AMERICAN UNIVERSITY OF BEIRUT

THE EFFECT OF HYGROSCOPIC MATERIALS ON THE INDOOR RELATIVE HUMIDITY OF SPACES CONDITIONED BY DX SPLIT UNITS

by
MOHAMAD OMAR KAMAL KATANANI

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Approved by:

____

Dr. Kamel Ghali, Associate Professor — Advisor
ME Department

____

Dr. Nesreen Ghaddar, Professor — Committee Member
ME Department

____

Dr. Mahmoud Hindi, Assistant Professor — Committee Member
ME Department

Date of thesis defense: February 16, 2011

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AN ABSTRACT OF THE THESIS OF

Mohamad Omar Kamal Katanani for Master of Mechanical Engineering
Major: Applied Energy

Title: The Effect of Hygroscopic Materials on the Indoor Relative Humidity of Spaces Conditioned by DX Split Units Systems

Hygroscopic materials are those that tend to absorb moisture from the air in periods of high relative humidity, and eventually release this moisture to the air when the latter is drier. This phenomenon of absorption and desorption, referred to as "moisture buffering", dampens the variation of indoor relative humidity. Since no energy is consumed to control the relative humidity, moisture buffering through the use of hygroscopic materials is a passive way of controlling humidity. An example of such hygroscopic materials is fabric curtains (such as wool and cotton ones). The installation of a hygroscopic curtain will help dampen the variation of relative humidity and thus enhance the indoor air quality and reduce the risk of mold growth. If it is required to carefully control the indoor humidity (Ex: dehumidifier), the use of hygroscopic materials can partially help buffer moisture, and reduce any electricity consumption required to remove the high latent load.

In this study, the transient moisture transfer between the curtain and indoor air was modeled and simulated when the curtain is placed in front of a wall. Experiments were conducted inside environmental chambers to validate the model and to test the ability of curtains to moderate indoor humidity and to compare the buffering moisture capacity of a hygroscopic curtain with a base identical case when a non-hygroscopic curtain (or no curtain) is installed. The experimental results of the curtain moisture uptake and the relative humidity inside the chamber compared well with the simulation results.

The model was used in simulating a case study, whereby the moisture buffering capacity of a cotton curtain was studied. The curtain was installed in a typical small office in the city of Beirut, Lebanon, conditioned by an accurately modelled DX Split-unit air conditioning system. It was found that, for a full day simulation, hygroscopic curtains maintained the relative humidity at less than 65% during part load operation compared to 70% in the case where no curtain was installed. It was concluded that both scenarios had the same energy consumption; at the time the Predicted Mean Vote for the case of the curtain was closer to "zero" than the case of no curtain by 0.1.

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NOMENCLATURE

Symbols

A	area (m^2)
c_i	wall layer thermal capacitance (J/kg·°C)
C_m	overall wall thermal capacitance (J/kg·°C)
C	air specific heat (J/kg·°C)
d	Gap distance between the wall and curtain (m)
COP	coefficient of performance
g	gravitational constant (m/s^2)
$HVAC$	heating ventilation and air conditioning
h	convective heat transfer coefficient (W/m^2·K)
h_m	mass transfer coefficient (kg/s.m^2.kPa)
h_{fg}	latent heat of vaporization (kJ/kg)
h_{ad}	Heat of adsorption (kJ/kg)
h_{rd}	Linearized radiative heat transfer coefficient(W/m^2.K)
$\dot{m}_a$	mass flow rate of air (kg/s)
P_{v-}	Water vapor pressure in the air (kPa)
Q_{sens}	load (W), (Btu/hr)
R	curtain regain (kg of absorbed water/kg of dry curtain)
R_d	curtain dry resistance(°C.m^2/W)
R_e	Curtain evaporative resistance(kPa.m^2/W)
r_i	element thermal resistance (K/W)

r_m	overall wall thermal resistance (K/W)
T	time (s)
T	temperature (°C)
V	volume (m^3)
W	Width of the space (m)
w	Humidity Ratio (kg_{water}/ kg_{air})

Greek Symbols

ρ	density (kg/m^3)
β	Volumetric thermal expansion of air(1/K)
υ	Kinematic viscosity (m^2/s)

Subscripts

a	Air
c	Curtain
db	Dry bulb
ent	Internal sensible loads
i,j,k	Internal, index
sens	Sensible capacity of the air conditioning system
W	Wall
wb	Wet bulb

CHAPTER 1

INTRODUCTION

The proper control of indoor humidity is important for human comfort, indoor air quality, building energy performance and durability, (Wilson et al 2003; Straub 2002). There are many reasons that justify the need to control humidity and to keep the indoor relative humidity within the allowed range. Human thermal comfort sensation is one of those important reasons (Harriman et al 2001; Toftum et al 1998). A high relative humidity causes the perspiration rates on the skin to be lower than it should be under dryer conditions. As a result, we feel warmer when the relative humidity is high than when it is low. A very low relative humidity might cause skin dryness, eye irritation, and respiratory infections. In addition, high relative humidity causes discomfort due to increased skin wettedness (the clinging effect of clothes), Harriman et al (2001) and a very low humidity may cause a feeling of discomfort due to the possible occurrence of electrostatic discharge. Controlling humidity is also important for the building structure as a whole, i.e., increasing the durability of the building structure (wood and steel materials) and prolonging the life service of the building. Besides the effects on human comfort and building rigidity, the indoor controlled set point relative humidity can have a significant impact on the building energy performance if the humidity is controlled by ventilation.

The control of humidity is accomplished by sub-cooling the air to a value less than the dew point temperature; remove a specific amount of the condensate, followed by reheating the air to the required supply temperature. Alternatively, one can use desiccant dehumidification as less energy requiring process (Griffiths et al 1989; Peng et al 1984). Both processes require energy to control the level of humidity and therefore both are classified as active methods for controlling the humidity. In the first process, not only the air-conditioning system has to be oversized to meet the sub-cooling requirement of the latent load, but also the operational energy of the air-conditioning system is increased due to the excess cooling and reheating. Also, the desiccant dehumidification whether accomplished by solid or liquid desiccants requires thermal energy to regenerate the desiccant. The desiccant dehumidification process is achieved by using what is called liquid solutions or solid materials (the desiccants) in order to absorb the moisture in the room air. Afterwards, the desiccants will require energy in order to release the absorbed moisture. The classification of whether this process is considered an active or a passive method depends on whether energy is required to regenerate the desiccant. If it requires a reactivation heater for example, the method is considered active. If it operates naturally by releasing absorbed moisture to a much less humid air, then it is considered as a passive way.

In this study, the moisture moderating capacity of the cloth curtains will be investigated by modeling and experimentation. The study will emphasize the role of hygroscopic curtains in passively dampening the moisture variation inside offices that are chracterized by a uniform high latent load use, and use DX air conditioning system to control the indoor office air temperature. For a DX system, the effect of moisture buffering of the hygroscopic curtain will be expected to be at its maximum during the part load operation of the air conditioning system. This is because, for a uniform moisture generation rate, the indoor humidity ratio is higher during the off design operating condition (part load period) rather than the peak load period. The reason is that during full load operation, the DX is on for most of the period to remove the high sensible load, and will thus be automatically removing the latent load. The shorter ON period of the DX air conditioing system during part load compared to the peak load operation will cause the humidity ratio inside the space to increase, given that there is a uniform moisture generation and there is no additional moisture sink (besides the HVAC system).

CHAPTER 2

LITERATURE REVIEW

The continuous increase in energy demand and costs, added to the associated environmental problems resulting from energy generation (ex. Global warming), pushed researchers to consider a more *passive* method for controlling the humidity: the use of indoor hygroscopic materials to assist, without any modifications, the air-conditioning system in eliminating the problems associated with high humidity levels, Rode et al (2004). Hygroscopic materials are those that tend to absorb moisture from the air in periods of high relative humidity, and eventually release this moisture to the air when the latter is drier. Household materials such as clothes, cotton, curtains, carpets, and wallpaper are hygroscopic ones. Through absorption and desorption, hygroscopic materials in the indoor environment are able to dampen the moisture variations of the indoors air, Svennberg (2006). During high latent loads, hygroscopic materials reduce the peaks of relative humidity levels due to their absorbing characteristics, Holm et al (2004), and during low latent loads, their desorption characteristics allow them to release the previously absorbed moisture into the indoor air, and thus keeping a more uniform indoor humidity level. Therefore, household hygroscopic materials buffer the moisture variation in a passive way and moderate the variation of indoor humidity, Osanyintola (2006). Every material is unique in the amount of moisture it can absorb from air for a given indoor temperature and relative humidity. Wool, for example, has a capacity to absorb a higher amount of water than cotton when exposed to air at the same temperature and relative humidity (Morton et al 1975; Ojanen et al (2004)). Studying moisture-buffering characteristics of indoor hygroscopic materials can be helpful in improving indoor air quality and saving energy in buildings.

Hygroscopic materials will reduce the operational energy consumption of the HVAC system if the latter is to control indoor relative humidity. If, however, the HVAC operation is controlled solely by temperature, then the energy consumption should not vary significantly, but the indoor air quality will be definitely enhanced. Researchers have conducted both theoretical and experimental studies on the the moisture storage capacity of the different building materials. For example, Simonson et al (2002) showed that the wooden materials inside bedrooms can reduce the peak humidity by 35% when the ventilation rate is 0.5 ach. The numerical study of Cerolini et al (2009) has shown that that the cellulose based material have a higher buffering performance and a faster adjustments to variations in indoor relative humidity when compared to non-cellulosic common building materials. Also Hameury et al (2005) considered the porous structures of the floor and walls to study their effect on buffering the indoor humidity levels. The moisture buffering capacity of building materials has been considered in the studies conducted by many researchers. However, the moisture buffering capacity of a *cloth curtain* as a potential indoor material for moderating the humidity without any additional energy cost has not been studied, as there is no significant related information in literature.

CHAPTER 3

MODELING METHODOLOGY

To assess the moisture buffering capacity of a cloth curtain during the operation of a DX air conditioning system, a mathematical model that can simulate the transient moisture uptake of the curtain is developed coupled with a building space and DX air conditioning models.

The moisture curtain model will be validated by conducting experiments inside AUB climatic chamber (Fig 1). The results will demonstrate the potential benefit of hygroscopic curtain in buffering the indoor moisture of air-conditioned office spaces in the absence of an HVAC control strategy for indoor relative humidity.

