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Investigation of the Influence of Ambient Conditions on the Thermodynamic Characteristics of Air as a Working Fluid for Gas Turbines

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ABSTRACT

The study focuses on estimating thermodynamic characteristics at constant pressure for ambient air as a working fluid for gas turbines. The objective of this paper is to carry out a thermodynamic analysis of the properties of air as a working gas for a power plant. Various values of relative humidity, as well as temperatures, were examined in this study. Code was written using EES (Engineering Equations Solver) to conduct the simulation. This code contains the necessary equation to compute the thermodynamic characteristics of the working fluid. According to the results, both temperature and relative humidity remarkably influence the specific heat capacity (C_p), isentropic exponent (γ_h) as well as the gas constant of air (R_h). According to the results, when the ambient air temperature is increased from 0 to 45 ℃ with constant relative humidity values of either 10% or 90%, the specific heat capacity increases by 5.01% and 17.6%, respectively. Furthermore, the isentropic exponent decreases by 1.07% and 4.5%, respectively. The results show that the gas constant of air increases with ambient air temperature and relative humidity. One can conclude that the ambient conditions have considerable influence on the thermodynamic characteristics of a gas turbine working fluid.

1. Introduction

A gas turbine is a heat engine in which hot flue gases produced by burning fuel, drive a turbine that is used to generate power. Every GT has three basic components, an air compressor, a turbine and a combustion chamber [1-4]. The turbine shaft is connected to a generator, which produces electrical energy through the rotation of an electrical generator shaft [5-8]. Gas turbines are essential for the production of energy, marine power plants, and airplanes [9-13]. They are an important component of every combined cycle power plant.

Gas turbine power output and efficiency can be utilized to evaluate its performance. Both are significantly influenced by the intake air temperature and relative humidity. A gas turbine's

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compressor is designed to work with a constant ambient air volume flow [14-17]. The mass flow rate into the turbine drops as the temperature of the surrounding air increases [18-22].

Reduced air mass flow directly impacts a gas turbine's power generation capability. Additionally, the specific compressor work is increased when the inlet air temperature is elevated [23-25]. Furthermore, increased temperature causes a decreased air density, which increases the heat rate and thus specific fuel consumption [26-27].

Several researchers studied how the environment affects gas turbine performance. In the Kingdom of Saudi Arabia, where gas turbines produce more than 50% of the total capacity, Saleh *et al.*, [28] examined the impact of ambient air conditions on gas turbine performance. In Ad Dammam, the ambient temperature can reach 45 ^{o}C, resulting in output power losses that can reach 16,139 kW. In Riyadh, the temperature can reach 43.87 ^{o}C, resulting in a maximum power loss of 15,246 kW in July. Alternatively, the maximum power gain is 4,220 kW when the temperature is 8.9 ^{o}C. A typical gas turbine performance operating in India was investigated by Mohd Saif *et al.*, [29] over a wide variety of ambient temperatures and pressure ratios. They found a 3.28% loss in thermal efficiency when the outside air temperature was increased from 283 K to 323 K. These losses increased by up to 3.97% with increased pressure ratios. Furthermore, it was found that a loss in net power of 3.87% occurs when the ambient temperature increases from 283 K to 323 K, and this loss increases to 4.46% with higher pressure ratios. In many different regions of Turkey, Hasan Erdem *et al.*, [30] utilized two models to investigate the effects of inlet air temperature on the energy output and fuel consumption of a typical gas turbine. They observed production losses in all locations with outside temperatures greater than 15 ^{o}C. This loss is between 2.87–0.71% in comparison with nominal production. Additionally, they reported that when cooling inlet air temperature is reduced to 10 ^{o}C, the output power is augmented by 0.37–7.59%. Wan *et al.*, [31] analyzed gas turbine performance under various temperatures considering the thermal characteristics of different working fluids. Their results showed that power generation decreased by 22.6% and efficiency decreased by 57.28% as ambient air temperatures changed from 5 to 35 ^{o}C. Additionally, the output of steam and gas turbines both dropped by 17.0% and 16.2%, respectively. A thermodynamic impact of intake air cooling systems was done by Sanjay *et al.*, [32]. He discovered that a simple gas turbine has lower net power output than an inlet air-cooled gas turbine. Furthermore, less work is required for compression when the air is cooler. Additionally, plant efficiency was improved by 4.88% and plant work increased by 14.77%. Enhancing micro-gas turbine performance was done by Comodi *et al*., [33] using an inlet air cooling vapor compression technique. The outcomes demonstrated a significant increase in electric power, exceeding 14 kWe. Alnasur *et al.,* [34] applied a fog system to cool air. This system enhanced output power by 25000-22000 kV h. Moreover, the thermal efficiency increases to 0.26. Majdi Yazdi *et al.*, [35] documented the effect of using inlet fogging, an absorption chiller, as well as heat pump systems to improve the performance by means of cooling inlet air in four different areas. According to their findings, inlet air conditioning systems may increase power output. Ehyaei *et al.*, [36] demonstrated that using turbine inlet air cooling methods improves turbine performance. According to their results, gas turbine power was increased while fuel consumption was reduced. In the current research, the effects of ambient conditions are studied for a single shaft turbine. Results for net power, thermal efficiency and specific fuel consumption were measured. In a study by Alaa *et al*., [37], the performance of a gas turbine was simulated thermodynamically and tested for a base case without any turbine inlet cooling systems and compared with the performance using evaporative cooler and absorption chiller. The results showed that a cooling system for the plant is needed to improve the performance. Moein *et al*., [38] carried out an exergy, exergoeconomic, and exergoenvironmental investigation of a gas turbine plant. Using an MOPSO algorithm, they documented that the ambient temperature has a significant influence on the exergy performance. Mohammad Reza Majdi Yazdi *et* *al.*, [39] carried out a comparison of gas turbine air cooling systems for several cities in Iran using exergy, energy, economic, as well as environmental analyses. Despite the important results achieved, they neglected the effects of relative humidity on the heat capacity of air.

