the heat of cooling can be approximately expressed by the area of the triangle and trapezoid:

$$\frac{-\Delta l}{q_c} = \frac{\Delta s\,(T_1 - T_{2s})}{\Delta s\,(T_1 + T_2)} \tag{3}$$

Assuming approximately T_1 = 1500K, T_2 = 1300 K, T_1 - T_{2s} = 150 K one obtains $\Delta l/q_c$= 5%.

The work performed in the reversible adiabatic stage and the heat of cooling per unit of the working fluid can be expressed as follows:

$$l_0 = c_{ps}(T_1 - T_{2s}) \qquad q_c = \zeta\, c_{pa}\, \Delta T_c \tag{4}$$

where: c_{ps}, c_{pa} : mean isobaric specific heat of the working fluid and cooling air in the considered temperature interval, ζ : ratio of the amount of cooling air to the amount of working fluid, ΔT_c : temperature increase of the cooling air inside the blade channels.

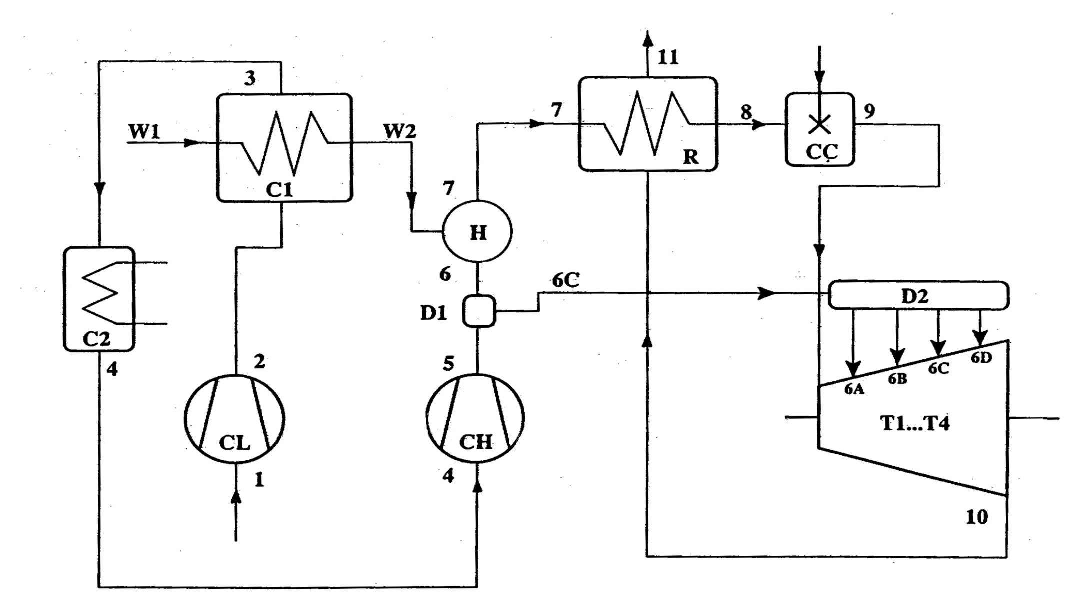

Figure 1. Scheme of the HAT power plant with cooling of turbine blades (CL, CH — low pressure and high pressure stage of the compressor, T1...T4 — stages of the turbine, C1 — interstage cooler and preheater of the injection water, C2 — external interstage cooler, CC — combustion chamber, H — humidifier of the compressed air, D — distributor of the cooling air, R — regenerative air preheater, 1...11 — streams of working fluid, W1,W2 — streams of injection water).

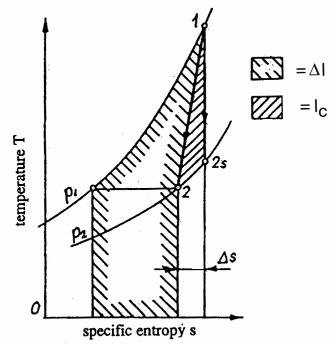

Figure 2. Influence of cooling of the blades on the work of turbine stage

For $c_{pa} \cong c_{ps}$; $\zeta = 0.07$; $\Delta T_c = 400$ K one obtains:

$$\frac{-\Delta l}{l_0} = \frac{-\Delta l}{q_c}\,\frac{q_c}{l_0} \cong 0.01 \qquad (5)$$

Hence the influence of cooling on the work performed in the stage is very small. The main deleterious effect of cooling is the decrease of temperature of working fluid before the next stage.

