Estimating the Effect of Moist Air on Natural Convection Heat Transfer in Electronics Cooling

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Introduction

Atmospheric air is a mixture of gases and water vapor as well as a number of pollutants. The amount of water vapor and pollutants vary from place to place. The concentration of water vapor and pollutants decreases with the increase of altitude from the sea level, and above an altitude of about 10 km, atmospheric air consists of only dry air [1]. The mixture of dry air and water vapor is known as moist air. Psychrometry is the study of the properties of moist air. At a given temperature and pressure, the dry air can only hold a certain maximum amount of moisture which is known as saturated air. The amount of water vapor present in the air is quantified by the term humidity. The term humidity is commonly refers to relative humidity. Relative humidity is defined as the amount of water vapor in a sample of air compared to the maximum amount of water vapor the air can hold at any specific temperature in a form of 0 to 100%.

In most of the numerical simulations, the properties of air (density, specific heat capacity, thermal conductivity and viscosity) are assumed as a function of temperature alone. The above assumption is true for dry air but in reality the air contains moisture also. Water vapor and dry air have different fluid properties; hence with increased relative humidity, mixture properties such as specific heat, viscosity, thermal conductivity and density vary accordingly.

A majority of outdoor electronic products work under natural convection with different humidity conditions. As changes in relative humidity affect the values of thermo-physical properties, such changes will also affect the natural convection heat transfer coefficients and hence the cooling of electronics. In this study, the air properties are discussed with reference to the dimensionless numbers which are used to characterize the natural convection heat transfer phenomena. A simple relationship for the Nusselt number in terms of air properties is derived from which the ratio of the Nusselt number for moist air to dry air is obtained. This article presents the equations for calculating the moist air properties. The comparison of the Nusselt number for dry and moist air for different relative humidity and temperature conditions is also presented to substantiate the effect of humidity on natural convection in thermal performance of electronic products.

Natural Convection and Dimensionless Numbers

When heat is carried by the circulation of fluids, due to buoyancy from density changes induced by heating itself, then the process is known as free or natural convective heat transfer. The heat transfer due to free convection is described by Newton's Law of Cooling,

$$Q = hA(T_w - T_\infty) \qquad (1)$$

The rate of heat Q transferred to the surrounding fluid is proportional to the object's exposed area A, and the difference between the object temperature T_w and the fluid free-stream temperature T_∞. The constant of proportionality h is termed the convection heat-transfer coefficient.

Over the years; it has been found that average free convection coefficient can be represented in the following functional form for a variety of circumstances:

$$Nu_f = C(Gr_f\, Pr_f)^n \qquad (2)$$

Typically, n = 1/4 and 1/3 for laminar and turbulent flows [2]. C is a constant.

The subscript f indicates that the properties (air) in the dimensionless numbers are evaluated at the film temperature.

Where,

$$T_f = \frac{T_\infty + T_W}{2} \tag{3}$$

$$Nu_f = \frac{hL}{k_{fluid}} \tag{4}$$

$$Gr_f = \frac{g\beta L^3 (T_W - T_\infty)\rho^2}{\mu^2} \tag{5}$$

$$Pr = \frac{\mu C_p}{k_{fluid}} \tag{6}$$

In the above equations, all the physical properties are assumed as function of temperature alone. But as discussed, the humidity may play a role.

Rearranging the Nusselt number equation,

$$Nu_{dry\,air} \propto \left(\frac{\rho_d{}^2 C_{pd}}{\mu_d k_d}\right)^n \tag{7}$$

$$Nu_{humid\,air} \propto \left(\frac{\rho_h{}^2 C_{ph}}{\mu_h k_h}\right)^n \tag{8}$$

And the ratio of humid air to dry air is,

$$\frac{Nu_{humid\,air}}{Nu_{dry\,air}} = \left(\frac{\rho_h}{\rho_d}\right)^{2n} \left(\frac{\mu_d}{\mu_h}\right)^n \left(\frac{C_{ph}}{C_{pd}}\right)^n \left(\frac{k_d}{k_h}\right)^n \tag{9}$$

Equation (9) quantifies the effect of humidity over dry air. The following topics describe the procedure to estimate moist (humid) air properties.

Methods for Estimating Moist Air Properties

It is difficult to estimate the exact thermo-physical property values of moist air as it is a mixture of several permanent gases and water vapor. However, the moist air obeys the perfect gas law with accuracy sufficient to engineering calculations up to 3 bar pressure. For higher accuracy, Goff and Gratch tables can be used for estimating moist air properties. These tables are obtained using mixture models based on fundamental principles of statistical mechanics that take into account the real gas behavior of dry air and water vapor. However, these tables are valid for a barometric pressure of 1 bar only. Even though the calculation procedure is quite complex, using the mixture models it is possible to estimate moist air properties at other pressures also. However, since in most cases the pressures involved are low, one can apply the perfect gas model to estimate psychrometric properties.