Although the temperature effects on gas turbine performance, such as power output and thermal efficiency, have been extensively studied, previous investigations did not examine the impacts of environmental factors on the thermodynamic characteristics of air, especially the specific heat capacity (C_p), isentropic exponent (γ_h) and the gas constant of air (R_h). Previous studies considered these values constant. However, they change in varying degrees according to different operating conditions. Additionally, previous studies took the effect of relative humidity on the mass flow value into account, but neglected the effect of relative humidity on other thermodynamic properties. Therefore, the aim of this study is to determine to what extent the thermodynamic properties of air can be considered constant and to what extent these properties are affected by relative humidity and temperature. In the current study, we present a thermodynamic analysis of the properties of air as a working gas for a power plant. Operating parameters of the active gas turbine used in the city of Jandar, in Syria, are simulated to evaluate the effect of ambient conditions on the values of $C_p, \gamma_h, \text{and } R_h$. Furthermore, the influence of ambient temperature and humidity is evaluated using a commercial computer program.

2. Mathematical Modelling

In general, specific heat capacity is an important physical property employed in thermodynamic estimates. Its determination involves calculating the change in gas specific enthalpy and determining the adiabatic exponent, among other parameters [40]. Specific heat, C_p, is constant throughout the thermodynamic processes in an ideal gas thermodynamic cycle. In practice, the specific heat changes depending on the operation to which it is being subjected, such as compression, heat addition, and expansion. In other words, the specific heat can be defined as the thermal energy needed to raise the temperature of a unit mass by one degree. Additionally, it represents heat storage and can be used to measure of a material's ability to store thermal energy [41]. The changes in the internal energy as well as enthalpy of a fluid during a process can be computed by utilizing specific heat values at the average temperatures.

As illustrated in Eq. (1), the specific heat for air can be written as a polynomial function of temperature.

$$C_p = a + b\left(\frac{T}{100}\right) + c\left(\frac{T}{100}\right)^{-2} \tag{1}$$

where T is temperature in K, while a, b and c are constants for a given gas. Their values are listed in Table 1. $C_p$is the specific heat at constant pressure with units of J/kg K. It can be seen from this equation that the specific heat capacity of dry air is only dependent on the temperature.

Table 2 shows the gravimetric or mass analysis composition of dry air. To calculate the specific heat, C_p, of air at a given temperature, Eq. (2) is used for each component of air employing the appropriate constants a, b, and c, which are given above. The specific heat, C_p, of air at a given temperature is then calculated as

$$C_p = 0.7553 \times C_{p.N2} + 0.2314 \times C_{p.O2} + 0.0128 \times C_{p.AR} + 0.0005 \times C_{p.CO2} \tag{2}$$

where $C_{p.N2}$, $C_{p.O2}$, $C_{p.AR}$ and $C_{p.CO2}$ are the specific heats for N_2, O_2, Ar and CO_2 at a specified temperature, respectively, and are estimated with Eq. (1) using data from in Table 1.

Table 1
Constants for evaluating the specific heat of air [42]

Coefficients	A	B	C	Molecular weight
O_2	936	13.1	–523	31.999
N_2	1020	13.4	–179	28.013
H_2O	1695	57.1	0	18.030
CO_2	1005	20.0	–1959	44.010
AR	521	0	0	39.948

Table 2
Gravimetric analysis of dry air composition [42]

Component	Mass fraction
O_2	0.2314
N_2	0.7553
CO_2	0.0005
AR	0.0128

The current study considers the C_p of dry air. The influence of humidity can be significant at high ambient temperatures since air includes water vapor, which must be considered when calculating the gas characteristics. The water vapor quantity needed for complete saturation of the air is typically used to represent the relative humidity of air. However, relative humidity is defined at a specified temperature as the ratio of the actual water vapor pressure of air to the water vapor pressure of saturated water. It may be calculated using Eq. (3)

$$\varphi = \frac{P}{P_s} \times 100 \tag{3}$$

where P is vapor pressure of water, P_s is the saturated water vapor pressure and φ is the percent relative humidity. The units for P and P_s are typically millibars (mb). The saturated vapor pressure, P_s, can be calculated from

$$P_s = 6{,}112 \times e^{\frac{17.67T}{T+234.5}} \tag{4}$$

where T is the ambient temperature in degrees Celsius. Therefore, the vapor pressure of water vapor is estimated given the ambient temperature and relative humidity.

