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MODELING AND SIMULATION OF THE EFFECT OF MOISTURE CONTENT AND AMBIENT TEMPERATURE ON GAS TURBINE POWER PLANT PERFORMANCE IN UGHELLI, NIGERIA

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ABSTRACT
The influence of ambient temperature and moisture content on the performance of Transcorp Power Plant, Ughelli, Delta State, Nigeria was investigated with the aid of a digital psychrometer. The simulation was done using codes developed on MATLAB R2017a and the results show that compressor power consumption increased by 1.65% for 0.7% rise in temperature, and 0.50% for 71.4% rise in moisture content. The specific fuel consumption also increased with increase in temperature where a 1.71% rise in ambient temperature resulted in a 0.15% rise in specific fuel consumption but it decreased by 0.49% for a 41.7% rise in moisture content. A 1.62% rise in temperature led to a 0.13% drop in net power output and a 29.4% rise in moisture content resulted in a 0.48% drop in net power output. Thus gas turbine plant operates optimally in areas with low ambient temperatures and high moisture content.

Keywords: *Work ratio, net power, efficiency, moisture content, specific fuel consumption, heat rate.*

1. INTRODUCTION

Gas Turbines have been used over the years to produce continuous electricity for rural and urban regions and it is viewed as the most fit solution to the increasing electricity demand. Therefore, it is necessary to discover ways to improve output from a gas turbine. Ensuring gas turbines run at maximum efficiency is a primary goal for operators. In the current economic climate, anything that increase productivity and hence profit is clearly welcome, as most of the operating cost of a gas turbine is the cost of the fuel.

The gas turbine compressor is design in such way that the air is compressed with a minimum work input while retaining relatively high efficiency and aerodynamic stability over the operating range [1].

The work outputs also depend on the maximum cycle temperature and pressure ratio, however, higher turbine inlet temperatures are still limited by turbine blade cooling requirement and metallurgical improvements [2]. A schematic diagram of a simple cycle, single shaft gas turbine and the corresponding temperature and entropy diagram is as shown in Figure 1 and Figure 2 respectively.

Operating parameters of ambient temperature, altitude, humidity termed climatic conditions affects the performance of gas turbine units [3][4]. These climatic conditions are worst still dynamic in nature, varying at different times of the day and season [5][6].

2. THERMODYNAMIC ANALYSIS OF THE GAS TURBINE PLANT

2.1. Psychrometrics Analysis of moist air

Psychrometrics define the thermodynamic properties of moist air under specified conditions of dry bulb temperature and wet bulb temperature and relative humidity. The amount of moisture in the air is readily obtained as [7]:

$$\omega = \frac{\dot{m}_v}{\dot{m}_a} \tag{1}$$

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where ω equals specific humidity in kg/kg dry air, $\dot{m}_v$ equals mass flow rate of water vapour in kg/s and $\dot{m}_a$ equals mass flow rate of dry air in kg/s. The specific humidity is normally expressed as [7]:

$$\omega = 0.622 \frac{P_v}{P_a} \tag{2}$$

According to Dalton's law; the total pressure of the atmospheric air is expressed as:

$$P = P_v + P_a \tag{3}$$

Where P equals barometric pressure, P_v equals partial pressure of water vapour and P_a equals partial pressure of dry air in N/m^2.
The relative humidity of the atmosphere is the ratio of the actual partial pressure of the vapour to the partial pressure of the vapour when the air is saturated at the same temperature. Mathematically, it is defined as [8]:

$$\phi = \frac{P_v}{P_{vs}} \tag{4}$$

It is also expressed as:

$$\phi = \frac{\dot{m}_v}{\dot{m}_{vs}} \tag{5}$$

where $\dot{m}_{vs}$ equals mass flow rate of saturated vapour, kg/s and P_{vs} equals partial pressure of saturated vapour, N/m^2.

