

Multiple-Objective Optimization of Gas Turbine Inlet Cooling Systems for Simultaneous Power Augmentation and Blade Life Preservation

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Abstract:

Gas turbine inlet air temperature varies significantly in hot and cold conditions, making performance sensitivity to ambient conditions critical, while ensuring that blade life is not compromised. Therefore, understanding how compressor inlet temperature, specific fuel consumption, and blade surface temperature affect gas turbine performance and blade life while achieving power augmentation and maintaining blade durability. A numerical investigation was conducted using Engineering Equation Solver (EES), actual GT13E2-2005 operating data and standard thermodynamic properties to evaluate gas turbine performance and blade life. The simulation results were tabulated, and Microsoft Excel was used to plot performance and blade-life trends to identify ways to improve power output while extending blade life. The key findings show that the application of inlet cooling helps in mitigating blade degradation resulting from increased turbine inlet temperature, high pressure ratio and high specific fuel consumption, while improving power output, thermal efficiency and blade life. Results indicate that inlet cooling increases air density, which significantly improves blade life, plant efficiency and power output, and slightly reduces specific fuel consumption. The results justified the focal aspect of this research which was to prolong blade life and improve the power output without necessarily increasing fuel consumption (minimizing fuel usage) while using inlet cooling.

Keywords — Simple gas turbine; inlet air cooling; blade life; compressor inlet temperature; power output, air density, blade surface temperature; specific fuel consumption.

I. INTRODUCTION

Gas turbines are commonly employed in industrial power generation due to their high-power density, rapid start-up capability, operational flexibility, and suitability for peak-load and distributed energy applications. However, their performance is strongly affected by compressor inlet air temperature, especially in hot climates where ambient conditions often exceed ISO reference values. As compressor inlet temperature increases, the inlet-air density decreases, reducing the compressor air mass flow rate and consequently lowering net power output, increasing specific compressor work, and worsening specific fuel consumption. This temperature sensitivity has been widely confirmed in previous

studies. Gas turbine output is governed primarily by the air mass flow rate admitted into the compressor, and higher compressor inlet temperature has been shown to reduce air density, net electrical output and cycle thermal efficiency while increasing heat rate and fuel consumption per unit power [1–3]. Kolp et al. [4] further demonstrated that reducing compressor inlet temperature can reduce heat rate and increase power output, confirming inlet air cooling as an effective strategy for recovering performance losses during hot-weather operation.

To mitigate temperature-induced performance degradation, several inlet cooling methods have been investigated, such as evaporative cooling, fogging, mechanical vapour-compression chilling, absorption chilling, and hybrid systems

[5]. Evaporative cooling is simple and cost-effective but is strongly dependent on ambient relative humidity, whereas mechanical chillers can provide deeper cooling at the expense of substantial auxiliary electrical power. Absorption chillers are especially attractive in applications with available waste heat because they can be driven by gas turbine exhaust gases, thereby reducing auxiliary electrical demand and improving suitability for power plants with recoverable exhaust heat [6–9]. Among these options, absorption chilling has received considerable attention because the system can be integrated with gas turbine exhaust heat. Al-Bortmany [6] described absorption chilling as an environmentally favourable inlet-cooling option because the thermal energy required by the generator can be recovered from flue gases that would otherwise be discharged to the atmosphere. This approach also avoids chlorofluorocarbon-based refrigeration. Nevertheless, its effectiveness is limited by the available generator heat input and refrigerant mass flow rate [10]. Despite these limitations, absorption chilling can cool compressor inlet air to a specified setpoint largely independent of ambient conditions, thereby improving gas turbine power output [11,12].

The performance benefits of absorption-based inlet cooling have been demonstrated under different climatic conditions. Nasser and El-Kalay [7] showed that exhaust-gas heat could reduce a 40°C ambient inlet temperature by about 10°C, producing an approximate 10% increase in power output. Bies et al. [11] investigated a double-effect lithium-bromide absorption chiller for compressor inlet air cooling, while Mohanty and Paloso [8] reported instantaneous power-output increases of 8–13%, with an average increase of about 11%, for a 100 MW gas turbine operating in Bangkok. For lithium-bromide-based absorption chillers, the minimum chilled-water temperature is typically about 6°C; therefore, the air-coil approach temperature often limits the practical compressor inlet air temperature to approximately 12°C [13]. Popli et al. [9] further argued that absorption chilling may be more suitable than mechanical chilling or evaporative cooling in some applications because mechanical chillers require significant electrical power, whereas evaporative cooling is highly dependent on humidity. The operating principle of absorption chilling supports this advantage. Similar to vapour-compression systems, inlet air is cooled by a refrigerant through cooling coils; however, the absorption system replaces the mechanical compressor with a circulating pump, substantially reducing electrical power consumption [9]. This makes absorption cooling especially relevant for gas turbine plants, where waste heat can be recovered for useful cooling rather than discharged to the environment. Beyond the choice of cooling technology, local climate strongly governs the magnitude of gas turbine performance loss. Mahmoudi et al. [2] observed that gas turbine performance is climate-sensitive because combustion air is drawn directly from ambient

