New approach to humidity control at hot humid climate

Ahmed A. Hanafy [a], Abd El-Hamid A. El-Sayed [b] and Akram Mohamed [b]
[a] *Maritime Research and Consultation Center Arab Academy for Science Technology and Maritime Transport*
[b] *Refrigeration and Air Conditioning, Faculty of Engineering, Alexandria University, Alexandria, Egypt*
e-mail: A_a_hanafy @yahoo.com

The ability of HVAC systems to maintain proper indoor humidity levels depends on the building load characteristics and the climate characteristics. The proper application, selection, sizing and operation of the HVAC equipment are the used keys to control humidity levels. This requires that the HVAC system meets sensible and latent heat loads, not only at design conditions (full load), but also over a broad range of off-design conditions (part loads). The analysis of building load characteristics sought to identify and describe key factors that affect humidity control. Bibliotheca Alexandria building is taken as a case study. The cooling and dehumidification are to be evaluated at the design conditions. The result of these evaluations will be used to identify the performance of the various humidity control options. The results indicate that the suggested new control approach:”The ability to separately control space temperature and humidity by varying the air flow rate based on the space humidity condition, varying chilled water temperature based on space temperature condition, while maintaining the required outdoor ventilation air flow rate,” is the optimum control to maintain proper indoor humidity levels while minimizing energy consumption.

تعتمد مقدرة أنظمة التكييف فى المحافظة على مستوى الرطوبة الداخلية على خواص أحمال المبانى والأحوال المناخية. إن الإختيار الصحيح لمعدات التكييف من حيث السعة وطريقة التشغيل يعتبر العامل الأساسى للتحكم فى الرطوبة. ويجب ان تكون معدات التكييف قادرة على مقابلة الاحمال المحسوسة والكامنة ليس فقط عند الشروط التصميمية (أقصى حمل تبريد) ولكن عند جميع الظروف. وتم أخذ مكتبة الإسكندرية كدراسة حاله حيث تم تحليل أحمال التبريد مع إختلاف ظروف التصميم الخارجية وإقتراح نظام تحكم فى الرطوبة معتمد على الفصل الجزئى فى التحكم بالنسبة لدرجة الحرارة والرطوبة مع المحافظة على معدلات التهوية المطلوبة ودراسة النتائج فى حالة تطبيقيه بالمكتبه.

Keywords: Humidity control, Dehumidification loads, Cooling coil, hot humid climate, Bibliotheca Alexandria

1. Introduction

The need for ventilation air has forced HVAC equipment (originally optimized for high efficiency in removing sensible heat loads) to remove high moisture load**.** Humidity control will certainly become a very important and will be a key for evaluation of HVAC equipment. In hot and humid climate, the most HVAC equipment latent capacity is insufficient to adequately dehumidify the air. There are many ways to treat this problem like electric reheat, desiccant system, and coil by pass, etc. these methods are not energy efficient systems. So the optimal control methods for indoor relative humidity have been studied.

Humidity problems can be found in many commercial building applications, including office buildings, libraries, hotels and other commercial facilities. In many HVAC applications, the cooling and dehumidification system is unable to properly meet the imposed load requirements of the building when the dehumidification load is high, either due to large internal moisture generation or high ventilation flow rate. The mismatch between the building latent load and the equipment latent capacity can degrade occupant's comfort.

The improved envelope design and ventilation requirements have a strong impact on humidity control. Consequently, additional ventilation in hot humid climates, increases the dehumidification load more than the sensible cooling, imposing a disproportionate demand on the HVAC equipment.

2. Modeling and case study

2.1. Dehumidification loads

The main source of latent load in commercial buildings is the moisture generated by occupants and ventilation flow rate. Ventilation only imposes loads on the building when the outdoor conditions are different from the indoor conditions. Fig. 1 shows the magnitude of the sensible and latent ventilation loads as a function of outdoor temperature and humidity for a building with indoor conditions of 23 °C and 50% RH for Alexandria, Egypt. The loads are expressed per unit volumetric airflow rate.

2.2. Cooling and dehumidification hours

Cooling and Dehumidification hours provide an overview of what is the optimal outdoor design condition for hot and humid climate. Statistics show that the number of dehumidification hours (for Alexandria, Egypt Vs. 23 °C and 50% RH indoor condition) is significantly higher than the number of cooling hours as shown in fig. 2.

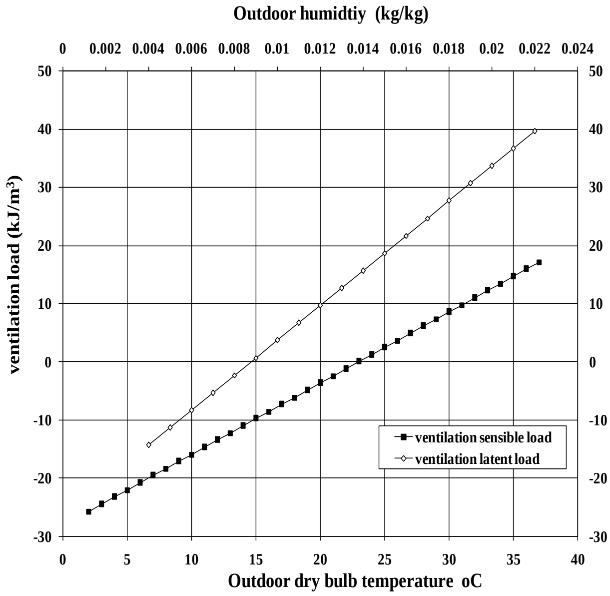

Fig. 1. Ventilation loads for Alexandria, Egypt vs. 23 °C and 50% RH indoor condition.