Degree of saturation is defined as:

$$\mu = \frac{\omega}{\omega_s} \tag{6}$$

where ω_s is the specific humidity at saturation. It is can also be computed from a knowledge

of relative humidity and pressures as:

$$\mu = \phi\left(\frac{P - P_{vs}}{P - P_v}\right) \tag{7}$$

The three most important variables on which every other properties of moist air depend are, the dry bulb temperature, wet bulb temperature and the barometric pressure or atmospheric pressure. These parameters are readily measured and are used to evaluate the partial pressure of water vapour using Carrier's equation, defined as [9]:

$$P_v = P_w - \frac{(P - P_w)(t_{db} - t_{wb})}{1547 - 1.44t_{wb}} \tag{8}$$

P_w equals partial pressure of saturated water, N/m^2, t_{db} equals dry bulb temperature in ^{o}C, and t_{wb} equals wet bulb temperature, ^{o}C.

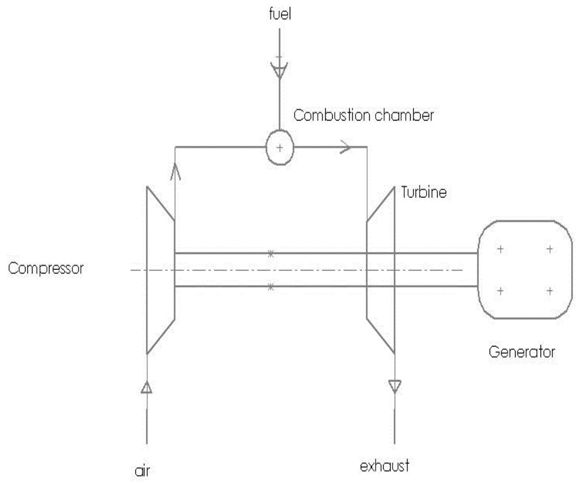

Fig. 1: Single Shaft Gas Turbine

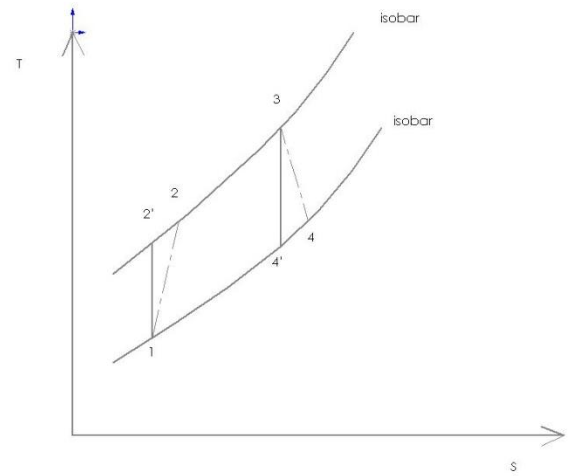

Fig. 2: Gas Turbine T – S Diagram

2.2. Performance criteria of the Gas turbine

2.2.1. Compressor Power consumption

The power delivered to the compressor by the turbine is defined as [10]:

$$P_c = \dot{m}_{ma} C_{pma}(T_{02} - T_{01}) \tag{9}$$

Where $\dot{m}_{ma}$ equals mass flow rate of moist air in kg/s and C_{pma} equals specific heat capacity at constant pressure of moist air, N/m^2, T_{02} equals compressed air exit stagnation temperature, ^{o}C and T_{01} equals inlet air stagnation temperature, ^{o}C.

But

$$\dot{m}_{ma} = \dot{m}_v + \dot{m}_a \tag{10}$$

In the light of equation (1)

$$\dot{m}_{ma} = \dot{m}_a(\omega + 1) \quad (11)$$

Substituting equation (11) in equation (9) yield:

$$P_c = \dot{m}_a(\omega + 1)\ C_{pma}(T_{02} - T_{01}) \quad (12)$$

The power consumption can be expressed directly as a function of humidity ratio and ambient temperature as [11, 12]:

$$P_c = \dot{m}_a(\omega + 1)\ C_{pma}T_{01}\left(\frac{T_{02}}{T_{01}} - 1\right) \quad (13)$$

The empirical relation to evaluate the specific heat capacity of moist air, C_{pma}, at constant pressure is expressed for the 200 K < T < 800 K [13] as:

$$C_{pma}(T) = 1.0189exp10^3 - 0.13784T + 1.9843exp10^{-4}T^2 + 4.2399exp10^{-7}T^3 - 3.7632exp10^{-10}T^4 \quad (14)$$

where

$$T = \frac{T_{02} + T_{01}}{2} \quad (15)$$

Using the isentropic relation, the temperature gradient across the compressor is obtained as [14], [15]:

$$T_{02} - T_{01} = \frac{T_{01}}{\eta_c}\left[\left(\frac{P_2}{P_1}\right)^{\frac{\gamma_a - 1}{\gamma_a}} - 1\right] \quad (16)$$

Where η_c is the isentropic efficiency of compression, P_2/P_1 is the compression pressure ratio, γ_a is the ratio of specific heat capacities of air.