surroundings. As ambient temperature and humidity increase, power generation and cycle efficiency decrease, with compressor inlet air temperature exerting the dominant influence on gas turbine performance [14]. A rise from ISO reference conditions of 15°C to hot-climate peak conditions of about 40°C can reduce power output by approximately 20%, while inlet cooling can recover part of this loss [12]. Cooling compressor inlet air to 4°C in hot climates has also been reported to increase specific power by up to 27% [12,15].

The severity of these losses has been reported across several operating regions and turbine fleets. Ameri and Hejazi [16] indicated that Iran operated over 170 gas turbine units with a total installed capacity of 9500 MW, from which approximately 1900 MW was unavailable during the hot summer period. A 1°C rise in inlet air temperature can reduce gas turbine power production [17,18], while ambient temperature can reduce rated output by 10–25%, depending on turbine type [19]. These findings are consistent with thermodynamic principles showing that gas turbine net power output varies directly with the air mass flow rate and inversely with ambient temperature. Thermal efficiency is also strongly influenced by compressor inlet temperature because lower inlet temperature increases air density and reduces compressor work, thereby improving power output and overall cycle efficiency. Rahman et al. [3] reported that higher compressor inlet temperature increases exhaust temperature and heat rate while reducing exhaust flow rate and net power output. This confirms that ambient temperature is directly proportional to exhaust temperature and heat rate but inversely proportional to exhaust flow rate and power output. Although lowering inlet cooling can improve performance, it introduces a design-sensitive constraint associated with the minimum safe compressor inlet temperature. Dawoud et al. [13] reported that condensate icing may occur when inlet temperature approaches freezing conditions. To avoid this risk, previous studies recommend maintaining a safe margin above freezing, with practical lower limits commonly ranging from about 4°C to 12°C depending on the cooling technology and chiller capacity [13,19–22]. In addition to power augmentation, inlet cooling has significance for turbine blade durability. Gas turbine blades undergo gradual deformation under sustained high temperature and stress and creep is widely reported as a major failure mechanism in turbine blade alloys [23–26]. Establishing the relationship between inlet cooling and creep life is therefore important for improving power output while preserving blade durability.

Despite the extensive literature on inlet cooling, most studies emphasize power output, heat rate, or thermal efficiency, whereas fewer studies integrate performance enhancement with turbine blade-life assessment. This creates a research gap because blade life is governed by blade metal temperature,

thermal stress, creep behaviour, and exposure time. Therefore, inlet cooling should be evaluated not only as a power-augmentation method but also as a durability-sensitive operating strategy. Therefore, this study aims to optimize inlet cooling conditions for simultaneous gas turbine power augmentation and blade-life preservation. The study links compressor inlet temperature, air density, power output, specific fuel consumption, blade surface temperature, and predicted creep life to determine how inlet cooling can enhance performance without imposing additional thermal damage on turbine blades.

II. MODELLING OF GAS TURBINE

A numerical modelling framework was developed to evaluate the integrated influence of inlet air cooling on gas turbine performance and turbine-blade creep life. The model was developed in Engineering Equation Solver (EES) because of its ability to solve coupled thermodynamic equations and provide built-in thermodynamic and heat transfer property functions. Actual GT13E2-2005 operating data, together with standard thermodynamic properties, were used as model

inputs to represent the gas turbine and inlet cooling system under representative operating conditions. The GT13E2-2005 turbine blade material was treated as a nickel-base superalloy because of its elevated-temperature strength, resistance to creep, and low-cycle fatigue capability. The integrated model was structured into sub-models for the inlet cooling system, compressor, combustor, turbine, and blade creep-life assessment. A vapour absorption inlet cooling system was considered, and a parametric analysis was performed by varying the compressor inlet temperature to determine its influence on inlet air density, compressor work, specific fuel consumption, thermal efficiency, net power output, blade surface temperature, and predicted blade life. Blade creep life was estimated using the Larson–Miller parameter, in which blade metal temperature is a principal input for predicting time to creep rupture [27]. The creep-life estimation was based on blade metal temperature, the material Larson–Miller master curve, and the assumed material constant, C. The modelling assumptions were limited by the availability and reliability of blade-material creep

data, the use of a constant C value, and the requirement to avoid excessive turbine temperature and stress, since sustained high temperature stress can promote crack initiation and propagation. The complete modelling sequence for the gas turbine, combined-cycle arrangement and inlet cooling system is summarized in Fig 1.

