Indoor humidity change with direct expansion (DX) air-conditioning (A/C) systems

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Abstract

DX A/C systems are widely used in small to medium size building and generally rely on On–Off control as a low-cost approach to maintain only indoor dry-bulb temperature. Continuously running the supply fan in a DX A/C system has a significant influence on the moisture removal effectiveness of DX A/C system. During an On–Off period, the air passing through the system's cooling coil may lead to the re-evaporation of the residual moisture on coil's finned surface. This paper discusses the effect of re-evaporation on the dehumidification performance of DX A/C systems by providing a broad literature review and experimental results. Experimental data are given by comparing the existing model of re-evaporation which was developed by Henderson.

1. INTRODUCTION

The use of DX A/C systems offers many advantages. They are simple in configurations, more energy efficient and generally cost less to own and maintain than chilled water-based large-scaled central A/C systems. Therefore, they find wider applications in buildings, in particular in small-to medium-sized buildings. In the US, according to the Department of Energy, packaged rooftop DX A/C systems consumed approximately 60% of the total amount of cooling energy used (Bordick & Gilbride, 2002). In Hong Kong, the annual total sales of DX residential A/C units were around 400,000 units in 1999 and 2000 (Zhang, 2002).

However, most DX A/C systems are equipped with single-speed compressor and supply fan, relying on On-Off cycling as a low-cost approach to maintain only indoor dry-bulb temperature, whereas the indoor air humidity is not controlled directly. Air dehumidification is usually only a by-product of an air cooling process. When a pre-set air temperature in a thermostat is reached, the compressor in an On-Off controlled DX A/C system is stopped and the dehumidification is also stopped. In a hot and humid climate like Hong Kong, the requirement for removing moisture from air can be often more demanding than removing sensible load. Therefore, indoor humidity may remain at a high level in the space served by an On-Off controlled DX A/C System.

The situation may become worse when the supply fan in a DX A/C system runs continuously while its compressor is On-Off operated. During an Off-period, due to the requirement of fresh air ventilation, the supply fan continues to draw the ambient air into room. As this warm and humid air blows over the surface of the cooling coil, if the moisture at the finned surfaces is not removed in time, the residual moisture will be blew off from the surface and entrained with the air flow, and most of them will be vaporized into the unsaturated air stream causing indoor humidity to rise. This phenomenon is called re-evaporation.

A high level of indoor humidity can cause discomfort for occupants for at lease two reasons: an uncomfortably high level of skin humidity and insufficient cooling of the mucous membranes in the upper respiratory tract by inhalation of humid or warm air (Toftum & Fanger, 1999). Therefore, indoor humidity level should be properly controlled.

A few works focus on the re-evaporation phenomenon directly. More relative information can be found in the research of analyzing cooling coil performance at part-load operation conditions. By investigating the effects of duty cycling on indoor humidity, Khattar *et al.* (1987) pointed out that continuous fan operation mode degraded the moisture removal rate by over 60% when compare to the fan on-off operation mode at a fraction of about 25% compressor run-time. The results of another experiment showed that 19% of the moisture collected during the compressor on-period entrained back to the space during the compressor off-cycling.

The continuous operation of supply fan also delayed the start-time of water draining off the cooling coil during on-period. Khattar *et al.* (1987) suggested that cooling coil was warm in continuous fan operation mode. It took a longer time to cool before the condensation. Another reason could be the moisture in the drain pan was evaporation, lowering its level during fan continuous operation mode.

Henderson (1990) also showed that a cooling coil quickly became an evaporative cooler once the coil was deactivated, providing sensible cooling while the coil was still at a lower temperature along with moisture addition, with no net enthalpy change across the coil.

A model representing the re-evaporation phenomenon was developed by Henderson & Rengarajan (1996). Based on several assumptions, the complex physical process of moisture evaporation from a deactivated cooling coil was represented by simple mathematical models, which were divided into three types: Exponential Decay Model, Linear Decay Model and Constant Evaporation Model. By analyzing the sensitivity of the models, Henderson & Rengarajan (1996) pointed out that the three different types of evaporation model considered for the analysis had very little impact on the latent heat ratio at part load (LHR) condition. The Linear Decay Model appeared the most physically realistic one. Douglas *et al.* (1998) also pointed out that the impact of continuous supply fan operation on latent capacity at part-load condition should be included in simulation programs.