Basic Gas Laws for Moist Air

The total pressure of a mixture of gases is made up by the sum of the partial pressures of the components in the mixture as known from Gibbs-Dalton's Law of Partial Pressures. According to this law, the total pressure in a mixture of gases can be expressed as:

$$P = \Sigma P_i \tag{10}$$

Applying this equation to moist air,

$$P = P_{tot} = P_{da} + P_{wv} \tag{11}$$

Important Psychrometric Properties

The psychrometric properties can be found in any basic air conditioning text book [3].

<u>(i). Dry bulb temperature (DBT)</u>:

Dry bulb temperature is the temperature of air measured by using a normal thermometer. The dry-bulb temperature is an indicator of heat content.

<u>(ii). Wet bulb temperature (WBT)</u>:

Wet bulb temperature is associated with the moisture content of the air. Wet bulb temperature can be measured with a thermometer that has the bulb covered with a watermoistened bandage with air flowing over the thermometer. Wet bulb temperatures are always lower than dry bulb temperatures but they will be identical with 100% relative humidity in the air.

(iii). Dew point temperature (DPT):
Dew point is the temperature at which water vapor starts to condense out of the air, the temperature at which air becomes completely saturated. Above this temperature, the moisture will stay in the air.

(iv). Saturated vapor pressure (p_{sat}):
It is the pressure of a vapor in equilibrium with its non-vapor phases. At any given temperature, for a particular substance, there is a pressure at which the gas of that substance is in dynamic equilibrium with its liquid or solid forms. This is the vapor pressure of that substance at that temperature.

The following equations are from the ASHRAE (American Society of Heating, Refrigeration and Air-Conditioning Engineers) Handbook of Fundamentals [4].

$$\ln(p_{sat}) = \frac{C_1}{T} + C_2 + C_3T + C_4T^2 + C_5T^3 + C_5T^4 + C_6T^5 + C_7\ln(T) \quad (12)$$

The regression coefficients C_1 to C_7 are given by:

Constants	Over ice (-100°C < T < 0°C)	Over water (0°C < T < 200°C)
C_1	-5.67E+03	-5.80E+03
C_2	-5.15E-01	-5.52E+00
C_3	-9.68E-03	-4.86E-02
C_4	6.22E-07	4.18E-05
C_5	2.07E-09	-1.45E-08
C_6	-9.48E-13	0
C_7	4.163502	6.545967

(v). Relative humidity (Φ):
Relative humidity is the ratio of the water vapor pressure (P_{wv}), to the vapor pressure of saturated air at the same temperature (P_{sat}), expressed as a percentage. Relative humidity is a relative measure. The moisture-holding capacity of air increases with air temperature. In practice, relative

humidity, indicates the moisture level of the air compared to the airs moisture-holding capacity. Relative humidity is normally expressed as a percentage. Thus, for saturated air, Φ is 100%.

$$\phi = \frac{P_{wv}}{P_{sat}} \tag{13}$$

(vi). Humidity ratio (W):

It is the ratio of the actual mass of water vapor present in moist air to the mass of the dry air. It can also be expressed with the partial pressure of water vapor,

$$W = 0.622 \frac{p_{wv}}{p_{tot} - p_{wv}} \tag{14}$$

(vii). Specific heat of humid air (C_{ph}):

The specific heat of humid air can be calculated using [5]:

$$C_{pm} = \left(\frac{1+1.858W}{1+W}\right) \tag{15}$$

(viii). Specific volume of humid air (v):

Specific volume is defined as the total volume of dry air and water vapor mixture per kg of dry air. From the perfect gas equation

$$v = \frac{R_a T}{P_a} = \frac{R_a T}{P_{tot} - P_{wv}} \quad m^3/\text{Kg dry air} \tag{16}$$

Where R_a is specific gas constant = 287.05 J/Kg K.

The specific heat, thermal conductivity and dynamic viscosity of humid air can be calculated as described by J. Zhang et al. [6].

