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FUNDAMENTALS OF SIMULTANEOUS HEAT AND MOISTURE TRANSFER BETWEEN THE BUILDING ENVELOPE AND THE CONDITIONED SPACE AIR

S.P.W. Wong
Fellow ASHRAE

S.K. Wang
Member ASHRAE

ABSTRACT

Study of moisture transfer between the building envelope and the conditioned space air is often focused on the moisture transfer due to the difference in water vapor pressure on the two sides of the building envelope. In actual practice, the influence of transient heat and moisture transfer between the building envelope and the space air during the cooldown, conditioning, and off period must be considered.

The results of site surveys have verified a considerable amount of space latent cooling load induced by the moisture transfer from the building envelope to the space air during the cooldown period. This paper discusses moisture migration within solids and proposes a physical model to describe the moisture transfer and a mathematical model to evaluate it.

This paper also investigates the empirical formulas and the dimensionless correlations that determine the convective heat transfer coefficient during forced convection and free convection. Also presented is the principle and a method of evaluating various parameters in the convective mass transfer. Finally, this paper proposes a mathematical model of a simultaneous heat and moisture transfer relationship and suggests a method to solve problems in this field.

OPERATING MODES OF NIGHTTIME SHUTDOWN IN AIR-CONDITIONED OFFICES

Most of the air systems in office buildings are shut down at night. Nighttime shutdown operation means that the air system will be operated primarily when the office is occupied. The operating period of the air system is usually shorter than a 24-hour cycle.

Figures 1 and 2 show the space temperature and the relative humidity measured and recorded by a thermo-hygrograph in a staff office of a Hong Kong educational institution during January and August 1984. The office has a total area of about 25 m^2 (269 ft^2), external windows 1.15 m (3.77 ft) high, and an external concrete wall 150 mm (5.9 in) thick.

A primary air fan coil unit system was installed for this office. A pneumatic thermostat modulates the water flow to the fan coil to maintain the required temperature. The fan coil is usually started at 8 a.m. and shut off at 10 p.m. after evening classes. Sometimes primary air may be supplied earlier.

In an office building, the operating modes of an air system that is operated intermittently can be divided into two categories, summer mode operation and winter mode operation.

Summer Mode Operation

In summer mode operation, both the space temperature and relative humidity decrease after the air handler or fan coil starts. Both the space temperature and the relative humidity increase when the air handler or fan coil ceases to operate. The reasons for the increase in humidity are related to moisture transfer to the space air from various external sources, as listed below:

- moisture transfer by the infiltrated hot and humid outdoor air through the elevator shafts, pipe shafts, and other vertical passages;
- moisture transfer by the outdoor infiltrated air through window cracks and gaps;
- moisture transfer from the wetted surface of the components in the air handler or fan coil through the return or supply ducts; and
- moisture transfer from the external wall.

Winter Mode Operation

In winter mode operation, space air temperature increases when the air system supplies warm air to the air-conditioned space. As the intake of outdoor air into the air system is at a low humidity ratio, the space relative humidity often drops to a value lower than 45%.

When the air handler or the fan coil ceases to operate, the space temperature drops due to the transmission loss. At the same time, the space relative humidity increases because the space temperature decreases.

From the point of view of space occupancy, the 24-hour cycle can be divided into an occupied period and an unoccupied period, as shown in Figures 1 and 2. On the other hand, for evaluating the operation of the air systems, the diurnal cycle is better divided into three periods:

(i) cooldown or warmup period,
(ii) conditioned period, and
(iii) off period.

The combination of the cooldown or warmup period and the conditioned period is called the operating period.

MOISTURE TRANSFER FROM THE BUILDING ENVELOPE

Study of moisture transfer between the building envelope and the conditioned space air is often focused on the moisture transfer across the building envelope induced by the difference

Samuel P.W. Wong is the Managing Partner of Associated Consulting Engineers, Hong Kong. **S.K. Wang** retired from Hong Kong Polytechnic and is now a Consulting Engineer.

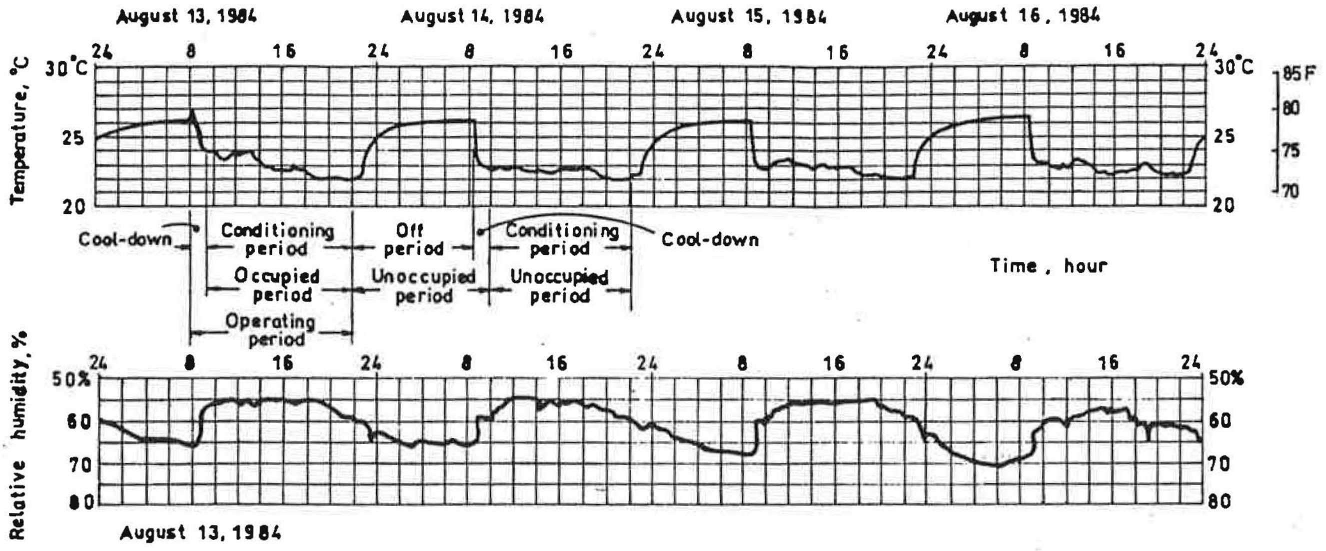

Figure 1 *Temperature and relative humidity curves for a staff office when the fan coil unit is shut down at night during summer*

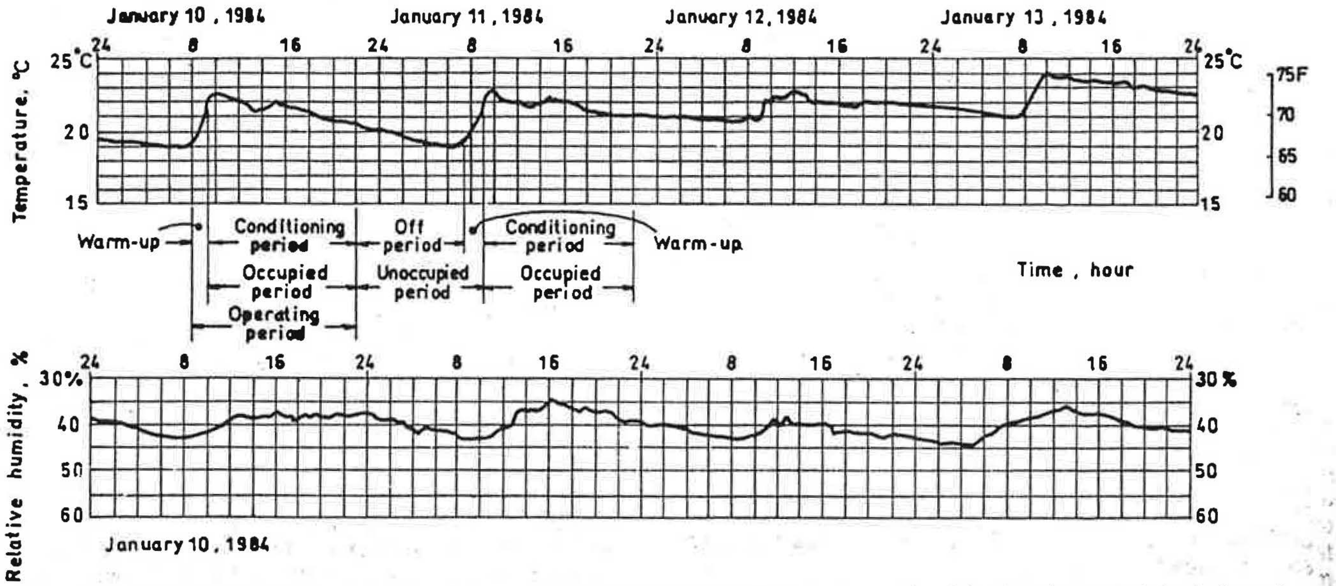

Figure 2 *Temperature and relative humidity curves for a staff office when the fan coil unit is shut down at night during winter*

in water vapor pressure on the two sides of the building envelope. Some of the literature overlooks the influence of moisture transfer from the space air to the building envelope and the included furnishings during the off period in the hot summer induced by the higher space relative humidity and temperature. There is also a moisture transfer in the opposite direction, that is, from the building envelope and furnishings to the space air during the operating period induced by the comparatively lower space relative humidity and temperature, as shown in the upper left corner of Figure 3.