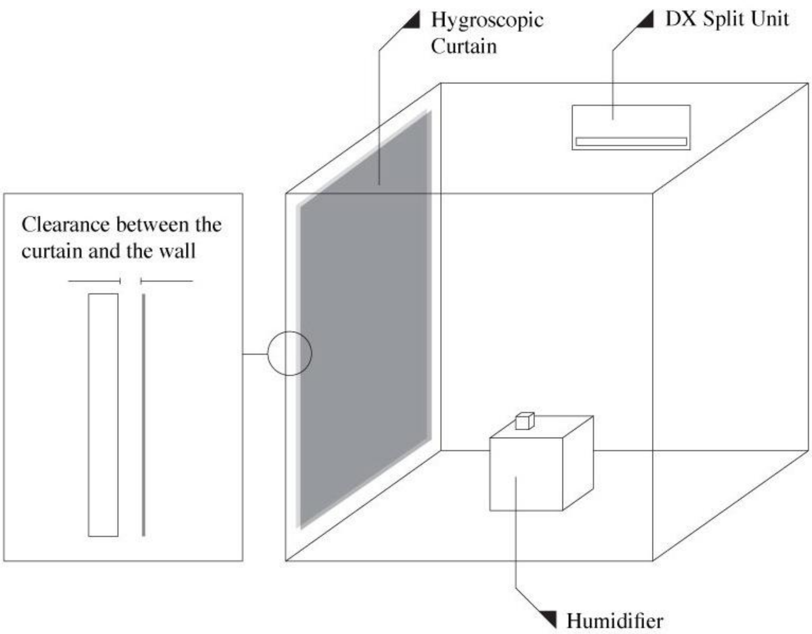

Figure 1: Schematic of the AUB indoor climatic chamber

3.1. Modeling the Space and the Moisture Interactions

Curtains are usually hanged infront of walls or windows for decoration or solar shading purposes, typically covering the whole area of the wall where they are located with small clearnce distances close to the floor and ceiling and therfore they divide the space air into two zones. Consequently, the air in the space is modeled as two lumped zones: one zone representing the air in front of the curtain (the main occupied room) and a second zone in the gap space between the curtain and the wall behind the curtain as shown in Figure 2.

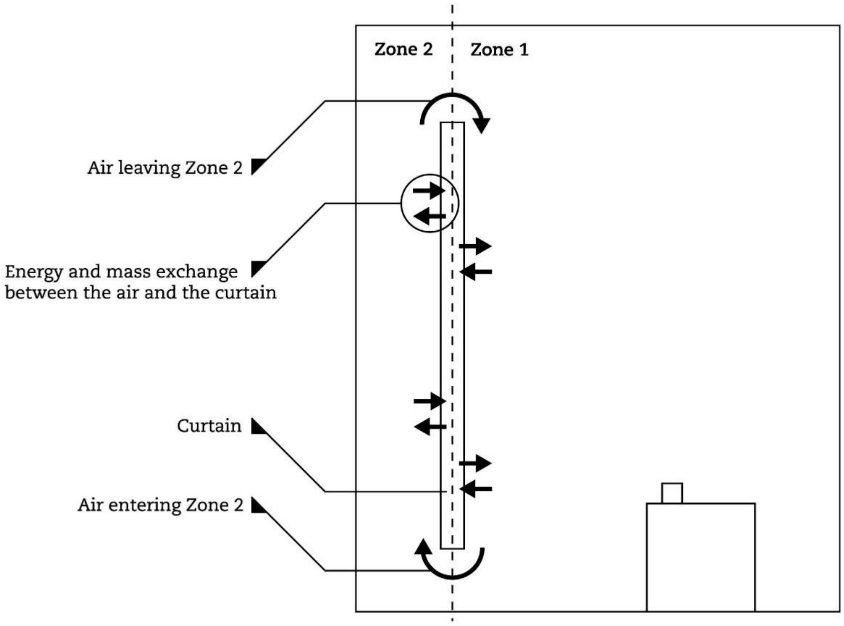

Figure 2: Schematic of the curtain's position and its interaction with the room air

The moisture balance for the air in zones 1 and 2:

$$m_{a,1}\frac{dw_1}{dt} = A_c\frac{(P_c - P_{v,1})}{h_{fg}\left(\frac{R_e}{2}+\frac{1}{h_m}\right)} + \dot{m}_{gen} - \dot{m}_{HVAC} + \dot{m}_{gap}(w_2 - w_1) \quad (1)$$

$$m_{a,1}\frac{dw_2}{dt} = A_c\frac{(P_c - P_{v,2})}{h_{fg}\left(\frac{R_e}{2}+\frac{1}{h_m}\right)} + \dot{m}_{gap}(w_1 - w_2) \quad (2)$$

Where m_a is the mass of air (kg), w is the humidity ratio (kg_{vapor}/kg_{air}), A_c is the curtain area (m^2), P_v is the water vapor pressure in the air (Pa), P_c is the pressure of the vapor in the curtain (Pa), h_{fg} is the latent heat of vaporization (kJ/kg), R_e is the evaporative resistance of the curtain (m^2Pa/W), h_m is the mass transfer coefficient (W/m^2Pa) and can be calculated from the Lewis relation LR (h_m/h_c = LR = 16.66 K/kPa), $\dot{m}_{gen}$ is the generated moisture of the occupants (kg/s), $\dot{m}_{HVAC}$ is the moisture removed by the HVAC system coil (kg/s), and $\dot{m}_{gap}$ is the air flow rate between the 2 zones in the gap (kg/s).

$P_{v,1}$ and $P_{v,2}$ can be determined from the ideal gas law function of the humidity ratios w_1 and w_2.

$P_{air} = \rho_{air}\left(\frac{\mathrm{R}}{M_{air}}\right) \mathrm{T_{air}}$, where

R is the universal gas constant = 8.314 $J.K^{-1}.mol^{-1}$ and M_{air} = 28.9 g/mol

$P_v = \left(\frac{w}{0.622}\right) \times \rho_{air}\left(\frac{\mathrm{R}}{M_{air}}\right) \mathrm{T_{air}}$

$P_c = \phi_c P_{sat}$, where ϕ_c can be obtained from the wool or cotton isotherm equilibrium curve for a certain value of the curtain Regain R (Fig 3), defined as the weight of the curtain fabric at a non-dry environmental conditions divided by the its dry weight, defined as:

$$R = \frac{m_{wet}-m_{dry}}{m_{dry}} \quad (3)$$

P_{sat} is the saturation pressure of the vapor in the curtain at a given curtain temperature. Note that the subscripts 1 and 2 refer to zones 1 and 2. The second equation is similar to the first one, except that it does not include the terms for moisture generation and removal.

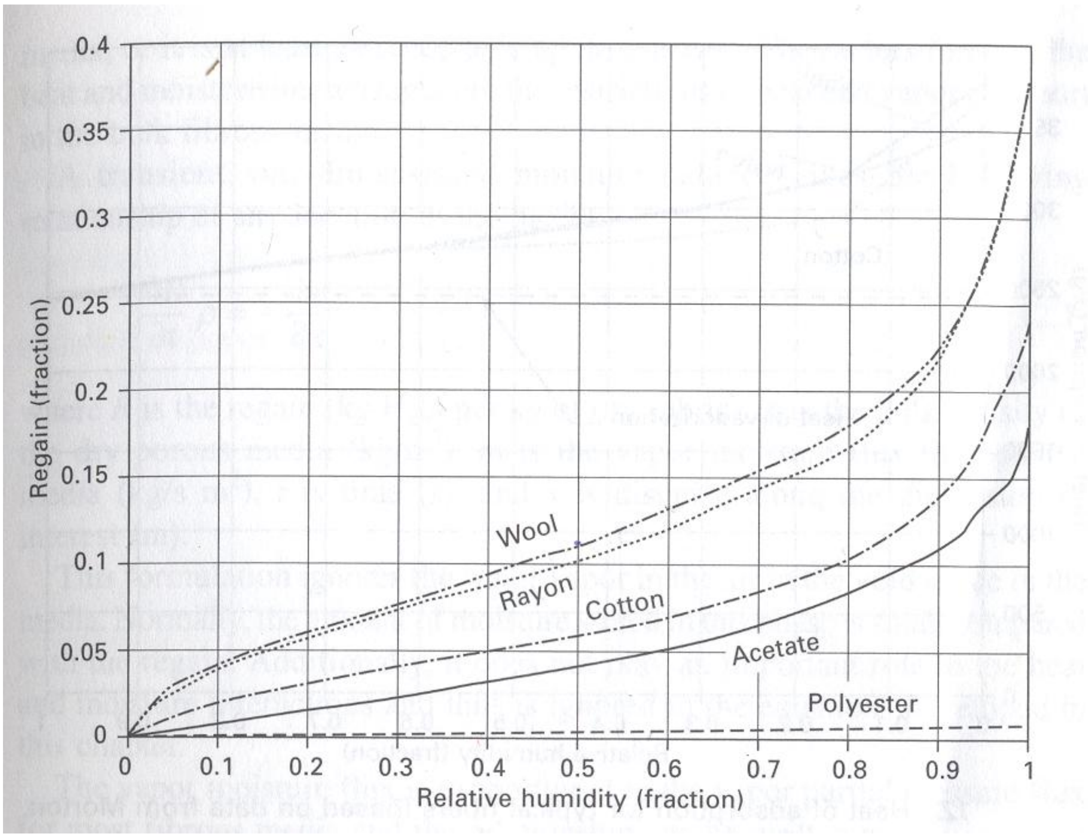

Figure 3: Regain of different fabric materials as a function of relative humidity

As for the energy balance of the 2 zones:

$$m_{a,1}c_{p,a}\frac{dT_{a,1}}{dt} = \tag{4}$$

$$\sum_{j=1}^{5} h_{a,j}A_{w,j}\left(T_{w,j} - T_{a,1}\right) + A_c\left(\frac{T_c - T_{a,1}}{\frac{1}{h_c}+\frac{R_d}{2}}\right) + \dot{m}_{gap}c_{p,a}\left(T_{a,2} - T_{a,1}\right) + \dot{Q}_{sens} - \dot{Q}_{HVAC}$$

$$m_{a,2}c_{p,a}\frac{dT_{a,2}}{dt} = \sum_{j=1}^{5} h_{a,j}A_{w,j}\left(T_{w,j} - T_{a,2}\right) + A_c\left(\frac{T_c - T_{a,2}}{\frac{1}{h_c}+\frac{R_d}{2}}\right) + \dot{m}_{gap}c_{p,a}\left(T_{a,1} - T_{a,2}\right) \tag{5}$$

In equation 4, the term in the left side of the equation represents the thermal capacitance of the air stored in zone 1. The five terms in the right side represent the heat transfer exchange with the five surfaces (3 walls, ceiling, floor), the heat transfer exchange with the curtain, the energy exchange at the open space between the two zones (below and above the curtain), the internal sensible energy released and the sensible load removed by the air conditioning unit, respectively. Notice the convective and diffusive resistances of the curtain in the second term of the right hand side of the equation.