Calculating gas thermal performance requires knowledge of the mass of water vapor. This is computed using the definition of specific humidity, which is the mass of water vapor contained in a dry air unit mass as

$$\omega = \frac{mass\ vapour}{mass\ air} \tag{5}$$

Eq. (5) can be depicted as the following by applying Dalton's Law of Partial Pressures

$$\omega = 0.622\ \frac{P}{Pa-P} \tag{6}$$

where Pa is the ambient pressure in millibars (mb).

The mass of water vapor is calculated using equations presented in this section knowing its relative humidity, ambient pressure, and temperature. This can be considered in the calculations for specific heat, where the C_p for humid air is

$$C_{p.h} = C_{p.d} \times mf_{air} + C_{p.water} \times mf_{water} \tag{7}$$

where
$C_{p.h}$ = specific heat for humid air at constant pressure.
$C_{p.d}$ = specific heat of dry air at constant pressure.
$C_{p.w}$ = specific heat of water vapor at constant pressure.
mf_{air} = quantity of dry air per kilogram of moist air.
mf_{water} = water vapor quantity in a kilogram of humid air.

According to [1], the mass flow rate of humid air is

$$\dot{m}_{h.a} = \dot{m}_{d.a} + \dot{m}_v \tag{8}$$

where $\dot{m}_{d.a}$ is the dry air mass flow, $\dot{m}_v$ is the water vapor mass flow rate. Psychrometric determination of the thermodynamic properties of moist air can be determined under specified dry and wet bulb temperatures as well as its relative humidity.

For specified conditions, the dry air rate is

$$\dot{m}_{d.a} = \frac{V * \rho_h}{(1+\omega)} \tag{9}$$

where ρ_h is humid air density and V is the volumetric flow rate (243 $m^3 s^{-1}$) [1]. Given Eq. (8)

$$\dot{m}_{h.a} = \dot{m}_{d.a}\ (1+\omega) \tag{10}$$

The humid-air isentropic exponent, γ_h, is

$$\gamma_h = \frac{C_{p.h}}{C_{p.h} - R_h} \tag{11}$$

The humid air gas constant, R_h, can be calculated as

$$R_h = \frac{8.3143}{MW_h} \tag{12}$$

where, MW_h is the humid air molecular weight given by [17]

$$MW_h = \frac{1}{\left(\frac{V_f}{18.015}\right) + \left(\frac{A_f}{28.79}\right)} \tag{13}$$

where V_f is the vapor mass fraction given by

$$V_f = \frac{\omega}{\omega + 1} \tag{14}$$

A_f is the dry-air mass fraction

$$A_f = 1 - V_f \tag{15}$$

3. Methodology

The thermodynamic characteristics of a gas turbine working fluid have been analyzed in the current work. The influence of ambient conditions on the thermodynamic properties of a gas turbine working fluid was investigated using and changing the ambient air temperature and relative humidity from 0 to 45 ^oC and 10%-90%, respectively. A software program, EES (Engineering Equation Solver), was utilized in this study for calculations of various parameters.

EES is a mathematical software package that is widely used to solve thermodynamic problems due to its capability of solving non-linear equations, in addition to having a library of precise thermodynamic properties of various and diverse working fluids. The program employs known mathematical terms for analyzing various thermodynamic phenomena. The computational procedure of the program is shown in Figure 1. First, the program reads the inputs for the mathematical model. The next step is assuming values of the ambient conditions. After determining the psychometric properties of ambient air, the program solves equations [Eq. (1), Eq. (2), Eq. (7), Eq. (9), Eq. (11) and Eq. (12)] simultaneously and prints the output.

