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Technical Notes

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Transient Condensation During Rapid Aircraft Descent

Michel Engelhardt*
BAE Systems, Inc., Greenlawn, New York 11740

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Nomenclature

A = altitude
C = specific heat
DR = descent rate
$h_{\rm in}$ = convective heat transfer coefficient inside the bay
k = thermal conductivity
L = window thickness in z direction
M = aircraft Mach number
m_a = mass of air in bay
m_v = mass of water vapor in bay
P = total air/water-vapor mixture pressure in bay
P_a = air partial pressure in bay
$P_{\rm amb}$ = ambient air pressure
P_g = saturation vapor pressure
P_v = vapor partial pressure in bay
$\ddot{q}$ = heat generation
RH = relative humidity
R_a = gas constant for air
R_v = gas constant for water vapor
s = speed (velocity) of air in the bay
T = window temperature
T_a = air/water-vapor temperature in the bay
$T_{\rm amb}$ = ambient air temperature
$T_{\text{win-in}}$ = inside window temperature
$T_{\text{win-o}}$ = outside window temperature
t = time
V = volume
z = z direction
α = thermal diffusivity
ρ = air density
ω = humidity ratio in bay
ω_T = humidity ratio near cold surface temperature

Subscripts

a-AVBL = air in the air/water-vapor boundary layer
T-AVBL = temperature in the air/water-vapor boundary layer

I. Introduction

THIS paper describes an analytical model to predict the transient condensation in a tactical aircraft unpressurized equipment bay. Condensation within an aircraft equipment bay is a function of an aircraft's flight profile and the local outside atmospheric temperature, pressure, and humidity. As the aircraft takes off and climbs, the air in the equipment bay leaks out of the unpressurized bay to the lower outside air pressure. When the aircraft reaches its operational ceiling, the equipment bay pressure is slightly above the local altitude atmospheric pressure due to the slightly higher boundary-layer air pressure and internal air movers (fans/blowers).

As the aircraft descends to a lower altitude, the atmospheric air pressure increases. Upon descent, the outside air starts to enter the equipment bay until the aircraft either levels off at an altitude or lands. As humid air enters into an equipment bay, it condenses onto surfaces for which the temperatures are below the dew point of the moist air. For many types of military electronic equipment and components, condensation is not desirable and must be controlled. Determining the effectiveness of condensation control management systems in airborne equipment bays requires a transient condensation model [1].

II. Transient Condensation Model

There are five interconnected parts to the transient condensation model. Part 1 is the transient temperature of the equipment located in the aircraft bay. Parts 2 and 3 are the transient aircraft descent rate and the atmospherics, respectively. Part 4 is the transient humidity present in the equipment bay air. Part 5 is the condensation that forms on surfaces. These model parts are integrated as the transient condensation model. The interconnected parts of the transient condensation model are described in the following paragraphs.

Part 1: Transient Equipment Temperatures. The transient temperature of electronic equipment located in an aircraft bay is obtained analytically [2]. The following equations provide an analytical model for computing the transient window surface temperature; however, any surface within the bay can be modeled [2].

The transient temperature of a heated window is obtained from the general conduction heat transfer equation:

$$\nabla^2 T + \frac{\ddot{q}}{k} = \frac{1}{\alpha} \cdot \frac{\partial T}{\partial t} \tag{1}$$

where

$$\alpha = k/\rho \cdot C \tag{2}$$

For a window surface with heat generation (e.g., electrical heating) and a one-dimensional heat transfer into the window in the z direction, the two boundary conditions (BC 1 and BC 2) and initial condition (IC) are as follows:

BC 1:

$$T(z=0) = T_{\text{win-o}}(t) = T_{\rm amb}(t) \cdot [1 + 0.17 \cdot M(t)^2]$$

BC 2:

$$k \cdot \frac{{\rm d}T(z=L)}{{\rm d}z} = h_{\rm in}(t) \cdot [T(z=L) - T_{\text{win-in}}(t)]$$

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*Senior Principal Engineer II, Mechanical Engineering, 450 Pulaski Road, Mail Stop 01-E10; michel.engelhardt@baesystems.com.

IC:

$$T(t=0) = T_{\mathrm{amb}}(t=0)$$

In BC 1, the aerodynamic recovery air temperature is used as the outside window temperature. For subsonic air flows, the aerodynamic heating on an external window corresponds to the outside boundary-layer recovery air temperature. In BC 1, both the local ambient air temperature [3] and the Mach number are functions of time and altitude. In BC 2, the internal heat transfer coefficient is a function of time and altitude. This is due to the decrease or increase in air density as the aircraft ascends or descends.