The equation of zone 2 is much similar to that of zone 1 except that it does not include the energy of the released mositure and the HVAC sensible load because the ducts are located in zone 1.

m_{air} is the mass of air (kg), $c_{p,a}$ is the specific heat of air at room temperature (1012 kJ/kg.K), T_a is the air dry-bulb temperature, h_{air} is the internal convective coefficient (5W/m^{2}K), A_w is the surface area of the wall /floor, T_w represents a wall surface temperature, T_c is the temperature of the curtain, R_d is the curtain dry resistance, h_c is the convective heat transfer between the curtain and the air, Q_{sens} is any generated sensible load in the room, and Q_{HVAC} is the sensible load removed by the HVAC.

3.2. Air Gap Modeling

It is important to model the amount of air-flow in the gap width behind the curtain (Fig 2). Assuming that the channel is narrow, the flow in the gap space can be considered Poiseuille. The flow enters at the bottom opening of the gap in the clearance between the curtain and the floor at T_{a1} and flows upward driven by the buoyancy forces. After comparing the dimensions of zone 2 with the Rayleigh number of the air, the air flow is assumed to be Poiseuille, and hence equation 6 may be used to calculate the mass flow rate of air (Bejan 1993):

$$\dot{m}_{gap} = \rho g \beta \frac{(T_{a2} - T_{a1}) d^3}{12v} w \quad (6)$$

Where ρ is the density of air (kg/m^3), g is the gravitational acceleration (m/s^2), β is the thermal expansion of air (K^{-1}), d is the gap width (m), w is the width of the room (m), ν is the kinematic viscosity of air (m^2/s).

3.3. Curtain Modeling

The mass balance of water vapor in the curtain is:

$$\rho_c t_c \frac{dR}{dt} = \frac{P_{v,1} - P_c}{\left(\frac{1}{h_m} + \frac{R_e}{2}\right) h_{fg}} + \frac{(P_{v,2} - P_c)}{\left(\frac{1}{h_m} + \frac{R_e}{2}\right) h_{fg}} \tag{7}$$

ρ_c is the density of the curtain (kg/m^3) and t_c is its thickness (m)

The energy balance for the curtain is:

$$\rho_c C_c t_c \frac{dT_c}{dt} = \rho_c C_c h_{ad} \frac{dR}{dt} + \frac{(T_{a1} - T_c)}{\frac{1}{h_c} + \frac{R_d}{2}} + \frac{(T_{a2} - T_c)}{\frac{1}{h_c} + \frac{R_d}{2}} + h_{rd}(T_{mrt} - T_c) + h_{rd}(T_{w1} - T_c) \tag{8}$$

h_{ad} is the heat of adsorption (kJ/kg, of the same order as h_{fg}), h_{rd} is the linearized radiative heat transfer coefficient between the air and the curtain (W/m^{2}K), T_{MRT} is the mean radiant temperature in the room (K), T_{w1} is the temperature of the wall behind the curtain (K). The regain R has a definite relation to the relative humidity of the water vapor through a property curve of the regain versus relative humidity (Morton et al 1975) (Fig 3).

3.4. Envelope Modeling

The thermal response of the space is modeled as a simple-lumped resistance-capacitor element as shown in Fig 4. The temperatures T_i, T_o and T_m are the inner, outer and uniform temperatures of the wall respectively. The overall element resistances are expressed in terms of the ratios *f* for resistances r in terms of the overall wall thermal resistance r_m and overall thermal capacitance C_m, (Laret et al, 1980):

$r_i = f\, r_m; \qquad r_0 = (1-f) r_m$ (9)

$C_m = \sum_{i=1}^{n} x_i \rho_i c_i$ (10)

Where x_i is the thickness of the wall layer i, c_i is the material capacitance of the wall layer i, h is the convective heat transfer coefficient, and ρ_i is the wall layer density. The external convective heat transfer coefficient is based on the external sol-air temperature Text to account for direct solar radiation for external walls, and air temperature for internal partition walls.

The *f* is based on the work of Laret et al (1980) for calculating the inner and outer resistances.

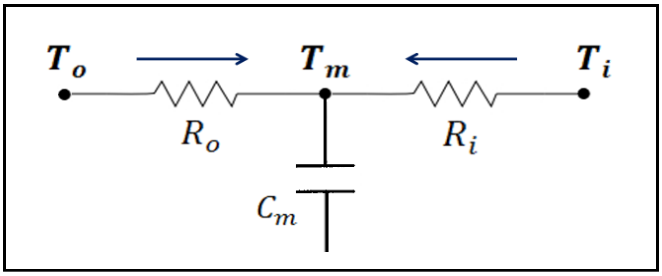

Figure 4: Simple Lumped resistance-capacitor construction element

CHAPTER 4

NUMERICAL SOLUTION

In order to simulate the buffering capacity of hygroscopic curtains inside a space air conditioned by a DX system, the following is needed:

- the space envelope construction material
- the external and internal sensible and latent load schedule
- the material and physical properties of the hygroscopic curtain
- the capacity table of the DX system (manufacturer's data) and its operation constraints (minimum on and off periods).

Starting from arbitrary initial conditions and using the Euler Forward explicit scheme with a time step of 1 second, the space walls inner surface temperature are computed. Also, depending on the indoor dry and wet bulb temperatures and the outside air temperature, the DX air conditioning total capacity and sensible heat ratio as well as the amount of condensate drain are computed. The six coupled variables T_{a1}, T_{a2}, w_1, w_2, R, Tc are then calculated.

The simulations are performed for a typical summer day in Beirut, Lebanon (July 31st, 2010) to obtain the space envelope and curtain thermal and moisture conditions over the whole day.

At the end of the 24 hours operational period, the initial conditions are recalculated and used again as an input in the cyclic simulations until convergence is achieved. Different time steps are chosen for the simulation 1 and 0.75 s over the whole simulation period. The higher time step of 1 s is found sufficient to produce a stable accurate solution.

CHAPTER 5

EXPERIMENTAL METHODOLOGY

5.1. Purpose

Experiments were conducted in the AUB environmental climatic chamber in the *Energy Lab* to validate the moisture buffering capacity model of hygroscopic curtains. It is always more credible to verify any devised equation by experimentation. This ensures that the assumptions made when modeling the phenomena are valid.

5.2. Dimensions and Makeup

The full scale climatic chamber is 3.4 meters x 3.4 meters and its height is 2.8 meters. The walls of the chamber are highly insulated with metallic sheet coverings in both sides of the wall to prevent moisture migration through the walls. The walls were constructed using typically used materials in Lebanon with a high level of insulation. A 9,000 BTU/hr DX split unit system with an indoor temperature control thermostat was installed on the west facing wall of the room.

5.3. Hygroscopic Curtain

The curtain was arbitrarily placed on the wall facing South, with a gap of 10 cm between the curtain and the South wall, dividing the room into 2 zones: In front of the curtain (main room) and behind the curtain to facilitate air flow between the 2 zones as in Fig 2. The curtain was given a wavy structure in order to increase its surface area and enhance the absorption mechanism (Fig 5).

Figure 5: The wavy curtain hung in the experimental room

The curtain used in the experiment is 100% untreated cotton having the following properties:

- thickness = 4.1 mm
- dry density = 350 kg/m^3
- dry thermal resistance R_d of 0.0158 m^2°C /W
- evaporative resistance R_e of 4.28 m^2Pa/W measured by a sweating guarded hot plate inside an environmental chamber

5.4. Wall Heating Blankets

In order to model outdoor summer conditions of Beirut, Lebanon, surface heating blankets were installed at this wall in order to bring its surface temperature to a one similar to that of an external wall. These polyester heaters have been covered by aluminum sheet coverings to ensure a uniform temperature over the whole wall surface area and to prevent any moisture absorption by the blanket fabric material. This is one of the attempts to minimize experimental errors induced by the heaters. The temperature was checked for homogeniety by the use of the thermal camera available at the AUB Energy Lab.

5.5. Moisture Generation

The generation of moisture due to occupants was modeled by locating a humidifier inside the room that will release a constant water vapor flow rate. The humidifier chosen for this experiment was "*Ultrastar F.B. Burg 8h-870e".* A fan was placed in the room directed toward the humidifier's nozzle outlet, in order to ensure distribution of the water vaport released.

5.6. Experimental Scenario

The AC was turned on initially in order to ensure steady state operation or to know the exact initial conditions of the experiment. The inside air temperature of the chamber was set at 23°C and the steady state relative humidity and temperature in the chamber were monitored and recorded via a data acquisition system, over a time span of at least 10 hours. An extra heating source supplying 700W was used in order to ensure that the AC turns on to remove this heat! Once the temperature and relative humidity in the chamber are no longer varying with time (i.e., reached steady state), the humidifier was remotely turned on to release moisture at a flow rate of 1.4 x 10^{-5} kg/s, which is the maximum value that can be released continuously from this humidifier. This value roughly corresponds to the moisture generated by one sedentary person. The DX system was also turned off to simplify the moisture balance equations, as it is hard to measure the exact mass of water removed by the cooling coil.

That being said, the moisture released by the humidifier ends up either in the air or absorbed by the curtain. Since the generated moisture flow rate is known, then using a relative humidity and an air temperature sensor,one can monitor the portion of this generated humidity that stayed in the air. The difference between these 2 numbers is more or less the water vapor absorbed by the curtain.

During the moisture release (90 minutes), the chamber air temperature and relative humidity, surface temperature of the walls, AC supply temperature and flow rate were recorded to assess the moisture buffering capacity of the curtain and the buffering period. The moisture release inside the chamber would mimic the situation when there is moisture load coupled with insufficiently low sensible load to bring the air conditioning system to the ON operating mode. The experiment would show the effect of a hygroscopic curtain in damping this surge of humidity.

