IMPLEMENTATION OF A MODEL FOR THE REDUCTION OF LATENT CAPACITY OF AN AIR-CONDITIONER AT PART-LOAD CONDITIONS WITH CONTINUOUS FAN OPERATION

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ABSTRACT

The moisture removal effectiveness of air conditioning systems is significantly degraded by continuously running the indoor distribution fan. Results of experiments documented in the literature show that during the equipment off-cycle, the cooling coil acts as an evaporative cooler, providing sensible cooling at the expense of increased air stream moisture content. The degradation in the latent capacity of the air-conditioner was found to increase with an increase in the equipment off-time. A model to predict the degradation of the performance of the air-conditioner was developed by previous investigators. This model accounts for the conditions of the air entering the coil, thermostat cycling rate, air-conditioner transient performance, and the moisture-retaining characteristics of the cooling coil. This model was not implemented within a building energy simulation program when it was developed. In addition, there has been a lack of information on the potential cumulative effect of the latent capacity degradation of the air-conditioner on comfort and equipment electricity consumption over the whole cooling season for different climates.

This paper covers the adaptation and implementation of the previous model as part of the air-conditioning model in the building energy analysis program ESP-r/HOT3000. The model is then used to study the effect of moisture evaporation from the coil, during compressor off-cycle with continuous fan operation, on the temperature and relative humidity inside a residential building during the cooling season. In addition, the effect of the equipment latent capacity degradation on the electricity consumption of the air-conditioner is investigated. The findings of the study confirm the negative impact of over sizing the air-conditioner on comfort conditions inside the space.

INTRODUCTION

Researchers such as Khattar et al. (1985) and Henderson (1990) have shown that the latent performance of air-conditioner is degraded when the circulation fan is in continuous mode and the compressor is off. Khattar et al. presented experimental data that shows clearly this degradation in the latent capacity. They found, for the installation they were testing, that when the compressor run-time fraction was 25%, the moisture removal rate of the air-conditioner was 60% lower in the fan constant mode compared to that in the fan auto-mode. During the transient performance of the air-conditioner, they reported that over 19% of the moisture collected during the compressor on cycle re-evaporates during the compressor off cycle.

Henderson (1990) showed that when the compressor is off, the cooling coil acts as an evaporative cooler. When the circulation fan runs constantly and the compressor is off, vapor condensate on the coil surface evaporates back into the air stream and is delivered to the conditioned space. The rate of degradation of the latent performance is mainly a function of the equipment run time fraction. The moisture that is carried back to the conditioned space can contribute to higher humidity levels, which can be associated with a decreased sense of comfort.

Henderson and Rengarajan (1996) developed a model for the prediction of the latent degradation of the air-conditioners and heat pumps when the circulation fan is in continuous mode. The model accounts for the effect of inlet air conditions to the coil, the thermostat cycling rate, equipment transient performance, and the moisture retention characteristics of the cooling coil. This model compared favorably to experimental data from Khattar et al. (1985). At the time the model was not implemented as part of an energy analysis tool to assess the impact of the degradation of the latent capacity on the comfort within the space and the energy consumption of the air-conditioner.

In this paper the previous model developed by Henderson and Rengarajan is incorporated within the ESP-r/HOT3000 building simulation engine. A summary of the model as formulated by Henderson and Rengarajan is first presented. Then the implementation of the model within ESP-r/HOT3000 is described. Simulations for the whole cooling season are then performed for Ottawa, Toronto, and New Orleans. The results are then used to assess the impact of the degradation of the latent capacity of the equipment on the comfort conditions within the space and on the energy consumption of the air-conditioner. The effect of over sizing the air-conditioner is also investigated.

DESCRIPTION OF THE LATENT CAPACITY DEGRADATION MODEL

A complete and detailed description of this model can be found in Henderson and Rengarajan (1996). One of inputs to the model t_{wet} is the ratio of the energy associated with the maximum condensate holding capacity of the coil to the latent capacity of the coil at rating conditions. Another model input γ is ratio of the moisture evaporation rate from the coil when the compressor goes off to the latent equipment capacity at rating conditions. Other inputs include the maximum cycling rate of the thermostat N_{max} and the equipment time constant τ. In the model a certain function is assumed for the variation with time of the evaporation rate of the moisture on the coil when the compressor first turns off. It is possible using this function to find the total amount of condensate left at some point in time after the compressor turns off. One major input to the model here is the maximum amount of condensate that the coil can hold before the water starts to drain.