In addition, the air temperature within the equipment bay can change to simulate the increase in bay air temperature due to heating the air while descending. While descending, heating the air decreases the relative humidity (RH) in the air and reduces condensation on the window and other critical surfaces that are located in the equipment bay.

Equation (1) is solved by the classical Duhamel superposition integral [4]. However, spreadsheet numerical solutions provide a more flexible means of solving this complex problem. This is due to the added complexity of accounting for the transient effect of RH and condensation as a function of altitude.

Parts 2 and 3: Aircraft Descent Rate and Atmospherics. As the altitude changes, the outside air temperature, pressure, and RH also change and affect the equipment bay air temperature, pressure, RH, condensation rate, and heat dissipation. The aircraft descent rate (DR) is defined as follows:

$$\mathrm{DR} = -\frac{\mathrm{d}A}{\mathrm{d}t} \tag{3}$$

The outside ambient air temperature and pressure are a function of altitude [1,3].

Part 4: Transient Humidity. Transient humidity equations are derived from first principles of thermodynamics for use in predicting the transient behavior of the air/water-vapor mixture during descent into a humid atmosphere. This paper presents two methods: method 1 for when the transient RH is known, and method 2 for when the transient humidity ratio is known.

Method 1 is used when an RH sensor is provided in the equipment bay or when the RH is defined within the bay. For a defined descent rate, Eq. (3), the RH is time dependent due to the change in the altitude and the ingestion of atmospheric humidity into the bay. Assuming that the air and vapor temperatures are in equilibrium at each altitude, the vapor pressure is obtained from the RH and the saturation pressure, P_g:

$$\frac{\partial P_v}{\partial t} = P_g(t)\cdot\frac{\partial(\mathrm{RH})}{\partial t} + \mathrm{RH}(t)\cdot\frac{\partial P_g}{\partial t} \tag{4}$$

The dew point temperature is the saturation temperature corresponding to the vapor pressure. Once the vapor pressure is obtained from Eq. (4), the dew point temperature is calculated from the following derived equation [5]:

$$\frac{\partial T_{\mathrm{dp}}}{\partial t} = 16.44\cdot\frac{\partial(\ell n\,\mathrm{RH})}{\partial t} + 0.995\cdot\frac{\partial T_a}{\partial t} \tag{5}$$

The dew point temperature formula, Eq. (5), is valid for an air/water-vapor-mixture temperature, T_a, between 5 and 45°C.

As the aircraft descends, an unpressurized bay reaches pressure equilibrium with its local altitude pressure within seconds. Consequently, it is a good engineering approximation to assume that the bay internal pressure is at the outside air pressure (plus any additional pressure change due to fans or nitrogen purging). The transient air partial pressure is then obtained by extending Dalton's Partial Pressure Rule to the transient domain:

$$\frac{\partial P_a}{\partial t} = \frac{\partial P}{\partial t} - \frac{\partial P_v}{\partial t} \tag{6}$$

The transient altitude-dependent humidity ratio [1] is derived from the ideal gas law and the definition of the humidity ratio:

$$\frac{\partial \omega}{\partial t} = \frac{R_a}{R_v\cdot P_a(t)}\cdot\frac{\partial P_v}{\partial t} - \frac{R_a\cdot P_v(t)}{R_v}\cdot\frac{\partial(\frac{1}{P_a})}{\partial t} \tag{7}$$

The mass of air is calculated from the transient form of the Ideal Gas Law:

$$\frac{\partial m_a}{\partial t} = \frac{V}{R_a\cdot T_a(t)}\cdot\frac{\partial P_a}{\partial t} + \frac{V\cdot P_a(t)}{R_a}\cdot\frac{\partial(\frac{1}{T_a})}{\partial t} \tag{8}$$

The transient mass of water vapor is then calculated from the humidity ratio and the mass of air:

$$\frac{\partial m_v}{\partial t} = \omega(t)\cdot\frac{\partial m_a}{\partial t} + m_a(t)\cdot\frac{\partial(\omega)}{\partial t} \tag{9}$$

Method 2 is used when the outside humidity ratio [1] is known. Two types of conditions can be applied. The first is to assume ingress of all the outside humidity into the unpressurized bay; the second condition accounts for only a portion of the humidity ingress into the bay due to absorption by desiccant or due to condensation/drainage in a heat exchanger followed by reheat. When the humidity ratio is known, the RH is obtained from the following expression:

$$\frac{\partial(\mathrm{RH})}{\partial t} = \frac{R_v}{R_a}\cdot\left[\frac{P_a(t)}{P_g(t)}\cdot\frac{\partial\omega}{\partial t} + \frac{\omega(t)}{P_g(t)}\cdot\frac{\partial P_a}{\partial t} + \omega(t)\cdot P_a(t)\cdot\frac{\partial(\frac{1}{P_g})}{\partial t}\right] \tag{10}$$

The equations provided in method 1 are used with Eq. (10) to compute all the condensation parameters.