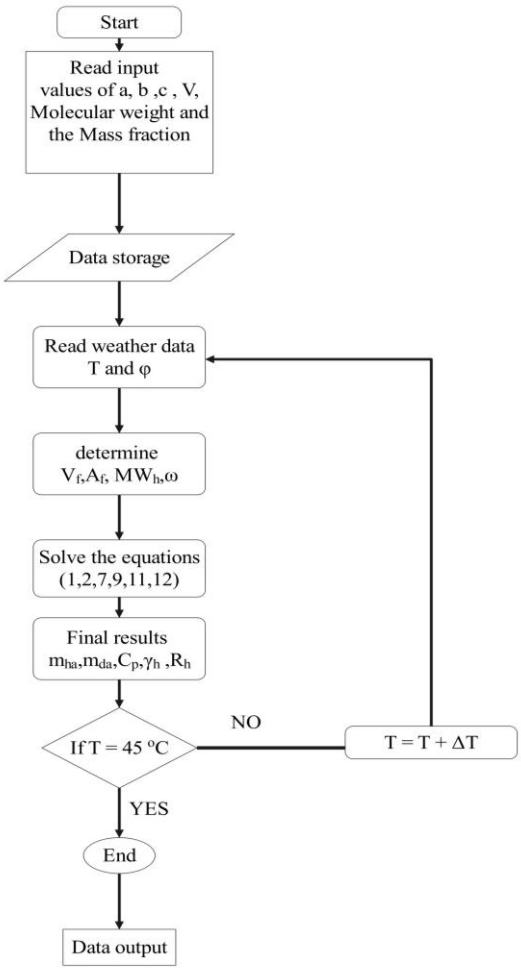

Fig. 1. Flowchart of the simulation steps used in the code

4. Results

Figure 2 represents the specific heat capacity for the dry air components at a relative humidity of 60%, which is common value. This figure is a result of Eq. (1) and Eq. (2). In Figure 2, the specific heat capacity for dry air components increases with air temperature, as expected. Only the specific heat capacity for argon gas remained constant at 0.521 kJ/kg °C from 0 °C to 45 °C. The specific heat capacity for O_2, CO_2 and N_2 increased from 901.6, 796.4 and 1030 to 930, 875 and 1045 J/kg °C, respectively. The specific heat capacity of dry air, $C_{p.d}$. increased from 995.7 to 1011 J/kg °C, an increase of 5.56%.

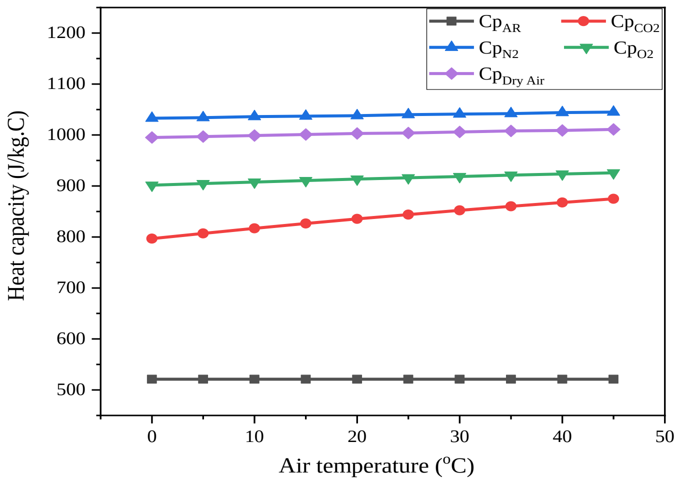

Fig. 2. Heat capacity for dry air components at a relative humidity of 60%

Figure 3 and Figure 4 show how ambient air influences wet air's specific heat capacity at relative humidity levels of 10% and 90%, respectively. It is notable that at high temperatures, the increased specific value is larger. At higher temperatures, air can hold more water vapor. Since water vapor has a higher specific heat compared to dry air, a higher specific humidity increases the heat capacity of air. It has been found that increasing relative humidity has a remarkable effect on specific heat capacity. The results reveal that when the ambient air temperature is increased from 0 to 45 °C at a constant relative humidity of 10%, the specific heat capacity increases by 0.05 kJ/kg °C, which represents an increase of 5.01%. This behavior can be attributed to a high humidity ratio value, as it increased to 0.0191 kg/kg. Moreover, at a relative humidity of 90%, the specific heat capacity increases by 0.177 kJ/kg °C, which represents an increase of 17.6%, while the humidity ratio increased from 0.0034 to 0.057 kg/kg, an increase of 16.1%. Dry air has a specific humidity 0 kg/kg, so water vapor has no effect.

Increasing C_p will directly increase the specific work of the compressor of the thermal plant since it is directly proportional to C_p. Thus, using cooling methods for reducing air temperature results in decreased C_p values, hence reducing the compressor specific work.

