Lowering the Intake Air Temperature of the Compressor to Improve the Efficiency of the Gas Turbine Unit at the Tashkent CHP Plant

Ulugbek Noraliyev, Kalibek Shamsiyev, Hasan Umarvayev and Ikrom Umarov
Tashkent State Technical University named after Islam Karimov, Universitet, Str,2, 100095, Tashkent,Uzbekistan
noraliyevulugbek@gmail.com, shamsievkalibek_1954@mail.ru, umarvayevkhasan@gmail.com, umarovikrom5814@gmail.com

Keywords: Outdoor Air Temperature, GTU, Compressor, Output Power, Efficiency.

Abstract: This study investigates the impact of ambient air parameters on the performance of a gas turbine unit (GTU) and evaluates methods for improving its efficiency through compressor inlet air cooling. Particular attention is given to the climatic conditions of Uzbekistan, where ambient temperatures vary widely from –25 °C to +45 °C, significantly affecting GTU operation. The analysis demonstrates that increasing ambient air temperature reduces air density, leading to a decrease in mass flow rate through the compressor and, consequently, a reduction in power output and overall efficiency. Based on ISO standard conditions (15 °C, 60% relative humidity), it is shown that a temperature increase of 10 °C results in an approximate 8% decrease in GTU power. Mathematical relationships describing the dependence of power and efficiency on inlet air temperature are derived and applied to the GTU operating at the Tashkent CHP plant. Three air cooling methods - absorption chillers, natural evaporative cooling, and forced evaporative cooling - are comparatively analyzed. The results indicate that, considering both technical and economic factors, forced evaporative cooling is the most practical and cost-effective solution. The proposed approach enables mitigation of performance losses and enhances the operational efficiency of GTUs under high-temperature conditions.

1 INTRODUCTION

According to ISO (International Organization for Standardization), for gas turbine units (GTUs), the standard outdoor air temperature is set at 15°C with a relative humidity of 60% and the GTU located at sea level. The rated power is determined under these standard conditions, leading to the following conclusions [1]:

- GTU performance increases as temperature decreases: For every 10°C drop in temperature below 15°C, the output power increases by 8% of the rated power. Conversely, for every 10°C rise in temperature above 15°C, the output power decreases by 8% of the rated power.
- The location of the GTU affects its power output: For every 300 meters above sea level, the output power decreases by 3.5%.
- Pressure losses in the system reduce power output: Exhaust gas pressure drops in filters, silencers, and outlet ducts amounting to 1 kPa cause a 2% decrease in output power.
- When the air temperature falls below 15°C, the GTU's power output correspondingly exceeds the rated value. However, the temperature of the air supplied to the compressor must not fall below 6°C. Below this temperature, water droplets in the air may turn into ice, which can enter the airflow path and cause damage to the equipment.

2 MATERIALS AND METHODS

For the model of the GTU at the Tashkent CHP Plant - 310TB95-977. The GTU was manufactured by HITACHI and has the following nominal operating parameters [2] (Table 1).

Table 1: Technical characteristics of the GTU at the Tashkent CHP Plant under nominal operating conditions.

Parameter	Value
Outdoor air temperature, °C	15
Ambient pressure, kPa	101.3
Operating load, %	100
Generator power output, MW	28.63
Thermal output for consumers as hot steam, MW	45.09
Total plant output, MW	73.72
Gas consumption, t/h	49.79
Gross efficiency, %	74.13

3 RESULTS AND DISCUSSIONS

A change in the outdoor air temperature T_1 at a constant gas temperature in front of the turbine T_3 leads to an increase in the power and rotational speed of the gas turbine unit (GTU), as well as an increase in the unit's output as T_1 decreases. At the same time, the efficiency of the unit also increases. Conversely, an increase in T_1 causes a decrease in the power and rotational speed of the GTU. Achieving the rated power of the GTU is only possible by increasing the gas temperature in front of the turbine above the design level.