(ix). Dynamic viscosity of humid air (μ_h):

The dynamic viscosity of humid air can be calculated using [7]:

$$\mu_h = \sum_{i=1}^{2} \frac{X_i \mu_i}{\sum_{j=1}^{2} X_j \varphi_{ij}} \tag{17}$$

Where,

$$\varphi_{ij} = \frac{1}{\sqrt{8}}\left(1+\frac{M_i}{M_j}\right)^{-1/2}\left[1+\left(\frac{\mu_i}{\mu_j}\right)^{1/2}\left(\frac{M_j}{M_i}\right)^{1/4}\right]^2 \tag{18}$$

M_{da}=28.97 mole and M_{wv}=18.015 mole

$$\mu_i = \mu_{0,i}\left(\frac{T}{T_0}\right)^{n_{\mu,i}} \tag{19}$$

Power law parameters for viscosity [8]

	μ_0 (Ns/m²)	T_0 (k)	n_μ (-)
μ_{da}	1.716×10^{-5}	273	0.666
μ_{vw}	1.12×10^{-5}	350	1.15

The mole fraction of dry air and water vapor may be found using,

$$X_{da} = \frac{P_{tot} - \varphi P_{sat}}{P_{tot}} \tag{20}$$

$$X_{wv} = \frac{\varphi P_{sat}}{P_{tot}} \tag{21}$$

(x). Thermal conductivity of humid air (k_h):

The thermal conductivity of humid air can be calculated using [9]:

$$k_h = \sum_{i=1}^{2} \frac{X_i k_i}{\sum_{j=1}^{2} X_j \varphi_{ij}} \tag{22}$$

Where,

$$k_i = k_{0,i}\left(\frac{T}{T_0}\right)^{n_{k,i}} \tag{23}$$

Power law parameters for thermal conductivity [8]

	k_0 (W/mK)	T_0 (k)	n_k (-)
k_{da}	0.0241	273	0.81
k_{vw}	0.0181	300	1.35

Steps to calculate the moist air properties

1. Calculate P_{sat} using Equation 12 for the given dry bulb temperature,
2. Calculate P_{wv} using Equation 13 for the given relative humidity,
3. Calculate W using Equation 14 using the above P_{wv} value,
4. Calculate Cp of humid air using Equation 15,
5. Calculate v of humid air using Equation 16. Inverse of v will give the humid air density.
6. Calculate μ of humid air using Equation 17-21.
7. Calculate k of humid air using Equation 18, 20-23.

Results and discussion

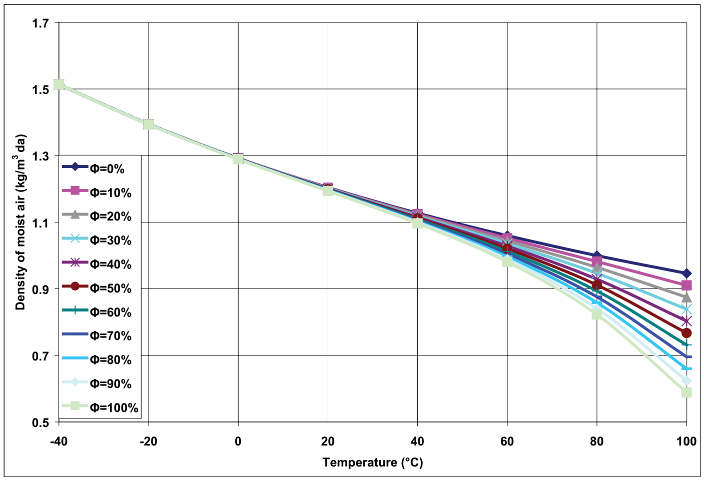


Figure 1. Density of moist air for different relative humidity and temperature

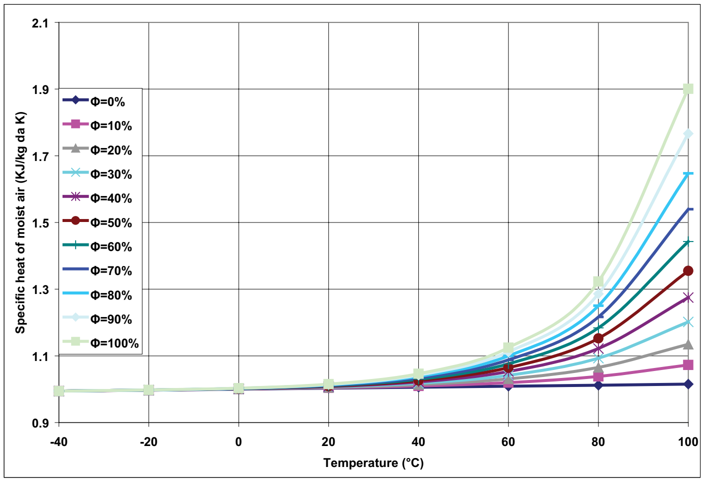


Figure 2. Specific heat of moist air for different relative humidity and temperature