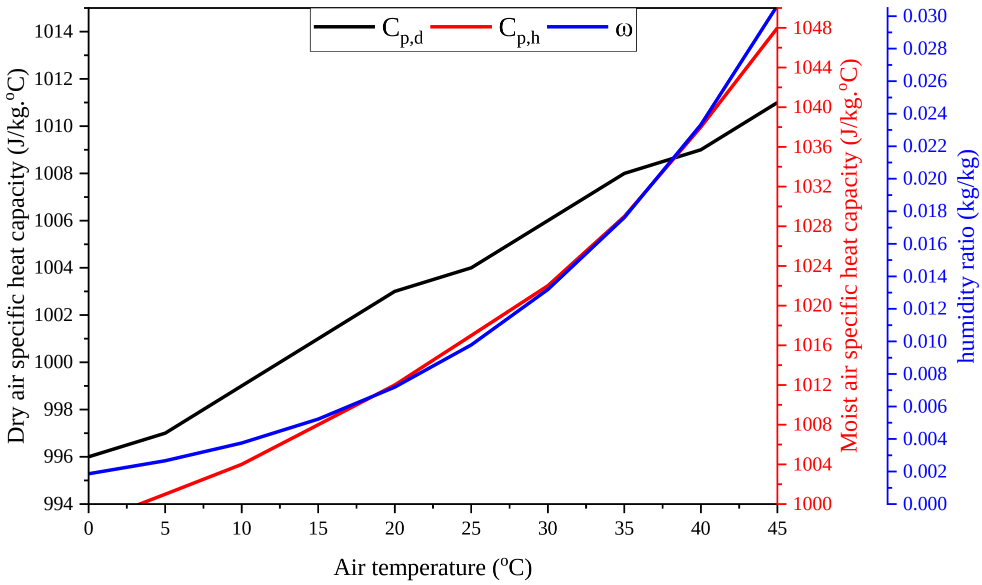

Fig. 3. Specific heat for dry and moist air at relative humidity of 10%

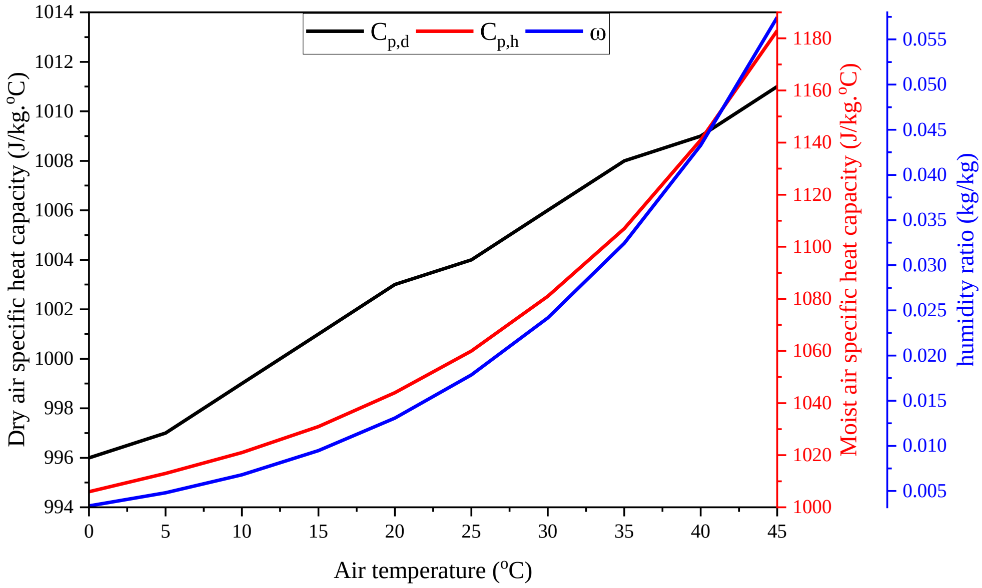

Fig. 4. Specific heat for dry and moist air at relative humidity of 90%

Table 3 presents the effect of C_p on the compressor specific work of the gas turbine block in the Jandar power plant when R_h and γ_h are held constant. The specific heat capacity has a significant influence on the specific work, especially at high relative humidities and temperatures, where the ambient air has the potential to carry greater amounts of water vapor. The specific work at a relative humidity of 90% increases by 10.8% at 45 °C compared to when C_p is constant. According to the results, at high values of relative humidity, the ambient temperature has bigger effect on C_p compared to the relative humidity. At constant relative humidity of 90%, C_p increased from 1013 to 1183 J/kg °C, which represents an increase of 16.7%. At constant relative humidity of 10%, C_p increased from 1000 to 1030 J/kg °C, which represents an increase of 3%. While at a constant ambient temperature of 40 °C and when the relative humidity increased from 10 to 90%, C_p increased from 1023 to 1143 J/kg °C, which represents an increase of 11.7%.

Table 3
Compressor specific work of the gas turbine unit under different conditions

Ambient temperature °C	Compressor Work (kJ)		
	C_p = Constant	C_p = variable relative humidity = 10 %	C_p = Variable relative humidity = 90 %
5	335.7	336	338.7
15	349	349.7	355.2
25	362.5	363.9	374.4
35	375.7	378.1	398.3
45	389	393.3	430.4

Figure 5 and Figure 6 demonstrate the effects of ambient air on mass flow of moist and dry air as well as water vapor entering the gas turbine at 10% and 90% relative humidities. As the ambient air temperature increases, both the dry and moist mass flow rates decrease while the water vapor mass flow increases due to a greater humidity ratio. These parameters very negatively impact gas turbine performance. An increased specific humidity reduces the air mass flow in the gas circuit since the atomic mass of H_2O is less than N_2 and O_2. For a given volume, moist air has a lower mass than dry air, so, it is less dense. The resulting lower density air reduces the humid-air mass flow rate entering the air compressor.