The high sensitivity of gas turbine units to changes in outdoor air temperature at the inlet of the axial compressor is due to two main factors. First, many modern GTUs have fixed flow areas of the gas turbine and axial compressor, which eliminates the possibility of regulating air flow while keeping the working fluid parameters constant. Second, modern units are characterized by a significant ratio of compression to expansion work, equal to $\lambda = \frac{h_K}{h_T} = 0.6 - 0.7$ in the nominal operating mode and $\lambda \approx 0.80$ under partial loads [3].

The influence of changes in the cycle's extreme temperatures on the characteristics of the gas turbine unit can be analyzed by studying the following relationships.

As known, the indicated power of a gas turbine unit (GTU) is determined by the following relationship:

$$N_i = N_{ik} - N_{iT}, \tag{1}$$

$$N_i = G_t C_{pm} T_1 \left(1 - \frac{T_3}{T_1}\right) - G_k C_{pm} T_1 \left(\frac{T_2}{T_1} - 1\right), \tag{2}$$

$$N_i = N_{ik} \cdot \frac{\lambda - 1}{\lambda}, \tag{3}$$

$$N_i = f(\theta, n, \pi_k) = f_1(\theta, n), \tag{4}$$

$$\theta = \frac{T_2}{T_1}, \qquad \pi_k = \frac{p_2}{p_1}, \tag{5}$$

where θ is the cycle boundary temperature ratio; n is the rotational speed of the GTU shaft; π_k is the compression pressure ratio across the axial compressor.

The above relationships show that the power of the GTU largely depends on the cycle boundary temperature ratio. The increment in GTU power for small deviations of the cycle boundary temperature ratio (Δθ) from the nominal value (θ_0) at a constant rotational speed (n = idem) is equal to:

$$\Delta N_i = \frac{\partial N_i}{\partial \theta} \cdot \Delta \theta. \tag{6}$$

At the same time, the change in the cycle boundary temperature ratio (Δθ) will vary depending on changes in each of the cycle boundary temperatures (T_1 and T_3):

$$T_1 = idem; \qquad \frac{\partial \theta}{\partial T_1} = \frac{1}{T_1}, \tag{7}$$

$$T_2 = idem; \qquad \frac{\partial \theta}{\partial T_1} = -\frac{\theta}{T_1}. \tag{8}$$

Therefore:

$$\frac{\partial \theta}{\partial T_1} = -\theta \, \frac{\partial \theta}{\partial T_3}, \tag{9}$$

$$\frac{\partial N_i}{\partial T_1} = -\theta \, \frac{\partial N_i}{\partial T_3}. \tag{10}$$

It follows from relation (10) that for any values of □k, |i,k, |i,T even a very small change in outdoor air temperature (T_1) can cause a change in GTU power several times greater than a change in gas temperature before the turbine (T_3).

To derive a calculated relationship for determining the change in unit power solely due to changes in outdoor air temperature, we adopt a number of simplifying assumptions: constant values of the relative efficiencies of the compressor and turbine, constant pressure ratios, constant gas temperature before the turbine, and constant working fluid flow rate through the unit. Only the outdoor air temperature changes. From (1):

$$\lambda = \frac{N_{ik}}{N_{iT}} \approx \frac{T_1}{T_3}, \tag{11}$$

$$\lambda_0 = \left(\frac{N_{ik}}{N_{iT}}\right)_0 \approx \frac{T_0}{T_3}, \tag{12}$$

$$\lambda = \lambda_0 \left(\frac{T_1}{T_0}\right), \tag{13}$$

where T_1 is the current value of the initial absolute outdoor air temperature; T_0 is the initial (design) value of the air temperature under nominal operating conditions (T_0 = 288.2 K); λ_0 is the power ratio of the compressor and gas turbine under the design operating mode of the gas turbine unit.