2.2.2. Combustion heat transfer

The air from the compressor is fed into the combustion chamber, where a quantity of fuel is introduced and ignited to release large quantum of energy to generate high temperature gaseous mixture. The stoichiometric ratio is approximately 15:1 but the actual fuel ratio would be in the region of 100:1 [16].

The heat transfer due to combustion in the combustion chamber by application of the steady flow energy equation is defined as:

$$Q = \dot{m}_g C_{pg}(T_{03} - T_{02}) \quad (17)$$

Where $\dot{m}_g$ equals mass flow rate of flue gases, kg/s, C_{pg} equals specific heat capacity of the flue gas, J/kgK, T_{03} equals turbine inlet temperature, °C.

But

$$\dot{m}_g = \dot{m}_{ma} + \dot{m}_f \quad (18)$$

Where $\dot{m}_f$ is the mass flow rate of fuel, kg/s. In the light of equation (11) and equation (17), the quantum of heat transfer can be computed as:

$$Q = \left(\left(\dot{m}_a(1 + \omega)\right) + \dot{m}_f\right)\ C_{pg}(T_{03} - T_{02}) \quad (19)$$

Knowing the fuel gas heat value (FHV), the mass flow rate of fuel flow rate, $\mathrm{m_f}$, can be obtained as;

$$m_f = \frac{Q/FHV}{\eta combustor} \quad (20)$$

Where ηcombustor is the combustor efficiency, Q is the quantity of heat supplied.

The specific fuel consumption, sfc, is determined as:

$$sfc = \frac{3600m_f}{P_c} \quad (21)$$

Where $\mathrm{P_c}$ equals net power output of the gas turbine. Application of the first law of thermodynamics in the combustor yields the energy balance as [17]:

$$\dot{m}_f FHV = \left(\dot{m}_a(1 + \omega) + \dot{m}_f\right)C_{Pg}(T_{03} - T_{02}) \quad (22)$$

2.2.3. Gross power output of turbine

The gross power output by application of the steady flow energy equation is expressed as:

$$P_t = \dot{m}_g C_{pg}\ (T_{03} - T_{04}) \quad (23)$$

From equation (17), the power output is then obtained as:

$$P_t = \left(m_a(1 + \omega) + m_f\right)C_{pg}\ (T_{03} - T_{04}) \quad (24)$$

Where $\mathrm{T_{04}}$ is the stack temperature, °C.

Using the isentropic relation, the temperature gradient across the turbine is obtained as:

$$T_{03} - T_{04} = \eta_T T_{03}\left(1 - \left(\frac{P_4}{P_3}\right)^{\frac{\gamma_g - 1}{\gamma_g}}\right) \quad (25)$$

Where η_T is the isentropic efficiency of the gas turbine and γ_g is the specific heat capacity of the products of combustion. The turbine inlet pressure, $\mathrm{P_3}$, can be calculated as [19]:

$$P_3 = P_2 - \Delta P_{co} \quad (26)$$

where, ΔP_{co} equals pressure drop in the combustor, and is usually within 2%-5% of P_2 [13].

And the net power developed by the plant is then:

$$P_n = P_t - P_c \quad (27)$$

Substituting equation (11) and equation (23) into equation (26), the net power developed by a gas turbine in terms of inlet temperature and specific humidity is [20]:

$$P_n = \left(m_a(1+\omega) + m_f\right)C_{pg}\,(T_{03} - T_{04}) - m_a(\omega + 1)\ C_{pma}(T_{02} - T_{01}) \quad (27)$$

The thermal efficiency is:

$$\eta_{th} = \frac{P_n}{m_f FHV} \quad (28)$$

The heat rate, HR, is obtained from [21]:

$$HR = \frac{Q}{P_n} \quad (29)$$

3. METHODOLOGY

The dry and wet bulb temperatures were obtained from the turbine plant environment with the aid of the microprocessor base digital psychrometer 8706, as shown in figure 3 from which the psychrometrics properties where computed.

