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Improving Gas Turbine efficiency by chilled water system

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ABSTRACT: The process in a gas turbine plant involves certain losses which can be divided into internal and external losses. In term of internal losses, the main factor is changing the state of working fluid. Since the temperature of atmospheric air may vary within a wide range, its variations can influence strongly the efficiency of gas turbine plants. With growing ambient air temperature, the specific volume of air increases, which can result in a larger work spent for air compression in the compressor. One of the most effective method for increasing the efficiency of gas turbine plants is to raise the gas temperature before the turbine. Since this temperature is the highest temperature in the cycle, this method is applicable for gas turbine plants of any scheme and type. However, there are some limitations on increasing gas temperature. The allowable temperature for reliable operation is between 1000 and 1400 k. However, decreasing ambient air temperature to increase the efficiency of gas turbine plants is easier and at low costs compared to rising gas temperature. As a decrease of 1°C temperature of inlet air increases the power output by 1%. In this paper our objective is improving the efficiency of gas turbine plants by decreasing ambient air temperature. To reach this we use a heat exchanger with different strip fins. The temperature of chilled water is constant on 7C°, but the ambient air temperature is variable between 20 and 36 C°. After designing process some graphs are presented, which give required mass flow rate to reach slightly ambient air temperature.

KEYWORDS: Gas Turbine, Strip Fins, Heat Exchanger, Chilled Water System.

1 INTRODUCTION

In gas turbines, the working fluid is a mixture of gases products of fuel combustion and air or a suitable gas (such as air) heated to a high temperature. The working fluid expends in the flow path of a gas turbine and its heat is transformed into the kinetic energy of the gas flow which is then converted into mechanical work on the shaft of a spinning rotor [1].

Gas turbine is finding ever wider application in power engineering. Due to their favorable start-up characteristics, gas turbine plants are often employed for carrying peak and semi peak loads and as standby units. In some cases, it is considered efficient to use them as base-load units at small power stations, with the heat of exhaust gases being utilized for heat supply [1]. In view of their law mass, small size, high mobility, and ease of maintenance gas turbines are used widely in automobile-hauled and railway mobile power plants.

Since the temperature of atmospheric air may be vary within a wide range, its variations can influence substantially the efficiency and power of gas-turbine plants. A gas turbine may experience a nominal power loss of about 7% when the intake temperature increases from 15 C° (at ISO conditions) t0 25 C° and even could reach up to 15 % lost when the ambient temperature is 36 C°.

2 SCHEMES AND CYCLES OF GAS TURBINE PLANTS

The T-S diagram of the combined thermodynamic and real cycles of a gas turbine plant with combustion at P-constant is shown in Figure of (1).

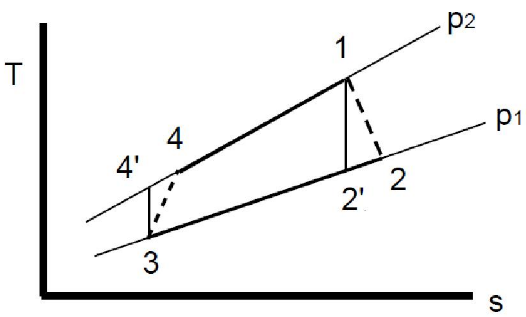

Fig. 1. T-S diagram of gas turbine cycle with combustion at P-constant

The thermodynamical cycle has been constructed under the following main assumptions; (1) the cycle is closed and is effected with a constant quantity of an ideal gas having a constant composition and constant specific heat; (2) all processes in the cycle are reversible, i.e. occur without thermal and hydraulic losses [4];(3) compression in the compressor and expansion in the turbine occur adiabatically. Since there are no losses, these process take place ate constant entropy. Hence, line 3'-4 describes the isentropic compression of air in the compressor which involves an increase of the air temperature and pressure from the initial value P_3 and T_3 to P_4 and T_4'. In the real cycle, however, compression is associated with internal losses in the compressor, so that the process line shifted towards increasing entropy (line 3-4).

Heat is supplied to the working fluid in the combustor along isobar 4-1. Line 1-2' denotes the isentropic expansion of the working fluid in the turbine [1]. In the real cycle with internal losses in the turbine, which increase the entropy, expansion occurs along line 1-2, so that the pressure decreases to P_2 and temperature to T_2 .