In order to calculate the temperature of working fluid before the next stage of turbine, the energy balance can be formulated taking into account the performed internal work. According to the result of Eq.5 it can be assumed, that this work equals to the work performed in an irreversible adiabatic stage:

$$\dot{n}_s c_{ps} T_{sd} + \dot{n}_c c_{pa} T_a = \dot{n}_s c_{ps} (T_{sd} - T_{sw}) + (\dot{n}_s c_{ps} + \dot{n}_c c_{pa}) T_w \qquad (6)$$

where, $\dot{n}_s$, $\dot{n}_c$: flow rate of the working fluid and cooling air at the inlet to the stage, T_{sd}, T_a: temperature of the working fluid and cooling air at the inlet to the stage, T_w: temperature of the working fluid and cooling air at the outlet, T_{sw} : temperature of working fluid which would appear at the outlet of an irreversible adiabatic stage.

The first term on the right hand of Eq.(6) expresses approximately the internal work of the stage. From Eq.(6) the temperature T_w of working fluid before the next stage can be derived.

3. Assumed Data and Calculation Method

The following data have been assumed:

- fuel: natural gas containing 92% CH_4, 0.7% C_2H_4, 0.6% C_3H_8, 0.7% C_4H_{10}, 6% N_2,
- temperature of combustion gases before the turbine 1200, 1300, 1400 ^oC,
- temperature of environment and fuel 20 ^oC,
- temperature difference between the combustion gases and compressed air at the hot and cold ends of the air preheater 30 K and 10 K,
- stage isentropic efficiency of turbine and each compressor 0.84 and 0.88,
- mechanical efficiency of machines 0.98,
- humidity of atmospheric air 80%,
- humidity content in fuel 0.0242 kmol/kmol dry gas,
- pressure losses in the combustion chamber 3.5%,
- pressure losses in the air preheater at the air side 3% and at the combustion gases side 4.5%,