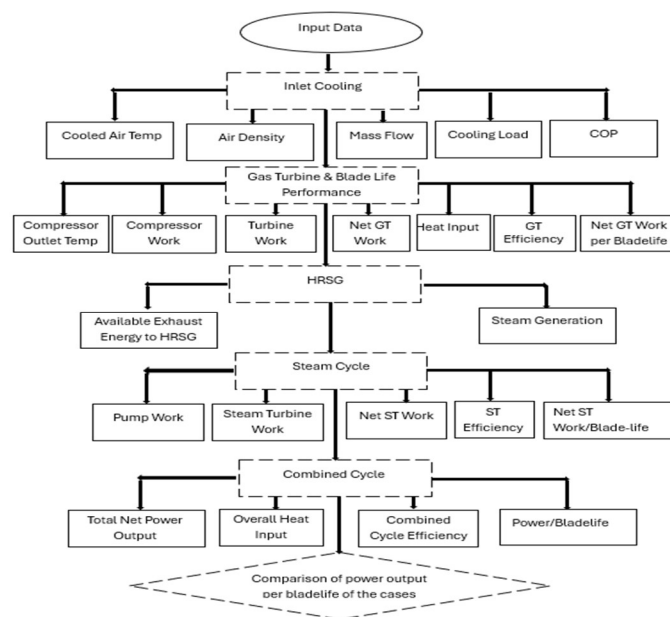


Fig 1 Flow chart of integrated gas turbine, combined cycle and inlet cooling.

2.1. Inlet Air Cooling Model

Real gas specific heat varies with both temperature and pressure. The absorption inlet air-cooling system reduces the ambient air temperature to a fixed setpoint, provided that the system has adequate capacity. Therefore, the temperature of air after chilling, i.e., the cooled temperature ($T_{inlet\ cool}$), is equal to the absorption chiller set temperature ($T_{Abs\ setpoint}$).

$T_{Abs\ setpoint}$ denotes the absorption chiller setpoint temperature, which commonly lies between 10 and 15°C; however, in this study, $T_{Abs\ setpoint}$ is taken as 12°C.

Equation (2.1) represents the thermal energy required to operate the vapour absorption (VA) cooling system at the selected coefficient of performance.

$$Q_{abs} = W_{input} \cdot COP = Q_{gen} \cdot COP = m_{air} C_p (T_{amb} - T_{Inlet\ Cool}) \quad (2.1)$$

the amount of cooled air entering the compressor is proportional to the density (higher density will lead to a higher mass flow rate) as shown in Equation 2.2.

$$\dot{m}_{air} = \rho \cdot (\text{Volumetric flow rate}) = \rho \cdot Q = \rho \cdot V A \quad (2.2)$$

where V denotes the fluid velocity and A denotes the flow cross-sectional area. Q_{abs} is the cooling load required by the vapour absorption (VA) system, corresponding to the heat removed from the chilled space.

$Q_{gen} = W_{input}$ is the generator heat input supplied to the absorption system. $(COP)_{VA}$ is the coefficient of performance of VA.

However, the cooled-air mass flow rate supplied to the

compressor is assumed to be equal to the ambient-air mass flow rate, provided that no leakage or bypass occurs within the inlet cooling system.

2.2. Gas Turbine Model

For the complete compressor, the steady-state energy equation is applied, assuming that inlet and exit properties such as pressure, temperature, and mass flow rate remain constant with time, as shown in Equation (2.3).

$$\sum \dot{m} \left(h_i + \frac{V_i^2}{2} + gZ_i \right) + \dot{Q}_c = \sum \dot{m} \left(h_e + \frac{V_e^2}{2} + gZ_e \right) + \dot{W}_c \quad (2.3)$$

where:

$\dot{m}$ denotes the mass flow rate, $\sum \dot{m} \left(h_i + \frac{V_i^2}{2} + gZ_i \right)$ represents the total energy entering the compressor, h_i is the inlet fluid enthalpy, V_i is the inlet velocity, corresponding to inlet kinetic energy $\frac{V_i^2}{2}$, while, gZ_i is the inlet potential energy, $\dot{Q}_c$ is the heat transfer associated with the compressor, and $\sum \dot{m} \left(h_e + \frac{V_e^2}{2} + gZ_e \right)$ represents the total energy leaving the compressor. The sub-script “e” represents exit.