Figure 3 shows the condition of space air and supply air and also the condition of air after it has gone through the supply fan in one of the air handlers installed in the Reserved Book Section of a library. The "state" points plotted on the psychrometric chart were determined from the measured dry-bulb and wet-bulb temperatures on August 8, 1977, from 8:20 a.m. to noon. All readings were measured from the extracted air by using aspiration psychrometers. In the upper left corner of Figure 3, the space air temperature and relative humidity were measured and recorded on the same day by a thermo-hygrograph. On the psychrometric chart of Figure 3, the dots represent the "state" points of the space air, the crosses the supply air, and the triangles the air through the supply fan in the air-handling unit. The coarser dots, crosses, and triangles indicate the condition of air when the air-handling unit had just been started at 8:30 a.m., and the thinner ones represent the "state" points nearer to noon.

The upper part of Figure 4 shows the difference in humidity ratio between the space air and the supply air in the Reserved Book Section. The middle part shows the space latent load, q_{rl}, and the ratio of space latent load to the cooling load, q_{rl}/q_{rc}. The bottom part shows the supply volume flow rate.

The Reserved Book Section was equipped with a variable-air-volume (VAV) system. The supply volume flow rate decreases gradually as the space temperature is reduced.

On May 17 and 18, 1988, the dry- and wet-bulb temperatures of the supply air at the slot diffuser outlet in the elevator lobby and at the return inlet of the air-conditioning equipment room were measured by the same aspiration psychrometer on one of the typical floors of a high-rise office building in Hong Kong. The conditioned area of this typical floor is 1120 m^2 (12,051 ft^2). All external windows have a height of 2.4 m (7.87 ft). The external concrete wall forms the external envelope of the ceiling plenum. A dual-duct VAV system with a single supply fan is used in this office building.

The calculated latent load during the cooldown period in the Reserved Book Section is about 18 kW (61,434 Btu/h). The calculated latent load for the typical floor of the high-rise building

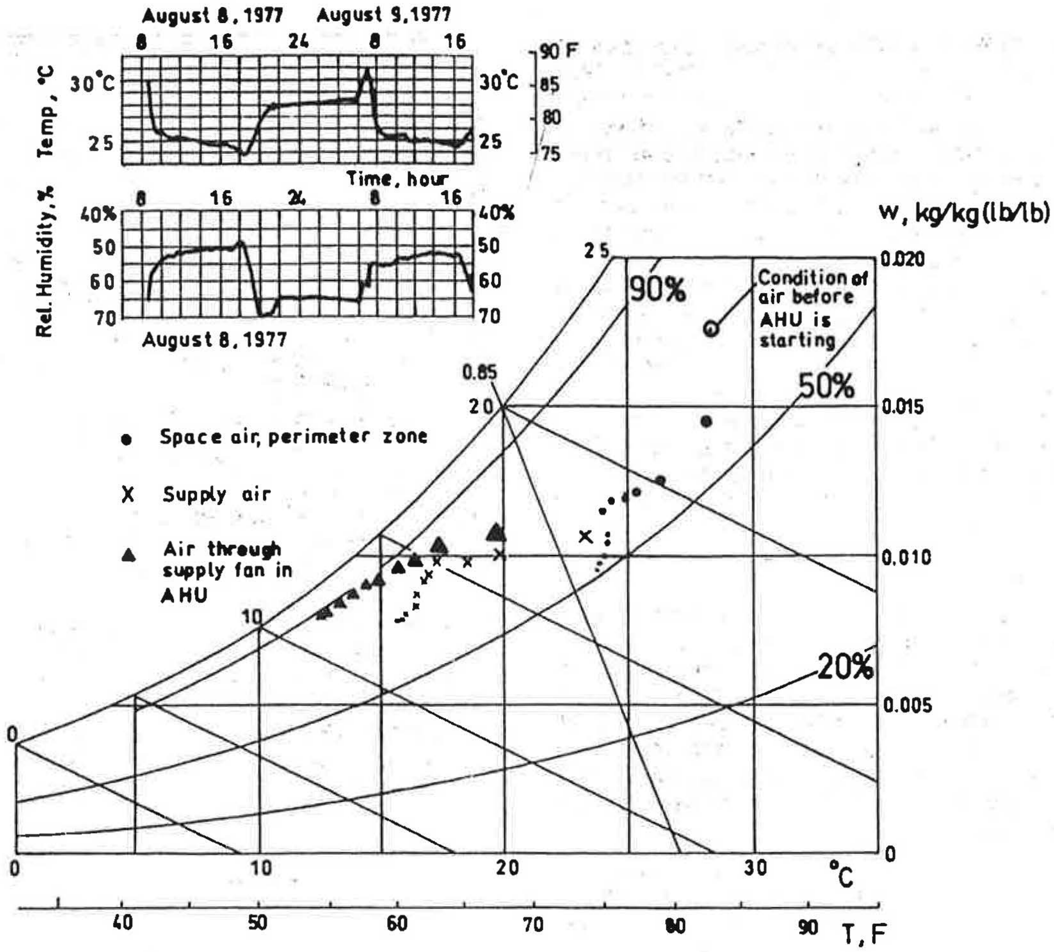

Figure 3 *Condition of space air, supply air, and air through supply fan in a library*