The specific power of the gas turbine as a whole can be considered independent of the outdoor air temperature at the compressor inlet. However, the compressor's output is directly dependent on the ambient air temperature, provided that the volumetric capacity of the compressor and the outdoor air pressure remain constant.

$$\frac{N_{e,T}}{N_{e,T,0}} \approx \frac{G_k}{G_{k.0}} = \frac{T_1}{T_0} \approx 1 - \frac{\Delta t_1}{T_0}. \tag{14}$$

From the relationship, (12) and (13) [4],

$$N_e = N_{e,T} - N_{e,k} = N_{e,T}(1 - \lambda), \tag{15}$$

$$N_e = N_{e,T,0}\left(1 - \frac{\Delta t_1}{T_0}\right)\left(1 - \lambda_0\left(\frac{T_1}{T_0}\right)\right). \tag{16}$$

In particular, for the nominal operating mode,

$$N_{e.0} = N_{e,T,0}(1 - \lambda_0). \tag{17}$$

From the relationship, (16) and (17),

$$\frac{N_e}{N_{e.0}} = \frac{\left(\frac{T_1}{T_0}\right)\left(1 - \lambda_0\left(\frac{T_1}{T_0}\right)\right)}{1 - \lambda_0}. \tag{18}$$

From the (16) and (17) the power ratio of the axial compressor and gas turbine under nominal operating conditions is typically $\lambda_0 \approx 0.65$. For the nominal operating mode of the Tashkent CHP GTU, $T_0 \approx 288$ K, and $N_{e.0}$ = 27 MW. Thus, we obtain the dependence of the Tashkent CHP GTU power on outdoor air temperature:

$$N_e(T_1) = 986{,}21 \cdot T_1 \cdot (288 - 0{,}65 \cdot T_1) \tag{19}$$

The reduced efficiency of the unit is related to the change in the relative power of the GTU by the ratio:

$$\frac{\eta_e}{\eta_{e.0}} = \frac{\frac{N_e}{N_{e.0}}}{1 - 0.75\left(1 - \frac{N_e}{N_{e.0}}\right)}. \tag{20}$$

For the nominal operating mode of the Tashkent CHP GTU, $T_0 \approx 288$ K, $\eta_{e.0} = 0.46$. From relations (19) and (20), we obtain the dependence of the Tashkent CHP GTU efficiency on the outdoor air temperature:

$$\eta_e = \frac{1481{,}6T_1(288 - 0{,}65T_1) \cdot 10^{-4}}{2863 + 2{,}96T_1(288 - 0{,}65T_1)}. \tag{21}$$

Using Eqs. (19) and (21), we obtain the dependence of the efficiency and power of the Tashkent CHP GTU on the outdoor air temperature, as shown in Figure 1.

During summer, when the outdoor air temperature reaches +45°C, the power output of the Tashkent CHP GTU decreases by 11% compared to the nominal operating mode (equivalent to 3 MW), while its efficiency decreases by 3%.

We propose three methods for cooling the air at the intake of the GTU compressor:

- Absorption chillers (ABHM);
- Natural evaporative cooling systems (NECS);
- Forced evaporative cooling systems (FECS).

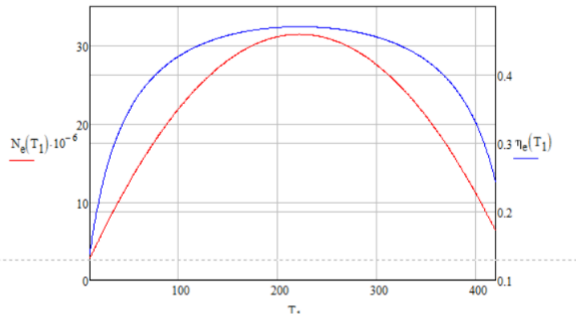

Figure 1: Dependence of the Tashkent CHP GTU power and efficiency on outdoor air temperature.