$\dot{W}_c$ denotes the compressor work input, which is the work required to compress the working fluid and raise its pressure. This term is commonly negative because it represents work supplied to the system.

$$\dot{W}_c = \dot{m}_{air} C_{pa} T_2 - \dot{m}_{air} C_{pa} T_1 \quad (2.4)$$

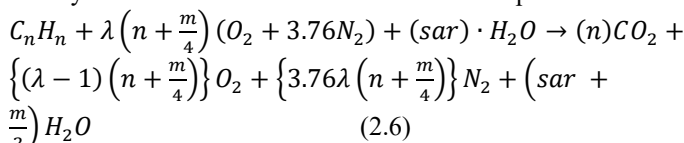
When gas turbine blade cooling is neglected, the compressor work ($\dot{W}_c$) [3] can be expressed as shown in Equation (2.5).

$$\dot{W}_c = \frac{C_{pa} \times T_1 \left[\frac{\gamma_a - 1}{\gamma_a} - 1 \right]}{\eta_m \times \eta_c} = \frac{C_{pa} \times T_1 \times R_{pa}}{\eta_m} \quad (2.5)$$

η_m represents the mechanical efficiency of the turbine-compressor assembly.

2.3. Combustor Model

The hydrocarbon fuel combustion reaction is represented as:



Applying the energy balance to the combustion chamber gives:

$$\dot{m}_a C_{pa} T_2 + \dot{m}_f \times LHV + \dot{m}_f C_{pf} T_f = (\dot{m}_a + \dot{m}_f) C_{pg} \times T_3 \quad (2.7)$$

where $\dot{m}_a$ denotes the air mass flow rate (kg/s), $\dot{m}_f$ denotes the fuel mass flow rate (kg/s), C_{pf} represents the fuel specific heat, T_f is the fuel temperature, T_3 = TIT denotes the turbine inlet temperature, and LHV represents the lower heating value. The flue-gas specific heat (C_{pg}) is defined according to Naradasu et al. [28] as:

$$C_{pg} = 1.8083 - 2.3127 \times 10^{-3} T_{g1} + 4.045 \times 10^{-6} T_{g1}^2 - 1.7363 \times 10^{-9} T_{g1}^3 \quad (2.8)$$

where $T_{g1} = T_4$ which is the inlet gases temperature of flue gas to the HRSG.

Rearranging Equation (2.7) gives the fuel-air ratio (f), as presented in Equation (2.9).

$$f = \frac{\dot{m}_f}{\dot{m}_a} = \frac{C_{pg} \times T_3 - C_{pa} T_2}{LHV - C_{pg} \times T_3} = \frac{C_{pg} \times T_3 - C_{pa} T_1 (1 + R_{pg})}{LHV + C_{pf} \times T_f - C_{pg} \times T_3} \quad (2.9)$$

however, f is dependent on fuel type, combustion temperature limits, turbine design and operating conditions. A rise in f will amount to a rise in turbine inlet temperature but may exceed material limits or lead to incomplete combustion.

2.4. Turbine Model

The turbine inlet temperature is determined using Equation (2.10).

$$T_3 = T_2 + \left(\frac{\eta_{cc} \dot{m}_f LHV}{\dot{m}_{air} C_{pg}} \right) \quad (2.10)$$

The gas turbine exhaust-gas temperature is calculated as follows:

$$T_4 = T_3 \left(1 - \eta_t \times \left[1 - \frac{1}{r_p^{\frac{\gamma_g - 1}{\gamma_g}}} \right] \right) = T_3 (1 - \eta_t \times R_{pg}) \quad (2.11)$$

$T_4 = T_{g1}$, where T_{g1} denotes the temperature of the gases entering the HRSG.

$$\eta_T = \frac{W_a}{W_i} = \frac{(h_3 - h_{4a})}{(h_3 - h_{4i})} = \frac{T_3 - T_4}{T_3 - T_{4i}} \quad (2.12) \text{ State}$$

4a represents the actual turbine exit condition, whereas state 4i represents the corresponding isentropic turbine exit condition.

The turbine shaft work (W_t) is given by Equation (2.13).

$$W_t = C_{pg} \times T_3 \times \eta_t \times R_{pg} / \eta_m \quad (2.13)$$

where “i” and “e” denote the inlet and exit conditions respectively, for either a turbine row or the complete turbine.