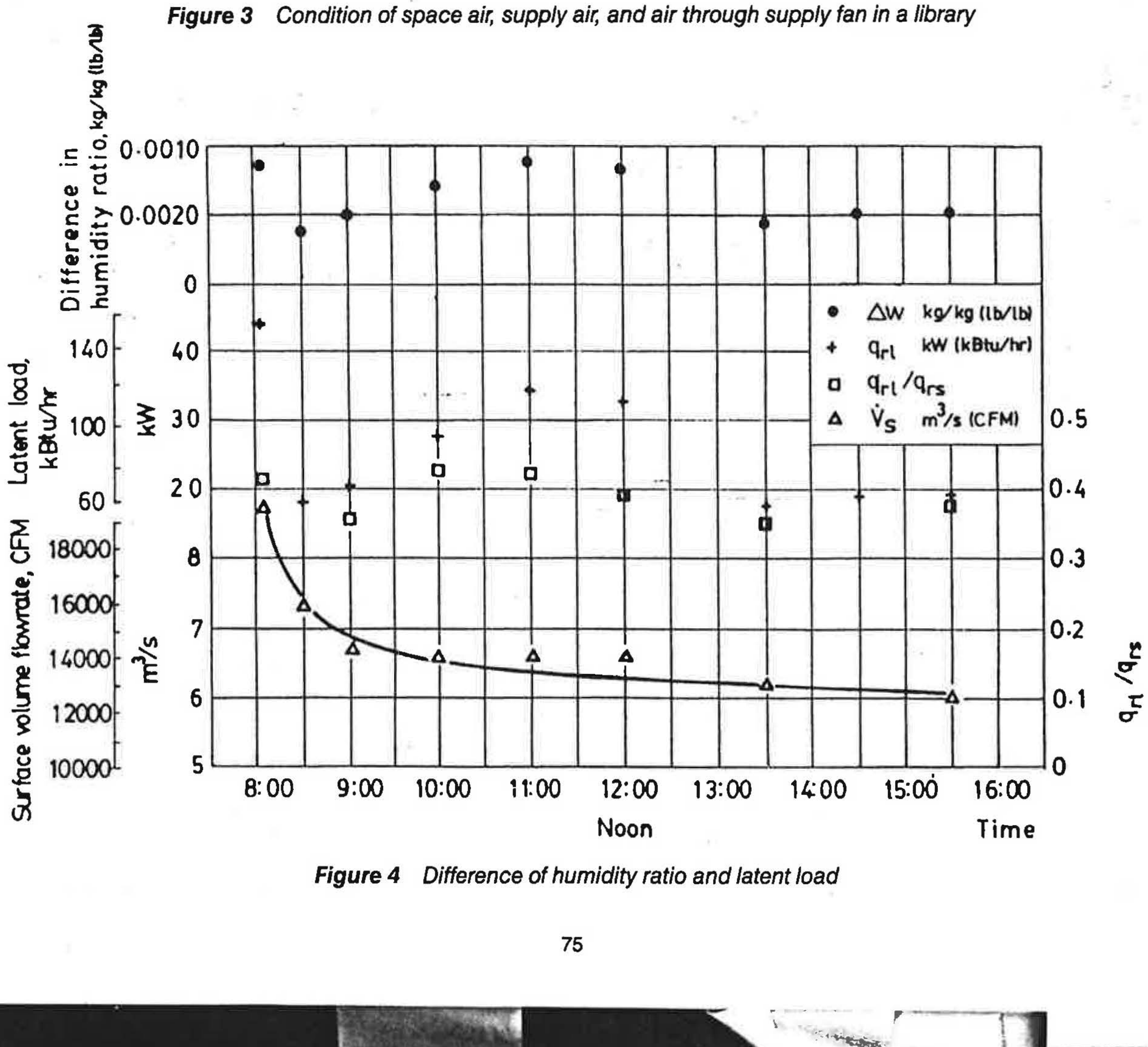

Figure 4 *Difference of humidity ratio and latent load*

is about 29 kW (98,977 Btu/h). The latent loads released by the occupants in both cases were less than 1.5 kW (5120 Btu/h). A test showed that positive pressure was maintained in the conditioned spaces during operating periods, that is, infiltrated air was negligible and, also, water surfaces were not found either in the Reserved Book Section or on the typical floor of the high-rise building. Therefore, a significant amount of latent load was induced by the moisture transfer from the building envelope to the conditioned space air.

Kusuda (1983) pointed out there is an apparent difference between the measured and the calculated indoor relative humidity if the associated moisture absorption and desorption phenomena are not considered. Fairey and Kerestegioclu (1985) pointed to the significant impact of moisture absorption and desorption of building materials on performance.

Moisture transfer from the building envelope and furnishings not only increases the space cooling load but also appreciably lowers the sensible heat factor of the space-conditioning process.

Three Types of Media

The term "moisture" indicates water in the form of vapor, liquid, and solid. Many building materials have a large number of interstices and pores that may or may not be interconnected. Moisture may be physically bound or mechanically attached to the surface of the interstices and the pores in which the vapor pressure of water is reduced. Special facilities are sometimes required to drive off the moisture from these surfaces. When evaluating moisture transfer, building materials can be classified into three types:

1. Capillary-porous media. There is a clearly recognizable pore space. The amount of physically bound water is negligible, that is, the medium is nonhygroscopic. The medium does not shrink when moisture is being driven from the medium. The enthalpy of wetting is negligible. Examples of this group are packings of sand, crushed minerals, polymer particles, bricks, and concrete.
2. Hygroscopic-porous media. There is also clearly recognizable pore space. This type of medium has a significant amount of physically bound liquid. When moisture is being driven from the surface of large pores, shrinkage often occurs. Enthalpy of wetting is significantly higher. Examples of this group are clay, wood, and natural textile fibers.
3. Colloidal (nonporous) media. There is no pore space. Evaporation of liquid water can take place only at the surface. All liquid water is physically bound. Examples of this group are nylon, glue, etc.

Sorption Isotherm

Moisture content is defined as the ratio of the mass of the moisture contained in the solid to the mass of the bone-dry solid.

A sorption isotherm is a constant temperature curve showing the moisture content of a material surrounded by ambient air at various relative humidities during an equilibrium condition, that is, the rate of condensation of water vapor onto the surface of the material is equal to the evaporation of water vapor from its surface.

Figure 5 shows the sorption isotherms for timber at temperatures between 20°C and 80°C (68°F and 176°F). When the ambient air is at a condition of very low relative humidity, moisture is usually held in a monomolecular layer bonded tightly to the surface of the finest pores and interstices. The binding energy depends on the nature of the surface, the structure of the material, and its chemical links.

At relative humidities between 20% and 80%, moisture is more loosely bound in polymolecular layers. Because the moisture is located in microcapillaries, the vapor pressure is depressed. The energy level is primarily the latent heat of condensation or evaporation. At relative humidities higher than

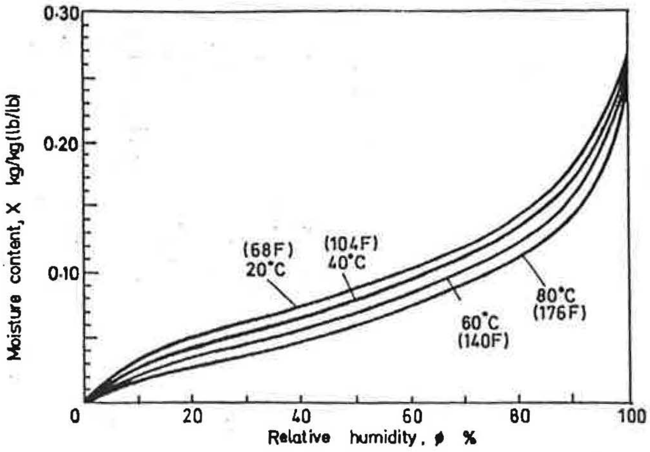

Figure 5 *Sorption isotherms for timber at temperatures between 20°C and 80°C (68°F and 176°F) (after Krisher)*

80%, the moisture is present in the large capillaries and pores and is relatively free for water molecules to leave the surface.

Temperature also has an influence on the moisture content of many materials. The effect of temperature on moisture content is shown in Figure 6. If the relative humidity of the ambient air remains constant, the moisture content will be lower at a higher temperature.

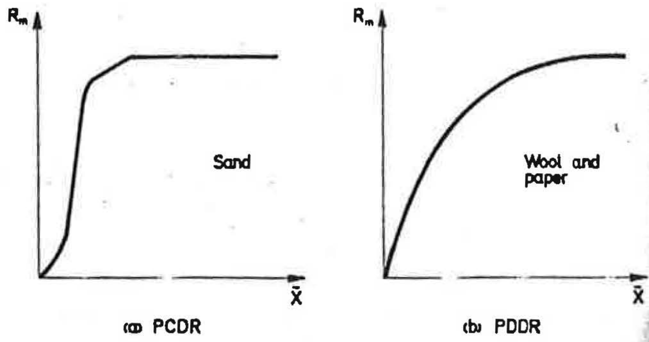

Figure 6 *Drying rate curves*

Many building materials show different absorption and desorption isotherms. The displacement between the absorption or desorption isotherms is called hysteresis. The hysteresis effect for building materials could be neglected for simplicity in this paper.

Heat of Sorption

When moisture is absorbed by a building material, heat is evolved. If liquid water is taken up by the material, energy similar to the heat of solution evolves. If water vapor is absorbed, then the heat evolution is similar to the latent heat of condensation, that is,

$$q_v = q_l + h_{fg}. \tag{1}$$

Strictly speaking, adsorption means moisture absorbed by the building material when the volume of the building material remains unchanged, and absorption describes the process of taking up moisture when the volume is changed, such as in swelling of wood.

Liquid Content and Vapor Content

Because density of liquid water is much greater than water vapor, the moisture content in building materials is usually in the form of liquid water. Consider a building material having a porosity of 0.65 and a density of 650 kg/m³ (40.63 lb/ft³). A moisture content of 0.01 means 0.01 × 650 = 6.5 kg (14.32 lb) of moisture in 1 m^3 (35.34 ft^3) of material. Even if all porous space is filled up with water vapor, its mass is only equal to $m_v = V\rho_{wv} = 0.65 \times 1 \times 0.598 = 0.389$ kg (0.8568 lb), which is a moisture content of only 0.0006. Here, ρ_{wv} is the density of water vapor. If part of the porous space is filled up with air, the mass of the moisture will be less.