CHAPTER 6

EXPERIMENTAL RESULTS

The recorded air temperature inside the chamber was close 23.5 °C with a relative humidity of 68.5%. The average walls (excluding the South wall) temperature varied between 23.5 °C and 24.8 °C, prior to turning the air conditioning system off. The temperature of the South wall was kept at a uniform temperature of 28 °C. During the OFF operation of the air conditioning system and the continuous moisture release, the air temperature of the insulated chamber increased at a very slow rate to reach a temperature of 25°C at the end of the two hour experimentation. Similarly, the surface temperature of the other walls of the chamber increased at a very small rate and at the end of the experiment the maximum temperature change was 0.3 °C. The temperature of the air and humidity in front and behind the curtain was continuously monitored and recorded for comparison with the numerical model results.

The instantaneous surface temperature of the walls and air were used as input into the curtain and air space models to obtain the relative humidity in front and behind the curtain. Figure 6 and 7 shows the variation of the experimentally measured and numerically predicted room relative humidity for both cases with time. The initial relative humidity is lower behind the curtain because of the warmer South facing wall surface.

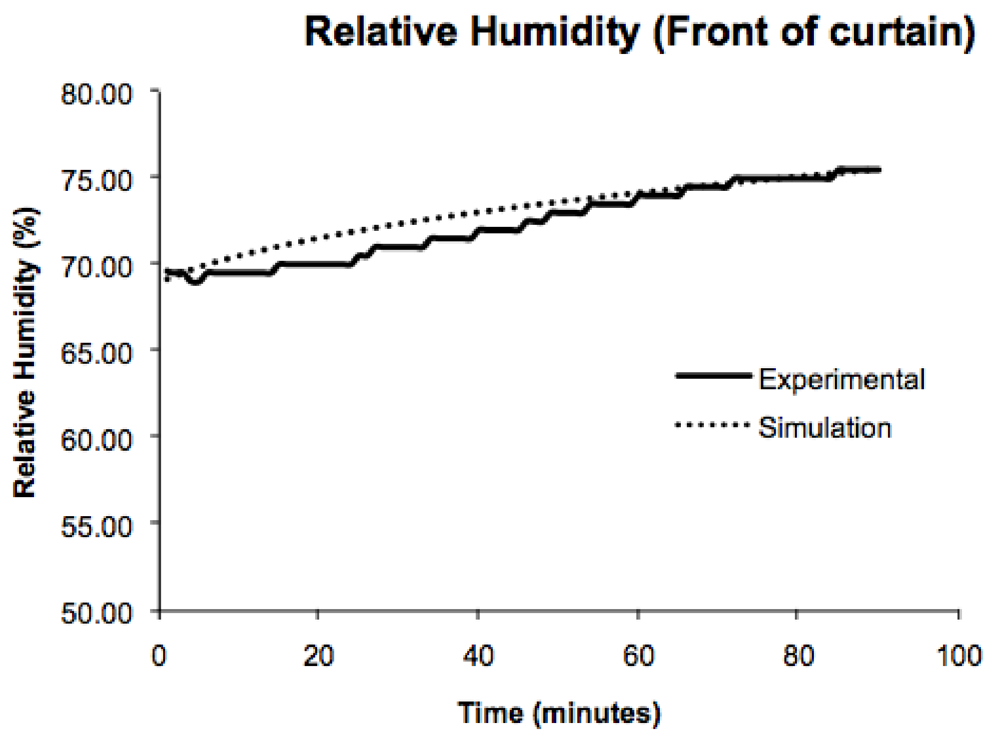

Figure 6: Comparison between the simulation and experimental data of the air relative humidity in the zone in front of the curtain

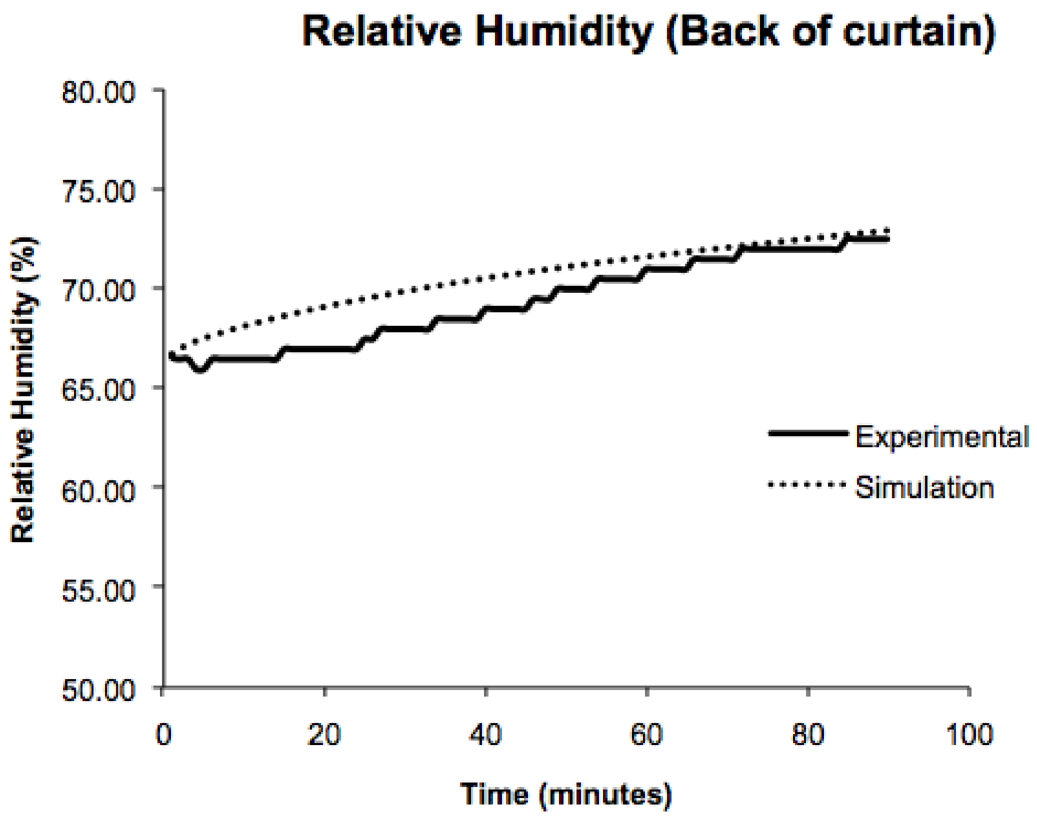

Figure 7: Comparison between the simulation and experimental data of the air relative humidity in the zone in the back of the curtain

The figures show good agreement between the model and the experimentation, however the experimental data were slightly lower than model especially at the onset of moisture release which could be related to the lumped approach of moisture modeling. The model assumes instantaneous mixing and homogenous moisture in the two zones where in reality there would be time lag for the moisture to distribute itself inside the space. A moisture balance check was made by weighing the curtain:

- at the end of the 10 hour pre-conditioning period (time t = -10 hrs)
- just before the moisture release (time t = 0), and
- at the end of the two hour experimentation (time t = 2 hrs)

The weighing was performed inside the environmental chamber to minimize any possible experimental error when exposing the curtain to the outside air.
In addition, the weight of the water humidifier tank was weighed before and after experimentation. The decrease in the amount of water in the humidifier tank was accounted in the weight increase in the curtain and the air humidity ratio (w) increase inside the chamber with a difference of about 5%.

CHAPTER 7

SCENARIO 1: 24 HOURS CASE STUDY

7.1. Purpose

The model has been developed earlier in the “Modeling of the Space and the Humidity Interactions”. The equations were verified in the experimental section. The experiment conditions were too specific, and mainly aimed to be as simple as possible in order to verify the equations and assumptions. For example, the duration was only 2 hours and the Air Conditioning system was not turned on.

As such, a case study is needed to model more realistic conditions for a full day with a proper model of the Air Conditioning system (DX Split Unit). This case study was simulated in order to model a typical office space in Beirut, Lebanon. Physical data for the office and the wall envelope were modeled, in addition to a typical occupancy profile and an internal sensible heat gain profile. The Outdoor Air Temperature and the Incident Irradiance on the office’s external wall were also simulated for a summer day in Beirut – July 31st, 2010. A specific HVAC unit system was modeled by entering the manufacturer’s data into the simulation.

The space modeled is a small office in a building in Beirut, Lebanon. The space dimensions are 5m x 5m x 3m (area of 25m^2 and a total volume of 75m^3). It has 4 vertical walls (3 of which are internal partitions adjacent to conditioned spaces and 1 is an external wall facing the South direction). The ceiling and the floor are considered as internal partitions, adjacent to conditioned spaces.

Typical wall U-values for Beirut, Lebanon were considered for the Case Study. The table below shows the wall make-up and properties of each layer:

Table 1: Wall Layer Properties

Layer Number	Layer Material	Thickness [mm]	Density [kg/m^3]	Specific Heat [kJ/kg-K]	Thermal Conductivity	R-Value [m^2-K/W]
1	Gypsum Board	20	800.9	1.09	0.161	0.09862
2	LW Concrete Block	100	608.7	0.84	0.38	0.26681
3	Face Brick	100	2002.3	0.92	1.33	0.07626

The overall thickness of the wall is 22 cm, and the overall U-value of the wall is 1.28 W/m^2-K.

7.2. Outdoor Conditions

Weather data was obtained from (www.weatherunderground.com) for Beirut, Lebanon on the day of July 31st, 2010. The outdoor dry bulb temperature varied between 27°C and 35°C, whereas the irradiance varied between a maximum of 420W/m^2 at noon and zero during the night, in a symmetric manner. Figures 8 & 9 in the Appendix show the hourly variation of the outside air dry bulb temperature and irradiance, respectively.

Note that the external conditions will not have a significant impact on the sensible load of the space, because only one of the 4 walls is exposed to the exterior, and the high internal sensible loads of an office overcome the sensible heat transfer through the external wall. However, the total gross capacity of a DX split unit system depends on the outdoor dry bulb temperaure, as this is the temperature of which heat is rejected. The less the outdoor dry bulb temperature, the easier it is for the compressor to reject the heat, and consequently the higher the capacity of the DX system.