3.1 Absorption Chillers (ABHM)

One option for cooling the outdoor air supplied to the GTU compressor is the use of absorption chillers (ABHM). The operating principle of ABHM is based on the ability of a lithium bromide solution to absorb water vapor. The cooled medium enters the evaporator, where it is cooled to the required temperature by the evaporation of the refrigerant (water) under vacuum. Water vapor from the evaporator enters the absorber, where it is absorbed by the aqueous lithium bromide solution. The diluted (weak) solution is fed into the generator, where it is evaporated using an external heat source (hot water, steam, exhaust gases, or other types of secondary energy resources). The greater the required temperature reduction of the air before the compressor, the more heat must be supplied to evaporate water in the generator to obtain a larger amount of concentrated solution (e.g., LiBr). The heat source for the ABHM generator can be turbine or boiler steam, as well as hot water at temperatures of 150–180°C. [7] The saturated solution from the ABHM generator returns to the absorber. The water vapor generated in the condenser is condensed in the ABHM condenser. In some cases, the main condensate can be used as the cooling medium for the ABHM condenser before supplying it to the GPK KU. The evaporator operates under vacuum, causing water to boil at low temperatures. The deeper the vacuum, the lower the vaporization temperature. Water vapor enters the absorber. In the absorber, the concentrated lithium bromide (LiBr) solution, being a strong absorbent of water, absorbs the vapor, turning into a diluted solution, which is pumped back to the generator [8]. The operating principle of ABHM is shown in Figure 2.

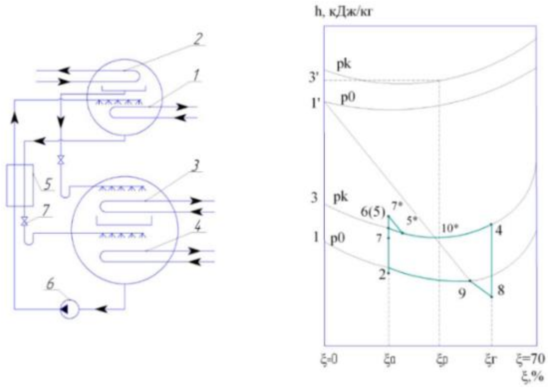

Figure 2: Schematic and cycle of a single-stage lithium bromide absorption chiller (ABHM): 1 – Generator;2 – Condenser; 3 – Evaporator; 4 – Absorber; 5 –Regenerative heat exchanger; 6 – Circulation pump; 7 – Throttle valve.

ABHMs require extended installation times and highly qualified personnel for both maintenance and repairs. Additionally, there is a risk of leaks of refrigerant harmful to human health [9]. The designed absorption chiller unit for the Tashkent CHP plant is shown in Figure 3.

A gas turbine power plant operates with a constant flow of outdoor air. When the temperature of the air entering the GTPP (Gas Turbine Power Plant) increases [12], its density decreases, reducing the mass flow rate of air in the compressor, which leads to a drop in the power output of the GTPP. By installing a cooling heat exchanger integrated into an absorption chiller-based cooling system (ABHM), the temperature of the air supplied to the GTPP's heat engine was reduced in some cases from +40°C to +15°C, resulting in an increase of electric power generation by approximately 30%. The process flow diagram of ABHM connection to the GTU is shown in Figure 4.

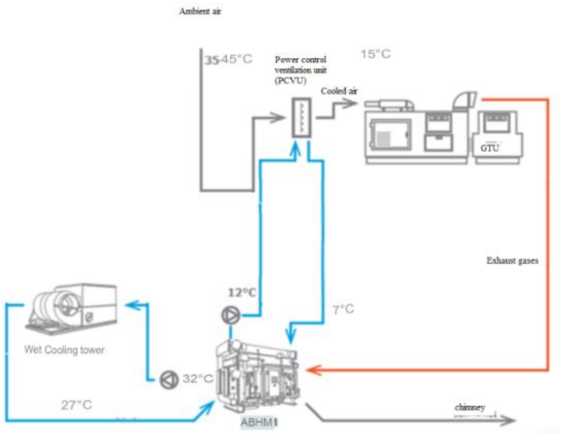

Figure 3: Schematic diagram of the designed absorption chiller unit (ABHM) at the Tashkent CHP plant.