A first order equation is used to describe the variation of the latent capacity of the unit from start up until steady-state conditions are reached. The variation of the latent capacity based on the assumed quasi-steady operation is shown in Figure 1. It is assumed that the cooling coil holds the maximum amount of condensate it can hold when the compressor goes off after satisfying a certain sensible cooling load in the conditioned space. During the time that the compressor is off, moisture will evaporate from the coil due to the continuous operation of the circulation fan. When the compressor starts again when there is a new call for cooling, moisture starts to condense until the time t_o when the total amount of condensate on the coil is again equal to the maximum condensate holding capacity of the coil. As a result, the condensation that happens up to this point does not contribute to any change in the space moisture content. The model then assumes that only the condensation that happens between time t_o and t_{on}, the total on-time of the equipment during a full cycle, is considered in determining the effective total latent capacity of the coil. The ratio of the modified latent heat ratio LHR of the equipment to the steady-state latent heat ratio LHR_{ss} is given by

$$\frac{LHR}{LHR_{ss}} = \frac{(t_{on} - t_o) + \tau(e^{-t_{on}/\tau} - e^{-(t_{on}-t_o)/\tau})}{t_{on} + \tau(e^{-t_{on}/\tau} - 1)} \qquad (1)$$

This equation is a slightly different version than the equation given by Henderson and Rengarajan (1996). In their equation the numerator in equation 1 is set to ($t_{on} - t_o$) which does not account correctly for the first order variation of the latent capacity of the equipment. This variation of the latent capacity of the equipment since startup is given by

$$\dot{q}_l = \dot{q}_{l,ss}(1 - e^{-t/\tau}) \qquad (2)$$

Therefore the total evaporation that takes place between time t_o and t_{on} is given by

$$q_{l,total} = \dot{q}_{l,ss}\left[(t_{on} - t_o) + \tau(e^{-t_{on}/\tau} - e^{-(t_{on}-t_o)/\tau})\right] \qquad (3)$$

This equation is then consistent with the numerator in equation 1. Note that it is possible for the numerator in equation 1 to be negative. In this case the actual latent heat ratio is zero. The air-conditioner is off for a long enough period of time that all the condensation that occurs during the unit on-time evaporates back to the air-stream. This happens when the predicted time t_o turns out to be grater than the predicted total on time of the equipment t_{on}.

In order to determine the ratio in equation 1 we need expressions for t_{on} and t_o. Henderson and Rengarajan provide an equation to find t_o iteratively given by

$$t_o^{j+1} = \gamma t_{off} - \left(\frac{\gamma^2}{4t_{wet}}\right)t_{off}^2 - \tau\left(e^{-t_o^j/\tau} - 1\right),\; t_{off} \le \frac{2t_{wet}}{\gamma} \qquad (4)$$

In this equation the unknown time t_o appears on both side of the equations. An initial estimate for t_o^j is used which is then used to get a new estimate for t_o^{j+1}. This equation is valid only when t_{off} is less than the time it takes for the total amount of condensate accumulated on the coil to evaporate back into the air stream. This time is denoted by $t_{off,max}$ and is given by

$$t_{off,\max} = \frac{2twet}{\gamma} \qquad (5)$$

The equipment on-time, used in equation 1, and off-time, used in equation 4, are given as a function of the run time fraction X by the following equations

$$t_{on} = \frac{3600}{4N_{\max}(1-X)} \quad (6)$$

$$t_{off} = \frac{3600}{4N_{\max}X} \quad (7)$$

The unit run time fraction X is related to the part-load ratio PLR and part load factor PLF through the equation

$$X = \frac{PLR}{PLF} \quad (8)$$

MODEL IMPLEMENTATION IN ESP-r/HOT3000

A complete description of the cooling model and its implementation in ESP-r/HOT3000 can be found in Haddad (2004). In the ESP-r/HOT3000 model the sensible cooling load of the conditioned zones is calculated in the loads simulation routine and then passed to the cooling model for each time step. Two of the user inputs to the model are the total cooling capacity and the sensible heat ratio at rating conditions. A correlation is then used to determine the total cooling capacity at any other coil inlet wet-bulb temperatures and ambient dry-bulb temperature. The sensible heat ratio at rating conditions is used to determine the steady-state sensible and latent capacities of the coil at other coil inlet conditions.