When the ambient air temperature increases from 0 to 45 °C at a constant relative humidity of 10%, the dry and moist air mass flow rates decrease by 40.8 and 39.7kg/s, respectively, while the water vapor flow rate increases by 1.141kg/s. At higher relative humidities, the decreased air mass flow rate was greater. At a relative humidity of 90%, when the ambient air temperature increases from 0 to 45 °C, the dry and moist air mass flow rates decrease by 59.3 and 46.6 kg/s, respectively, while the water vapor flow rate increases by 12.718 kg/s.

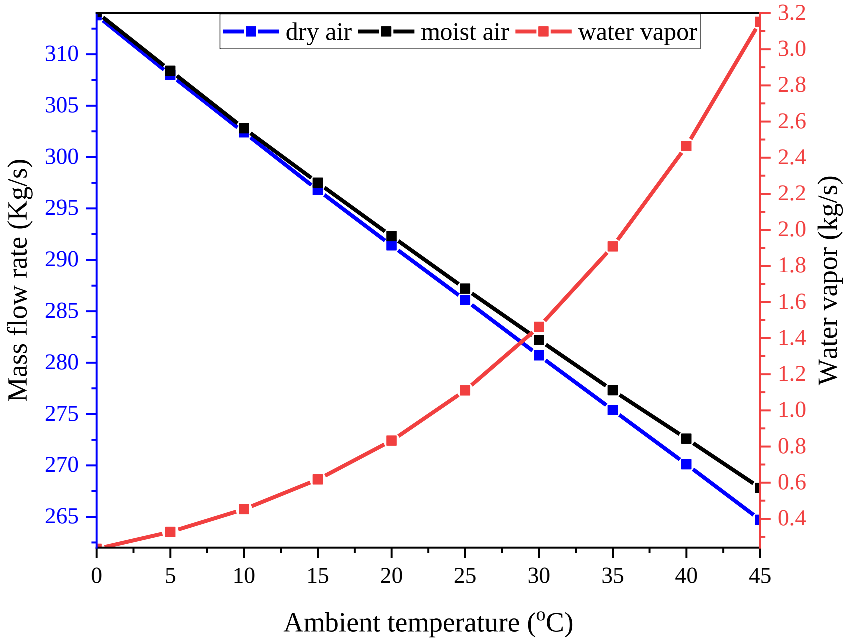

Fig. 5. Variation of mass flow of moist and dry air as well as the water vapor with ambient air at a relative humidity of 10%

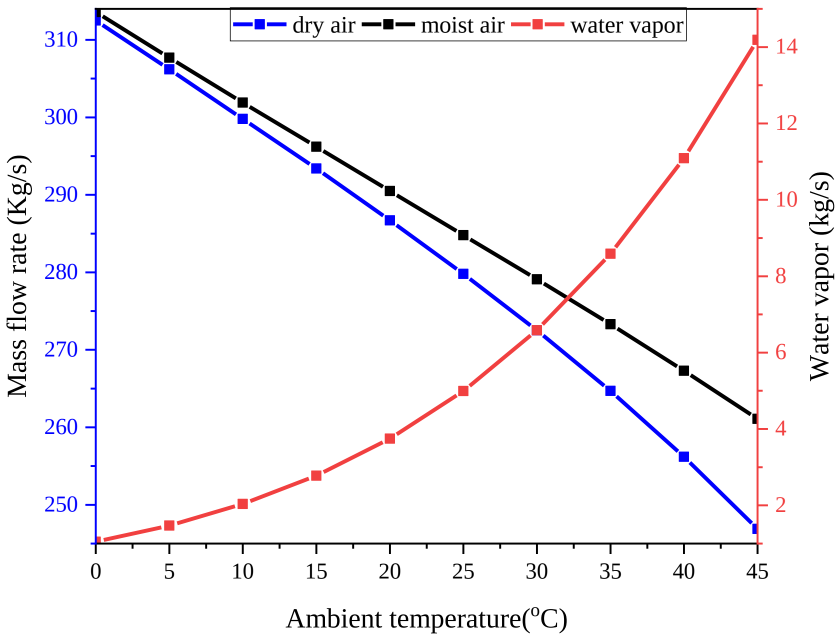

Fig. 6. Variation of mass flow of moist and dry air as well as the water vapor with ambient air at a relative humidity of 90%

Figure 7 and Figure 8 show a variation of dry air and water vapor mass fraction with ambient air at 10% and 90% relative humidities, respectively. According to these results, between 0 and 45 °C, with a 10% relative humidity, there is a 1.11% decrease in the dry air mass fraction, while the vapor mass fraction increased by the same value. Furthermore, when the relative humidity is 90%, the dry air mass fraction decreases by 5.1%, while the vapor mass fraction increases by the same amount. High vapor mass fraction values imply an increased moist content that results in a reduced air mass flow rate.