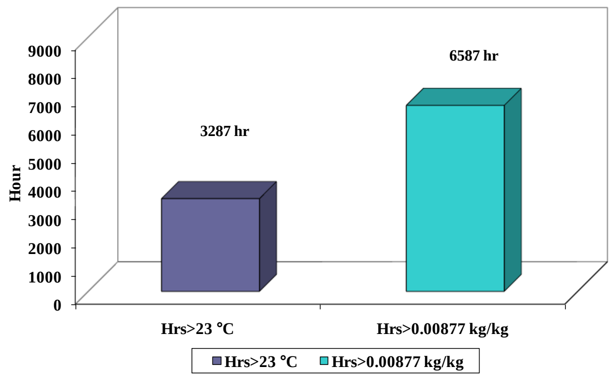

Fig. 2. Cooling and dehumidification hours for Alexandria, Egypt vs. 23 °C and 50% RH indoor conditions.

In order to improve accuracy in tropical climates, the new ASHRAE extremes were calculated on an annual, rather than a seasonal basis for all locations. The new design data has been addressed in Chapter 26 of the [5] ASHRAE Handbook of Fundamental. Based on 0.4% dew point (23.33 °C DP and 27.78 °C DB) outdoor design conditions, there are quite a few hours above the moisture design level (0.0182 kg/kg) that are being used for the latent load calculation unlike 0.4% dry bulb temperature.

Table 1 indicates that the ventilation loads are important factors in determining the SHR imposed on the HVAC equipment especially in case of 0.4% dew-point temperature when it is taken as a design condition. The calculations also imply that dehumidification requirements can be a very large portion of the overall building loads.

Table 1
Ventilation loads at different design conditions for Alexandria Vs. 23 °C and 50% RH indoor conditions

Outdoor design conditions	Percent cooling (Sensible heat ratio) %	Percent dehumidification %
0.4% dry bulb temperature (32.22 °C DB, and 21.67 WB)	54.09	45.91
0.4% dew point temperature (23.33 °C DP and 27.78 °C DB)	17.15	82.85

To avoid moisture problems, it's important to understand how well the HVAC system will dehumidify at both full load and off-design conditions, and to know how to improve its performance.

2.3. Computational procedure for indoor relative humidity

The simulation is based on the system characteristics and the zone design requirements. For each zone, the inputs include:

- Zone sensible and latent loads.
- Zone set point temperature.
- Minimum zone supply air flow.
- System characteristics include:
- Supply air temperature set point.
- Entering water temperature of cooling coils.
- Minimum flow of outside air.

Computational procedure for indoor relative humidity is [7].

1. Calculate ventilation sensible and latent loads.
2. Calculate required supply air volume flow rate.

$$Vs = \frac{Qzs}{1.224\ (ZDB - Ts)}, \qquad (1)$$

Where:
Ts is the supply air temperature °C.
ZDB is the zone dry-bulb temperature °C.

3. Calculate cooling coil entering conditions
4. Assume an initial air humidity leaving cooling coil.
5. Calculate indoor humidity ratio from latent loads.

$$RAW = CCLAW + \frac{QzL}{3010Vs}$$

Where:
RAW return air humidity ratio
CCLAW cooling coil leaving air humidity ratio.
QzL zone latent load.

6. Calculate mixed air dry bulb temperature, humidity ratio and wet bulb temperature.
7. Calculate cooling coil correction factors.

If operating conditions are different (cooling coil air entering conditions) the correction factors should be used.

8. Calculate cooling coil load.

$$QCSA = QCS * f_s. \qquad (2)$$

$$QCTA = \frac{Q_{CSA} * Q_{CT}}{Q_{CS}}. \qquad (3)$$

Where:
fs Sensible cooling coil correction factor (function relating available capacity at different operating conditions) .
QCSA Available sensible cooling coil capacity at different operating conditions.
QCS Sensible cooling coil capacity at design conditions.
QCTA Available total cooling coil capacity at different operating conditions.
QCT Total cooling coil capacity at design conditions.

9. Calculate cooling coil leaving air temperature and humidity ratio.

$$CCLADB = CCEADB - \frac{Q_{CSA}}{1.224\ *\ Vs}. \qquad (4)$$

Where:
CCEADB cooling coil entering air dry bulb temperature.
CCLADB cooling coil leaving air dry bulb temperature.

10. Calculate zone dry bulb and humidity ratio.

2.4. Case study

The main reading area of Bibliotheca Alexandria was taken as a case study. So the indoor relative humidity was calculated for two cases [0.4% dry bulb (32.22 °C DB, and 21.67 WB) – 0.4% dew point temperature (23.33 °C DP and 27.78 °C DB) outdoor design conditions] at all outdoor conditions.

Tables 2 presents the relative humidity control performance of air handling unit at each outdoor weather bin (joint frequency bin) for two cases.

In examining table 2, the following may be concluded:

- Air conditioning system based on 0.4% dew point design tended to maintain a lower average relative humidity than system based on 0.4% dry bulb temperature design.
- Air conditioning system based on 0.4% dew point design maintains relative humidity lower than 50% about 7978 hrs in the year (91% of the time), Conversely, air conditioning system based on 0.4% dry bulb design temperature maintains relative humidity lower than 50% about 5457 (60% of the time).

Maintaining proper humidity levels within Bibliotheca Alexandria is an ever present concern. Air conditioning leads to continuous moisture flow from the outside to the conditioned interior, although Bibliotheca Alexandria conditioning systems are controlled by a thermostat responding to sensible heat.

As explained in fig. 4, the system which is designed according to dew-point outdoor design condition is maintaining lower indoor humidity level than the other design systems, although it can not maintain proper humidity levels at all outdoor conditions.