The models generated were converted into MATLAB programs, that generate pictorial representation of solutions to help in understanding the performance of the gas turbine plants under varying conditions of the atmosphere. The basic gas turbine parameters employed are those of Transcorp Power Plant in Ughelli, Delta State, Nigeria. The simulation assumes ideal conditions for the turbine and compressor work transfer processes.

Fig. 3: Microprocessor 8706 Digital Psychrometer

4. RESULTS AND DISCUSSION

4.1. Psychrometric Graphics

Figure 4 shows a plot of dry and wet bulb temperature against time of the day. The variability of the environment is quite evident with changes in time. The changes are largely due to the earth rotation. Using Carrier's equation, the variability of the partial pressure of vapour with changes in time is obtained as shown in figure 5.

Figure 6 shows a plot of variability of moisture content with changes of time due to the earth rotation while figure 7 is a plot of variability of degree of saturation with changes in time occasioned by the earth rotation.

As clearly seen the atmospheric conditions is dynamic and as such gas turbine performance metric is expected to be dynamic in nature.

4.2. Gas turbine performance Metric

4.2.1. Compressor Power Consumption

Figure 8 represents a plot of compressor power consumption against dry bulb temperature at various humidity ratio.

The plot shows that the compressor power increases as the dry bulb temperature increases, and also increases with an increase in moisture content. This is because as temperature rises, the molecules in air separate further and are more energetic, thus requiring more power to compress them. Similar results were obtained by [13, 21].

4.2.2. Turbine Net Power Output

Figure 9 shows plot of net power output against dry bulb temperature at various humidity contents. The plot shows that the net power output of the turbine decreases with an increase in the dry bulb temperature, and also increases with an increase in moisture content. Higher dry bulb temperature leads to a lower air density and thus a higher compressor work.