Therefore, the gas turbine net work, or specific work output (W_{Gnet}), is expressed as:

$$W_{Gnet} = W_t - W_c = C_{pg} \times T_3 \times \eta_t \left[1 - \frac{1}{r_p^{\frac{\gamma_g - 1}{\gamma_g}}} \right] - \frac{C_{pa} \times T_1 \left[\frac{\gamma_a - 1}{\gamma_a} - 1 \right]}{\eta_c} = T_1 \left(\frac{C_{pg} \times \eta_t \times \theta \times R_{pg} - C_{pa} R_{pa}}{\eta_m} \right) \quad (2.14)$$

The turbine output power (P) is calculated using Equation (2.15).

$$P = \dot{m}_a \times W_{Gnet} \quad (2.15)$$

The specific fuel consumption (SFC) is defined by Equation (2.16).

$$SFC = \frac{3600 f}{W_{Gnet}} \quad (2.16)$$

The heat supplied is also expressed as represented in Equation (2.17) (Ibrahim and Rahman, 2012).

$$Q_{add} = C_{pgm} (T_3 - T_1 [1 + R_{pa}]) = f \times LHV \quad (2.17)$$

The gas turbine thermal efficiency (η_{th}) is expressed in Equation (2.18).

$$\eta_{th} = \frac{W_{Gnet}}{Q_{add}} \quad (2.18)$$

the heat rate (HR) can be expressed as shown in Equation (2.19).

$$HR = \frac{3600}{\eta_{th}} \quad (2.19)$$

The recoverable heat contained in the gas turbine exhaust stream is expressed in Equation (2.20).

$$Q_{av} = \dot{m}_g C_{pg} (T_{g1} - T_{g4}) h_{1f} \quad (2.20)$$

2.5. Creep Life Model

Metal creep life is estimated using the Larson-Miller parameter (LMP), which is inversely related to total stress (i.e., LMP $\propto$

$\frac{1}{\sigma})$). The creep-life formulation is based on the power-law relationship linking temperature and stress. This power law combines the Arrhenius and Norton equations, as presented in Equation (2.21).

$$\dot{\epsilon} = A\sigma^n e^{(-Q/RT_b)} \quad (2.21)$$

but Monkman-Grant equation is defined as

$$\dot{\epsilon}^\alpha t_r = C \quad (2.22)$$

adding Equations (2.21) and (2.22) and taking logarithm of both sides gives Equation (2.23).

$$\frac{Q}{R} = T_b (\log[t_r] + C) \quad (2.23)$$

Therefore, when LMP is defined as a function of von Mises stress, blade creep life can be expressed as shown in Equation (2.24).

$$\tau_r = 10^{\frac{LMP}{T_b} - C} \quad (2.24)$$

where LMP denotes the Larson-Miller parameter, expressed as a function of the combined mechanical and thermal stresses in MPa. It is also commonly referred to as the material master curve.

T_b is the operating blade metal temperature in Kelvin, and t_r is the creep-rupture time in hours, representing blade creep life.

$\dot{\epsilon}$ is the minimum creep rate, and α is the applied stress, obtained from the slope of $\log t_r$ plotted against $\log \dot{\epsilon}$, with $\alpha = 1$.

n is the stress exponent, Q is the activation energy, R is the universal gas constant, and A is a material parameter.

C is the material constant. This can be gotten through GT blade super alloy specimens creep test and the value ranges from 17-23. However, 20 is usually used in most engineering applications and as such in this study.

III. RESULTS AND DISCUSSION

This section presents and interprets the simulation results obtained from the inlet-cooling and blade-life models. The discussion focuses on how changes in compressor inlet temperature influence air density, specific fuel consumption, compressor work, power output, blade surface temperature, and predicted blade life.

3.1. Inlet Cooling and Blade Life Model

This section summarizes how inlet cooling improves gas turbine performance and blade life by increasing inlet air density and mass flow rate, reducing compressor work and specific fuel consumption, and lowering blade surface temperature. These effects support higher power output, better fuel utilization, and longer predicted blade life, as discussed in Figures 2–7.

3.1.1. Inlet Cooling on Air Inlet Density:

Figure 2 shows that inlet cooling, represented by a reduction in compressor inlet temperature, raises the density of the inlet air and increases the air mass flow rate. The denser air increases the quantity of air supplied to the compressor, thereby enhancing gas turbine power output and cycle thermal efficiency. Consequently, inlet cooling supports higher net power generation with better fuel utilization.

3.1.2. Inlet Cooling on Specific Fuel Consumption:

Figure 3 shows that inlet cooling reduces specific fuel consumption. Cooler air is denser, allowing more air mass to be supplied to the compressor for the same inlet volume. Although the increased air mass may require additional compression and combustion energy, the corresponding improvement in turbine power output exceeds the relative increase in fuel input. Therefore, inlet cooling improves fuel utilization by lowering the fuel required per unit of generated power.