The process in areal gas-turbine plant involves certain losses which can be divided into internal and external. Internal losses are closely associated with changes of the state of working fluid. The internal efficiency of the plant is:

$$\eta_i = \frac{\tau\left(1-\frac{1}{\beta^m}\right)\eta_{ri} - \left(\beta^m - 1\right)\left(1/\eta_c\right)}{\tau - 1 - \left(\beta^m - 1\right)\left(1/\eta_c\right)}\eta_{cc}^{th} \qquad (1)$$

Where $P_1/P_2 = P_4/P_3 = \beta$ is the compressor pressure ratio, and $T_1/T_3 = \tau$ is the degree of temperature rise in the cycle. η_c , η_{cc}^{th} and η_i are compressor efficiency, thermal efficiency of combustor and turbine efficiency respectively.

Since the temperature of atmospheric air may be vary within a wide range, its variations can influence substantially the efficiency and power of gas-turbine plants [5-7]. With a increase of T_3 , the specific volume of air increases, resulting in a larger work spent for air compression in the compressor. The mass flow rate of air, and therefore, the capacity then decreases.

3 PLATE-FIN HEAT EXCHANGER

Plate-fin heat exchangers are a mixture of flat plates and corrugated fins in a sandwich construction. Heat is transferred from the hot stream through the fin interface to the parting sheet and through the alternating layer fins into cold stream.

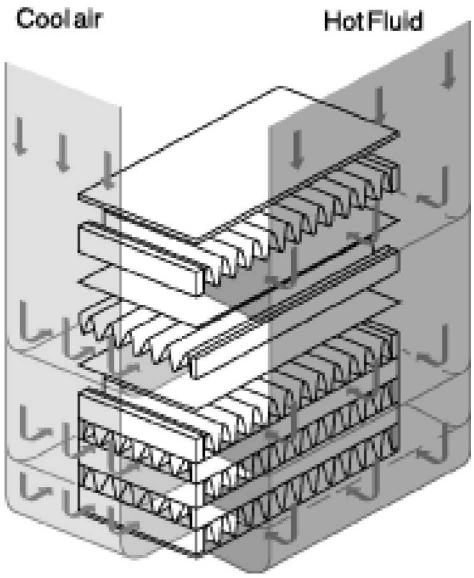

Fig. 2. Plate-fin heat exchanger

To determine flow areas, Knowing " σ " as one of the fin geometric parameter is required because:

$$\alpha_i = \frac{b_1}{b_1 + b_2 + 2a}\beta_i \tag{2}$$

$$A = \left(\frac{\alpha.D_h}{4}\right)A_{fr} \tag{3}$$

Here, α is the ratio of the total surface on one side to total volume, and β is the ratio of heat transfer area to volume between plates. Also, A_{fr} is frontal surface area and is introducing by,

$$A_{f,1} = H \times W \tag{4}$$

$$A_{f,2} = H \times D \tag{5}$$

Pressure drop for both streams is described by,

$$\Delta P_1 = \frac{G_1^2}{2\rho_{in,1}}\left[\left(1 + K_{C,1} - \sigma_1^2\right) + 2\left(\frac{\rho_{in,1}}{\rho_{out,1}} - 1\right) + \left(f_1 \times \frac{S_1}{A_1} \times \frac{\rho_{in,1}}{\rho_{m,1}}\right) - \left(1 - \sigma_1^2 - k_{e,1}\right) \times \frac{\rho_{in,1}}{\rho_{out,1}}\right] \tag{6}$$

The terms of these equations are entrance losses, flow acceleration, care friction, and exit losses respectively. The K_C and K_e values depend on the cross-sectional flow geometry, σ and Reynolds number. Noticeably, the entrance and exit losses are normally less than 10% of the total core loss.

The convective coefficient can be introduced by,

$$h = j.G.C_p.\mathrm{Pr}^{\frac{-2}{3}} \tag{7}$$

We can use use, here, some dimensionless parameters such as:

Colburn factor: $j = St.\mathrm{Pr}^{2/3}$ (8)

Stanton number: $St = \frac{Nu}{\mathrm{Re}.\mathrm{Pr}}$ (9)

Nusselt number: $Nu = \frac{h.D_h}{K}$ (10)

Prandtl number: $\Pr = \dfrac{C_p \cdot \mu}{K}$ (11)

Reynolds number: $\mathrm{Re} = \dfrac{G.D_h}{\mu}$ (12)

The fin efficiency correlation for the air and chilled water side is:

$$\eta_{fi} = \frac{Y_{o,fi}}{2\left(b_i + S_i\right)Lh} \frac{\tanh m_{fi} b_i + \left(2Y_{o,s} / Y_{o,f}\right)\tanh m_s S_i}{1 + \left(2Y_{o,s} / Y_{o,f}\right)\tanh m_s S_i \tanh m_{fi} b_i} \tag{13}$$

On the other hands, overall passage efficiency is:

$$\eta_{w,1} = 1 - \left(1 - \eta_{f,1}\right) \times \frac{S_{f,1}}{S_1} \tag{14}$$

So, now calculating heat transfer coefficient is possible by,

$$\frac{1}{U_1} = \frac{1}{h_1 \eta_{w,1}} + \frac{S_1}{h_2 \eta_{w,2} S_2} \tag{15}$$

4 RESULT AND DISCUSSION

The parameters for energy rate balance in the plate-fin heat exchanger between hot ambient air and chilled water are according to Table of (1). Based on Karim research [3] the best case occurs when we use fin of "1/8-19.82(D)" in hot side and "1/8-20.06(D)" in cold side.