Table 2
Indoor relative humidity for main reading area

Design condition	Cooling coil			No. of hours: RH%>50	Max RH%	Min RH%
	Sensible	Latent	Total			
0.4% dry bulb (32.22 °C DB, and 21.67 WB)	102.086	27.6	121.024	3303	61.36	41.26
0.4% dew point temperature (23.33 °C DP and 27.78 °C DB)	93.748	40.204	133.95	782	56.31	35.23

Figs. 3 and 4 summarize the variation of the indoor air relative humidity for the two design conditions at all outdoor conditions, respectively.

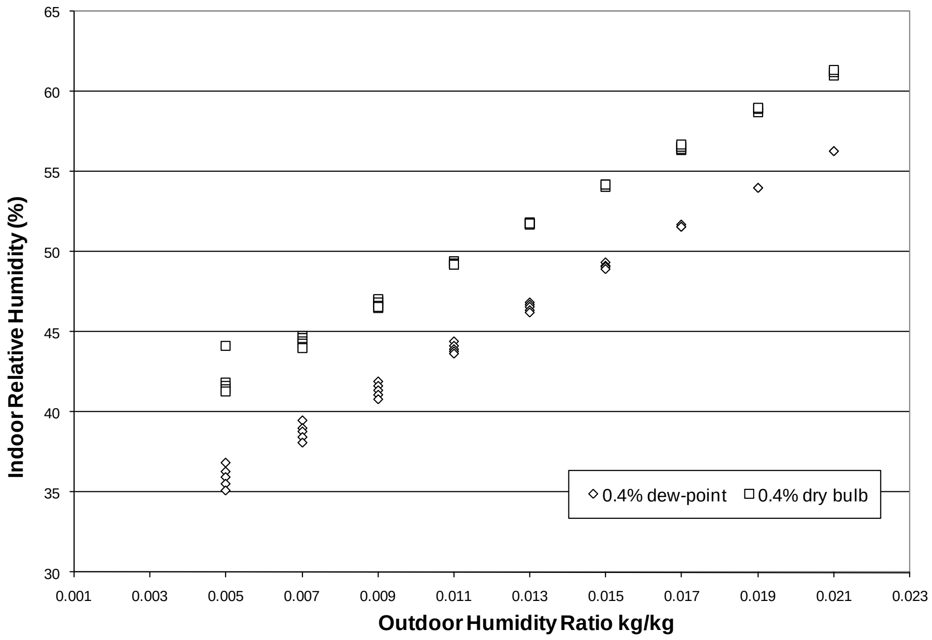

Fig. 3. Impact on indoor relative humidity for AHU-8.

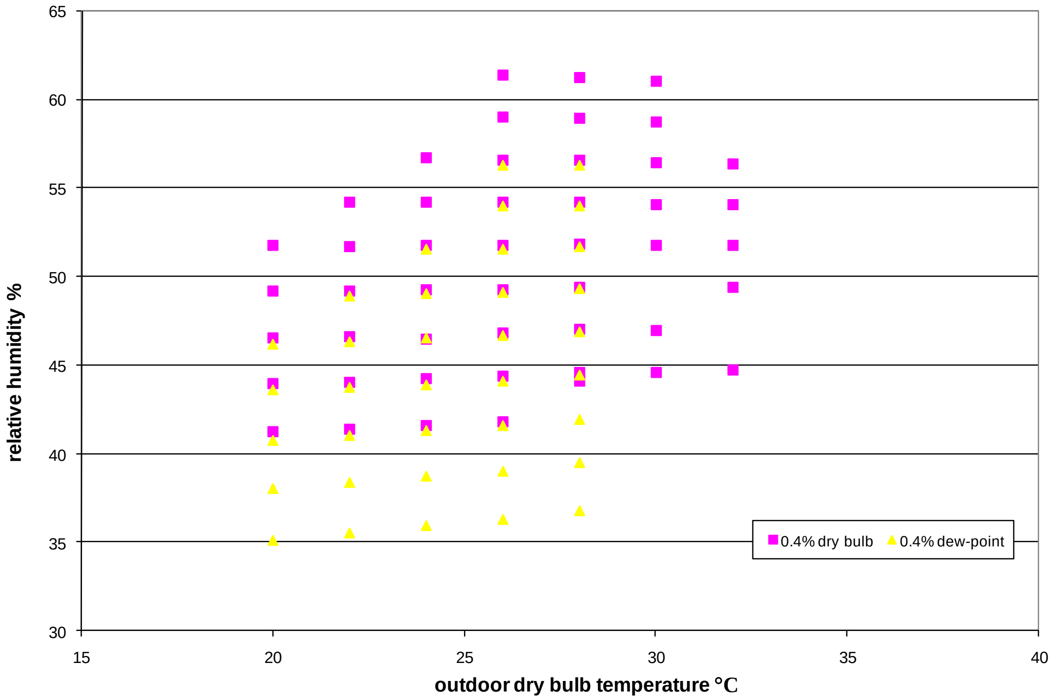

Fig. 4. Variation of the indoor relative humidity for the two cases.

From the investigation, there are some problems for the HVAC system of Bibliotheca Alexandria, which are summarized in the following points:

- Space humidity has not been actively controlled and has often been described as coincidental.
- Cooling coils that are controlled to maintain the dry bulb temperature in the zone often operate without adequate latent capacity at peak latent load conditions.

The prevailing way to solve this problem is overcooling and installing reheat coils to heat the air backup, resulting in simultaneous heating and cooling, and thus increasing energy consumption which is done in Bibliotheca Alexandria.

Over sizing the cooling coil can not prevent this shortfall in latent capacity if the system control is based solely on sensible conditions (space dry-bulb temperature) which are clear from the calculations carried out. In many cases, over-sized HVAC equipment has worse operational efficiency than properly sized equipment as the case of "AHU-8" because it operates at lower part-load conditions more than necessary.