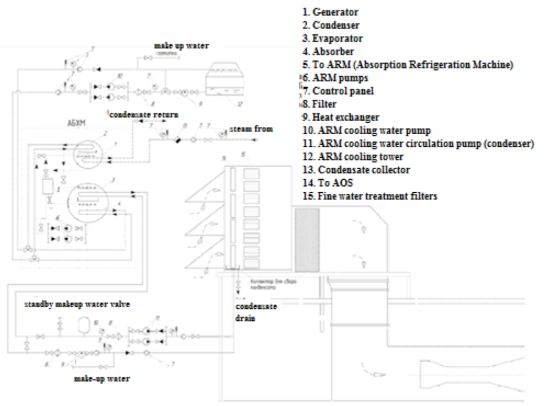

Figure 4: Process flow diagram of ABHM connection to the GTU [5].

Advantages [10]:

1) Highest cooling capacity (down to +12.5°C).
2) Ability to bypass the dew point of outdoor air during the cooling process.
3) Theoretical possibility of utilizing condensate generated during air cooling.
4) Extremely low makeup water consumption for the internal cooling circuit.

Disadvantages:

1) Highest technical complexity of system implementation.
2) Highest cost of the technical solution. The specific cost of ABHM at current prices ranges from 120–180 USD/kW, excluding the cost of the cooling tower and auxiliary equipment.
3) Significant space required for new equipment.
4) Necessity to create an independent water circulation circuit.
5) Significant expansion of the auxiliary equipment area required.
6) Annual maintenance by certified specialists and winterization are necessary.

3.2 Forced Evaporative Cooling Systems (FECS)

The operating principle of such systems is based on the physical phenomenon of air cooling through the evaporation of water droplets [11]. This method is most similar to the principle of natural evaporative cooling systems (NECS), with the key difference being that instead of passive water flow forming a thin film through which incoming air passes, nozzles are used to spray water and create fine droplets that subsequently evaporate. Forced evaporative cooling uses a small amount of water injected at high pressure. Injection is provided at specific points in the air intake through specialized nozzles to create a cooling effect. Air cooling, in turn, reduces the energy consumed by the compressor drive.

Water droplets are generated by spraying water through nozzles. The efficiency of this system directly depends on droplet size. Minimal droplet size ensures an extensive heat and mass transfer surface between the water sprayed into droplets and the cooled air. During heat and mass transfer, water droplets evaporate, extracting thermal energy from the air. As evaporation occurs, air temperature decreases, while relative humidity and moisture content increase. It is important to note that the design goal of such a system is to supply an amount of water that ensures complete evaporation of all droplets. Any water droplets entering the compressor intake are unacceptable. In other words, the amount of water supplied for cooling is limited by the relative humidity of the cooled air. When relative humidity reaches 100%, the droplet evaporation process stops. Thus, the maximum achievable cooling is determined by the relative humidity and temperature of the air supplied to the FECS [6]. The nozzle layout of the FECS is shown in Figure 5. The process flow diagram of FECS connection to the GTU is shown in Figure 6.

Advantages:[13]

1) Relative simplicity of technical implementation.
2) Lowest equipment installation costs.
3) Minimal aerodynamic resistance.
4) Lowest weight load on the PCVU structure.
5) No complex maintenance required.

Disadvantages:

1) Cooling capacity is limited by the wet-bulb temperature.
2) Irreversible water carryover into the compressor.
3) Higher requirements for the droplet capture system.
4) Requires high-quality chemically purified water.
5) Short nozzle lifespan (approximately 2 years with regular use).

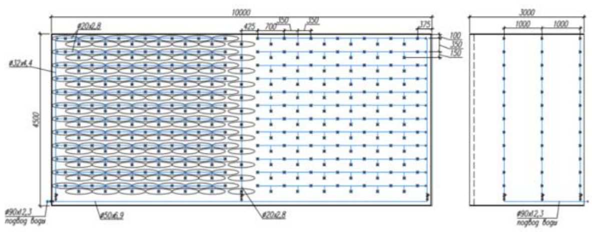

Figure 5: Nozzle layout diagram of the FECS.