Table 1. Operating conditions [3]

Description	Air	Chilled water
Inlet temperature	varies from 23C to 36 C	maintained at 7C
Outlet temperature	constant at 20 C	maintained at 13C
Mass flow rate	19 kg/s	Must be determined
Specific heat	1.005 kJ/kg.K	4.18 kJ/kg.K

Table 2. Sizing combination for strip-fins [3]

No	Air	Water
1	1/8 - 20.06 (D)	1/8 - 13.95
2	1/8 - 20.06 (D)	1/8 - 19.82 (D)
3	1/8 - 20.06 (D)	1/8 - 16.00 (D)
4	1/8 - 19.82 (D)	1/8 - 16.00 (D)
5	1/8 - 19.82 (D)	1/8 - 13.95
6	1/8 - 16.00 (D)	1/8 - 13.95
7	1/8 - 19.82 (D)	1/8 - 20.06 (D)
8	1/8 - 13.95	1/8 - 16.00 (D)
9	1/8 - 13.95	1/8 - 19.82 (D)
10	1/8 - 16.00 (D)	1/8 - 20.06 (D)
11	1/8 - 13.95	1/8 - 20.06 (D)
12	1/8 - 16.00 (D)	1/8 - 19.82 (D)

After selecting fins, the mechanical design process will start. At this stage, it is necessary we assume a value for water mass flow rate. This assumption must be based on previous experiences. If the results were not satisfactory, it would change. Here, to improve gas turbine efficiency, a plate-fin heat exchanger has been used, where the just variant is the mass flow rate of water. Naturally, this mass flow rate must be according to ambient air temperature. To reach this an Algorithm has been written, and consequently different graphs (For different ambient air temperature) are extracted.