Other common energy saving practices, such as supply air or chilled water reset with outside air temperature, will further reduce coil latent capacity under part-load conditions. The design of a cooling system capable of providing a suitable low sensible heat ratio (SHR) is important, but even this will not usually be adequate in humid climates under the increased ventilation rates specified by ASHRAE Standard 62 [2]. Also, in humid climates like in Alexandria, ventilation air must be continually dehumidified to yield acceptable moisture control.

3. Results and discussions

Other methods for humidity control and achieving good ventilation performance at the same time are analyzed.

3.1. Direct dehumidification"Dedicated Outdoor Air Systems (DOAS)"

When it is necessary to maintain a low relative humidity "as book store for example", both sensible and latent heat capacities must be controlled directly from both zone temperature and zone relative humidity. It is common practice in commercial conditioned buildings to combine ventilation make-up air with return from the building, and distribute the conditioned air to the interior space. One way to directly control dehumidification is to individually treat the return air and outdoor air streams before mixing them. This can be accomplished by dedicated outdoor air systems, which condition the outdoor ventilation make-up air separately from the return air of the conditioned space see fig. 5. By separately conditioning the make-up air so that excessive ambient humidity is removed.

Handling the treatment and distribution of the ventilation make-up air and return air from the occupied space with separate, parallel systems offers a number of potential advantages over conventional VAV systems (most common system):

- The ventilation make-up air system can be sized and operated to provide the ventilation air flow rate required by ASHRAE Std. 62 regardless of the interior temperature.
- The predominant humidity load in most commercial buildings at most climate areas is the humidity brought in with the ventilation make-up air. Consequently, the entire humidity load for the building can be handled efficiently
- With the ventilation make-up air separately conditioned the entire building humidity load handled in the process; the re-circulated indoor air conditioning system can then be operated to maintain temperature control.
- The combination of a DOAS with sensible cooling only VAV systems saves energy by reducing total ventilation air flow and by handling sensible cooling load efficiently.
- DOAS systems provide superior indoor humidity control over a wide range of outdoor temperature and humidity levels.

Table 3 summarizes the coil cooling loads of the separate air treatment paths in AHU-8 in Bibliotheca Alexandria.

Based on "0.4% dry bulb temperature (32.22^{o}C DB, and 21.67 WB)" as an outdoor design condition, total coil load, rises from 121.03 kW to 127.4 kW, is split between the outdoor coil (49.1 kW) and the return coil (78.29 kW). For "0.4% dew-point temperature (23.33 °C DP and 27.78 °C DB)" as an outdoor design condition, total coil load is increased from 133.95 kW to 138.3 kW. The load is split between the outdoor coil (63.7 kW) and the return coil (74.6 kW). In this case, each coil must be sized for its individual peak load.

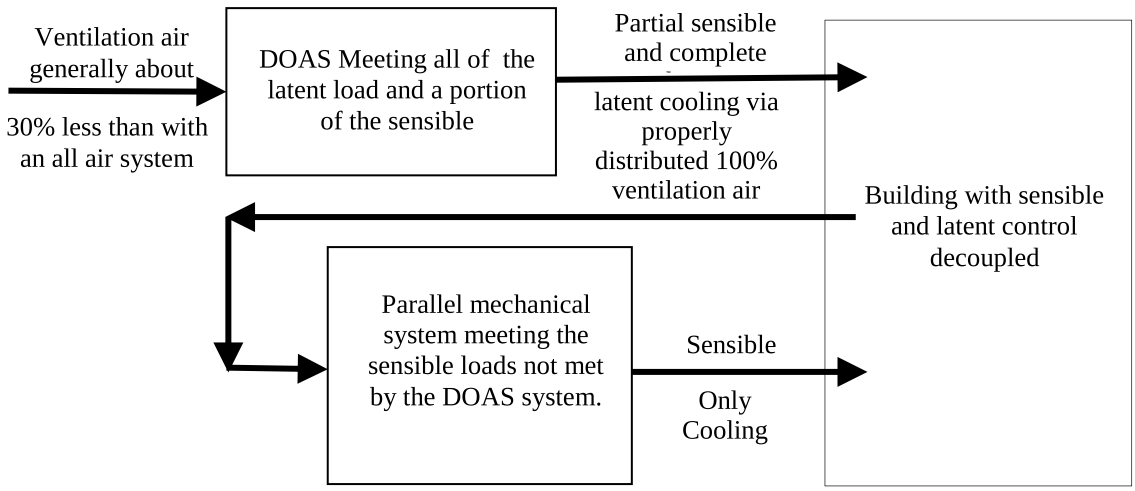

Fig. 5. Dedicated Outdoor Air Systems (DOAS).

3.2. VAV approach control system

The ability of the HVAC equipment to maintain adequate control of indoor relative humidity level depends on the relative comparison between the SHR of the space load and the SHR of the cooling coil. As shown in fig. 6, if the SHR of the cooling coil is greater than that required by the loads, indoor relative humidity will rise above the set point of 50% RH. So, an analysis of the sensible and latent loads imposed on the cooling coil has been performed for the main reading area at all outdoor conditions.

To make a good control of the indoor relative humidity, the gap between the SHR of the zone loads and SHR of the cooling coil must be reduced. So, a new approach to VAV control depending on the partial decoupling of the sensible and latent control was developed. In the new control approach, air flow rate is modulated in order toachieve the desired discharge temperature, which can be varied in order to adjust sensible heat ratio. While this does not provide as complete a decoupling of latent and sensible heat loads as would an active desiccant dehumidification system does, it adds some control flexibility.

It reverses the way in which air volume is adjusted. Under VAV conventional control, the zone thermostat affects the throttling of the terminal box valve, and the supply fan is modulated to deliver the required flow rate.