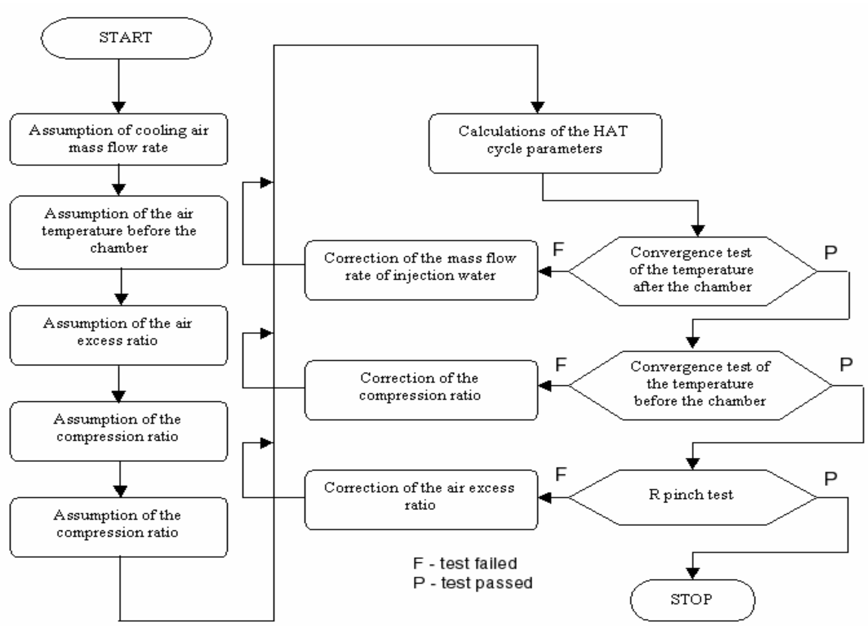

Figure 3. Block scheme of the calculations program

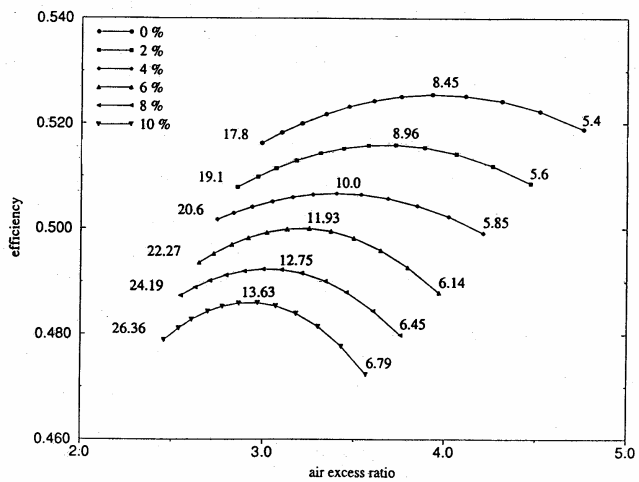

Figure 4. Efficiency of the power plant at the temperature of combustion gases 1200 °C

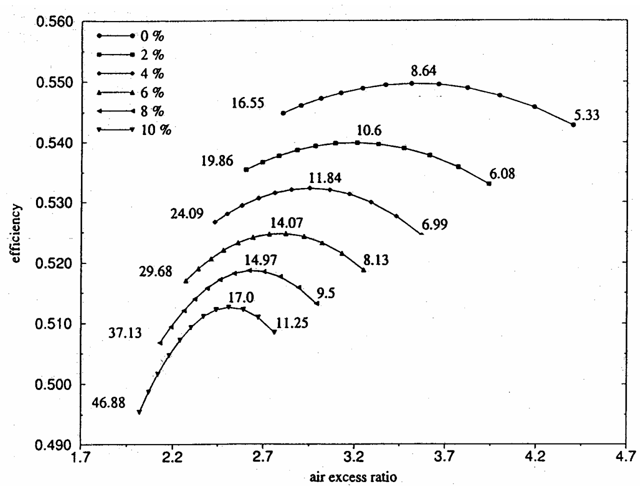

Figure 5. Efficiency of the power plant at the temperature of combustion gases 1300 °C

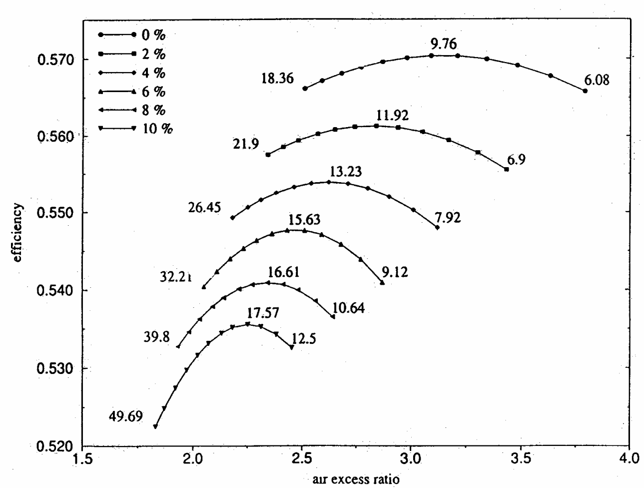

Figure 6. Efficiency of the power plant at the temperature of combustion gases 1400 °C