THEORIES OF MOISTURE MIGRATION IN SOLIDS

Many theories have been proposed to predict moisture migration within solids. Liquid diffusion is an early theory that considered liquid diffusion as the principal kind of moisture flow in solids. Experiments showed that moisture can flow in response to a vapor pressure gradient and against a moisture concentration gradient. Discrepancies between calculated moisture distribution and experimental values were frequent. The criticism of this theory is mainly due to the assumption that liquid movement is the only mechanism of moisture transfer in solids.

Capillary theory introduced the concept that the capillary potential is the driving force for capillary liquid flowing through the interstices and flowing over the internal surfaces of a solid. In a capillary liquid flow, moisture flow may be possible in the direction of increasing concentration. Unfortunately, capillary theory does not take into account the water vapor flow at low relative humidities in solids.

Luikov, in 1934, proposed the phenomenon of moisture thermal diffusion, and claimed that the temperature gradient is also a factor influencing moisture transfer in solids. Luikov assumed that the fluxes due to vapor diffusion and liquid diffusion consisted of two parts: one due to the concentration gradient and the other due to a temperature gradient. A concept of mass transfer potential is introduced to describe the fact that moisture may possibly transfer from a body with lower concentration to one having a higher concentration.

In 1957, Philips and De Vries also developed equations describing moisture and heat transfer in porous materials under combined moisture and temperature gradients. Their approach is a mechanistic one and assumes that moisture moves in solids by both vapor diffusion and capillarity.

Recently, Matsumoto (1988) recommended the water chemical potential as the moisture flow potential for the multi-layer structure of a wall. The relation between water chemical potential and capillary flow must be clarified before this approach can be applied.

One proposed analysis for the calculation of moisture migration considers the flow to be hydraulic, under the influence of hydrostatic forces when the materials are saturated and a vapor flow produced by vapor pressure differences in unsaturated materials. A simple vapor flow theory is adopted in unsaturated building materials. Such an approach overlooks the fact that the moisture content of building material is mostly liquid water by mass when the relative humidity of the ambient air is greater than 50% and also that the liquid fraction of moving moisture in porous materials has a higher mass at higher moisture contents, according to Luikov (1966). Moreover, the penetration of air through a building material without cracks and pin holes is often negligible because there is a comparatively lower pressure difference. A simple vapor theory cannot explain and evaluate the moisture transfer between building materials and the space air when the air system is operated in the nighttime shutdown mode.

PROPOSED MODEL

The proposed model of moisture flow in building material in a nighttime shutdown air-conditioned space has the basic concepts of Krischer's and Berger and Pei's theories. This model is based on the following assumptions:

1. Liquid flow is induced by capillary flow and concentration gradients; vapor diffusion is induced by vapor pressure gradients.
2. Heat transfer within building materials occurs by conduction through the solid and by latent heat from phase changes.
3. During the transport process, the moisture content, the partial vapor pressure, and the temperature are always in equilibrium at any location of the building material.
4. For moisture contents larger than saturated sorptional content, the vapor pressure is equal to the saturation value.
5. Vapor pressure gradients can be determined from moisture contents by means of sorption isotherms.
6. Fick's law is applicable.
7. All mass and heat transfer coefficients are constants.
8. Only a one-dimensional flow across the building material is considered.
9. Building materials are homogeneous.
10. Airstreams through the building material are negligible.

Mathematical Model of Moisture Transfer

For a one-dimensional moisture flow in a building material, the mass flux, J_m, in kg/s·m^2 (lb/h·ft^2), can be expressed as

$$J_m = -\rho\left(D_{lv}\frac{\partial X}{\partial x} + D_T\frac{\partial T}{\partial x}\right). \tag{2}$$

In a capillary-porous medium, in the absence of osmotic force, the relationship between the liquid water movement and the concentration gradient can be expressed as

$$\frac{J_m}{\rho} = \frac{K}{\rho_l g}\left(\frac{\partial p_c}{\partial x}\right)_T + K\cos\theta = \frac{K}{\rho_l g}\left(\frac{\partial p_c}{\partial \bar{X}}\frac{\partial \bar{X}}{\partial x}\right) + K\cos\theta \tag{3}$$

and, hence, the mass diffusivity,

$$D_{lv} = \frac{K}{\rho_l g}\left(\frac{\partial p_c}{\partial \bar{X}}\right)_T \tag{4}$$

and also, the normalized moisture content is

$$\bar{X} = X - X_e / X_i - X_e. \tag{5}$$

Equations 3 and 4 indicate that the true driving force of liquid flow in a building material is the capillary pressure, p_c. The moisture gradient relates to the capillary pressure gradient in a complicated way.

During the heat and moisture transfer processes in a conditioned space, the temperature gradient in most of the building material is usually very small, less than 0.03°C/mm (1.372°F/in.). Also, D_T is comparatively smaller than D_{lv}. Therefore, it is simple to neglect $D_T(\partial T/\partial x)$ in Equation 2. Then, Equation 2 is simplified to:

$$J_m = -\rho D_{lv}\, \partial X / \partial x. \tag{6}$$

For transient one-dimensional mass transfer, Fick's second law gives

$$\partial X/\partial t = D_{lv}\, \partial^2 X/\partial x^2 \tag{7}$$

which is analogous to a one-dimentional transient heat transfer,

$$\partial T/\partial t = \alpha\, \partial^2 X/\partial x^2. \tag{8}$$

Mass Diffusivity

In Equations 2, 6, and 7, the mass diffusivity, or the mass diffusion coefficient, actually is the sum of mass diffusivity of liquid water, D_l, and mass diffusivity of water vapor, D_v, that is:

$$D_{lv} = D_l + D_v. \tag{9}$$

Many researchers found that mass diffusivity is not a constant. It varies with the moisture content and also with the temperature. For mass diffusivity of liquid water, the higher the moisture content, the greater will be D_l. The relative humidity of an air-conditioned space during summer mode operation usually varies between 50% and 70%. For simplicity, the mass diffusity is considered as a constant in the moisture transfer calculations.

Convective Heat Transfer

For convective heat transfer, the heat transfer rate, q_c, can be expressed in the form of Newton's law of cooling:

$$q_c = h_c A (T_s - T_\infty) . \tag{10}$$

The magnitude of the convective heat transfer cofficient, h_c, depends on the nature of fluid flow, the velocity of fluid flow over the surface, the nature of the surface, and the temperature difference between the surface and the bulky fluid, etc.

Forced Convection

The surfaces of the building envelope in an air-conditioned space are usually flat surfaces. When the air system is operating, the conditioned air will be forced to flow over the inside surfaces of various building envelopes, such as external walls, windows, partition walls, ceilings, and floors. Hence, during the operating period, a forced convective heat transfer occurs. For forced convection:

1. Even at an air velocity of 0.5 m/s (100 fpm) and a characteristic length of the flat plate L = 3.6 m (11.8 ft), the Reynolds number of the fluid flow at the surface of the building envelope is:

$$\mathrm{Re}_L = \frac{vL}{\nu} = \frac{0.5 \times 3.6}{1.62 \times 10^5} = 1.11 \times 10^5 < 5 \times 10^5 .$$

Hence, the fluid flow is laminar. Then, for a flat plate in laminar flow, the relationship is

$$\mathrm{Nu}_L = 0.664\, \mathrm{Re}_L^{(1/2)} \mathrm{Pr}^{(1/3)} .$$

Hence, for an air velocity of 0.15 m/s (30 fpm) at 25°C (77°F),

$$h_c = 0.664 \frac{k}{L} \mathrm{Re}_L^{(1/2)} \mathrm{Pr}^{(1/3)}$$
$$= 0.664 \frac{0.0255}{3.6} \left[\frac{0.15 \times 3.6}{1.62 \times 10^5} \right]^{(1/2)} (0.71)^{(1/3)}$$
$$= 0.766\ \mathrm{W/m^2 \cdot K}\ (0.135\ \mathrm{Btu/h \cdot ft^2 \cdot °F}) .$$

2. A computer program for estimating energy consumption sponsored by the U.S. Department of Energy recommended in the DOE-2 Reference Manual a convective heat transfer coefficient of 2.1A h_c = 3.87 W/m²·K(0.68 Btu/h·ft²·°F) for a vertical wall.
3. The ASHRAE Task Group on Energy Requirements for Heating and Cooling of Buildings (ASHRAE 1985a) proposed a value of h = 11.36 W/m²·K (2 Btu/h·ft²·°F) for the calculation of a space load in energy calculation, as

$$h = h_c + h_r \tag{11}$$

where h_r represents the radiative heat transfer coefficient and is about 5.9 W/m²·K (1.039 Btu/h·ft²·°F) at room temperature.