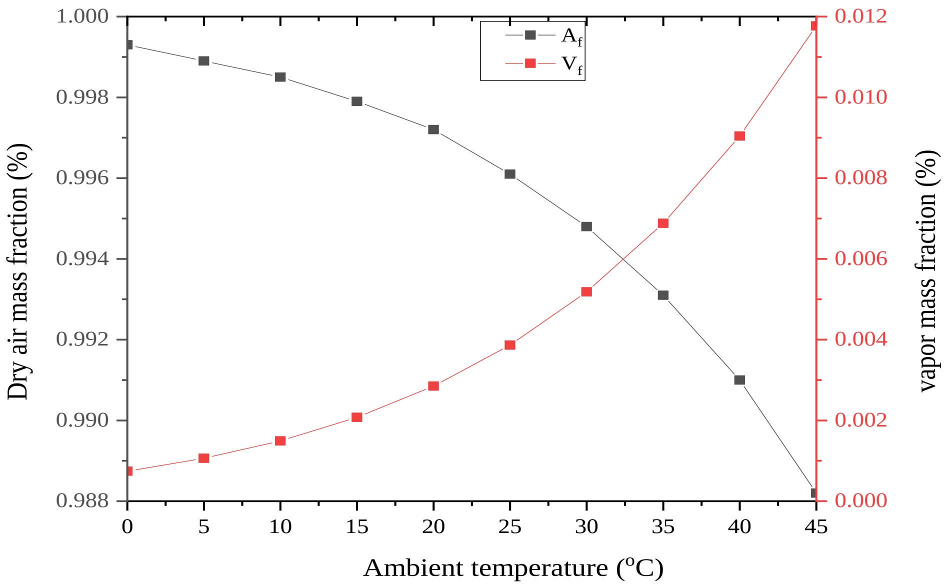

Fig. 7. Variation of dry air and the water vapor mass fraction with ambient air at a relative humidity of 10%

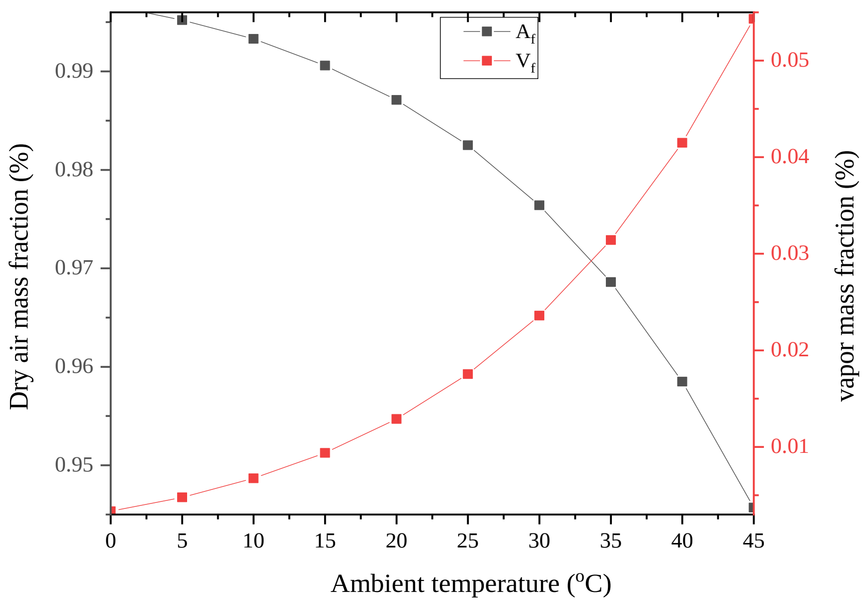

Fig. 8. Variation of dry air and the water vapor mass fraction with ambient air at a relative humidity of 90%

The isentropic exponent changes with temperature and relative humidity, as shown in Figure 9. It was found to decrease with increasing ambient air temperature and relative humidity. When the ambient air temperature increases from 0 to 45°C at a constant relative humidity of 10%, the isentropic exponent decreases by 0.015, which represents a 1.07% reduction. Moreover, at a relative humidity of 90%, the isentropic exponent decreases by 0.063, which is a 4.5% decrease.