In the ESP-r/HOT3000 model the sensible and the latent load of the space are used with the total cooling capacity to determine the part-load ratio of the equipment for the time step as given in the following equation

$$PLR = \frac{Total\_Cooling\_Load}{Total\_Cooling\_Capacity} \quad (9)$$

The part-load factor in ESP-r/HOT3000 is then determined based on the value of PLR using the equations

$$PLF = \frac{PLR}{COOL\_EIR} \quad (10)$$

where:

$$COOL\_EIR = a + b \times PLR + c \times PLR^2 + d \times PLR^3$$

(11)

Equation 11 is based on curve fitting of equipment part-load performance data.

The values of PLR and PLF can then be used to find X from equation 8. The equipment on-time and off-time can also be determined now from equations 6 and 7 where the maximum number of thermostat cycles is a user input. In case the off-time of the unit is greater than time needed for all the moisture accumulated on the coil to evaporate, then t_{off} is set to $t_{off,max}$ before using it in equation 4. It is then possible to find the time t_o iteratively from equation 4. In case the time t_o is found to be larger than the total on-time t_{on}, then the new latent heat ratio LHR of the equipment is set to zero. Otherwise the degraded latent heat ratio is evaluated from equation 1.

In the ESP-r/HOT3000 model the total cooling load in equation 9 is the sum of the sensible load and the latent load for the time step. The latent load in turn is derived from the sensible load and the sensible heat ratio SHR for the time step based on the following equation

$$Latent\_Load = \frac{1-SHR}{SHR} Sensible\_Load \quad (12)$$

Given the degradation in the latent heat ratio, predicted by equation 1, as a result of the moisture evaporation when the compressor is off, there is a reduction in the actual latent load of the space given by

$$Latent\_load\_\mathrm{Re}duction = \left(\frac{1-SHR}{SHR} - \frac{LHR}{SHR}\right) Sensible\_Load$$

(13)

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A fraction of the reduction in the latent load evaluated in equation 13 is then included as a latent gain in the ESP-r/HOT3000 moisture balance, for the time step, for each of the zones served by the HVAC system. This fraction is equal to the ratio of the supply airflow to the zone to the total airflow through the cooling coil. In addition, it is known that the cooling coil acts as an evaporative cooler when the compressor is off (Henderson and Rengarajan (1996)). Therefore the latent gain to the space given in equation 13 is associated with an equal amount of sensible cooling to the conditioned spaces. Therefore, a sensible cooling term equal to the reduction in the latent load is also included the air point energy balance in ESP-r/HOT3000 for each of the zones served by the HVAC system.

An additional user input to the ESP-r/HOT3000 model is the COP of the air-conditioner at rating conditions. A correlation is used to adjust the value of the COP for any other coil inlet wet-bulb temperature and ambient dry-bulb temperature. The COP accounts for the power consumption of the compressor and the outdoor fan of the condenser. Given the total cooling capacity, COP, PLR, and PLF for the time step, it is then possible to determine the electricity consumption of the air-conditioner for the time step:

$$Electricity\_Consumption = \frac{Total\_Cooling\_Capacity}{COP} \times \frac{PLR}{PLF} \times \Delta t$$

(14)