Therefore, the convective heat transfer coefficient is

$$h_c = 11.36 - 5.9 = 5.46\ \mathrm{W/m^2 \cdot K}\ (0.962\ \mathrm{Btu/h \cdot ft^2 \cdot °F}) .$$

4. *ASHRAE Fundamentals* (ASHRAE 1985b) recommends a simplified empirical formula by McAdams for the evaluation of the convective heat transfer coefficient of vertical planes when the air velocity flows over the surface at $v < 5$ m/s (1000 fpm).

$$h_c = 5.6 + 3.9v \tag{12}$$

At an air velocity v = 0.15 m/s (30 fpm),

$$h_c = 5.6 + 3.9(0.15)$$
$$= 6.19\ \mathrm{W/m^2 \cdot K}\ (1.09\ \mathrm{Btu/h \cdot ft^2 \cdot °F}).$$

5. According to the actual measured convective heat transfer coefficients for airflow velocities near the wall surface at the sixth floor level (highest level) around the building corner using a heat-flow meter by Sato et al. (1972), the empirical relationship between h_c and air velocity near the wall surface, v, can be expressed as:

$$h_c = 7 + 10v . \tag{13}$$

For an air velocity of 0.15 m/s (30 fpm),

$$h_c = 7 + 10(0.15)$$
$$= 8.5\ \mathrm{W/m^2 \cdot K}\ (1.497\ \mathrm{Btu/h \cdot ft^2 \cdot °F}).$$

From the above calculations and analysis, for the same air velocity of 0.15 m/s (30 fpm), the convective coefficient h_c will vary from 0.766 W/m²·K (0.135 Btu/h·ft²·°F) to 8.5 W/m²·K (1.497 Btu/h·ft²·F).

Based on the results of a series of four tests of heat and moisture transfer in a test chamber (which will be introduced in more detail in another paper, "Simulation of Simultaneous Heat and Moisture Transfer by Using a Finite Difference Method and the Verified Tests in a Test Chamber"), the space sensible load, q_{rs}, is equal to the heat extraction rate, q_{ex}, when it is at thermal equilibrium at time t:

$$q_{rs} = \sum_{i=1}^{n} h_{ci} A_i (T_{s,i} - T_r) + q_{ap} + q_{oc}$$
$$= \dot{V}_s \rho_a c_{pa} (T_r - T_s)$$
$$= q_{ex} . \tag{14}$$

The sensible heat extraction rate, q_{ex}, at noon for these four tests when the motor speeds of the supply fan were 200 rpm, 405 rpm, 650 rpm, and 805 rpm are calculated and shown in Table 1. Also, the space sensible load due to the convective heat transfer from the light troffer, partition walls, ceiling, and floor is calculated by Equations 12 and 13 and is given in Table 1.

By comparing the calculated sum of convective heat transfer and appliance load, $\sum_{i=1}^{n} q_{ci} + q_{ap}$, in Table 1 with the measured sensible heat extraction rate, q_{ex}, the calculated results using the empirical formula $h_c = 7 + 10v$ are shown to be more reasonable and the differences between q_{ex} and $\sum_{i=1}^{n} q_{ci} + q_{ap}$ are comparatively smaller. The rates of convective heat transfer from the light troffer, q_{trof}, are approximately the same when $h_c = 7 + 10v$ was used, whereas the q_{trof} calculated from the empirical formula $h_c = 5.6 + 3.9v$ decreased gradually as the air velocities along the surface of the light troffer, v_{trof}, were increased. Such a consequence is obviously not reasonable.

Free Convection

When the air system in an air-conditioned space ceases to operate during the off period, the fluid flow in the space depends entirely on the motive force from the density difference between the airstreams and results in free convection.

Nansteel and Greif (1981) suggested an empirical equation to calculate the convective heat transfer coefficient for vertical walls during free convection:

$$h_c = 2.03 (T_{sa}/H)^{0.22} \tag{15}$$

where $T_{sa} = (T_s - T_\infty)/2$.

If H = 2.32 m (7.6 ft) and $(T_s - T_\infty)$ = 0.3°C (0.54°F), then

$$h_c = 2.03\,(0.15/2.32)^{0.22}$$
$$= 1.11\ \mathrm{W/m^2 \cdot K}\ (0.196\ \mathrm{Btu/h \cdot ft^2 \cdot °F})$$

In chapter 3 of the 1985 *ASHRAE Fundamentals*, the following empirical formulas were recommended:

For horizontal plates facing upward when heated or downward when cooled,

$$h_c = 1.32\,(T_{sa}/L)^{0.25} . \tag{16}$$

For horizontal plates facing downward when heated or upward when cooled,

$$h_c = 0.59\,(T_{sa}/L)^{0.25} . \tag{17}$$

Convective Mass Transfer

Convective mass transfer involves the transport of moisture or other fluids between a boundary surface and the moving fluid flowing over the surface. Analogous to the convective heat transfer as shown in Equation 10, the rate of moisture transfer, $\dot{m}_m$, can be evaluated as:

$$\dot{m}_m = h_m A_m (C_{m \cdot s} - C_{m \cdot r}) \text{ kg/s(lb/h)} \tag{18}$$

or

$$\dot{m}_m = \rho_a h_m A_m (w_s - w_r) \text{ kg/s (lb/h)}. \tag{19}$$

In Equations 18 and 19, w_s indicates the corresponding humidity ratio of the solid surface that will form a humidity ratio difference $w = w_s - w_r$ and give the same actual mass transfer rate. Mass concentration of moisture is $C_m = \rho_a w$.

Prior to the determination of h_m, w_s, and A_m, the mechanism of moisture transfer on the surface of the building envelope should be discussed.

MECHANISM OF THE MOISTURE TRANSFER FROM THE SURFACE OF A BUILDING MATERIAL

When water vapor is evaporated from the surface of a building material that is covered with a film of liquid, there must be a saturated air film in contact with liquid, and the vapor pressure at the surface is the saturated vapor pressure at the saturated air film corresponding to the surface temperature. Hence, the humidity ratio at the surface of the building material must be the humidity ratio corresponding to the saturated pressure at the surface temperature.

For an air-conditioned space in which the air system is operated in the nighttime shutdown mode, the space relative humidity may vary in the range of 50% and 60% between operating and off periods. Hence, the surface of the building material is not wetted. The degree of filling the pore spaces with the liquid water depends upon the relative humidity of the ambient air and the surface temperature of the building material.

At a specific time, the equilibrium of moisture transfer between the surface of the building material and the ambient air determines the degree of filling of various sizes of pores and interstices, from the finest microcapillaries to the largest pores. The higher the relative humidity of the ambient air, the higher will be a balanced vapor pressure between the vapor pressure in the pores and interstices in the surface of the building material and the ambient air. The higher the surface temperature of the building material, the greater will be the water vapor pressure in the pores and interstices.

Work must be done to increase the surface of a liquid; if a surface is curved, the pressure of the convex side of the curved surface is less than that of the concave side. Hence, when moisture fills up various sizes of pores and interstices, the water vapor pressure of the convex side of the interface, that is, the side where water molecules will be adsorbed in successive layers or be held in pores and microcapillaries, is lower than the concave side of the interface. The degree of depression depends on the diameter of the pores and microcapillaries. The smaller the diameter, the lower will be the vapor pressure. The transport of liquid water from the inner part of the building material to the surface is primarily by capillary flow.

For a nighttime shutdown air system, the complexity of the mechanism is due to liquid water and vapor filling up the pores and interstices at the corresponding vapor pressure during the off period at time $t = 0$ as opposed to the vapor pressure during the operating period at time $t + \Delta t$. The difference in water vapor pressure, $\Delta p_{s\text{-}r}^{t+\Delta t} = \Delta p_s^{t+\Delta t} - \Delta p_r^{t+\Delta t}$, can be more conveniently represented in terms of the difference in humidity ratio, $\Delta w_{s-r}^{t+\Delta t} = w_s^{t+\Delta t} - w_r^{t+\Delta t}$.