Table 3
Coil cooling loads of the separate air treatment paths in AHU-8

	0.4 % dry bulb temperature				0.4% dew-point temperature			
	Outdoor coil	Return coil	Total Separate paths	Conventional VAV	Outdoor coil	Return coil	Total Separate paths	Conventional VAV
Cooling coil total load (kW)	49.1	78.29	127.4	121.03	63.7	74.6	138.3	133.95

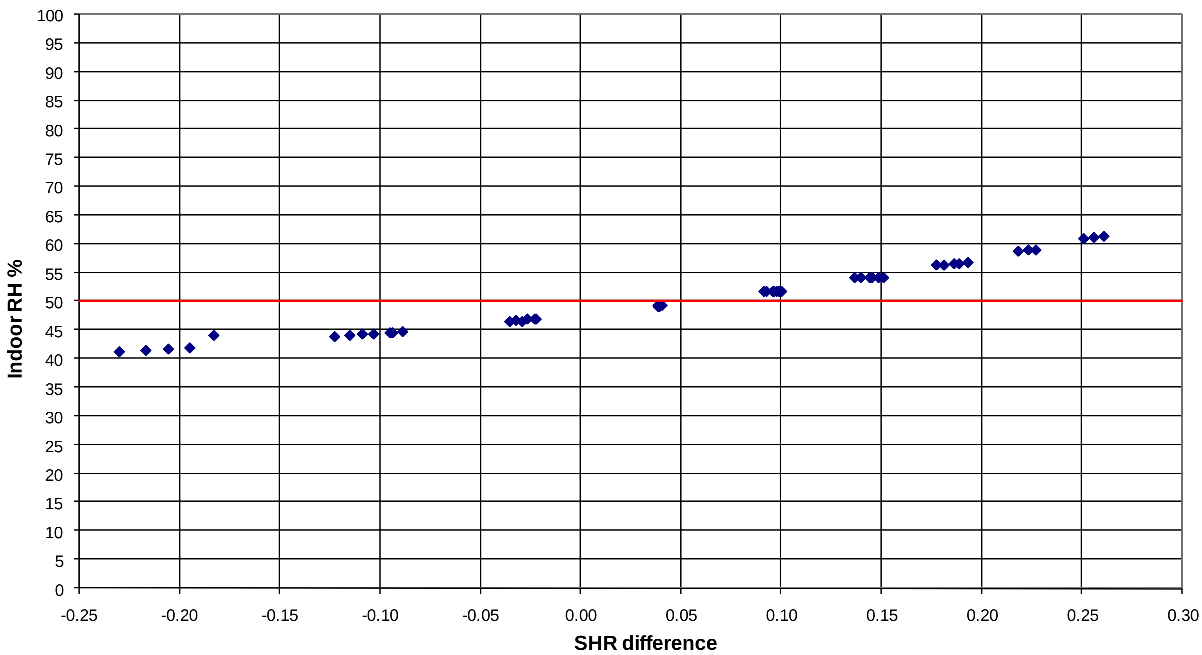

Fig. 6. Indoor relative humidity of the main reading area (installed AHU-8) Vs. the difference between SHR of the space load and the SHR of the cooling coil.

The cooling coil capacity is adjusted (through chilled water valve modulation) to control the supply air temperature. In the new approach, the chilled water temperature is adjusted based on the space temperature, and the supply fan responds by adjusting airflow to maintain a desired supply air temperature to control space relative humidity. The supply temperature set point varies depending on the relative humidity of the space to the space set point. When the space needs more dehumidification and less sensible cooling, the supply fan speed will drop to reduce airflow so that the low supply air temperature can be achieved. When the space requires less dehumidification, the supply fan speed will increase leading to an increase in the airflow. This method of control allows partial decoupling of the sensible and latent control of the space, allowing for use of reduced airflow to enhance dehumidification when it is appropriate, but operating with high airflow when there is less moisture load. Table 4 illustrates the statistics of indoor relative humidity in a new approach control.

As shown in figs. 7 and 8, the humidity controls of the new approach are more sensitive than the conventional control, due to the ability to reduce the gap between the SHR of the loads and that of the cooling coil. Even so, humidity fluctuations were not entirely avoided but the range of fluctuations was smaller than the range in the conventional VAV control. In the new control, the air flow modulation and chilled water temperature control must be set up with moderate speed response to avoid instability. Fig 11 compares indoor relative humidity between installed air handling unit No. (8) And air handling unit which design based on 0.4% dry bulb temperature (32.22 °C DB, and 21.67 WB). It shows that the humidity control is better in an installed air handling unit than in 0.4% dry bulb design air handling unit but energy use of the installed design is higher than that in the 0.4% dry bulb design.

Table 4
Statistics of relative humidity

	Installed - VAV control	0.4% dry bulb - VAV control
Maximum	52.68	52.9
Minimum	37.55	42.73
Annual Mean	48.54	50.9
Range	15.13	10.17

4. Conclusions

1. HVAC designers should consider and deal with peak moisture in their calculations by understanding weather behavior. This understanding would enable more economical designs and better humidity control.
2. In the hot humid climate cities like as Alexandria, the cooling coil latent load and often the total cooling load peak occur when the outdoor dew-point temperature (and not the dry-bulb temperature) is highest.
3. Sizing up the cooling coil to accommodate the highest total load would not necessarily prevent a shortfall in the latent capacity of the system if control of the latter is based solely on sensible conditions.
4. The issue of equipment over-sizing becomes a bigger problem for every hour of the year when building heating and cooling requirements are less than design conditions. Based on the research conducted, it appears that most of the HVAC equipment in Bibliotheca Alexandria was over-sized to handle pick-up loads and to provide an extra factor of safety. It is thus easy to conclude that HVAC systems employed there are very unlikely to operate at full load.
5. When humidity control is of a concern as in the case of Bibliotheca Alexandria, system performance should be analyzed under sensible and latent design conditions while taking into account the moisture control needs of the application.
6. Based on this study, it is reasonable to conclude that controlling the space temperature and humidity by varying the air flow rate based on the space humidity condition, and varying the chilled water temperature based on the space temperature condition, while maintaining required outdoor ventilation requirements, is a superior control strategy to conventional control methods. This approach seems to work at keeping the humidity levels in check under all outdoor conditions.