3.1.3. Gas Turbine Power Output on Inlet Cooling:

Figure 4 shows that inlet cooling markedly increases gas turbine power output by increasing air mass flow rate, reducing compressor work, and promoting more effective combustion. Since turbine work depends strongly on turbine inlet temperature and working-fluid mass flow rate, the increased supply of denser inlet air contributes to higher turbine output. As compressor inlet temperature decreases, thermal efficiency also increases, confirming the beneficial role of inlet cooling in power augmentation.

3.1.4. Inlet Cooling on Compressor Work Output:

Figure 5 shows that inlet cooling reduces specific compressor work. Cooler air has a lower specific volume and therefore requires less compression energy at the same pressure ratio. The decrease in compressor work allows a greater fraction of turbine work to be converted into useful net output. Thus, lowering compressor inlet temperature improves overall cycle performance by reducing compression demand and increasing net power output.

3.1.5. Inlet Cooling on Blade Life:

Figure 6 shows that inlet cooling improves predicted gas turbine blade life by reducing thermal loading and supporting more stable operating conditions. The effect is indirect because inlet cooling mainly changes compressor inlet conditions, air density, mass flow rate, and combustion behaviour, which in turn influence blade surface temperature and creep- life response. The main mechanisms are summarized below:

- i. Improves mass flow rate and combustion efficiency, allowing the turbine to deliver the required power level without excessive turbine inlet temperature. This helps prevent over-firing, which is a major contributor to blade overheating and thermal fatigue.
- ii. Lowers blade metal temperature, thereby significantly extending blade creep life.
- iii. Maintains more stable operating temperatures by reducing thermal gradients, particularly during start-up, shutdown, and load-change conditions.
- iv. Reduces thermal cycling, which causes crack initiation and propagation in blade over time

Since cooler and denser air improves air-fuel ratio and combustion control or stability, inlet cooling helps in minimizing hot spots that cause localized blade damage.

3.1.6. Inlet Cooling on Blade Surface Temperature:

Figure 7 shows that reducing compressor inlet temperature lowers blade surface temperature. At a constant turbine inlet temperature, inlet cooling raises air density and air mass flow rate, increases net power output, and reduces thermal loading on the blade. When fuel-flow control is maintained, a lower compressor inlet temperature can also reduce the thermal severity imposed on turbine blades. Since blade surface temperature strongly influences creep damage, this reduction supports longer predicted blade life.