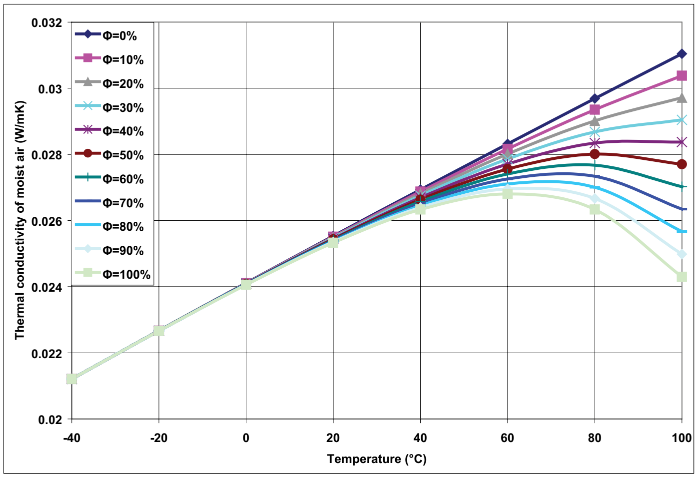


Figure 3.Thermal conductivity of moist air for different relative humidity and temperature

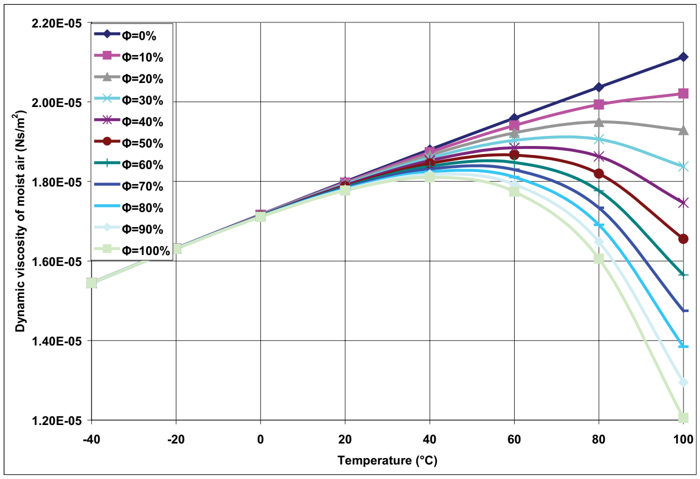


Figure 4. Dynamic viscosity of moist air for different relative humidity and temperature

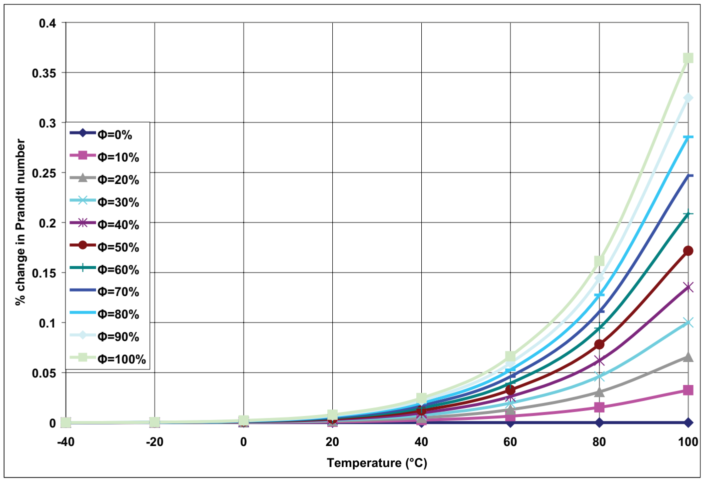


Figure 5. Percent change in Prandtl number for different relative humidity and temperature