Humidity Ratio of the Surface of the Building Material

In the calculation of moisture transfer between the building structure and the space air in an air-conditioned space, it is more convenient to apply Equation 19 to evaluate the rate of moisture transfer, $\dot{m}$. In Equation 19, w_s is the humidity ratio at the surface of the building material corresponding to the water vapor pressure and the moisture content at the surface at a specific temperature, that is,

$$w_s = f(X, \phi_a, w_{ss}, T).$$

If it is assumed that w_s is influenced primarily by the moisture content of the building material, X°, in the off period, then the corresponding relative humidity of the ambient air, ϕ°, can be determined from the sorption isotherm of the building material. The humidity ratio, w_s, can be determined from the following relationship:

$$\phi w_s^{t+\Delta t} = \phi^\circ w_{ss}^{t+\Delta t} \tag{20}$$

where $w_{ss}^{t+\Delta t}$ is the humidity ratio of the saturated air corresponding to the surface temperature of the building material at time $t+\Delta t$. When the surface temperature of the building material is in a range between 0°C and 30°C (32°F and 86°F) w_{ss} can be calculated from a polynomial relationship:

$$w_{ss} = a_1 + b_1 (T_s) + c_1 (T_s)^2 + d_1 (T_s)^3 \tag{21}$$

where

$a_1 = 3.768 \times 10^{-3}$
$b_1 = 3.0517 \times 10^{-4}$
$c_1 = 4.648 \times 10^{-6}$
$d_1 = 3.787 \times 10^{-7}$.

Contact Area

In Equation 19, the contact area between the space air and the liquid water at the surface of the building material, A_m, is a function of moisture content, X. A precise evaluation of A_m will be rather complicated. If the surface area of the building material is represented by A_s, then a rough estimate gives:

$$X = A_m / A_s. \tag{22}$$

Then the rate of moisture transfer calculated by Equation 19 can be modified as:

$$\dot{m}_m = \rho_a h_m X A_s (w_s - w_r). \tag{23}$$

Period of Constant Drying Rate (PCDR) or Period of Decreasing Drying Rate (PDDR)

During the evaporation of liquid water from the surface of the building material, if the space temperature and relative humidity are maintained at nearly constant values, the normalizing drying characteristic may show a period of constant drying rate, PCDR, or a period of decreasing drying rate, PDDR, as shown in Figures 6a and 6b.

The normalizing drying characteristic, R_m, is defined as the ratio of the mass drying rate in (X) to the mass drying rate corresponding to the initial moisture content in (X_i).

$$R_m = \dot{m}(X) / \dot{m}(X_i) \tag{24}$$

Many researchers found that sand, silica-brick mix, and ceramic plate display a PCDR curve as shown in Figure 6a, while paper, wool, and similar materials show a PDDR curve as in Figure 6b. Based on the actual evaporated liquid water from the acoustic tile and the wood floor in the test chamber, which will be described in more detail in another paper, "Simulation of Simultaneous Heat and Moisture Transfer in a Test Chamber by Using a Finite Difference Method and the Verified Tests in a Test Chamber," the latent load during the operating period is shown to perform as a PCDR curve (see Figure 7).

Mass-Transfer Coefficient h_m

The Chilton-Colburn analogy relates the heat and mass transfer in the form:

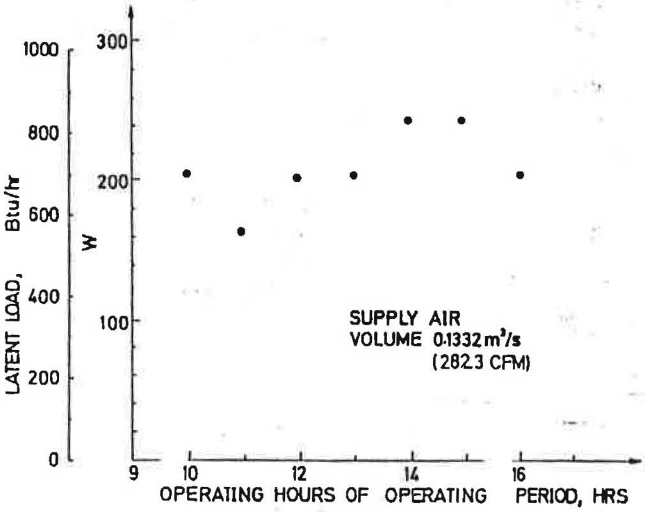

Figure 7 *Latent load vs. operating time in a test chamber*

$$j_H = j_D$$
$$(h_c/v_\infty c_{pa})\,\mathrm{Pr}^{(2/3)} = (h_m/v_\infty)\,\mathrm{Sc}^{(2/3)}$$
$$h_c/c_{pa}\,(\nu/\alpha)^{(2/3)} = h_m\,(\nu/D_{AB})^{(2/3)}. \quad (25)$$

The analogy is valid for liquid and gases when $0.6 < \mathrm{Sc} < 2500$ and $0.6 < \mathrm{Pr} < 100$.

For air temperature at 25°C (77°F):

Mass diffusivity of air-water	D_{aw}	$= 2.63 \times 10^{-5}$ m²/s (2.83×10^{-4} ft²/s)
Kinematic viscosity	ν	$= 1.62 \times 10^{-5}$ m²/s (1.74×10^{-4} ft²/s)
Thermal diffusivity	α	$= 2.27 \times 10^{-5}$ m²/s (2.44×10^{-4} ft²/s)
Density	ρ	= 1.15 kg/m³ (0.719 lb/ft³)
Specific heat of moist air	c_{pa}	= 1020 J/kg·K (0.243 Btu/lb·°F)

and

$$\mathrm{Pr} = (\nu/\alpha) = 1.62 \times 10^{-5}/2.27 \times 10^{-5} = 0.714$$

also

$$\mathrm{Sc} = (\nu/D_{aw}) = 1.62 \times 10^{-5}/2.63 \times 10^{-5} = 0.615.$$

Then Equation 23 is valid for heat and moisture transfer at a space air temperature of T_r = 25°C (77°F). Therefore,

$$h_m = h_c\,1/c_{pa}\mathrm{Sc}^{(2/3)}\;\mathrm{Pr}^{(2/3)}$$
$$= h_c(0.714)^{(2/3)}/1.15 \cdot 1020 \cdot (0.615)^{(2/3)} = 0.000945\,h_c. \quad (26)$$

Condensation of Water Vapor into Liquid Water from the Moving Vapor

When water vapor in the moving fluids is condensed into liquid form and stays inside the pores and interstices of the building material during convective mass transfer, condensation of the vapor may occur in any of the pores and interstices on the surface of the building material. Hence, the rate of moisture transfer when condensation of water vapor occurs can be expressed as:

$$\dot{m}_m = \rho_a h_m A_s\,(w_r^{\bullet} - w_s) \quad (27)$$

where w_s is now the corresponding humidity ratio at the surface of the building material based on its moisture content.

From the sorption isotherm, the relative humidity ϕ_s corresponding to the moisture content at the surface of the material can be determined by

$$\phi_s = w_s/w_{ss}.$$

Then,

$$w_s = \phi_s w_{ss}. \quad (28)$$

Simultaneous Heat and Moisture Transfer

For problems encountered in the transient heat and moisture transfer at the external wall, partition wall, floor, and ceiling in an air-conditioned space, it is often necessary to determine the temperature and moisture distributions inside the building envelope, to evaluate the heat and mass fluxes, and then to calculate the rate of heat and moisture transfer from or to the building envelope.