7.3. Internal Loads

There are two types of loads on the space in any occupied space:

- Sensible load: is the load that causes a rise in the surrounding temperatures. Examples include the human sensible heat released (approximated by 70W for a sedentary person), area lights, computer equipment, and miscellaneous process loads that do not include generation of water vapor.
- Latent load: is the load that causes an increase in the humidity ratio of the surrounding air. Examples are people, with a sedentary person generating at around 1.46×10^{-5} kg/s. Coffee machines evaporate a lot of water, and are consequently considered as a latent load to the space.
- If the outside air is hot and humid (such as in August), any infiltration is also considered a considerable sensible and latent load on the space.

The mass flow rate of generated moisture had 2 distinct values: a constant maximum during the occupied period of 24×10^{-5} kg/s (7:00AM to 7:00PM) and a low minimum value of 1.44×10^{-5} kg/s during the non-occupied period (7:00PM to 7:00AM), to account for any infiltration effect. Six full-time occupants were assumed to occupy the office, with each releasing 1.46×10^{-5} kg/s of moisture, adding up to 8.76×10^{-5} kg/s. the total 24×10^{-5} kg/s took into account the generation due to equipment and envelope infiltration.

As for the sensible load, each of the occupants dissipates 70W of sensible heat to the surroundings, adding up to 420W. The total sensible load is given the values of 600W, 1200W, and 2000W (peak) during the occupied region, and 200W during night-time to account for any equipment that is ON 24/24.

7.4. HVAC System

The HVAC system modeled consists of a DX Split Unit system, which is typical for many offices due to its low cost in the market and high local controllability of thermal conditions. A Variable Air Volume system, for example, generally does not allow a control of the air temperature.

A “1 ton” market DX split unit system was chosen from TRANE®:

- The indoor evaporator unit: MCX-512G1
- The outdoor condenser unit (compressor) TTK 512P1

A screen shot of the data sheet for this unit is provided below (Table 2). This unit has a capacity of 12,300 BTU/hr, equivalent to 3,605 W (~1 ton) at rated conditions, hence the number 12 (for 12,000 BTU/hr) in the unit’s nomenclature.

Table 2: DX Unit - Manufacturer's Data (TRANE)

Outdoor D.D	I.D W.B	Gross Cap.	SENS CAP. AT ENTERING D.B. TEMP. 72	74	76	78	80	Compr. kW
85	61	11.3	8.5	9.2	9.9	10.6	11.0	0.91
	65	12.2	7.0	7.8	8.5	9.2	9.9	0.95
	67	12.7	6.3	7.0	7.7	8.4	9.2	0.97
	71	13.6	4.7	5.4	6.1	6.9	7.6	1.01
95	61	11.0	8.4	9.1	9.8	10.4	11.0*	0.99
	65	11.9	6.9	7.6	8.3	9.0	9.8	1.03
	67	12.3	6.1	6.8	7.6	8.3	9.0	1.05
	71	13.2	4.6	5.3	6.0	6.7	7.5	1.09
105	61	10.5	8.1	8.8	9.5	10.0	10.6*	1.08
	65	11.4	6.6	7.3	8.0	8.7	9.5	1.12
	67	11.8	5.8	6.5	7.3	8.0	8.7	1.14
	71	12.7	5.0	5.0	5.7	6.5	7.2	1.19
115	61	10.1	7.8	8.5	9.2	9.8	10.3*	1.17
	65	10.9	6.4	7.1	7.8	8.5	9.2	1.21
	67	11.3	5.6	6.3	7.0	7.8	8.5	1.23
	71	12.2	4.1	4.8	5.5	6.2	6.9	1.28

The rated conditions are the following:

- The rated outdoor dry bulb temperature is 35°C

- The rated inlet wet bulb temperature is 19.4°C (entering the coil). For 100% recycled air systems, this is the same as the set wet bulb temperature of the zone.
- The rated flow rate is 400CFM
- The rated Coefficient of Performance is 2.9

The thermostat setpoint for this Scenario was taken to be 26°C with a temperature variation of ± 1°C.

The compressor cannot possibly cycle on and off at very high frequencies, because such a behavior will definitely reduce its lifespan. Accordingly, a minimum time on and a minimum time off are specified by the compressor's controller. The minimum time on is 5 minutes, and the minimum time off is 3 minutes.

If the compressor is On, it will not turn off until 2 conditions are met:

- At least 5 minutes have passed
- The inside temperature decreases to below 25°C

Similarly, if the compressor is Off, it will not turn on until:

- At least 3 minutes have passed
- The inside temperature increases above 27°C

When a 12,300 BTU/hr unit is specified, this does not mean that it will run at this capacity all the time! This is the rated capacity, which is only achieved at the rated conditions listed above. Similarly, the sensible heat ratio and the coefficient of performance are not constant.

To begin with, the outdoor dry bulb temperature and the coil entering wet bulb temperature must be known in order to know the gross total thermal capacity of the unit and the COP. Note that many manufacturers specify the Electrical Input Ratio (EIR) instead of the COP, with EIR = 1/COP (kW_{elec} / $kW_{thermal}$).

The gross capacity incorporates both the sensible and the latent ones. The sensible capacity depends on the entering dry bulb temperature of the air. The latent capacity is then subtracted from the total gross capacity.

For example, for a fixed inside wet bulb temperature of 19.4°C, the gross capacity of the DX system decreases from 12,700 BTU/h to 11,300 BTU/hr when the outside dry bulb temperature increased from 29.4°C to 46.1°C. For a country like Lebanon, the temperature profile does not reach temperatures as high as 46°C, and the gross capacity for the date of 31st of July will not vary more than 4%. As such, this effect is neglected in order to facilitate interpolation from the manufacturer's data table. This gross total capacity varies according to a bi-quadratic equation incorporating the outside dry bulb and the inlet wet bulb temperatures. Below is an example of such equation:

$$Q/Q_{rated} = a + bT_{wb} + cT_{wb}^2 + dT_{db} + eT_{db}^2 + fT_{wb}T_{db} \quad \text{, where} \qquad (11)$$

a = 0.40731210

b = 0.04517144

c = 0.00008412

d = 0.00140582

e = -0.00003830

f = -0.00046771

As for the Energy Input Ratio (EIR) of the DX unit (which is the inverse of the COP), another biquadratic equation is given by:

$$EIR/EIR_{rated} = a + bT_{wb} + cT_{wb}^2 + dT_{db} + eT_{db}^2 + fT_{wb}T_{db}\text{ , where} \qquad (12)$$

a = 0.72724128

b = -0.02055985

c = 0.00075095

d = 0.01355680

e = 0.00040789

f = -0.00086178

Note that for a 100% recycled air (which is the case modeled), the Sensible Heat Ratio (SHR) varies with the room indoor dry bulb temperature for a given total gross capacity. A correlation may be fitted to these data points. Alternatively, the SHR may be calculated from the coil bypass factor BF and the apparatus dew point temperature (T_{ADP}). The BF method is not used in this study.

7.5. Results

This section will investigate the effect of hygroscopic curtains on both the air temperature and relative humidity of the office space. In addition, a qualitative explanation of the moisture buffering process during the 24-hour operation of the DX air conditioning system operation will be detailed.

Two main simulations were carried out for *Scenario One* to obtain the temperature and relative humidity transient values during a typical summer day in Beirut, Lebanon (July 31st 2010): one in the presence of the hygroscopic curtain and another without a curtain.

In order to get accurate results independent of the initial conditions, each simulation was performed for 4 consecutive days, and data for the fourth day was adopted. The fourth day started and ended with the same values, no matter what initial conditions of the temperature and humidity were inputted. The transient temperature profiles for the two cases (i.e. in the presence of and in the absence of a curtain) are very similar with no appreciable difference. Figure 8 shows the cyclic steady state temperature variations for any of the 2 simulations, along with the curtain temperature, discussed later .

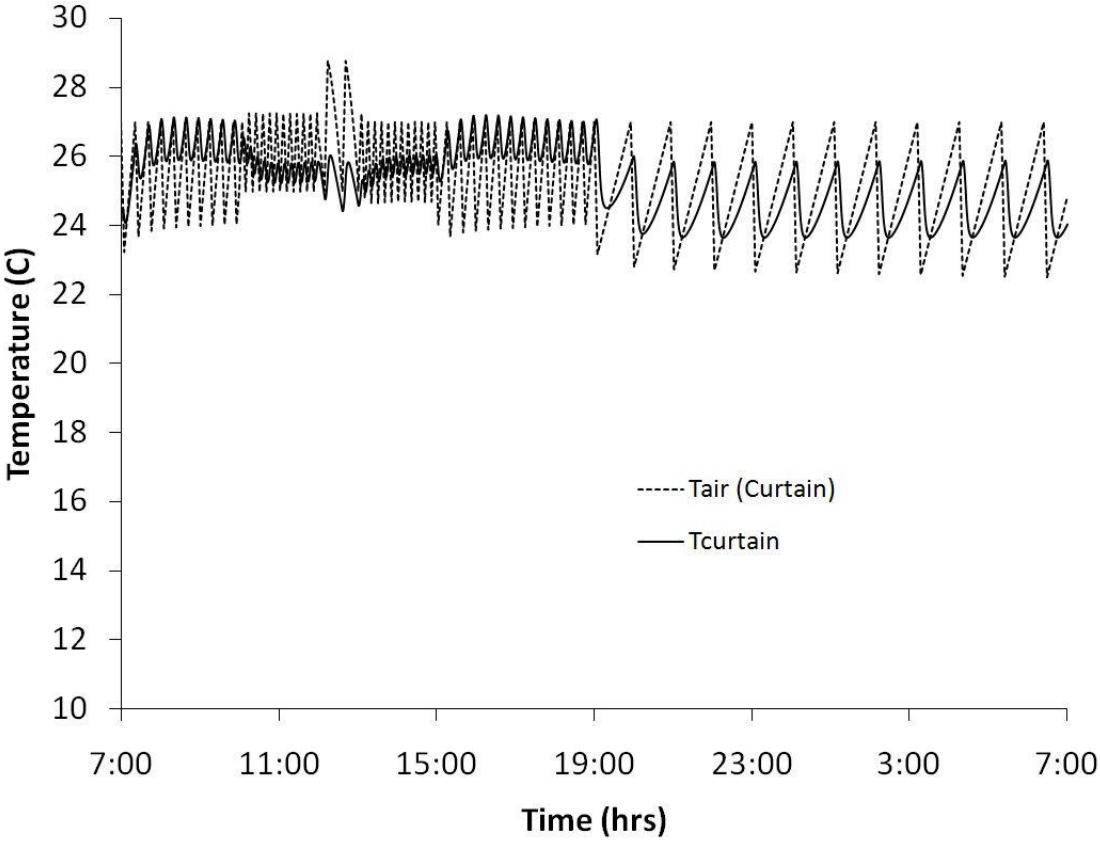

Figure 8: Curtain & air temperature variations for the whole day

The office temperature in both the occupied and non-occupied periods was maintained very close to 25.2°C since the ON/OFF operation of the DX system is controlled by the indoor air temperature. In the occupied period, more cycling is observed because of the higher sensible load that raises the air temperature to the upper limit. On the other hand, less cycling is observed in the non-occupied period in addition to larger temperature variation because the sensible load is lower, meaning that when the AC is OFF, reaching the upper setpoint temperature require more than the minimum time OFF (3 minutes) to be achieved.