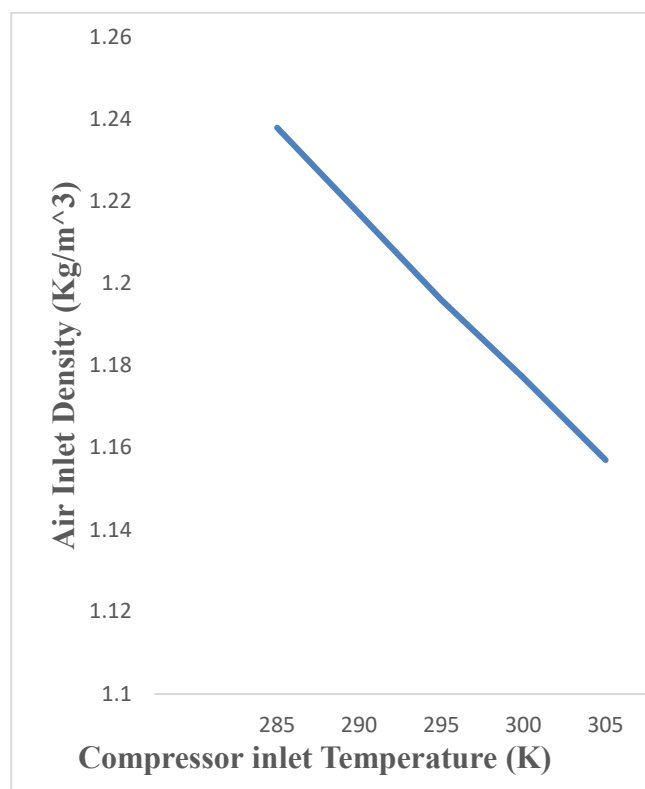


Fig. 2 Effect of inlet cooling on intake air density

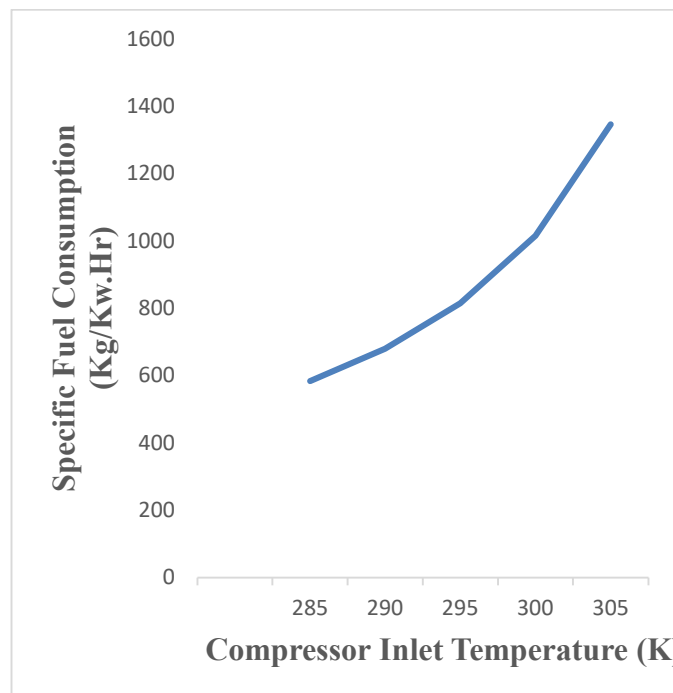


Fig. 3 Effect of inlet cooling on specific fuel consumption

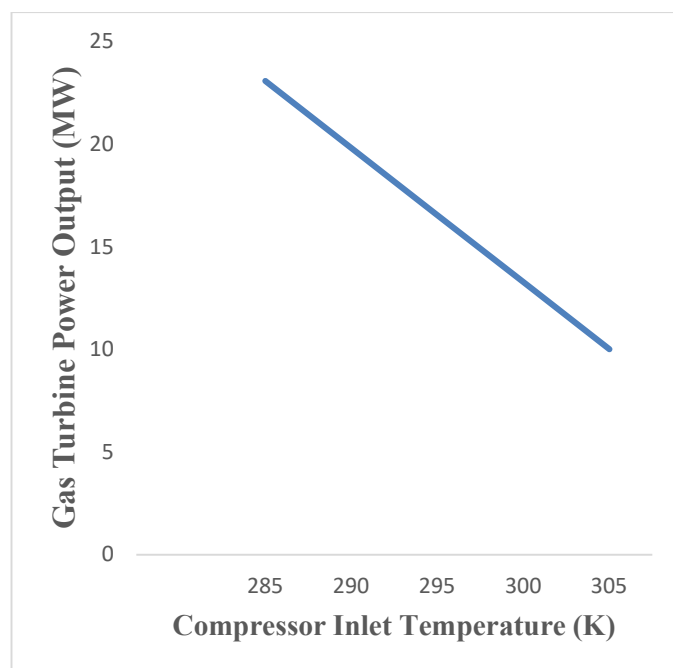


Fig. 4 Effect of inlet cooling on gas turbine power output

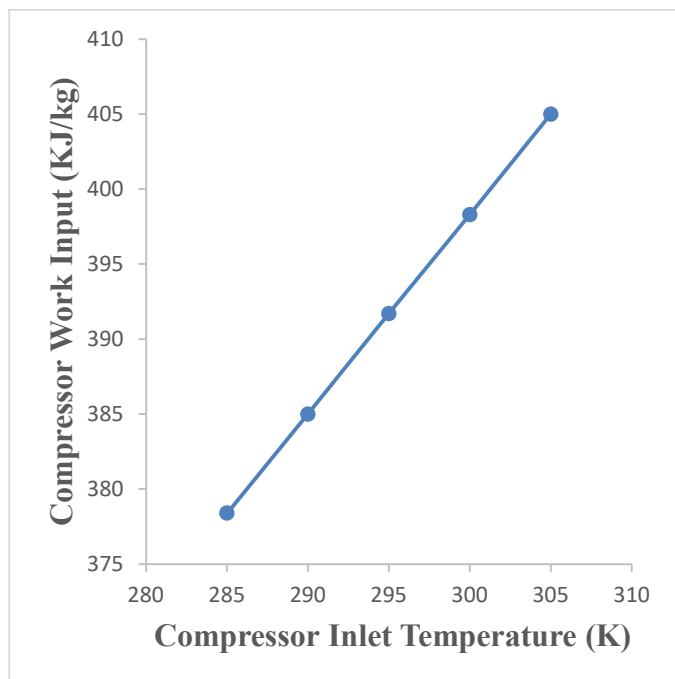


Fig. 5 Effect of inlet cooling on compressor work input

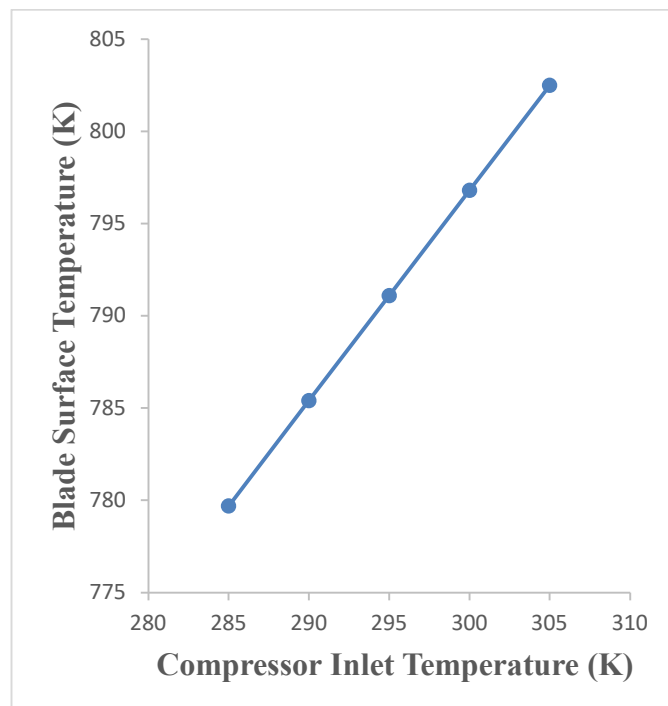


Fig. 7 Effect of inlet cooling on blade surface temperature

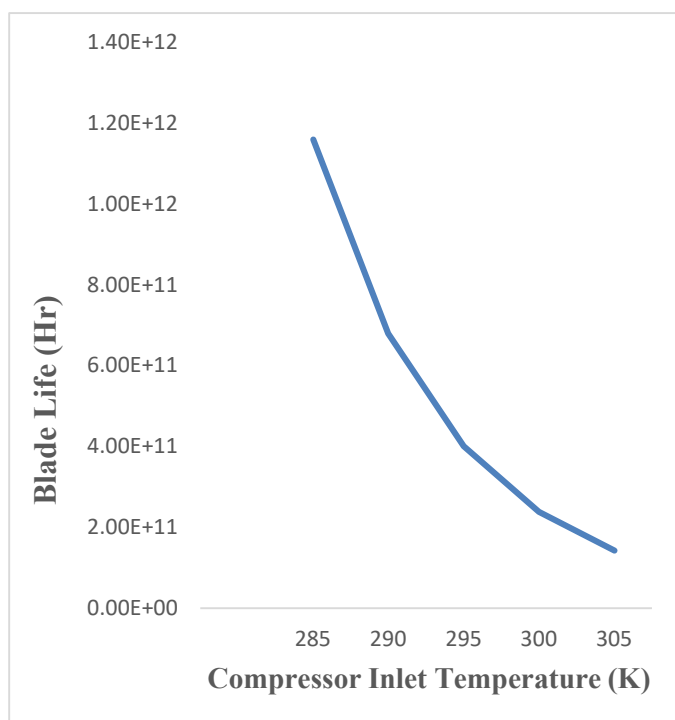


Fig. 6 Effect of inlet cooling on predicted blade life

IV. CONCLUSIONS

This study investigated the influence of inlet air cooling on simple gas turbine performance and turbine blade life using a numerical model developed in Engineering Equation Solver. The results demonstrate that lowering the compressor inlet temperature increases inlet-air density and the air mass flow rate, thereby enhancing power output and thermal efficiency. Inlet cooling also decreases compressor work and specific fuel consumption, indicating improved fuel utilization.

The blade-life model further shows that inlet cooling reduces blade surface temperature and improves predicted creep life. This confirms that inlet cooling provides a dual benefit: it enhances gas turbine output while helping to preserve turbine blade durability. The findings support the use of inlet air cooling as a practical retrofitting strategy for gas turbines operating in hot climates, particularly where higher power output is required without excessive fuel consumption or increased thermal loading of turbine blades.

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Appendix A

Table 1 Impact of Turbine Inlet Temperature on Gas Turbine Performance and Blade Life

T ₃	t _r	Z ₁	SF C	W _t	W _{Gnet}	T _b	P _G T
10 60	1.339 E12	1.72 4E-09	58 3.7	42 5.3	46.1 7	75 4.7	23.0 85
10 90	3.028 E12	1.57 3E-08	28 2.9	47 5.1	95.2 6	76 9.7	47.6 31
11 20	7.249 E11	9.95 6E-08	18 6.7	52 5	144. 4	78 4.7	72.1 76
11 50	1.831 E11	5.28 2E-07	13 9.3	57 4.8	193. 4	79 9.7	96.7 21
11 80	4.865 E10	2.49 3E-06	11 1.1	62 4.6	242. 5	81 4.7	121. 266