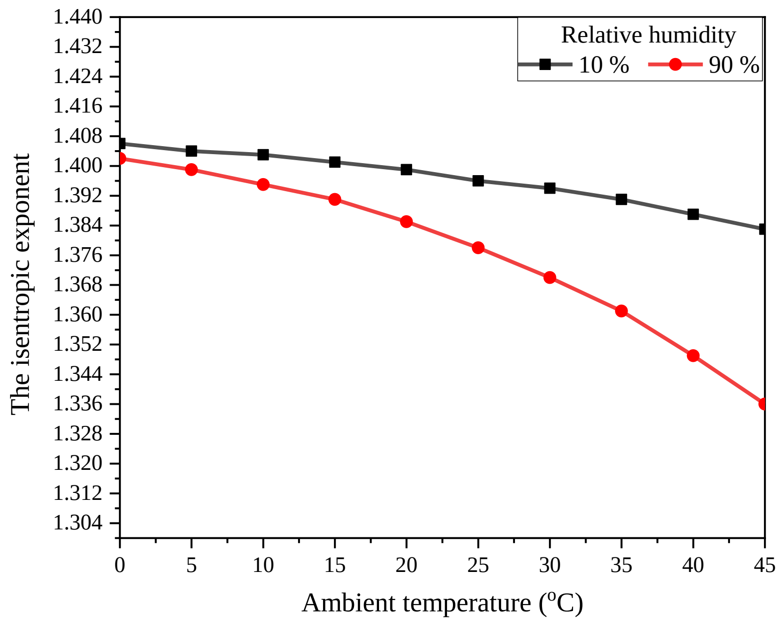

Fig. 9. Variation of the isentropic exponent with ambient air temperature

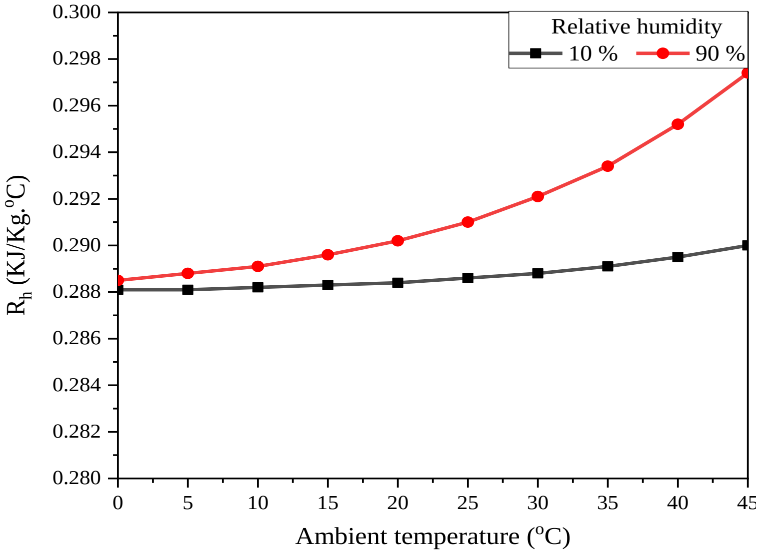

Fig. 10. Variation of the gas constant of air with ambient air temperature

Fluctuations of the air gas constant with temperature and relative humidity are shown in Figure 10. R_h increases with both the relative humidity and temperature of the surrounding air. Increasing R_h suggests that the molecular weight of the surrounding air is decreasing, which immediately causes a decreased mass flow rate. When the ambient air temperature increases from 0 to 45°C at a constant relative humidity of 90%, R_h increases by 0.0086, which represents a 3% change. Additionally, at a relative humidity of 10%, the change is only 0.001 KJ/Kg°C which is considered very small and can be neglected.

4. Conclusions

Using various inlet air flow temperatures and relative humidities, this thermodynamic study of a gas turbine's working fluid found that

i. The specific heat capacity, isentropic exponent, and gas constant are all affected by temperature and relative humidity.
ii. The specific heat capacity of air can be increased by increasing either or both the specific humidity and air temperature. At high values of temperature and humidity, the specific heat capacity can increase by 17.6%.
iii. With increasing ambient air temperature and relative humidity, the isentropic exponent decreases. At an ambient air temperature of 45°C at a constant relative humidity of 10%, the isentropic exponent decreases by 1.07% and 4.5% at a relative humidity of 10 % and 90%, respectively.
iv. The gas constant of air increases with ambient air temperature and relative humidity. At an ambient air temperature of 45°C and constant relative humidity of 90%, R_h increases by 0.0086.
v. Changes in the thermodynamic properties of air are more severe at higher temperatures.
vi. At low ambient temperatures, the increase in specific humidity, and thus, the change in the air's mass of vapor is minimal. The water vapor flow rate increases by 1.141 kg/s at high temperatures and low relative humidity while the dry and moist air mass flow rates decrease by 40.8 and 39.7 kg/s, respectively.

vii. At high ambient temperatures, the impacts of humidity are seen in performance estimates.
viii. Since specific humidity changes very little with relative humidity at low ambient temperatures, the changes in air characteristics are also small. For example, the change in R_h is 0.001 KJ/Kg°C.

Thus, this type of research work is a significant opportunity to understand the thermal performance of ambient air using modelling.

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