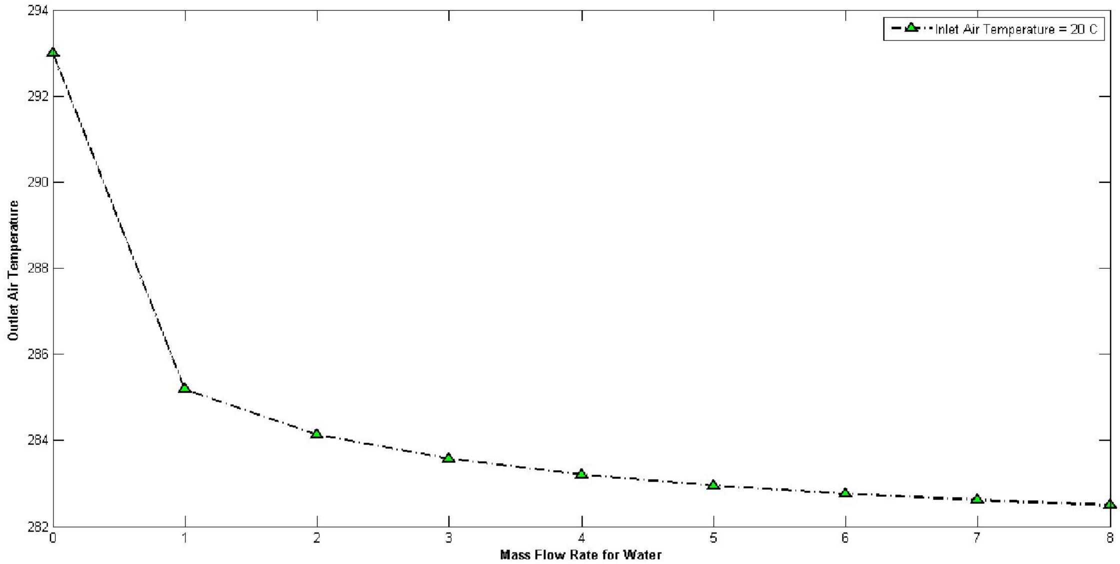

Fig. 3. Outlet air temperature vs mass flow when ambient air temperature is 20 C

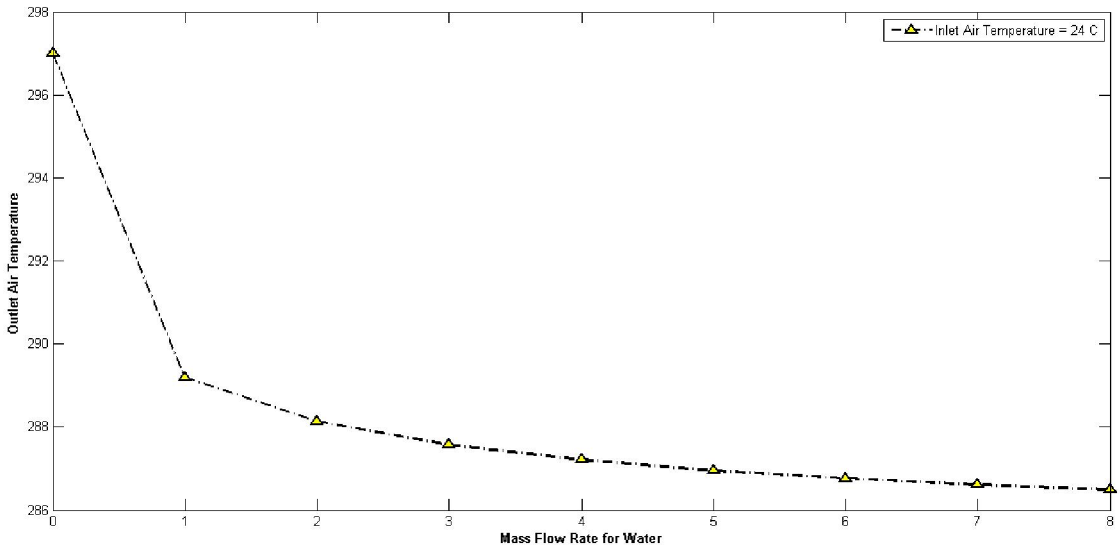

Fig. 4. Outlet air temperature vs mass flow when ambient air temperature is 24 C

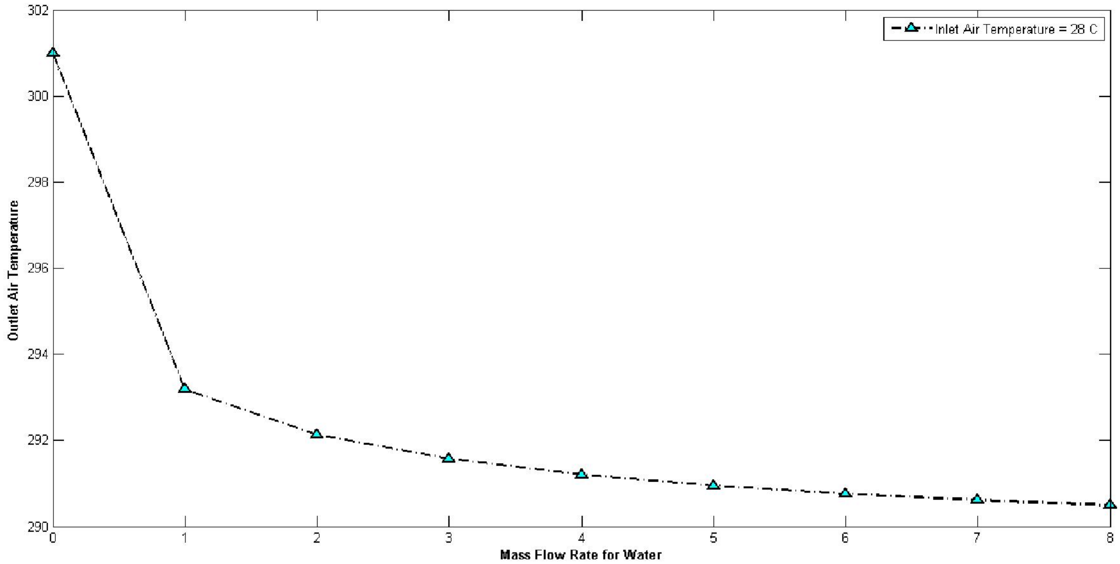

Fig. 5. Outlet air temperature vs mass flow when ambient air temperature is 28 C

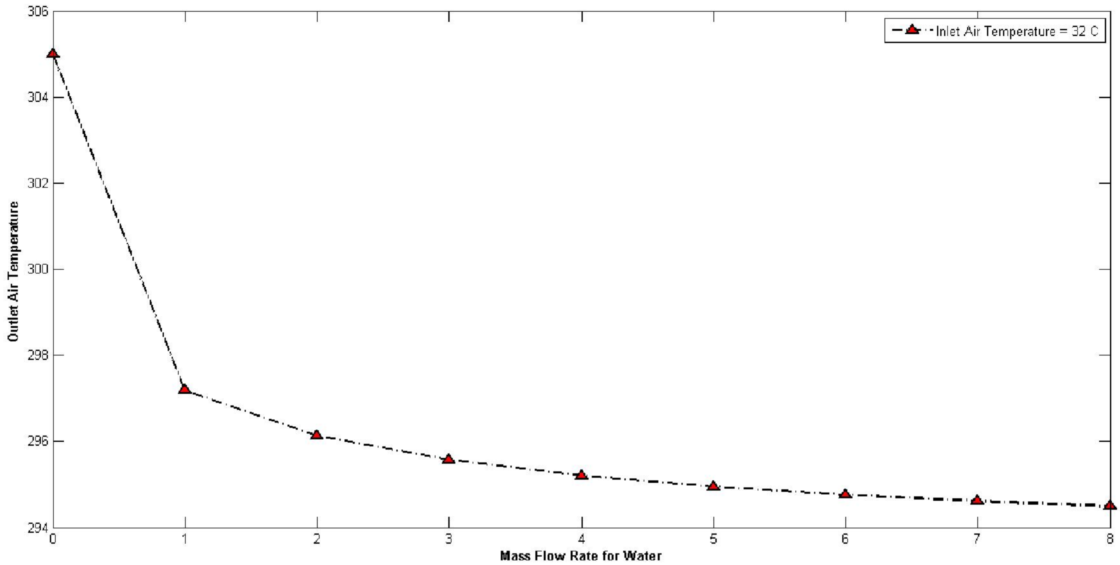

Fig. 6. Outlet air temperature vs mass flow when ambient air temperature is 32 C

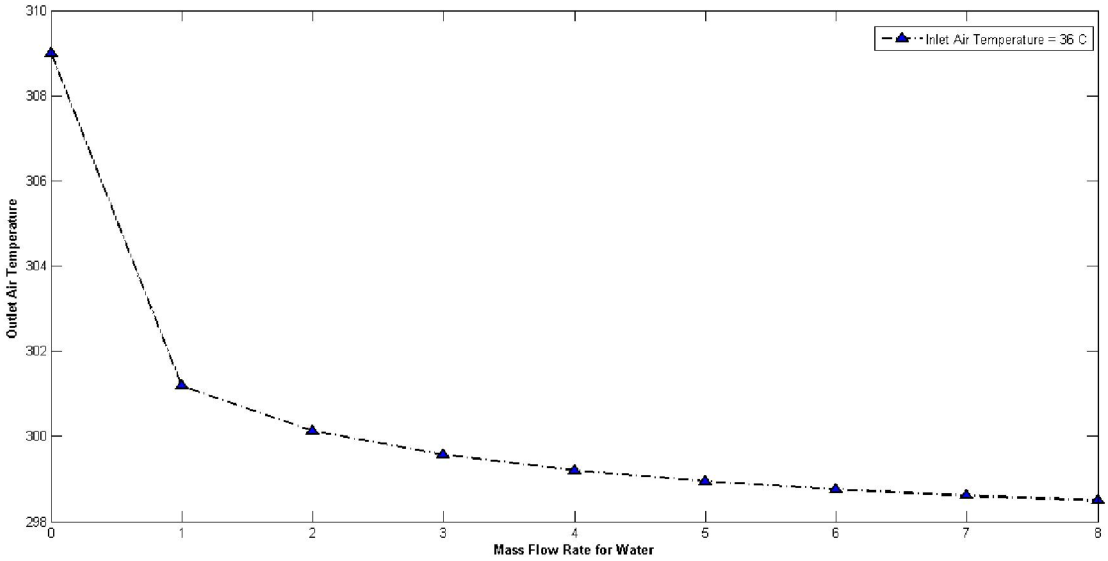

Fig. 7. Outlet air temperature vs mass flow when ambient air temperature is 36 C

This graph can be used in industries. For example, if the ambient air temperature be 24 C we can use Fiq.(4). Vertical axe denotes the outlet air temperature. In other words, to reach required outlet air temperature we must select water mass flow rate from horizontal axes. The interesting point is that there is a key mass flow rate in each diagrams, which means after this mass flow rate the outlet air temperature would not change with mass flow rate, and here, the outlet air temperature is not a function of mass flow rate and is strongly dependent to fin geometries. So, to have an optimal heat exchanger in order to reach maximum gas turbine cycle efficiency we must focus on both mass flow rate effects and fin geometries.

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