Latent capacity degradation models had been incorporated into a detailed building simulation model FSEC 2.0 by Shirey and Rengarajan (1996). By considering the re-evaporation phenomenon in a small office building in Miami, the simulation results showed that the re-evaporation increased relative humidity level nearly 10% during morning hours on a hot summer day.

From previous research work, it can be seen clearly that the re-evaporation phenomenon has a significant influence on the dehumidification performance of DX A/C systems. Experimental work is required to provide information on this neglected aspect.

2. EXPERIMENTAL RIG

An experimental rig was set up to resemble a typical DX A/C system. It mainly consisted of two parts, a DX refrigeration plant and an air-distribution sub-system. The schematic diagram of the experimental rig is shown in Figure 1.

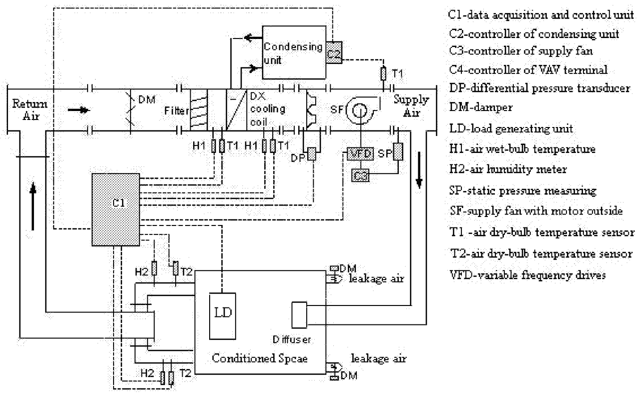

Figure 1 Schematic diagram of the experimental rig

2.1 DX refrigeration plant

The major components in the DX refrigeration plant included a variable-speed rotor compressor, an Electronic Expansion Valve (EEV), an air-cooled tube-plate-finned condenser and a high-efficiency louver-fin-and-tube DX evaporator. The evaporator is located inside an air-distribution sub-system to work as an air cooling and dehumidifying coil in the air side of the DX A/C system. The design air face velocity for the DX cooling coil is 2.5m/s. The nominal output cooling capacity from the DX refrigeration plant is 9.9kW. The actual output cooling capacity from the DX refrigeration plant can however be modulated from 15% to 110% of the nominal capacity. The schematic diagram of the DX refrigeration plant is shown in Figure 2 and the geometry of the cooling coil is shown in Figure 3. Both condenser fan and compressor were driven by variable-frequency drives (VFD). Working refrigerant of the DX plant was R22.

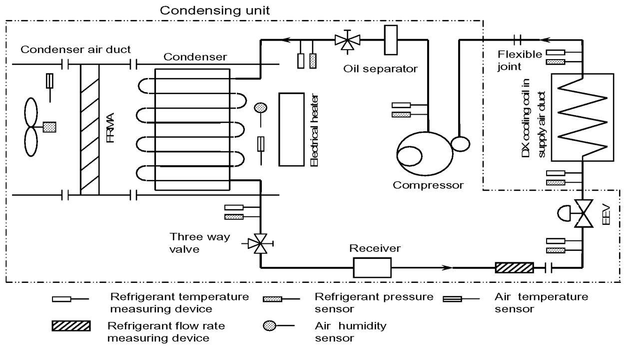

Figure 2 Schematic diagram of the DX refrigeration plant

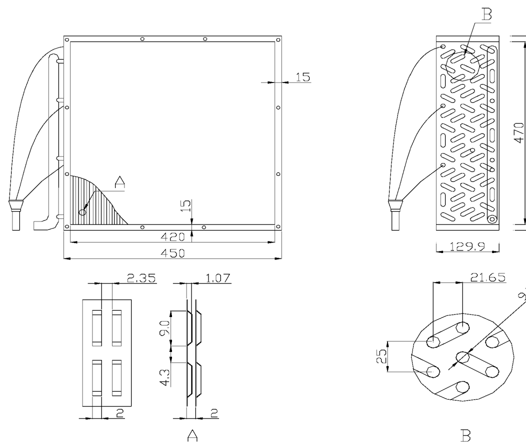

Figure 3 Geometry of the cooling coil (units in mm)

2.2 Air-distribution sub-system

The air-distribution sub-system included an air-distribution ductwork with return air dampers, an air filter, a variable-speed centrifugal supply air fan and a conditioned space. The size of the conditioned spaces was 7.8m (L) × 3.8m (W) × 2.8m (H). Inside the space there was a sensible heat and moisture load generating unit (LGU). The unit was intended to simulate different cooling loads and sensible heat ratio.