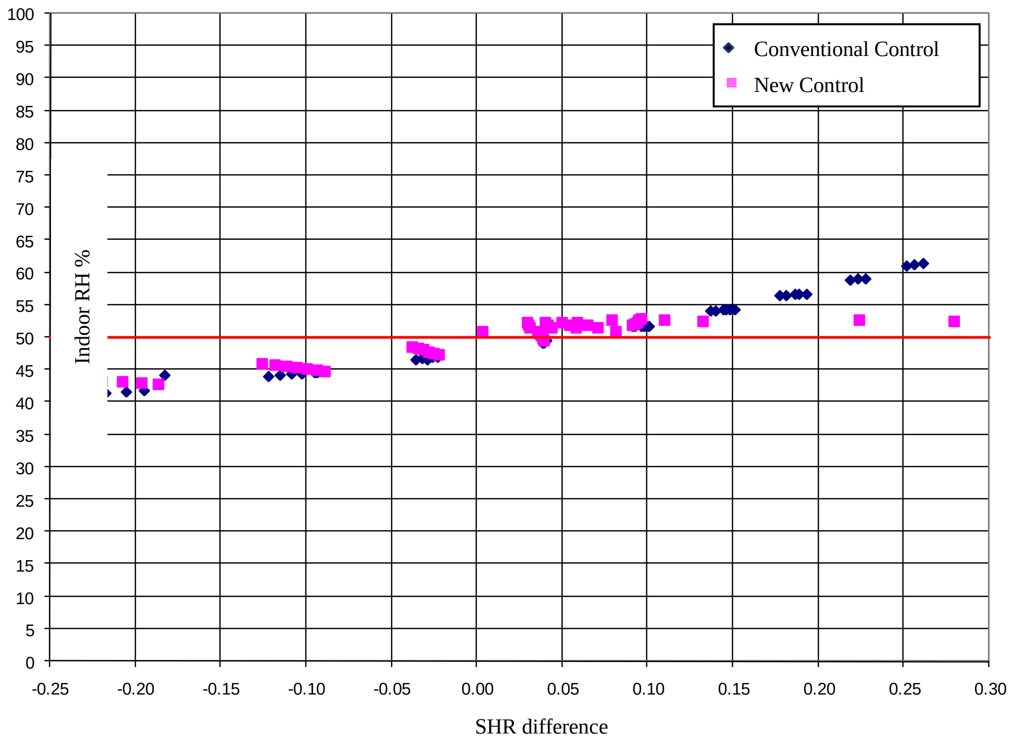

Fig. 7. Comparison of indoor relative humidity vs. SHR difference in new and conventional control for AHU-8 (0.4% dry bulb temperature design condition).

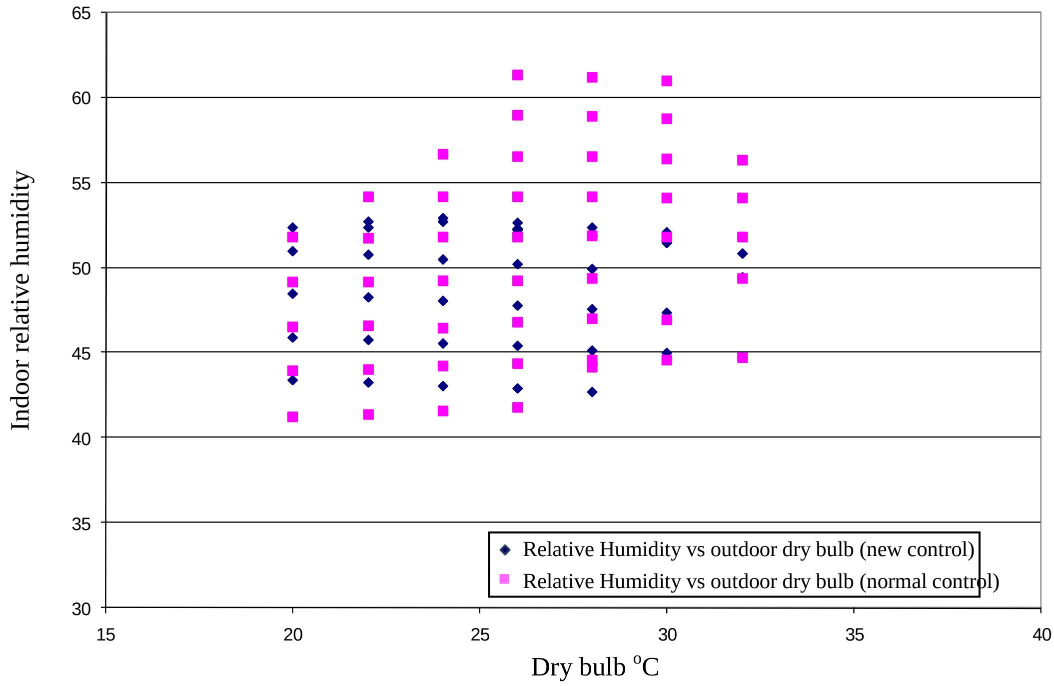

Fig. 8. Comparison of indoor relative humidity vs. outdoor dry bulb in new and conventional control for AHU-8 (0.4% dry bulb temperature design condition).