The effect of the hygroscopic curtain is more pronounced on the relative humidity as is shown in Figure 9. The simulation results clearly demonstrate that when compared to the no curtain case, the use of a hygroscopic curtain results in:

- lower indoor relative humidity during the occupied period
- in higher relative humidity during the non-occupied period

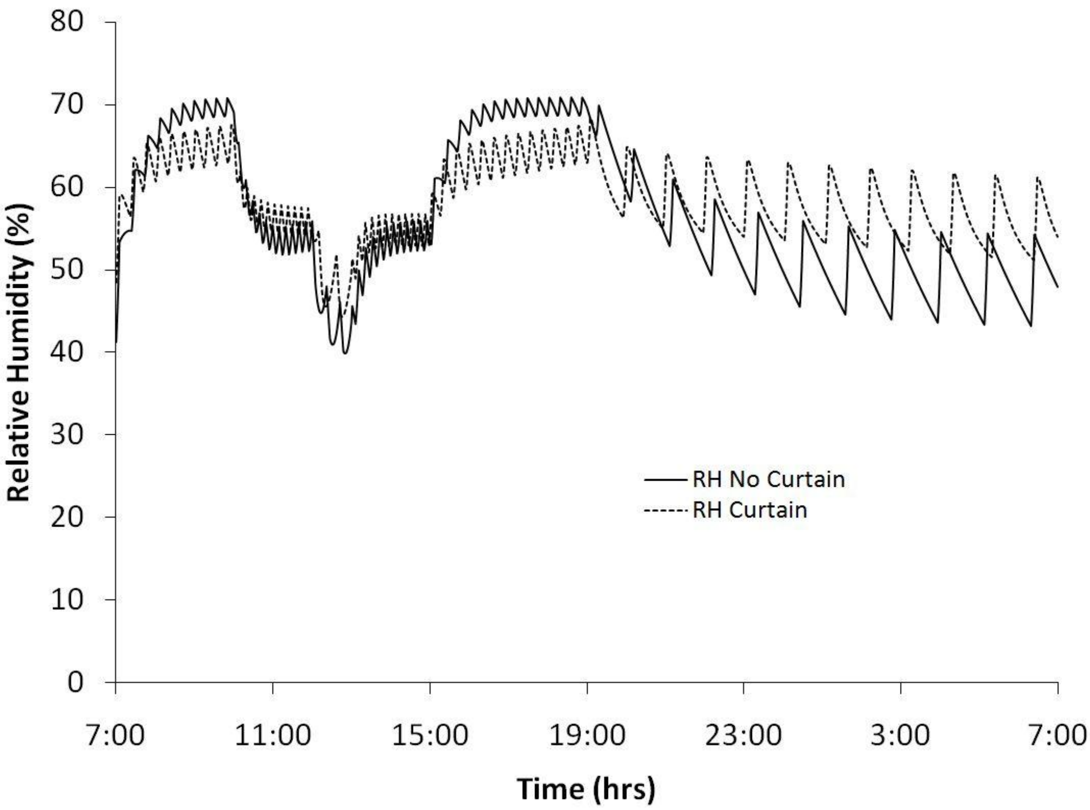

Figure 9: Relative humidity variations for the whole day

For the case where no curtain was installed, the internal relative humidity ranged from a minimum of 40% during the peak load and 70% during the part load of the occupied period, and fluctuated around 50% during the non-occupied period. As for the case where the curtain was present, the average relative humidity ranged from a minimum of 45% during the peak load and 67% during the part load of the occupied period, and fluctuated around 57% during the non-occupied period.

During the part load operation the hygroscopic curtain buffers the fluctuation of relative humidity and reduces its variation. During the peak load, however, the curtain has little effect because the air conditioning system is capable of controlling both the temperature and relative humidity. In the non-occupied period, the air conditioning system regenerates the moisture in the hygroscopic curtain and lowers its regain. This process of desorption of the curtain's moisture raises the relative humidity during the non-occupied evening period. The desorped moisture will stay in the space all night untill the HVAC system operates for longer periods during higher sensible loads in the day, where it will cause a load on the space for the first couple of hours of the second day. Chances "throwing away" the desorped moisture will be investigated in *Scenario Two*.

The moisture buffering process of the hygroscopic curtain can also be explained in Figures 8 and 10. At the start of the occupied period (7:00AM), the curtain regain is low (Fig 8) with a potential to absorb moisture if the curtain is subjected to an increase in the relative humidity. When the generated moisture increases during the occupied part load period, the following happens:

- The humidity ratio will increases
- Partial vapor pressure of air will increase

- The curtain regain will increase
- The curtain temperature will increase due to the $h_{adsorption}$ (~ h_{fg}) (Fig 10)

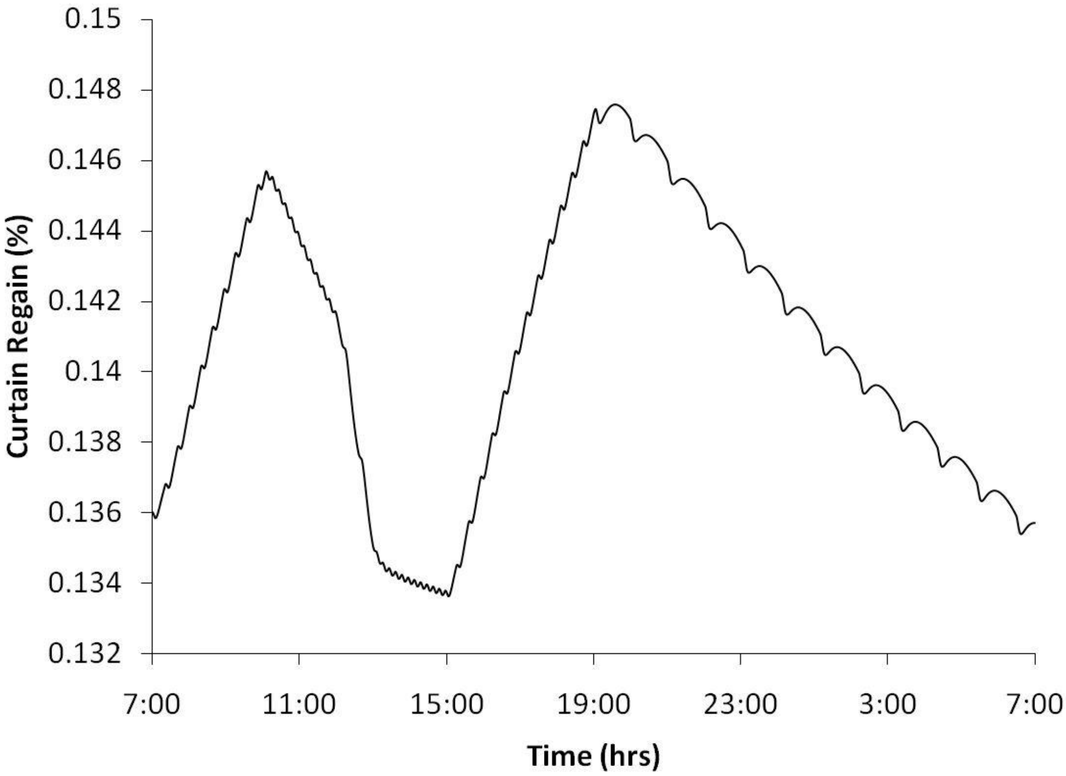

Figure 10: Curtain Regain variation for the whole day

During the peak occupied period, the AC will be running for the most of the time for a larger periods due to the space sensible load being closer to the AC rated power. Accordingly, the exact inverse of the steps above will happen.

Even though the curtain releases the absorbed moisture in the peak period, the capacity of the DX system is capable of maintaining acceptable relative humidity inside the office.

During the non occupied period, the low moisture generation aided by the DX air conditioning system running for small periods of time, desorbs the curtain and lowers its temperature as seen in Figures 8 and 10.

The daily gross capacity of the DX system was almost the same for both the curtain and the non-curtain cases (~20kWh), divided into 13 kWh of sensible energy consumption and 7 kWh of latent energy consumption. It should be noted that the results obtained are specific for the selected test case; however the experimental and modeling results clearly demonstrate the effect of hygroscopic curtains in moderating the moisture inside offices during the part load operation of the DX system. The energy savings that may result from the moisture buffering of hygroscopic curtains could be improved if night ventilation is used to get rid of the desorped humidity rather than operating the AC system or leaving the desorped humidity until the next occupied day to be removed by the AC. The night ventilation option is described next.

CHAPTER 8

SCENARIO 2: NIGHT PURGING

8.1. Purpose

The purpose of the second scenario is to model the effect of what we call "night purging", whereby outdoor air is introduced to the office during the night in order to remove the sensible and latent storage effects of the space.

The sensible storage is due to the external wall of the room that stores heat due to the fact that it had been exposed for high external dry bulb temperatures and sunlight radiation. The external walls are generally made of concrete blocks, which have a considerable heat capacity. Accordingly, the stored energy during the day will be released to the space during periods of low external temperatures. This is known as the sensible storage effect of materials.

By analogy, the latent storage effect occurs due to the hygroscopic curtain absorbing moisture during periods of high latent load (occupants during the day), and releasing this moisture to the space during the night (when the office is no longer occupied).

The disadvantage of the storage effect is that the released heat and moisture will cause direct space loads during the next day of operation. That is, the HVAC system will have to remove the load from the occupants and equipment in addition to the heat and moisture released during the night, resulting in a higher AC consumption, and possibly a slightly lower level of thermal comfort if the storage effect is significant and not accounted to when the HVAC system was sized.