DESCRIPTION OF TEST HOUSE

The ESP-r/HOT3000 house model used for testing this new algorithm is based on the reference research house at the Canadian Centre for Housing Technology (CCHT) in Ottawa, Ontario, Canada (Figure 2). The house is a typical two storey Canadian home built to the R-2000 construction standard. The above grade floor area of this four bedroom house is 210 m^2. The walls are 50mm x 150mm wood frame construction with RSI 3.5 fibreglass batt in the framing. The ceiling is insulated to RSI 8.8 and the 2.4m high basement is insulated on the interior to RSI 2.1. The windows are double glazed with insulated spacers, argon gas fill, low-e coated and vinyl frames. The doors are metal with interior foam insulation and good perimeter seals. Although the R-2000 air leakage requirements are 1.5 ach at 50 Pa pressure difference, we have increased the infiltration rate to 3.0 ach @ 50 Pa to ensure higher latent loads for testing.

The house model consists of two conditioned zones and two unconditioned zones. All of the above grade living space is designated as the "main" zone and is controlled by a single thermostat that is set to 21°C for heating and 25°C for cooling. The second conditioned zone is the basement living space which is controlled to the same temperatures as the main zone. The attached garage and attic space are zoned but unconditioned and their temperature floats in response to outside conditions and the influence of the attached conditioned spaces.

Internal loads due to occupants, lighting, equipment and appliances have been set to approximate typical residential usage profiles with peak gains occurring in the morning and evening hours. We are most interested in the response of the air conditioning system to house latent loads so we have set occupancy to four people and have assumed reasonable latent loads for dishwashing, clothes washing, cooking and DHW use (Aydinalp et al. 2002).

The house is cooled by a central air-to-air heat pump (AAHP) system located in the basement. The inside coil is located downstream of the distribution fan which circulates conditioned air through a ductwork network to all rooms of the house. The system ARI rated cooling capacity is 9 kW and the sensible heat ratio (SHR) of the cooling coil is 0.72. The fan is run in continuous mode for this evaluation. The fan size and airflow rate are determined by the AAHP model.

The four important inputs to the moisture evaporation model are N_{max}, t_{wet}, γ, τ. Henderson and Rengarajan (1996) estimated values for N_{max} of 3.0, for t_{wet} of 200 sec, and for γ of 0.6 for the air-conditioning system they tested. These same values are used in the present study. In addition the time constant τ is set to 60 sec.

RESULTS AND DISCUSSION

Figure 3 and 4 show the space temperature, relative humidity (RH) and relative humidity frequency distribution for the city of Ottawa before the air conditioning coil re-evaporation model is activated. The results are shown for the period from May 1st until August 31st. The maximum RH for the summer period in the main zone is 55.4% and the mean RH for the same period is 39.0%. The average space dry-bulb temperature is 24.9°C. In this case, the highest frequency for the space relative humidity is in the bin 37-39%.

Figure 5 and Figure 6 show the space temperature, relative humidity and relative humidity frequency distribution for the city of Ottawa after the air conditioning coil re-evaporation model is activated. The maximum RH for the summer period in the main zone increases by 11.6% to 67.0% and the mean RH for the same period increases by 5.8% to 44.8%. The average space dry-bulb temperature remains unchanged. In this case the highest frequency for with RH is associated with the bin 43-44%. It is clear from these results then that the evaporation of the moisture from the coil during the compressor off-time leads to an increase in the RH of the space. This type of increase in the space relative humidity could potentially lead to uncomfortable conditions inside the house.

Figure 7 shows the before and after hourly graphs for the main zone relative humidity for July 15th. The air conditioning coil re-evaporation model clearly shows that running the distribution fan continuously increases the average space relative humidity by 4-6% during this summer day without a significant effect on the space dry-bulb temperature.The effect of including the moisture evaporation model on the electricity consumption of the air-conditioner is shown in Table 1 for Ottawa, Toronto, and New Orelans.