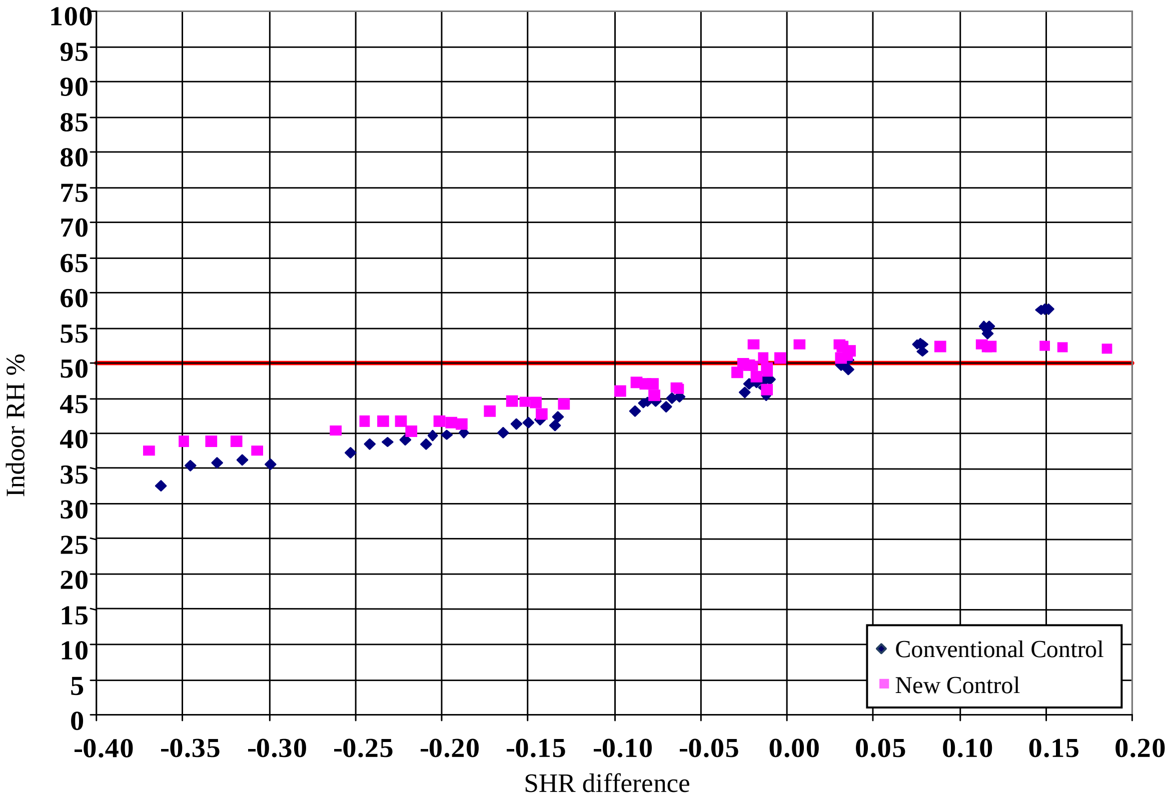

Fig. 9. Comparison of indoor relative humidity vs. SHR difference in new and conventional control for AHU-8 (Installed AHU).

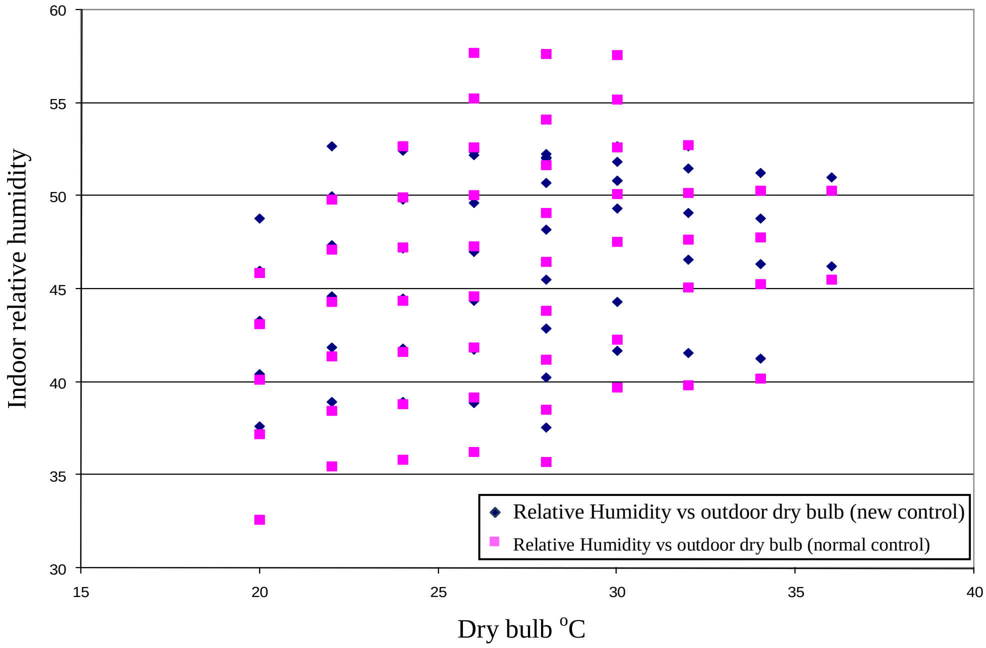

Fig. 10. Comparison of indoor relative humidity vs. outdoor dry bulb in new and conventional control for AHU-8 (Installed AHU).