For a one-dimensional transient heat flow in an external wall, the conduction equation (Equation 8) gives

$$\partial T/\partial t = \alpha \partial^2 T/\partial x^2.$$

Based on the heat balance per unit area at the outside surface of this external wall, one of the boundary conditions, $T(0, t)$, can be determined, as shown in Figure 8:

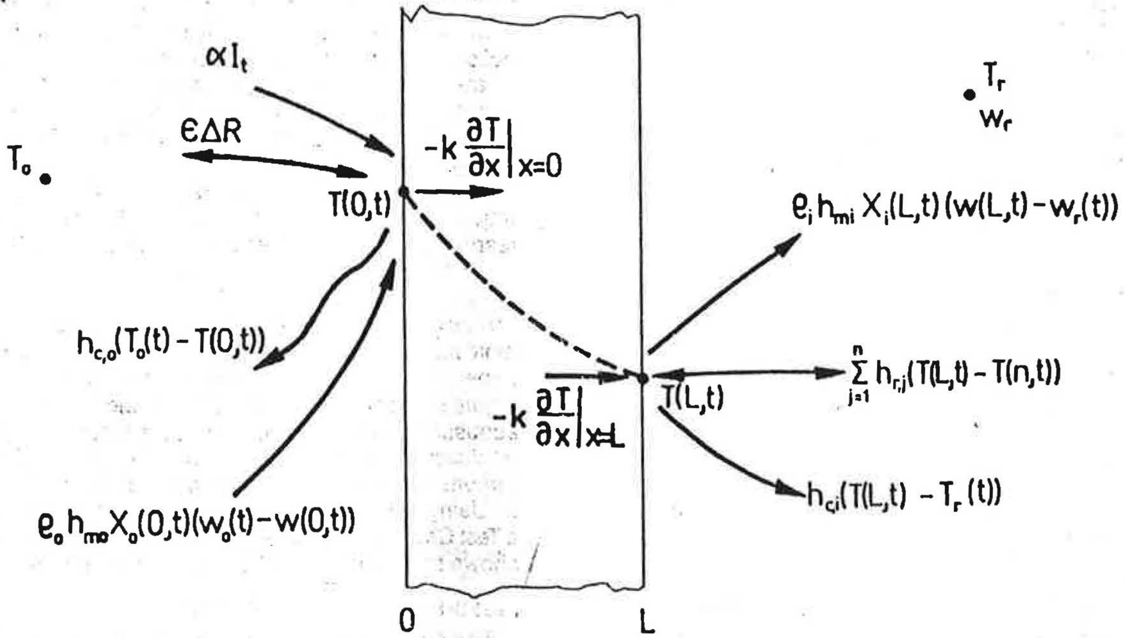

Figure 8 *Boundary conditions of an external wall*

Solar radiation absorbed	+	Radiation exchange between outside surface and the sky and surroundings	+	Latent heat of condensation of liquid water condensed on the outside surface	=	Convection from the outside surface	+	Inward conduction.

That is,

$$\alpha I_t + \epsilon \Delta R + \rho_o h_{mo} X(0, t) [w_o(t) - w(0, t)] h_{fg} = h_{c,o}[T_o(t) - T(0, t)] - k\, \partial T/\partial x|_{x=0}. \quad (29)$$

For each unit area of the inside surface of this external wall, another boundary condition, $T(L,t)$, can be found from the following relationship (see Figure 8):

Inward heat conduction	=	Convection from the inner surface	+	Radiation exchange between inner surface and other surfaces	+	Heat required for the evaporation of the liquid water from the inner surface.

$$-k\partial T/\partial x\,|_{x=L} = h_{c,i}[T(L,t) - T_r(t)] + \sum_{j=1}^{n} h_{r,j}[T(L,t) - T(n,t)] + \rho_i h_{mi} X_i(L,t)\, [w(L,t) - w_r(t)]\, h_{fg} \quad (30)$$

In Equations 29 and 30, mass flow rates of moisture transfer times the latent heat of vaporization become a part of the energy balance. Also, the moisture content at the outer surface, $X(0,t)$, and at the inner surface, $X(L,t)$, are the boundary conditions of the one-dimensional transient mass transfer equation as shown in Equation 7.

Again, the term $w(L,t)$ is also a function of $T(L,t)$ and $X(L,t)$.

The initial condition of $T(x,0)$ in Equation 9 can be assumed to be a uniform temperature: $T(x,0) = T_o$.

A possible solution of $T(x,t)$—with the boundary conditions determined from Equations 29 and 30 and linked to transient mass transfer—is by using numerical solutions, that is, by using a finite difference method.

Solution of the transient mass transfer partial differential Equation 7, $X(x,t)$, is similar to the transient heat transfer.

Moisture Transfer at the Interface between Two Different Building Materials

When a piece of building material A is in contact with another piece of building material B of different properties, as shown in Figure 9, there is often a moisture transfer at the interface plane i. It is also possible that moisture may transfer from material A having a low moisture content, X_{iA}, to material B having a higher moisture content, X_{iB}.

Luikov (1966) constructed a scale of moistness W, in °M, which is analogous to the temperature scale T, in °C (°F). A term mass capacity C_m'' has also been proposed as:

$$c_m = (\partial x/\partial W)_T \quad (31)$$

which is analogous to heat capacity

$$c_p = (\partial H/\partial T)_p. \quad (32)$$

If two materials are in contact at hygrothermal equilibrium, the degree of moistness will be the same at the interface for these two materials.

Matsumoto et al. (1988) recommended water chemical potential (Gibb's free energy) as the moisture flow potential at the interface of two building materials $\mu_{iA} = \mu_{iB}$.

Many scientists in analyses of the theories of moisture absorption of textile fibers proposed that the water molecules are attached to particular sites. They may be tightly bound to

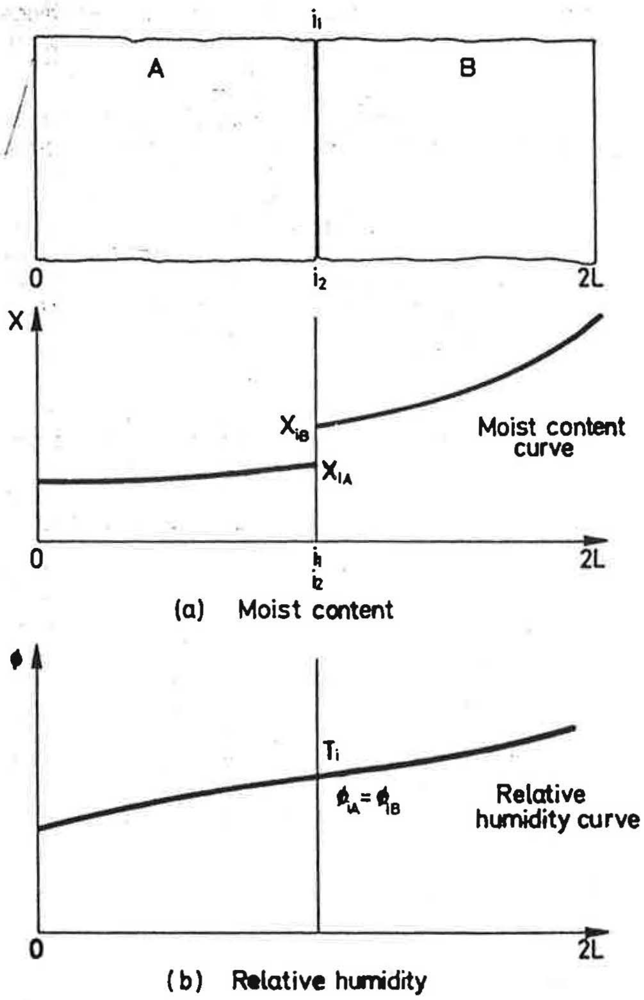

Figure 9 *Moisture content and relative humidity curves when two pieces of material are in contact*

the hydrophilic groups or tightly bound to the surface in a monomolecular layer. There may be further absorption of water by the mixing of molecules of different types, having no limitation about the position of molecules, that is, the solution theory. Such absorbed water molecules are attached more loosely.

Either by using the solution theory or by adopting indirect attachment, an equilibrium always exists between the dissolved water and the water vapor in the ambient air. In this case, the vapor pressure of the water molecules not tightly bound with the material is balanced with the water vapor pressure in the ambient air and, hence, the relative humidity at a specific temperature.

For any material having a moisture content X, there is always a corresponding relative humidity ϕ from the sorption isotherm at a specific temperature. For material A, the relative humidity at the interface of temperature T_i corresponding to a moisture content X_{iA} is ϕ_{iA}. If there is a hygrothermal equilibrium at the interface i, the relative humidity of material B at the same temperature T_i and at a different moist content X_{iB} must have the same relative humidity ϕ_{iB} from the sorption isotherm of material B, as shown in Figure 9.

Hence, at the interface, although the moisture content of material A, X_{iA}, may be lower than that in material B, X_{iB}, there is a possibility of moisture transfer from material A to material B.

CONCLUSIONS

1. The results of site surveys in tropical areas show there exists a large amount of space latent cooling load induced by

moisture transfer from the building structure and furnishings to the conditioned space air during the operating period. The ratio of latent load to the space cooling load may be as high as 0.3 to 0.4 during the cooldown period.