Table 1: Cooling electricity consumption of the air-conditioner for three different cities (total cooling capacity = 9 kW)

	Electricity	Electricity

Weather City	Consumption (kWh) Model Off	Consumption (kWh) Model On
Ottawa	8,356	8,233
Toronto	8,492	8,366
New Orleans	14,973	14,713

The results in Table 1 show a 3.5%, 1.5% and 1.7% reduction in cooling energy use for Ottawa, Toronto, and New Orleans, respectively, as a result of using the moisture evaporation model. These reductions in the equipment electricity consumption are much smaller than the increase associated with running the indoor circulation fan in continuous mode instead of intermittent mode. When the moisture evaporation model is on, the predicted wet-bulb temperature of the air entering the cooling coil is higher then when the model is off. The total equipment cooling capacity increases with an increase in the inlet wet-bulb temperature of the air. As a result, the part-load ratio of the air-conditioner is smaller and therefore the electricity consumption goes down as shown in Table 1.

The cooling capacity was increased in two steps to investigate how cooling system over-sizing affects the coil loads and the resulting summer relative humidity. Table 2 shows the results of these runs. Although there is virtually no difference in the maximum RH achieved in the main zone, the average summer RH increases by 1% to 2% with each 50% increase in cooling system capacity. Air conditioning system over-sizing does not appear to have a significant effect on the predicted moisture level inside the space, when the moisture evaporation model is on, when the SHR is set to 0.72. However, the size of the equipment is found to have a large effect on the electricity consumption. When the size increases by a factor of two, the electricity consumption jumps from 8,233 kWh to 12,301 kWh for the total cooling season. It is clear that the increased size has a major impact on the equipment part-load performance.

It is to be noted here that the inputs to the model t_{wet}, γ, and τ are maintained constant, at 200 sec, 0.6, and 60 sec respectively, when the equipment size is increased. The parameter t_{wet}, as indicated previously, is the ratio of the energy associated with the condensate on the coil at the maximum holding capacity to the equipment latent capacity at rating conditions. The assumption here then is that when the capacity of the equipment increases by a certain fraction, the maximum condensate holding capacity increases by the same fraction. The other model input γ is the ratio of the initial evaporation rate from the coil to the latent capacity at rating conditions. When the equipment size increases, the airflow rate through the coil increases proportionally which is expected to increase the initial evaporation rate through the coil. At the same time the latent capacity of the equipment increases when total capacity increases. It is expected then that γ changes with increasing equipment capacity. However, the assumption of constant γ used in this study is deemed acceptable since Henderson Rengarajan (1996) show that their model is weekly dependent on γ and τ.

Table 2: Effect of cooling capacity on space relative humidity with moisture evaporation model on

Cooling System Capacity (kW)	Main Zone Maximum RH (%)	Main Zone Average RH (%)	Electricity Consumption (kWh)
9.0	66.98	44.82	8,233
13.5	66.78	46.95	10,289
18.0	66.53	48.06	12,301

The cooling coil sensible heat ratio (SHR) was also used to test the sensitivity of the air conditioning model changes. In this case, when the size of the equipment is doubled from 9 to 18 kW, the average humidity of the space increases 37.7 to 44.4%. The effect then of equipment over-sizing increases with a decrease in the coil sensible heat ratio. Table 3 shows the maximum and average relative humidity for the two values of the sensible heat ratio for the case when the equipment size is 9 kW.

Table 3: Sensible heat ratio (SHR) effect on the zone relative humidity (total cooling capacity = 9 kW)

Cooling Coil SHR	Main Zone Maximum RH (%)	Main Zone Average RH (%)
0.72	66.98	44.82
0.60	63.32	37.70

As expected, a decrease in the cooling coil sensible heat ratio increases the latent capacity of the coil and its ability to extract moisture from the return air. The maximum and average space RH is always higher with the moisture evaporation model included regardless of coil SHR and capacity.

All of the previous results confirm that there is a noticeable increase in the space relative humidity when the indoor circulation fan runs continuously and the moisture evaporation from the coil is accounted for in the simulation. This increase in the moisture content of the space can be offset by using a smaller size for the air conditioner to reduce the equipment off-time. However, the decrease in the space relative humidity associated with using a smaller unit will also be associated with the space temperature set point not being during part of the cooling season. In this case the trade off between lower humidity levels and higher temperatures can still lead to comfortable conditions inside the space. As a result, there can be an advantage for using a smaller air-conditioner size especially given the potential electricity consumption savings associated with this. Additional work is planned in the future to assess the effect of equipment under-sizing on space comfort and equipment electricity consumption.