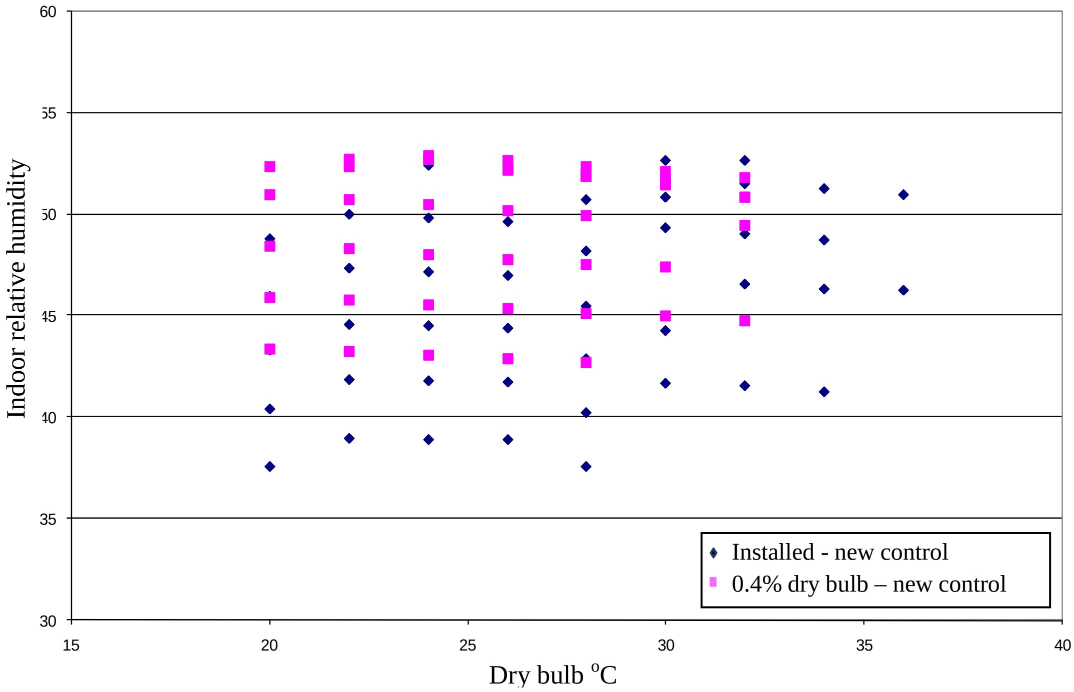

Fig. 11. Comparison of indoor relative humidity at installed and 0.4% dry bulb design for AHU-8 (new control).

Nomenclature

CCEADB	Cooling Coil Entering Air Dry Bulb temperature °C,
CCLADB	Cooling Coil Leaving Air Dry Bulb temperature °C,
f_s	Sensible cooling coil correction factor,
Q_{CS}	Sensible cooling coil capacity at design conditions (kW),
Q_{CSA}	Available sensible cooling coil capacity at different operating conditions, (kW),
Q_{CT}	Total cooling coil capacity at design conditions (kW),
Q_{CTA}	Available total cooling coil capacity at different operating conditions, (kW),
Q_{ZS}	Zone sensible load (kW),
T_s	Supply air temperature, °C,
V_s	Total air flow rate, (l/s), and
ZDB	Zone Dry Bulb temperature, °C.

Terminology

AHU	Air Handling Unit,
SHR	Sensible Heat Ratio, and
DOAS	Dedicated Outdoor Air Systems.

References

[1] Hugh I. Henderson and P.E. Jr. "The Impact of Part-Load Air-Conditioner Operation on Dehumidification Performance: Validating a Latent Capacity Degradation Model", IAQ and Energy, American Society of Heating, Refrigerating, and Air Conditioning Engineers, Inc, pp. 115-122 (1998).

[2] Stanley A. Mumma and Kurt M. Shank "Achieving Dry Outside Air in an Energy-Efficient Manner", ASHRAE Transactions, pp. 23-28 (2001).

[3] John Murphy "Dehumidification Performance of HVAC Systems", ASHRAE Journal, March, America Society of Heating, Refrigerating and Air Conditioning Engineers, Inc (2002).

[4] Lew Harriman "New Peak Moisture Design Data, in the ASHRAE Handbook of Fundamentals" (1997).

[5] ASHRAE Handbook of Fundamental, America Society of Heating, Refrigerating and Air Conditioning Engineers, Inc (1997).

[6] ProServie Prof. Moustafa Elsayed "Cooling Load of Bibliotheca Alexandria Report", (2001).
[7] David E. Knebel "Simplified Energy Analysis Using the Modified Bin Method", American Society of Heating, Refrigerating, and Air Conditioning Engineers Inc (1983).
[8] Barry M. Cohen and Douglas R. Kosar "Bin Data for Air Conditioning Calculations", http:www.binmaker.com/humissue.htm
[9] L.K. Norford, A. Rabl and R.H. Socolow "Control of Supply Air Temperature and Outdoor Airflow and its Effect on Energy Use in a Variable Air Volume System", ASHRAE Transactions, Vol. 92, pt 2B, publ. by ASHRAE, Athanta, GA, USA, pp. 30-45 (1986).
[10] Hugh I. Henderson, Danny Parker and Yu J. Huang "Improving DOE-2's RESYS Routine: User Defined Functions to Provide Mor Accurate Part Load Energy Use and Humidity Predictions" http://www.fsec.ucf.edu/bldg/pubs/resys/index.htm
[11] Conte-Richard "Reducing Annual Energy Consumption by Proper Selection of HVAC Equipment", Proceedings of the Intersociety Energy Conversion Engineering Conference V2, NJ: IEEE, pp. 1428-1431.
[12] Thomas E. Durbin and Michael A. Caponegro, "Desiccant Dehumidification Technology" http://test.wbdg.org/ccb/DOE/TECH/ugdppdf

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