As such, it would be of great benefit if the impact of the storage effect is minimized. That is, if there is a possibility to “throw” this stored energy outside the room before the HVAC system is operational the second day, we would be saving energy by reducing the space sensible and latent load.

One way of getting rid of the storage effect is by “night purging”, which consists of introducing outside fresh air during non-occupied periods to the space. However, one cannot introduce hot and humid air to the space, as this will worsen the inside conditions! The introduced fresh air must have:

- a lower dry bulb temperature to remove the sensible heat, AND
- a lower moisture content (humidity ratio) to decrease the relative humidity inside the space

8.2. HVAC System Operation

Night purging is accomplished through the use of a separate fan that supplies the zone with untreated fresh outdoor air. The DX Split unit operates for 12 hours between 7:00AM and 7:00PM, when the office space is occupied. After that, the DX system is turned off, and the night ventilation fan is turned on for 12 hours. An airflow rate of 200CFM is sufficient for our case.

8.3. Applicability

Note that in order to ensure effective “Night Purging”, the introduced air must be at a lower temperature and a lower humidity ratio than the inside air. This is crucial in order to get rid of the storage effect.

As such, the dry bulb temperature of the introduced air should be below 27°C. This temperature may be exceeded provided that this doesn't occur immediately before occupancy. In other words, if the introduced air enters at 30°C from 9:00PM to 11:00PM, this will not cause a problem, as the inside air will have sufficient time to cool during later hours.

As for the entering air's humidity ratio w, the requirements are a bit more stringent. It is not advised that a very humid air enter the space, as condensation might occur, and consequent problems in the building structure will arise (such as mold formation). The humidity ratio of the introduced air should be below $0.010 kg_w/kg_a$. Higher humidity ratios (up to $0.014 kg_w/kg_a$) may be introduced during early hours of night purging. Note that it is better to have the air humidity as less as possible, as this will improve the relative humidity inside the office during the first three hours of occupancy (7:00AM to 10:00AM), where the sensible load is relatively lower and the relative humidity is higher. A lower humidity ratio (and consequently a lower wet-bulb temperature) results in a lower HVAC energy consumption, according to the manufacturer's datasheet (Table 2)

Actual weather data for Beirut city, Lebanon, was obtained from www.wunderground.com. The cooling season in Beirut extends from April to November (around 8 months). By examining the weather data (dry bulb temperature and relative humidity), it is clear that outdoor temperatures and relative humidity during the months of June, July, August, and September are very high and night purging will actually worsen the case!

Data for April, May, October, and November seem suitable for the phenomenon of Night Purging. Detailed plots for relative humidity for the month of April are provided in the appendix, along with Table 3, which summarizes the results.

8.4. Results

Simulations were performed for the same office space of the first scenario. The total simulation time was 48 hours.

During the first 12 hours (7:00AM to 7:00PM), all conditions and results were the same as those for the first scenario.

Night purging was applied during the second 12 hours (nonoccupied office from 7:00PM to 7:00AM). This night purging caused the inside air to be at a state close to the external environment. The temperature follows the same variation curve, with the inside temperature being slightly higher than the outside due to a small sensible load that exists at night. The relative humidity is also a bit higher inside due to the desorption of the curtain (the latent load in the space is lower).

During the third 12 hours (the day after the night purging), the same loads were applied as in the first day. The outside air moisture content was relatively low during the night and the state of the inside air was brought close to the that of the outside air. As a result, the occupied period of the space started off with a lower moisture content. This phenomenon remained active for a couple of hours, especially that the DX system was operating under part load, meaning that it was not ON all the time, and the moisture buildup was higher than in the peak load operation. After this period, the night purging will not have a significant effect on the air conditions, as the high internal loads will take over!

The last 12 hours were similar to the second 12 hours, whereby night purging is also applied in order to enhance the conditions of the 3rd day (not modeled).

Figures 11, 12, and 13 compare the simulation results of the month of April, for the air Temperature, Relative Humidity, and curtain Regain, respectively. The plots for the days before and after the night purge are superimposed.

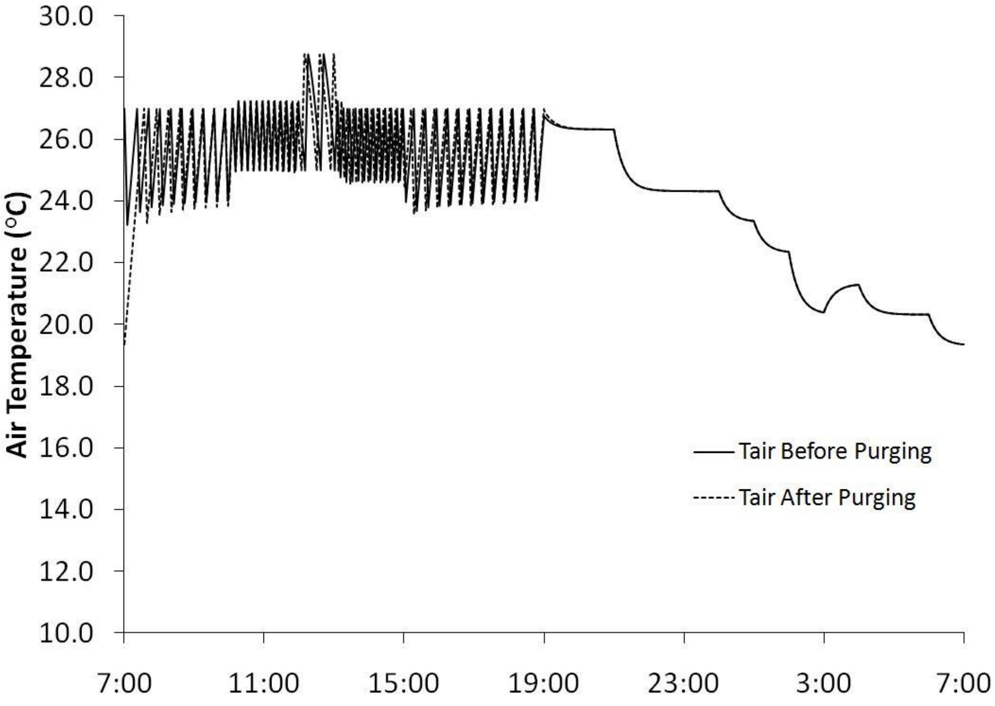

Figure 11: Air temperature before and after night purging - April

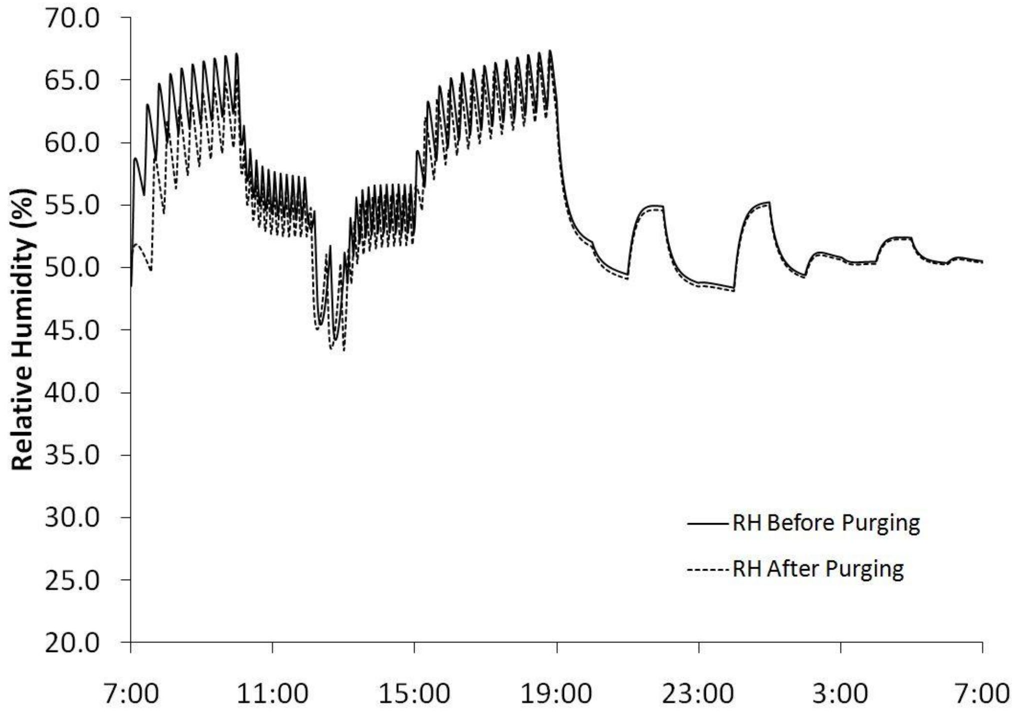

Figure 12: Relative Humidity before and after night purging – April

Figure 12 shows that the relative humidity after the night purging was applied decreased by around 3% during the first 3 hours of occupancy. After that, both cases were almost the same as the higher internal loads took over!

Note that Figures 11 and 12 show how the inside air follows the pattern of the outside one during the night time period. Night purging is able to allow the internal air conditions to closely reach those of the external air.