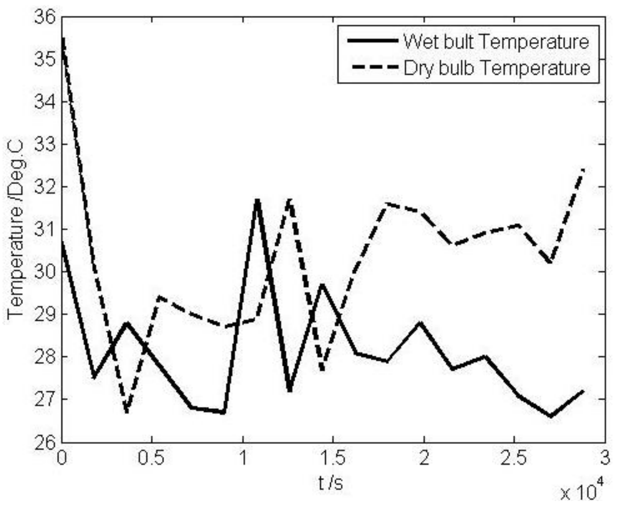

Fig. 4: Plot of Dry and Wet bulb temp. vs time

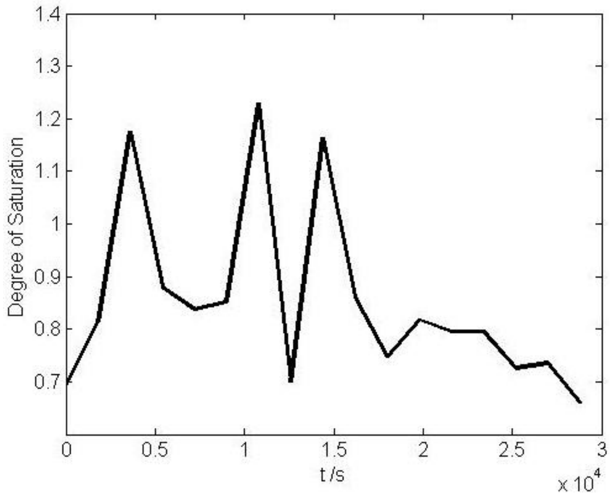

Fig. 7: Plot of Degree of saturation vs time

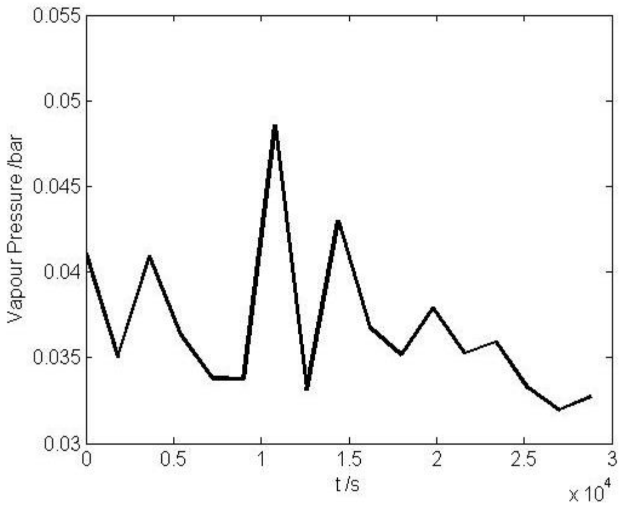

Fig. 5: Plot of Vapour pressure vs time

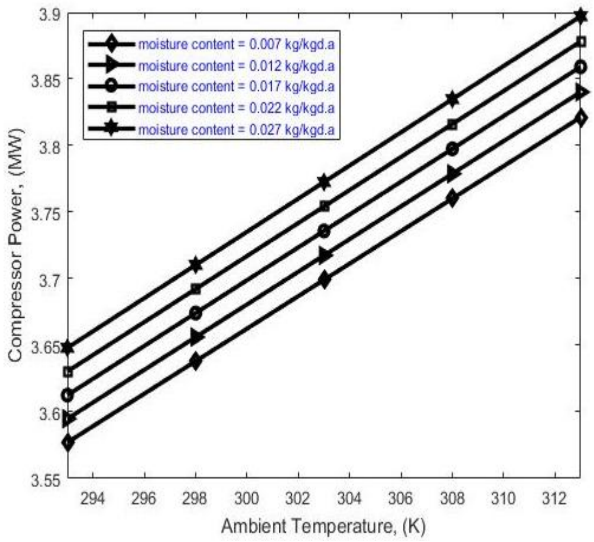

Fig. 8: Compressor power consumption vs dry bulb temperature

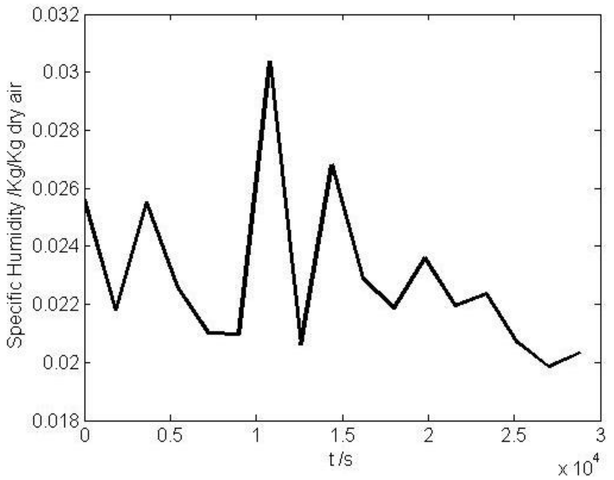

Fig. 6: Plot Specific humidity vs time

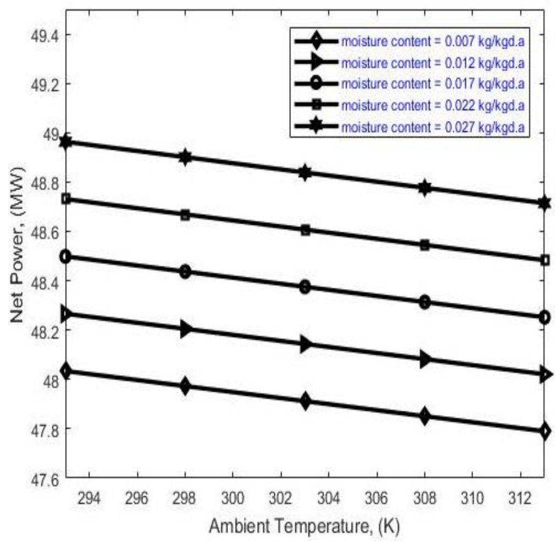

Fig. 9: Turbine net power output vs dry bulb temperature

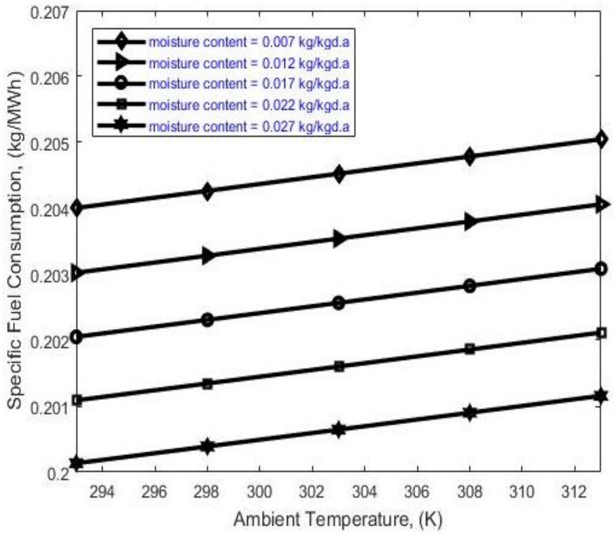

Fig. 10: Specific Fuel Consumption vs dry bulb temperature

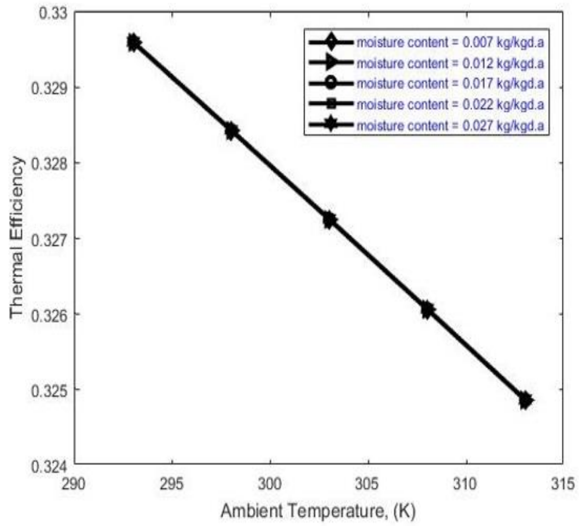

Fig. 11: Thermal Efficiency vs dry bulb temperature

Although increasing the moisture content leads to an increase in both turbine power and compressor power, the increase in turbine power is more than that of the compressor power thus leading to an increase in net power. Similar results were obtained by [1, 13, 22].

4.2.3. Specific Fuel Consumption

Figure 10 shows a plot of specific fuel consumption against dry bulb temperature at different humidity ratios.

The plot shows that the Specific Fuel Consumption increases as the dry bulb temperature increases but decreases as moisture content increases. The air mass flow rate entering the compressor increases with a decrease in dry bulb temperature and decreases with an increase in moisture content, similar results were obtained by [1], [21], [23].

4.2.4. Thermal Efficiency