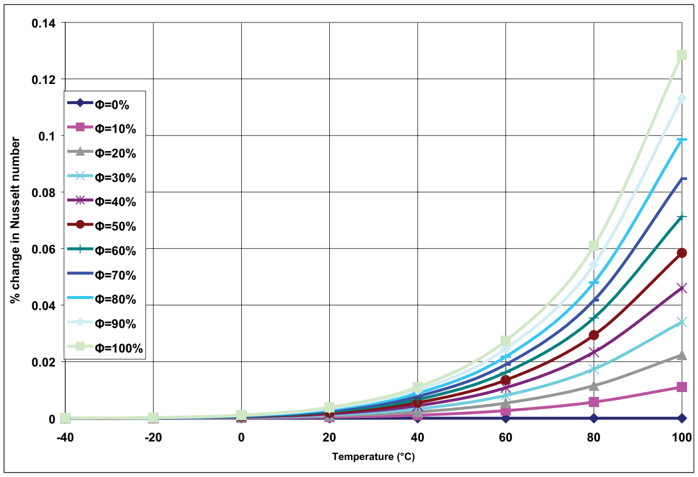


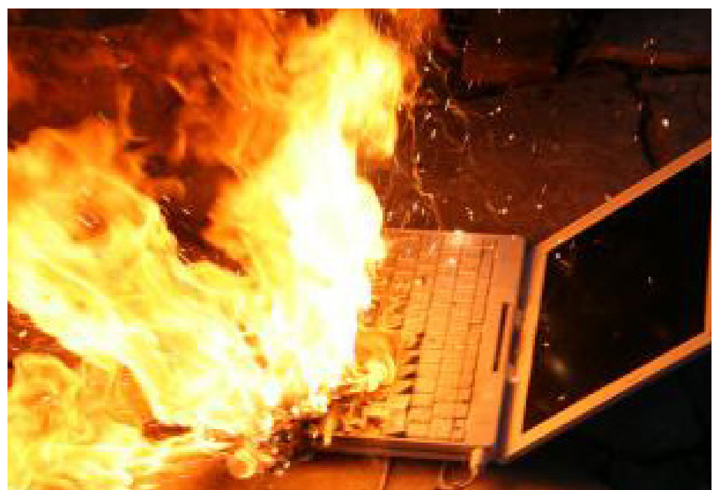
Figure 6. Percent change in Nusselt number for different relative humidity and temperature

The percent change in Prandtl number is calculated as and the same, way the % change in Nusselt number also calculated.

Figures 1 to 6 show the effect of relative humidity on fluid properties. As expected, it is found that relative humidity has a significant effect on Nusselt and Prandtl numbers, especially at high temperature. In general, it is observed that as the relative humidity increases, the Nusselt and Prandtl numbers increase. The density of humid air decreases with the increase in relative humidity, as the mass fraction of water vapor is less than that of air. Also, the decrease in density of the mixture increases the convection currents producing a larger Nusselt number. The specific heat of humid air increases with an increase in humidity. As humidity increases, the water content in the mixture increases which in turn increases the specific heat of the mixture. The thermal conductivity of the humid air decreases with an increase in moisture content as the thermal conductivity of water vapor is less than dry air and with an increase in temperature the mole fraction of water vapor and the thermal conductivity of dry air increases. The dynamic viscosity of the humid air decreases with increase in relative humidity, as the water content in the mixture increases with increase in humidity. Recall that the above fluid properties constitute the prandtl and Nusselt number. So the Prandtl and Nusselt number increases with an increase in relative humidity and temperature.

Conclusions

The effect of relative humidity and surface temperature on the Nusselt number was studied using simple Nusselt number correlation and moist air properties. The relative humidity was varied from 0 to 100% and the temperature



was changed from -40°C to 100°C. It is observed that, variation of relative humidity and temperature resulted in a change of thermo-physical properties and hence increased the Nusselt number in the order of 13%, compared to dry air in laminar flow conditions for 100°C, 100% humid air. The effect of humidity is significant at higher temperatures, as in the case of automotive electronics which experiences an ambient close to 100°C.

Nomenclature:

Q	Heat transfer rate
T_w	Wall temperature
T_∞	Free stream temperature
Nu_f	Nusselt number at film temperature
Pr	Prandtl number
Gr	Grashof number
L	Characteristic length
K_{fluid}	Thermal conductivity of fluid
g	Acceleration due to gravity
β	Volumetric thermal expansion coefficient = $1/T_f$
ρ	Density
μ	Dynamic viscosity
C_p	Specific heat capacity
P	Pressure of the mixture
P_i	Partial pressure of gas i
P_{tot}	Total barometric pressure
P_{da}	Partial pressure of dry air
P_{wv}	Partial pressure of water vapor
p	Saturated vapor pressure of water [Kpa]
K^{sat}	Temperature [K]
M	Molar mass, mole

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Ashok Kumar serves as a thermal specialist in the automotive electronics group at BOSCH. He is responsible for thermal analysis of Automotive Electronics components both at package as well as system level. Prior to joining BOSCH, he worked as a research scientist in Electronics Packaging division at SAMEER-CEM, Chennai. His interests are in the area of phase change heat transfer and package thermal-mechanical design. Ashokkumar received his B.E in Mechanical Engineering from Barathiyar University, Coimbatore, India in 1998 and M.E in Refrigeration and Air conditioning from Anna University, Chennai, India in 2004.

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