2.3 Instrumentation and data acquisition system

The rig was fully instrumented for measuring all operating parameters, which might be classified into three types: temperatures, pressures and flow rates. All measuring instrument and data logging system were computerized.

There were five sets of air temperature and humidity measuring sensors located in the air-distribution sub-system of the experimental rig. To minimize the influence of uneven distribution of air temperatures, pressures and humidity ratios inside the air duct, standard air-sampling devices as recommended by the ISO Standard 5151 were provided to ensure measuring accuracy (ISO, 1994). Relative humidity of air was indirectly measured by derivation of air dry-bulb and wet-bulb temperatures. The temperature sensors for air and refrigerant were of platinum Resistance Temperature Device (RTD) type, using three-wire Wheatstone bridge connection and with a pre-calibrated accuracy of ±0.1°C. Refrigerant pressures in various locations in the DX refrigeration plant were measured using pressure transmitters with an accuracy of ±0.13% of full scale reading. The atmospheric pressure was measured by a barometer having an accuracy of ±0.05 kPa. Air flow rate measuring apparatus was constructed in accordance with ANSI/ASHRAE Standard 41.2, consisting of nozzles of different sizes, diffusion baffles and a manometer with a measuring accuracy of ±0.1 % of full scale reading (ASHRAE, 1987).

A computerized data acquisition unit was provided in this experimental rig. It provided 48 channels for monitoring various types of operating parameters. The direct current signal obtained from various measuring devices and sensors were scaled into their real physical values of the measured parameters using a logging and control supervisory program.

3. RESULTS

The measured indoor air temperature and relative humidity (RH) in a space was taken for this experiment. It was served by a DX A/C with its compressor on-off controlled and its supply fan running at a constant speed. The indoor dry-bulb temperature was set at 24°C with a dead-band of ± 1°C, and indoor RH was unspecified. For safety reason, compressor would only be re-started with a 5-minutes delay. The space sensible heat ratio (SHR) was set at 0.8 representing a normal latent cooling load operating condition. As it is designed for this experiment, while the indoor air temperature was controlled at between ~ 22.4°C and 26.1°C partially due to the presence of the 5-minutes delay, indoor RH significantly fluctuated between ~ 57% and ~ 77%, with an average of ~ 64%. This suggests that under on-off control, the quality of indoor thermal parameters is hardly acceptable, with significant fluctuations. This is to say when the set-point of indoor dry-bulb temperature is satisfied, the compressor in a DX A/C is shut down, rather than working at part load. When the indoor dry-bulb temperature is not satisfied, the compressor is operated continuously. It is the basic principle of On-off control. Hence its cooling coil will keep dehumidifying at operation, but it will promote residual moisture from re-evaporation at shut off condition. The need of setting indoor air temperature lower for better dehumidification during on-time becomes a must. If indoor air dry-bulb temperature has to be maintained at a low level to ensure adequate dehumidification, the energy consumption of a DX A/C system would increase. Wastage of energy for cooling is always happening in humid region like Hong Kong because dehumidification is far more important than cooling as the relative humidity in Hong Kong in hot summer time is always higher than 75%. Figure 4 shows the transient sensible and latent energy change of air passing through the evaporator during off-period. It can be seen that the sensible energy change is almost a mirror effect of the absolute value of latent energy change. This agrees well with the conclusion of Henderson model (Henderson, 1990).