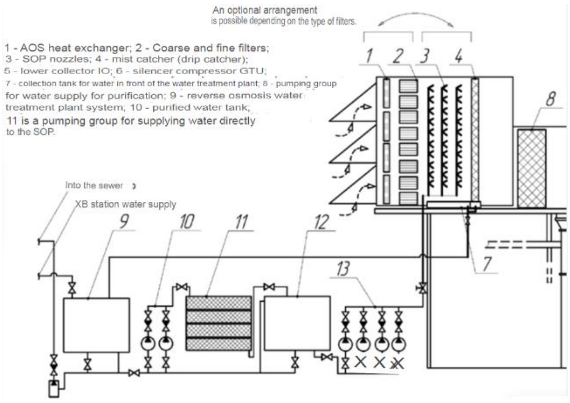

Figure 6: Process flow diagram of FECS connection to the GTU.

3.3 Natural Evaporative Cooling Systems

The essence of evaporative cooling is as follows. According to kinetic theory, water molecules are in random thermal motion, with their velocities varying widely [14]. Those molecules with the highest velocity (greatest kinetic energy) escape into the space above the water surface (evaporation). Only molecules near the water surface, whose velocity component normal to this surface is sufficiently high to overcome molecular attractive forces, can break away from the water. Water molecules that escape from the surface may collide with air molecules and be pushed back, resulting in condensation.

The practical implementation of this principle in gas turbine systems is realized through natural evaporative cooling systems (NECS), as shown in Figure 7, where the PCVU is combined with evaporative elements to reduce the inlet air temperature.

Furthermore, the process flow and integration of NECS into the combined cycle power plant (CCPP) configuration are presented in Figure 8.

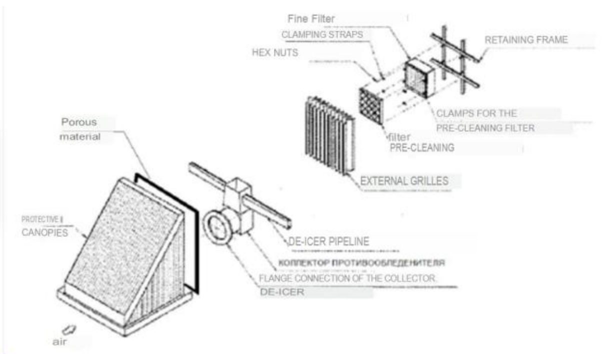

Figure 7: Diagram of the PCVU Combined with the NECS.

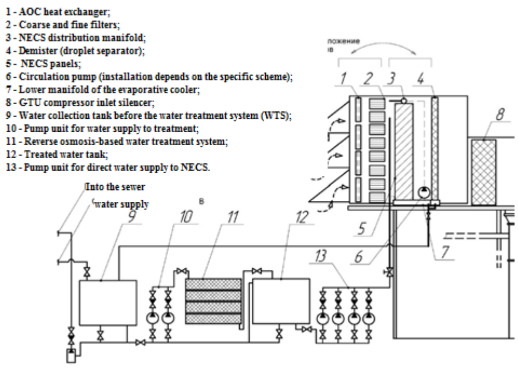

Figure 8: Process flow diagram of NECS connection to the CCPP systems.

Advantages:
1) Simplicity of technical implementation.
2) No complex maintenance required.
3) Water treatment requirements are less stringent compared to FECS.
4) Relatively easy dismantling of evaporative panels for winter (without GTU shutdown).
5) Auxiliary NECS equipment occupies minimal space. The installation does not create significant aerodynamic resistance in the PCVU pathway.

Disadvantages:
1) Cooling capacity is limited by the wet-bulb temperature.
2) Irreversible water carryover into the compressor.
3) Relatively high project implementation cost due to limited competition among manufacturers in this niche.
4) Higher requirements for the droplet capture system.
5) Scarcity of companies specializing in such projects.

4 CONCLUSIONS

This study examined the influence of ambient air temperature on the performance of a gas turbine unit (GTU) and evaluated practical methods for improving its efficiency under high-temperature conditions. The analysis confirmed that an increase in inlet air temperature leads to a decrease in air density, which reduces the mass flow rate through the compressor and results in a decline in both power output and overall efficiency of the GTU.