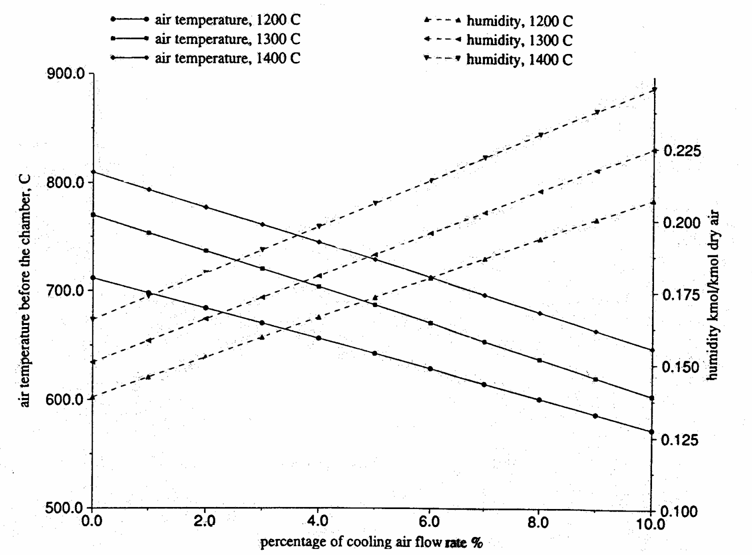

Figure 7. Characteristic parameters in the points of maximum efficiency

- pressure losses of air in the interstage cooler of the compressor 2.5% and in the external cooler 2.5%,
- heat losses in the air preheater 1%,
- heat losses in the combustion chamber 0.5%,
- temperature of the injection water at the inlet to the interstage cooler 15 °C,
- internal and mechanical efficiency of the injection pump 0.77 and 0.95,
- efficiency of the electric generator 99%.
- minimum temperature difference between air and water in the interstage cooler 10 K,
- air temperature before second stage of compressor 51 °C,
- efficiency of the electric generator 99%.

The internal efficiency of the entire turbine is greater than the efficiency of particular stages, because of the recovered exergy of internal energy dissipation (hydrolic friction). The stage isentropic efficiency has been so assumed that the efficiency of the entire turbine equals that given by the producers.

The block scheme of the calculation is presented in *Figure 3*. The R pinch test denotes the test of the admissible temperature difference between the heat exchanging gases in the regenerative air preheater. An iterative calculation procedure is necessary. The temperature of the preheated air is assumed in first step. At a given temperature of combustion gases before the turbine and given temperature of preheated air the amount of injected water can increase if air excess becomes smaller. However when the air excess decreases, the temperature difference between the combustion gases and the air at the cold end of the air preheater also becomes smaller. Hence the limitation of the air excess and of the amount of injected water appears. The optimal compression ratio increases when the air excess becomes smaller. Changing the temperature of the preheated air one can determine its optimal value ensuring the maximum efficiency of the plant.
Calculations have been made for the flow rates of cooling air between 0 and 10% of the flow rate of compressed air. It has been assumed that the turbine has three cooled stages. The mixing of cooling air with combustion gases appears after the guide vanes of the first stage, after the rotor blades of the first stage and after the second and third stages of the turbine. The data about the consumption of cooling air are not accessible. Lazaretto and Segato (1999) evaluated in their example the friction of cooling air to be 14% of the compressed air.

The calculation performed by means of the computer Pentium 166 MHz lasted 5 hours. The application of the computer HP735 shortened the calculation to 2 hours.

4. Results of Calculations

The results of calculations are presented in *Figures 4 - 7. Figures 4,5* and *6* present the dependence of the efficiency on the air excess ratio and flow rate of cooling air. Values of compression ratio are shown in characteristic points of the curves. In *Figure 7* the humidity of air and the temperature of preheated air appearing at maximum efficiency are presented.

5. Conclusions

1) The decrease in the efficiency of plant due to the cooling of blades, expressed in percentage points, has similar values for all the considered values of temperature before the

turbine. When the flow rate of cooling air amounts to 10% of the compressed air, the efficiency of the plant decreases by about 3.5 percentage points. This deleterious result is equivalent to the decrease of temperature of combustion gases before the turbine by about 150 ^{o}C.

2) The increase in the amount of cooling air evokes the decrease of the optimal air excess and increase of the optimal compression ratio.

3) The increase in the amount of cooling air makes it possible to increase the humidity content in the compressed air after injection.

Nomeclature

c_p isobaric specific heat capacity,
i specific enthalpy,
l work per unit of the working fluid,
n amount of substance in kmol,
q heat per unit of the working fluid,
s specific entropy,
T absolute temperature,
Δ symbol of the increase.

Acknowledgement

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