Part 5: Condensation Rate. If the air/water-vapor mixture comes into contact with a surface for which the temperature is below the dew point, then condensation occurs. Equation (1) provides the temperature, T, of an internal bay surface (e.g., a window), and Eq. (5) provides the dew point, T_{dp}. If $T < T_{\mathrm{dp}}$, then condensation occurs.

Assume an air/water-vapor boundary layer (AVBL) close to the equipment surface. As a conservative assumption, the AVBL is at the equipment surface temperature whereas the bulk air/water-vapor mixture, which flows over the equipment surface, is at the bay air temperature. Also, assume that condensation forms on the equipment surface from the AVBL, assuming a quasi-constant-pressure process during each time step.

The saturated mixture at the window internal temperature, T, is expressed as

$$\left.\frac{\partial P_v}{\partial t}\right)_T = \left.\frac{\partial P_g}{\partial t}\right)_T \tag{11}$$

The AVBL humidity ratio is determined from Eqs. (6) and (7); however, the vapor pressure is determined based on the cold equipment surface temperature, Eq. (11). The humidity ratio determines how much water vapor is held in the AVBL. The AVBL humidity ratio is then obtained from the following equations:

$$\frac{\partial P_a}{\partial t} = \frac{\partial P}{\partial t} - \left.\frac{\partial P_v}{\partial t}\right)_{T-\mathrm{AVBL}} \tag{12}$$

$$\left.\frac{\partial\omega}{\partial t}\right)_{T-\mathrm{AVBL}} = \frac{R_a}{R_v\cdot P_a(t)}\cdot\left.\frac{\partial P_v}{\partial t}\right)_{T-\mathrm{AVBL}} - \frac{R_a\cdot P_v(t))_{T-\mathrm{AVBL}}}{R_v}\cdot\frac{\partial(\frac{1}{P_a})}{\partial t} \tag{13}$$

The amount of water vapor that condenses on the equipment surface is equal to the difference between the bulk humidity ratio, Eq. (7), and the humidity ratio, Eq. (13), due to cooling the air/water-vapor mixture within the AVBL. Based on this difference, the following condensed mass of water is then determined:

$$\left.\frac{\partial m_v}{\partial t}\right)_{T-\mathrm{AVBL}} = (\omega(t) - \omega_{T-\mathrm{AVBL}}(t)) \cdot \frac{\partial m_{a-\mathrm{AVBL}}}{\partial t} + m_{a-\mathrm{AVBL}}(t) \cdot \frac{\partial(\omega - \omega_{T-\mathrm{AVBL}})}{\partial t} \tag{14}$$

The mass of air in the AVBL is a function of the volume of the air/water-vapor mixture close to the surface of the equipment. To determine the volume of air in the AVBL (for use in computing the mass of air in the AVBL), consider a 0.5-in.-thick AVBL over the equipment surface. The volume of the AVBL is then computed from the surface area and the thickness of the AVBL. When the air/water-vapor mixture flows through a heat exchanger, the cold plate spacing is typically narrow; therefore, the entire volume of air that flows through the heat exchanger is considered. In the case of humid airflow through a heat exchanger, the bulk air/water-vapor mixture and the AVBL are modeled as one. The aforementioned method is numerically solved [6] on a spreadsheet.

III. Conclusions

A transient condensation model is provided to determine the effectiveness of condensation control management systems in airborne equipment bays. The five-part transient condensation model determines the condensation in aircraft equipment bays during aircraft dive and descent mission legs. The transient condensation model includes the aircraft descent rate, the equipment bay air transient RH, and the transient condensation rate on cold surfaces. The significant aspect of the transient condensation model is that it accounts for the transient change in partial pressure during descent and humidity profiles as a function of altitude.

References

[1] Engelhardt, M., "Aircraft Equipment Bay Transient Condensation Model While Descending into a Humid Atmosphere," AIAA Paper 2008-1296, Jan. 2008.

[2] Engelhardt, M., "Thermal Control of an Airborne Electronics Bay," AIAA Paper 2007-1217, Jan. 2007.

[3] Anon., "Climatic Information to Determine Design and Test Requirements for Military Systems and Equipment," Department Of Defense Standard MIL-STD-210C, June 1997.

[4] Aparci, V. S., *Conduction Heat Transfer*, Addison–Wesley, Reading, MA, 1966.

[5] Keenan, J. H., and Keyes, F. G., *Thermodynamic Properties of Steam*, Wiley, New York, 1936.

[6] Akai, T. J., *Numerical Methods*, Wiley, New York, 1994, ISBN 0-471-57523-2.