2. There is also moisture transfer from the space air to the building structure and furnishings during the off period of hot summer days in tropical areas because of the higher space relative humidities and temperatures due to infiltration.
3. Simple vapor theory overlooks the fact that the moisture content of the building material is primarily in the form of liquid water when the relative humidity of ambient air is greater than 50%. Also, the liquid fraction of moisture transfer in building material has a higher percentage of the total mass transfer at higher moisture content and higher relative humidities.
4. For convective mass transfer, it is more convenient to use the humidity ratio difference as the driving potential for calculating moisture transfer from the building envelope to the space air. During the evaporation of liquid water into water vapor at the surface of the building material, the surface humidity ratio should be considered as the humidity ratio corresponding to the moisture content and the related relative humidity of the ambient air that gives the same moisture content on the sorption isotherm of the building material at a specific temperature.
5. The rate of moisture transfer from the building structure and furnishings to the space air, that is, the building latent load, during the operating period in summer depends primarily on the difference in space relative humidity and temperature between the off period and the operating period, the structural characteristics, surface temperatures, moisture contents of the building materials, and the velocity of air flowing over the surface of the building envelope.
6. For moisture transfer at the interface between two different building materials, A and B, the relative humidity of the sorption isotherm of material A having a corresponding moisture content at the interface must be equal to the relative humidity of the sorption isotherm of material B having another corresponding moisture content at the interface.
7. In order to reduce the space latent load induced by moisture transfer from the building structure and furnishings to the space air during the cooldown period in tropical areas, it is recommended that infiltration through elevator shafts, pipe shafts, duct shafts, and the cracks around windows be reduced during the off period and also that carpets or furnishings used have comparatively lower moisture content at normal space relative humidity.

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NOMENCLATURE

A = area, m² (ft²)
A_m = contact area between the moisture and the ambient air, m² (ft²)
c_m = mass concentration of moisture, kg/m³ (lb/ft³)
c_m = mass capacity, 1/°M
c_{pa} = specific heat of moist air at constant pressure, kJ/kg·K or J/kg·K (Btu/lb·°F)
D_{aw} = mass diffusivity for air-water, m²/s (ft²/s)
D_{lv} = mass diffusivity of liquid and vapor, m²/s (ft²/s)
D_T = mass diffusivity due to temperature gradient, m²/s·K (ft²/s·°F)
g = gravitational acceleration, m/s² (ft/s²)
H = height of the surface, m (ft)
h = mean heat transfer coefficient, W/m²·K (Btu/h·ft²·°F)
h_c = mean convective heat transfer coefficient, W/m²·K (Btu/h·ft²·°F)
h_{fg} = latent heat of vaporization or condensation, kJ/kg (Btu/h·lb)
$h_{r,i}$ = radiative heat transfer coefficient of surface i, W/m²·K (Btu/h·ft²·°F)
h_m = convective mass transfer coefficient, m/s (ft/s)
h_r = mean radiative heat transfer coefficient, W/m²·K (Btu/h·ft²·°F)
I_t = total intensity of solar radiation, W/m² (Btu/h·ft²)
J_m = mass flux, kg/s·m² (lb/h·ft²)
j_D = j factor for mass transfer
j_H = j factor for heat transfer
K = permeability of liquid flow, m/s (ft/s)
k = thermal conductivity, W/m·K (Btu/h·ft·°F)
K_{sc} = coefficient of structural characteristics
L = length, m (ft)
$\dot{m}$ = mass flow rate, kg/s (lb/s or lb/h)
$\dot{m}_m$ = rate of moisture transfer, kg/s (lb/s or lb/h)
p_c = capillary pressure, Pa (lb_f/ft^2)
q_c = rate of convective heat transfer, W or kW (Btu/h)
q_l = heat evolved when unit mass of liquid water is absorbed, kJ/kg (Btu/h·lb)
q_{rc} = space cooling load, W or kW (Btu/h)
q_{rl} = space latent load, W or kW (Btu/h)
q_{rs} = space sensible load, W or kW (Btu/h)
q_v = heat evolved when unit mass of water vapor is absorbed, kJ/kg (Btu/h·lb)
R = difference between longwave radiation incident on the surface from the sky and surroundings and the radiation emitted from the surface, W/m² (Btu/h·ft²)
R_m = ratio of normalizing drying characteristic
T = temperature of the building material, °C (°F)
T_∞ = temperature of the undisturbed bulky fluid remote from the heat transfer surface, K (°F)
t = time, s or h
V_s = supply air volume flow-rate, m³/s (cfm)
v = velocity, m/s (fpm of ft/s)
w = humidity ratio, kg/kg (lb/lb)
W = scale of moistness, °M
X = moisture content, kg/kg (lb/lb)
$\bar{X}$ = normalized moisture content, kg/kg (lb/lb)
x = displacement in x direction, m
α = thermal diffusivity, m²/s (ft²/s), coefficient of absorption
ϵ = hemispherical emittance of the surface
θ = angle with vertical, degrees
ν = kinematic viscosity, m²/s (ft²/s)
ρ = density or density of solid and moisture, kg/m³ (lb/ft³)
ρ_l = density of the liquid, kg/m³ (lb/ft³)
ϕ = relative humidity, % or dimensionless

Subscripts

a = air
ap = appliance
i = number of surface, such as surface i; initial; inner
l = liquid
lv = liquid and vapor
o = outer, outdoor
oc = occupant
r = space, room
s = surface
ss = saturated vapor
t = at time t
v = vapor

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TABLE 1
Verification of Convective Heat Transfer Coefficient (SI units)

Motor speed of supply fan,	rpm	200		405		650		805	
Supply volume flow rate,	m^3/s	0.0412		0.0814		0.1332		0.1711	
$T_r^t - T_s^t$	°C	6.3		5.0		3.4		3.4	
$q_{ex} = \dot{V}_s \rho_a c_{pa}(T_r^t - T_s^t)$	W	318		498		554		712	
$h_c = A + Bv$		h_c=5.6+ 3.9v	h_c=7+ 10v	h_c=5.6+ 3.9v	h_c=7+ 10v	h_c=5.6+ 3.9v	h_c=7+ 10v	h_c=5.6+ 3.9v	h_c=7+ 10v
Heater + Data logger	W	316	316	316	316	316	316	316	316
v_{trof}	m/s	0.054	0.054	0.18	0.18	0.35	0.35	0.49	0.49
$h_{c.trof}$	W/m^2K	5.8	7.5	6.3	8.8	7.0	10.5	7.5	11.9
$\Delta T = T_{s.trof} - T_r$	°C	6.4	6.4	5.5	5.5	4.4	4.4	3.6	3.6
$q_{trof} = h_{c.trof} A_{trof}(T_{s \cdot trof} - T_r)$	W	130	169	121	169	108	162	95	150
V_{wc}	m/s	0.16	0.16	0.34	0.34	0.62	0.62	0.85	0.85
$h_{c.wc}$	W/m^2K	62	8.6	6.9	10.4	8.0	13.2	8.9	15.5
$\Delta T = T_{wc} - T_r$	°C	-0.2	-0.2	0.1	0.1	0.2	0.2	0.2	0.2
$q_{wc} = h_{c.wc} A_{wc}(T_{wc} - T_r)$	W	-38	-53	21	32	50	82	56	96
v_{fl}	m/s	0.16	0.16	0.34	0.34	0.62	0.62	0.85	0.85
$h_{c.wc}$	W/m^2K	6.2	8.6	6.9	10.4	8.1	13.3	8.5	14.3
$\Delta T = T_{fl} - T_r$	°C	-0.6	-0.6	-0.2	-0.2	0.05	0.05	0.1	0.1
$q_{fl} = h_{c.fl} A_{cfl}(T_{fl} - T_r)$	W	-52	-72	-19	-29	6	9	12	20
v_g	m/s	0.48	0.48	1.0	1.0	1.73	1.73	2.43	2.43
$h_{c.g}$	W/m^2K	7.5	11.8	9.5	17.0	12.3	24.3	15.1	31.1
$\Delta T = T_g - T_r$	°C	0	0	0.2	0.2	0.3	0.3	0	0
$q_g = h_{c.g} A_g (T_g - T_r)$	W	0	0	3	5	5	10	0	0
$q_{rs} = \sum_{i=1}^{n} q_i + q_{ap}$	W	356	360	442	493	485	579	451	582

+ wc indicates partition wall plus ceiling; trof represents light troffer; fl denotes floor; g indicates window glass