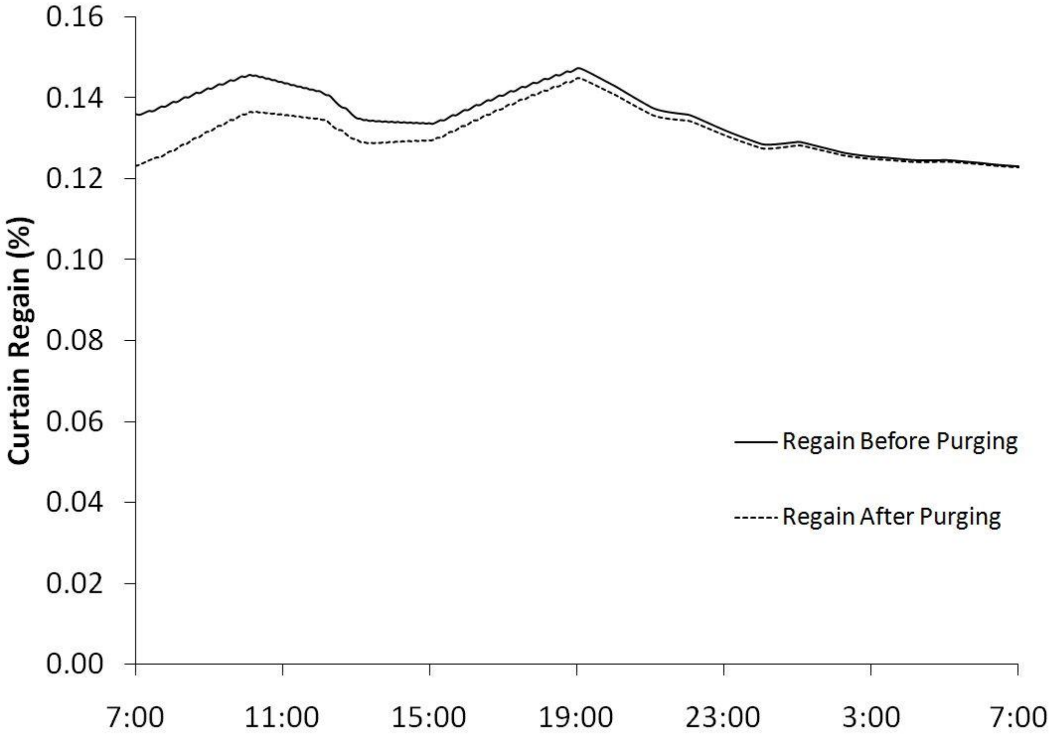

Figure 13: Curtain Regain before and after night purging - April

Figure 13 shows the that the curtain regain during the first accupied period of the day after purging was lower by 1% in value (from 13.5% to 12.5%).

The phenomenon of night purging is supposed to reduce the energy consumption of the HVAC. However, as noted above, any reduction in the relative humidity or temperature (and consequently the energy consumption) would occur during the first three hours of operation (part load), where the sensible load is relatively lower and the relative humidity is higher.

From the DX unit manufacturer’s data, it is noted that the AC energy consumption is directly proportional to the entering wet bulb temperature. As a result, if the space relative humidity decreases, then the wet bulb temperature will decrease too (for the same temperature), reflecting a reduction in energy consumption!

The Night Purging process is not a "Zero Energy" process! Instead, there is an energy consumption related to the fan that will supply the zone with outdoor air. If a high efficiency fan is specified with a Specific Fan Power (SFP) of 0.0006 kW/CFM, then a 120W rated fan should be specified to ensure the 200 CFM of outdoor air. If the ventilation fan is operating for 12 hours/day, then the energy consumption is 1.44 kWh (120W x 12 hours) Also note that it is assumed that natural exhaust can take place easily. If the office envelope was very tight, then another Exhaust fan will be required, with the same CFM/Power rating. This is not taken into consideration in this scenario.

8.4.1. Energy Consumption

Two simulations were carried on for each of the four months, one without a curtain and another with a hygroscopic curtain installed. A typical day in the middle of the month was chosen, and hourly values of temperature and relative humidity were inputted. It was concluded that, for the simulations with a curtain:

- The daily energy consumption reduced from 17 kWh to 16 kWh during the month of April (a saving of 1 kWh/day)
- The daily energy consumption reduced from 17 kWh to 16.6 kWh during the month of May (a saving of 0.4 kWh/day)
- The energy consumption reduced from 17 kWh to 15.7 kWh during the month of October (a saving of 1.3 kWh/day)
- The energy consumption reduced from 17 kWh to 15.3 kWh during the month of November (a saving of 1.7 kWh/day))

Table 3 (below) summarizes the above results:

Table 3: Improvement of %RH & Energy Consumption after night purging

	April	May	October	November
Decrease in Relative humidity (first 3 hrs)	3%	1%	5%	7%
Energy consumption per day	1 kWh/day	0.4 kWh/day	1.3 kWh/day	1.7 kWh/day

The energy saving is balanced by the increase in energy consumption due to the ventilation fans. Accordingly, energy savings is not very significant when adapting night purging. Note that if natural ventilation can be applied for the office space, then one can confidently say that the phenomenon of night purging results in a lower net total energy consumption!

Apart from the energy consumption, the lower level of relative humidity when night purging is applied will result in a better comfort level in the occupants and help protect the integrity of the building (condensation problems, mold formation and growth, damages to the building components). The effect on the occupant comfort is studied in the following section.

8.4.2. Occupant Comfort

Several correlations were developed by researchers to measure the human thermal comfort. Many parameters affect the thermal comfort. A model by Fanger lists 6 main parameters:

- Air temperature
- Mean radiant temperature
- Relative Humidity

- Air velocity
- Metabolic rate
- Clothing thermal resistance

Other researchers such as "Rohles and Nevins" developed what they call a Predicted Mean Vote (PMV), which is a thermal comfort sensation scale developed back in the 1970s. The scale is from -3 to +3, with -3 being cold, 0 being neutral, and +3 being hot. The general acceptable range for thermal comfort is between -1 and +1. The PMV scale was developed after tests were conducted incorporating the air dry-bulb temperature, humidity level, sex, and length of exposure. This correlation assumes young adult subjects with sedentary activity and a clothing resistance factor of 0.5 clo, with the Mean Radiant Temperature being equal to the air temperature, and the air velocity being less than 0.2 m/s.

The equation of the PMV according to this model is given by:

$PMV = a \times T_{air} + b \times P_v + c$, where (13)

T_{air} is the dry bulb temperature of air, P_v is the water vapor pressure in the air, and the coefficients a, b, and c are given by Table 4 below, depending on the occupant sex and the exposure period:

Table 4: Coefficients a, b, and c (used in Eq. 13)

Exposure Period, hr	Sex	Coefficients: English Units, t (°F), P_w (psia) a^*	b^*	c^*	SI Units, t (°C), P_w (kPa) a^*	b^*	c^*
1.0	Male	0.122	1.61	−9.584	0.220	0.233	−5.673
1.0	Female	0.151	1.71	−12.080	0.272	0.248	−7.245
1.0	Combined	0.136	1.71	−10.880	0.245	0.248	−6.475
3.0	Male	0.118	2.02	−9.718	0.212	0.293	−5.949
3.0	Female	0.153	1.76	−13.511	0.275	0.255	−8.622
3.0	Combined	0.135	1.92	−11.122	0.243	0.278	−6.802

For the *Scenario 2 Case Study*, the PMV was calculated by the equation above. Note that the PMV was only calculated for the first 3 occupied hours, as this time is the one that experienced the most differences in relative humidity.

When night purging was implemented, the relative humidity during the first 3 hours of the second day decreased by 3% to 7% relative to the first day (before the purge), depending on the month which month was studied.

For example, when the relative humidity decreased from 67% to 60%, the PMV decreased by 0.1. This decrease is not too significant as the PMV scale is from -3 to +3. However, a 7% reduction in relative humidity is sensed by the occupant and may help protect the integrity of the space (condensation, corrosion, or mold formation).

CHAPTER 9

SIGNIFICANCE

The phenomenon of absorption and desorption of curtains in combination with DX unit operation has not been studied thoroughly before. This study gives an insight about the physics of such process, their magnitude, the relationship among all the variables, and how to conduct an experiment to verify the claims.

Two cases were always compared: a room with no curtain and a room with a hygroscopic curtain.

Initially, one might think that 5% reduction in relative humidity by the hygrpscopic curtain case over the non curtain case is not significant. However, it is only 1 curtain that is being modeled here!

Think about another curtain, about a carpet, fabric furniture, and other hygroscopic materials in an office. If all these are to be modeled, then the phenomenon of moisture buffering will be much stronger and reductions of, for example, 15% in relative humidity will not be uncommon.

CHAPTER 10

CONCLUSION

After the results shown above, one can conclude that the introduction of hygroscopic curtains has a considerable effect on the indoor relative humidity inside conditioned spaces, especially during part load operation, where the indoor humidity is usually higher than that in the full load operation.

Mathermatical equations were formulated based on the physics of moisture absorption and based on basic energy balance. The mathematical model was verified by experiments which were conducted in conditions very similar to the simulation. The results of the experiments compared well with those of the simulation, meaning that the assumptions of the model were not far from reality.

After verifying the model, the same equations were used to simulate 2 case study scenarios for a typical small office space in Beirut, Lebanon, conditioned by a specific DX Split Unit system with a temperature control. It was seen that, when compared to the case of no curtains, the installation of a hygroscopic curtain decreased the relative humidity of the space by 5% during part load. Furthermore, if night purging was implemented for certain months, further reductions of up to 7% can be additionally achieved during the first 3 hours of the day, along with daily energy consumption reductions of up to 1.4 kWh of the total of 20 kWh (~8.5% less).

10.1. Recommendations

Installing a hygroscopic curtain proved to be certainly beneficial when it comes to reducing the indoor relative humidity. As such, it is advised to install such curtains in occupied spaces where the humidity reaches high levels. Moreover, if one has the choice between the type of fabric for the curtain, wool is to be chosen, as it has the highest buffering capacity of all fabrics, followed by cotton. Polyester has almost no buffering capacity.

In addition, it is recommended to adapt the night purging option whenever possible. Note that the approximation of the months was based on weather data for 1 year only, and if the night purging is applicable for 1 night during the month, then it is not necessary that it can be done for every night. As such, it is advised to install a simple controller that measures the air dry bulb temperature and the relative humidity, and decides accordingly if fresh air is valid to be introduced to the zone without treatement.

10.2. Future Work

The study above had many assumptions, and the scenarios simulated represented special cases at some times. Further works can definitely be implemented on the topic of moisture buffering and control. A few suggestions are:

Studying the transient operation of the DX unit's ON/OFF cycles. This is because when the compressor turns from OFF to ON, the AC capacity does not jump instantaneously from zero to the specified capacity. Instead, it will increase exponentially till it reaches the operating capacity.

In this study, HVAC fans were assumed to be OFF when the compressor turned OFF. In most AC systems, the fan remains even when the compressor is OFF. This may cause evaporation of the water droplets on the cooling coil to the air and their re-introduction to the air stream flowing to the zone.

The effect of curtains on the energy consumption during the heating season may also be studied. This could be because the curtain heats up as it absorbs humidity, and eventually come into equilibrium with the surrounding air. This may probably act as a heat source and decrease the heating load.

A type of HVAC other than DX units may also be studied, that can be equipped with a humidity controller instead of a temperature one.

And last but not least, the buffering capacity of other hygroscopic materials such as furniture, carpets, or wallpaper should be studied to determine further reductions in relative humidity.

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