Figure 11 shows a plot of thermal efficiency against dry bulb temperature at different humidity contents. Clearly the thermal efficiency decreases as dry bulb temperature increases, but shows little changes with changes in moisture content with extrapolation of the efficiency curves. This increase of efficiency as a result of decrease in moisture content and ambient temperature is due to the low compressor power consumption by the compressor, similar results were also obtained by [21, 23].

5. CONCLUSIONS

Ensuring gas turbine run at maximum efficiency is a primary goal for operators. From an environmental point of view, it is important that turbines run efficiently and emit fewer emissions. The gas turbine was critically analysed from the point of view of environmental conditions of dry bulb temperature and moisture contents. The results pointed clearly that compressor power consumption increased by 1.65% for 0.7% rise in ambient temperature, and 0.50% for 71.4% rise in moisture content. The specific fuel consumption also increased with increase in ambient temperature, a 1.71% rise in ambient temperature resulted in a 0.15% rise in specific fuel consumption but it decreased by 0.49% for a 41.7% rise in moisture content. A 1.62% rise in ambient temperature led to a 0.13% drop in net power and a 29.4% rise in moisture content resulted in a 0.48% drop in net power output. The effect of variation of heat rate, work ratio and thermal efficiency is more significant with changes in ambient temperature than in humidity ratio.

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