CONCLUSIONS

Henderson Rengarajan (1996) developed a model for the for moisture evaporation from the cooling coil during compressor off-time when the indoor circulation fan is on continuously. This model is implemented in the ESP-r/HOT3000 simulation engine. The results show a noticeable increase in the relative humidity level of the space when the moisture evaporation model is used. For instance, for the city of Ottawa the space average RH increases by 5.8% during the cooling season when the model is included. Doubling the size of the air-conditioner leads to an increase in the space average relative humidity and the electricity consumption of the equipment. The increase in the space RH is more pronounced for lower values of the sensible heat ratio of the coil.

The increase in the space relative humidity with the moisture evaporation from the coil can be reduced by using a smaller air-conditioner. However, the decrease in this case in the space moisture content will be associated with the space set point not being met for a certain period of the cooling season. More work is needed in the future to study the effect of the reduced equipment size on the space conditions and the associated decrease in the electricity consumption of the unit. Additional work is also needed to study the effect of operating conditions, such as airflow rate, equipment capacity, and cooling coil type on the moisture evaporation model input parameters.

NOMENCLATURE

COP Coefficient of performance of air-conditioner

COOL_EIR
Ratio of power input at part-load to power input at steady-state

Electricity_Consumption
Electricity consumption of the air conditioner (Whr)

LHR Latent heat ratio after accounting for evaporation from coil

LHR_{ss} Latent heat ratio at steady-state operation

N_{max} Maximum number of cycles of the thermostat

PLF Equipment part-load factor

PLR Equipment part-load ratio

$\dot{q}_l$ Latent capacity of coil at any point in time after startup (W)

$\dot{q}_{l,ss}$ Latent capacity of coil at steady-state operation (W)

$q_{l,tot}$ Total latent energy between time t_o and t_{on} (J)

t_o Time after compressor startup that amount of moisture that evaporates from coil during off time is condensed again (sec.)

t_{off} Total off time of the air-conditioner during a full cycle of operation (sec.)

t_{on} Total on time of the air conditioner during one full cycle of operation (sec.)

t_{wet} Ratio of the energy of the maximum condensate holding capacity of the coil to the latent capacity of the coil at rating conditions (sec.)

Total_cooling_capacity
Sensible + latent cooling capacity of coil

Total_cooling_load
Sensible + latent load on the coil

X Equipment run time fraction

Δt Simulation time step (sec)

γ Ratio of the moisture evaporation rate when the compressor goes off to the latent equipment capacity at rating conditions

τ Time constant of air-conditioner (sec.)

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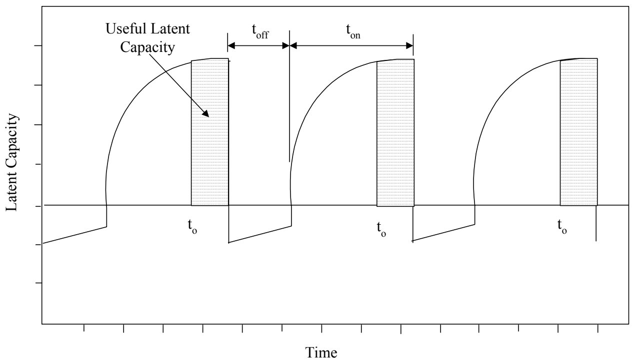

Figure 1. Variation of the latent capacity of the air-conditioner during quasi-steady operation showing effect of moisture evaporation during compressor off-time

Figure 2. CCHT reference house used to create the testing model in ESP-r/HOT3000 (www. ccht-cctr.gc.ca)

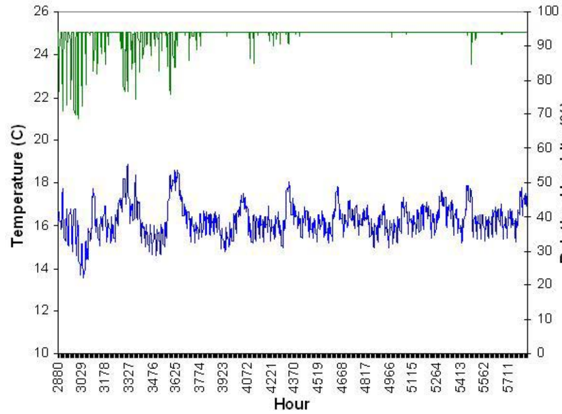

Figure 3. Main zone space temperature (left scale, upper plot) and relative humidity (right scale, lower plot) before the a/c re-evaporation model is activated.