Based on the developed mathematical relationships, it was shown that the GTU operating at the Tashkent CHP plant experiences a noticeable performance drop at elevated temperatures, with power losses reaching up to 11% during peak summer conditions. This highlights the critical importance of inlet air cooling for maintaining stable and efficient operation.

A comparative assessment of three cooling methods - absorption chillers, natural evaporative cooling systems, and forced evaporative cooling systems - was conducted. The results indicate that while absorption chillers provide the highest cooling capacity, their implementation is associated with significant capital and operational costs. Natural evaporative systems offer a simpler solution but are limited in efficiency.

Among the considered options, forced evaporative cooling systems demonstrate the most favorable balance between technical effectiveness and economic feasibility. Their relatively simple design, low installation cost, and sufficient cooling performance make them the most practical solution for improving GTU efficiency under the climatic conditions of Uzbekistan.

REFERENCES

[1] M. Al-Ibrahim and A. Varnham, “A review of inlet air-cooling technologies for enhancing the performance of combustion turbines,” Appl. Therm. Eng., vol. 30, no. 14–15, pp. 1879–1888, 2010, doi: 10.1016/j.applthermaleng.2010.04.025.

[2] O. Arai et al., “Characteristics and applications of Hitachi H-25 gas turbine,” Hitachi Rev., 2008. [Online]. Available: https://www.hitachi.com/ICSFiles/afieldfile/2008/10/28/r2008_06_006.pdf.

[3] P. Porshakov, A. A. Apostolov, and V. I. Nikishin, Gas Turbine Units in Gas Pipelines. Moscow, Russia: Neft i Gaz Publishing, Gubkin Russian State University of Oil and Gas, 2003, pp. 54–62.

[4] M. Boyko, B. V. Budzulyak, and B. P. Porshakov, “Status and development prospects of the country’s gas transportation system,” Univ. News Oil Gas, no. 7, pp. 64–74, 1997.

[5] L. S. Timofeevskiy, Ed., Refrigeration Machines. Politekhnika Publishing, 1997, pp. 66–74.

[6] S. M. Andoniev, Evaporative Cooling of Metallurgical Furnaces. Metallurgiya Publishing, 1970, pp. 53–64.

[7] M. P. Boyce, Gas Turbine Engineering Handbook, 4th ed. Oxford, UK: Butterworth-Heinemann, 2011, pp. 53–64.

[8] S. Pourhedayat et al., “A comparative and critical review on gas turbine intake air cooling technologies,” Therm. Sci. Eng. Prog., 2023, doi: 10.1016/j.tsep.2023.101828.

[9] H. M. Kwon et al., “Gas turbine performance enhancement by inlet air absorption chilling,” in Proc. ASME Turbo Expo, 2016, doi: 10.1115/GT2016-56539.

[10] Y. S. H. Najjar, “Enhancement of performance of gas turbine engines by inlet air precooling (absorption chillers),” Appl. Therm. Eng., 1996, doi: 10.1016/1359-4311(95)00047-H.

[11] R. C. da Costa et al., “A technical-economic analysis of turbine inlet air cooling,” Res. Soc. Dev., vol. 10, no. 13, 2021, doi: 10.33448/rsd-v10i13.21763.

[12] N. Koosha et al., “Energy, exergy, economic, and environmental (4E) analysis of gas turbine fogging across climate zones,” Appl. Therm. Eng., 2025, doi: 10.1016/j.applthermaleng.2025.126828.

[13] K. Mohapatra et al., “Analytical investigation of parameters affecting evaporative inlet cooling of gas turbines,” Arab. J. Sci. Eng., 2013, doi: 10.1007/s13369-013-0527-z.

[14] Z. T. Liu et al., “The effects of inlet air heating on gas turbine efficiency under partial load,” Energies, vol. 12, no. 17, p. 3327, 2019, doi: 10.3390/en12173327.