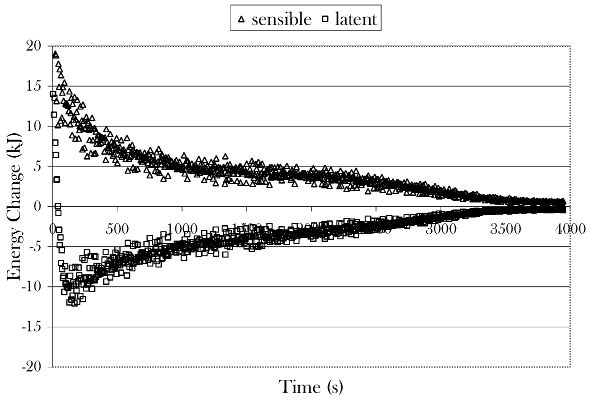

Figure 4 Transient sensible and latent energy change of air passing through an evaporator during off-period

Figure 5 shows the transient re-evaporation rate during off-period. It can be seen that the curve can be approximated to an exponential decay curve in the range from the highest point to a time of 1,000 second, a linear decay curve in the range from 1,000 second to 3,000 second and again an exponential decay curve in the range from 3,000 second to 4,000 second. These results, to a certain extent, agree well with Henderson model (Henderson & Rengarajan, 1996). The Linear Decay Model appeared the most similar one among the three models. However, at the beginning and the end of re-evaporation, it seems the exponential decay model is more applicable.

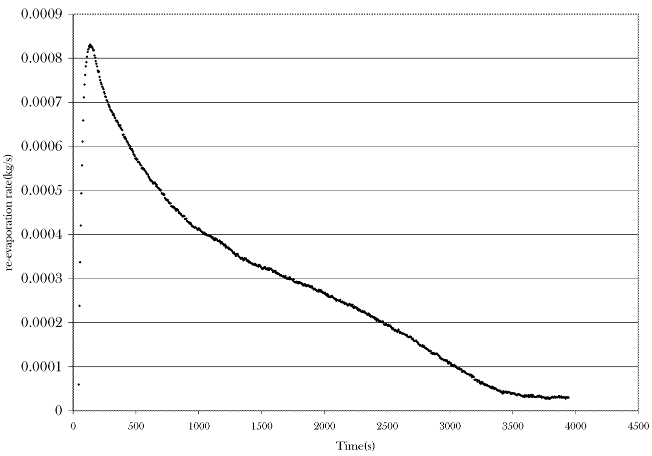

Figure 5 Transient re-evaporation rates during off-period

The alternation of the condensation type may dominate the change of re-evaporation mode. There are two types of condensation on the finned surfaces of cooling coil as pointed out by Xu *et al.* (2006). They are film condensation and drop-wise condensation. At the beginning of re-evaporation, the condensate retained may fully cover the fin surfaces as film state. During the re-evaporation process, the thickness of condensation film decreased. When the thickness decreased to a certain value, condensate retained could not maintain the film state. The surface tension force would make it changing to drop-wise state. This is because the re-evaporation mode changes from exponential decay mode to linear decay mode as the alternation appeared at 1,000 second. At the end of re-evaporation, most finned surfaces became dry, except some condensate retained at the louver gaps. At the gaps, only one side of condensate was in touch with the air as in the film state. The re-evaporation mode changed to the exponential decay mode again.

4. CONCLUSION

DX A/C systems are widely used in the past because of its simple construction and easy operation. They perform well in relatively narrow temperature ranges and also work best with fairly consistent load. As the increased complexity of refrigeration requirement and indoor thermal environment, single speed On-off DX A/C systems could not satisfy some specific applications. The use of variable speed compressor DX A/C becomes more and more popular in small to medium size refrigeration plants due to quick responsive characteristic, but it remains at a relatively high cost. A better understanding of single speed On-Off DX A/C is still useful for improving indoor environment.

By analyzing the dehumidification performance of the DX A/C system at part-load condition, it shows a clear relation between the re-evaporation phenomenon and the latent degradation of the DX A/C system. The most popular model for representing the re-evaporation was developed by Henderson & Rengarajan (1996). The evaporation rate and mode is changing at different stages and conditions. The concept of using H-L speed control algorithm (Xu *et al.*, 2008) gives an alternative for better humidity control, having many advantages over traditional On-Off control DX A/C systems while variable speed compressor is not required. However, more work should be carried out, with particular respect to selecting appropriate different speeds, for both compressor and supply fan under different load conditions as these would directly impact on control performance and energy efficient. More future experimental work under different space load combinations would provide a deeper insight of this new control algorithm. This project provides experimental results for a better understanding of re-evaporation at cooling coils and gives insight to the dynamic performance of re-evaporation phenomenon.

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