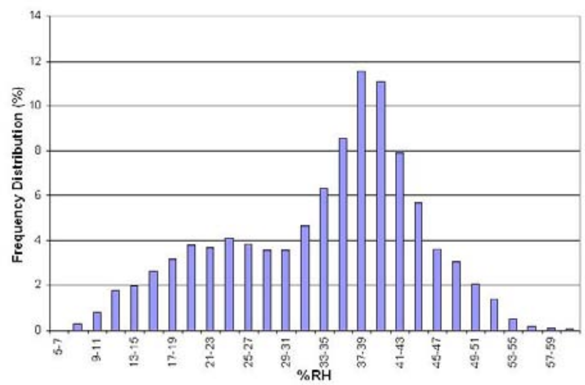

Figure 4. Main zone RH histogram before the a/c re-evaporation model is activated.

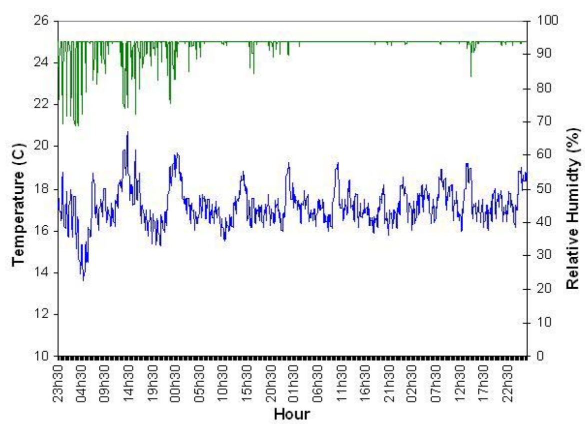

Figure 5. Main zone space temperature (left scale, upper plot) and relative humidity (right scale, lower plot) after the a/c re-evaporation model is activated.

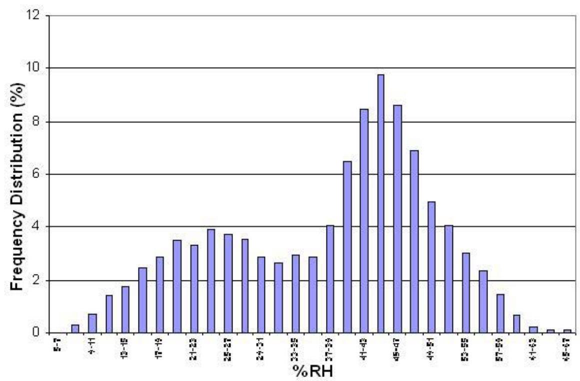

Figure 6. Main zone RH histogram <u>after</u> the a/c re-evaporation model is activated.

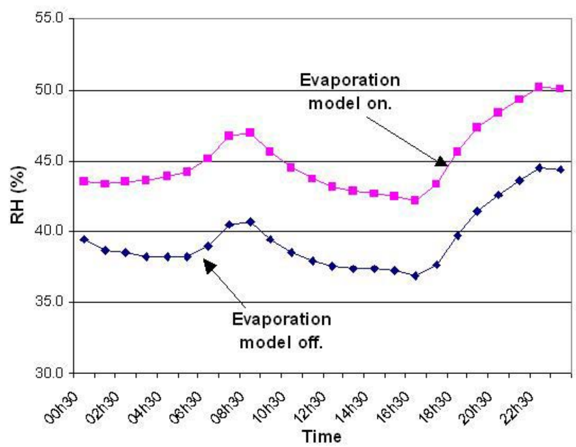

Figure 7. Main zone temperature profiles for July 15 (Ottawa) before and after evaporation model is activated.