Discussions for Chicago Technical and Poster Session Papers

This is a compilation of the written questions and comments submitted to authors by attendees at the 2012 ASHRAE Winter Conference in Chicago, Illinois. All authors were given the opportunity to respond.

The questions/comments and authors' responses are published with the papers in the hardbound volume of *ASHRAE Transactions*, Volume 118, Part 1.

CH-12-001

Heat Pump Water Heater Technology Assessment Based on Laboratory Research and Energy Simulation Models

Kate Hudon **Bethany Sparn** **Dane Christensen, PhD** **Jeff Maguire**
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JR Anderson, Principal, Anderson Engineering: How do systems compare to solar and electrical performance?

Kate Hudon: Heat pump water heater (HPWH) performance is compared to the performance of an electric resistance water heater in Tables 11 and 12 of the paper. Basically, we found that source energy savings ranges from about 30% to 65%, depending on the climate. The performance of solar water heaters (SWHs) is not presented in the paper. However, analysis shows that SWHs will save about the same percent of source energy as HPWHs, when compared to electric resistance water heating.

CH-12-011

Effect of Hybrid Ventilation System on Indoor Environment and Annual Cooling Load in a High-Rise Building

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Yuichi Takemasa, PhD, PE
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Masahiro Katoh **Masaya Hiraoka**

Lixing Gu, Principal Research Engineer, Florida Solar Energy Center: The paper addresses that natural ventilation (NV) and HVAC system operation are independent. This should not be a good approach. It is possible that the conditioned air will go to the openings directly, so the conditioned air is wasted when both work independently. A good practice is that the NV and HVAC should be interlocked.

Katsuhiro Miura: I appreciate your question on my research. I agree with your idea that the air-conditioned air may go outside through ventilation opening.This kind of inefficient operation would occur when we selected outlet diffusers with poor ability of entraining room air and discharged conditioned air directly to ventilation openings.We could realize this situation with nozzles. But the situation will change when adopting diffusers with good ability of entraining air or discharge conditioned air to other directions from the openings. The conditioned air mixes with room air well before going out through the openings and the inefficiency will reduce very much. We selected diffusers and decided the discharge direction to prevent such inefficiency in ST building.

CH-12-012

Ventilation Rate Investigations in Minnesota Bars and Restaurants

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Sriram Somasundaram, Manager, Building Energy Systems & Technologies Group, Pacific Northwest National Laboratory: How many times was each venue visited so that you could get different occupancy patterns in each of them?

Martha Hewett: Each venue was visited approximately 36 times, but these measurements, from the longer two hour visits, were made primarily on Friday and Saturday nights and so were at high occupancy levels.

Carlos Lisboa, Engineer, BLC Navitas, LDA: To validate the assumptions made in the calculation of the calculated ventilation rates, namely the CO_2 generation rate by internal sources, did the team measure directly the ventilation airflow in ducts or diffusers/grilles in any of the places analyzed?

David Grimsrud: No, direct measurements of flows in ducts were not made in this set of measurements. These measurements were made without direct involvement of the owners or managers of the restaurants visited. However, the technique has been used in other settings where duct flow measurements and tracer decay measurements were also used as comparisons. See, for example, Grimsrud et al., 2011. In this study of three large retail stores, the ratio of ventilation rates measured by the TAB contractor and the mass-balance model using CO_2 concentrations was 1.09 +/–0.23; the ratio of the tracer gas decay measurements of ventilation rates to the mass-balance model results was 0.94 +/–0.13.

CH-12-033 (RP-1345)

Effect of Condensation Temperature and Water Quality on Fouling of Brazed-Plate Heat Exchangers

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Noma Park, Chief Research Engineer, LG Electronics: 1) What is the definition of Langelier Saturation Index? 2) How severe is the fouling resistance when it reaches the asymptotic value as compared to other heat transfer resistance?

Lorenzo Cremaschi: 1) The Langelier Saturation Index (LSI) is defined as LSI = pH – pH_s. LSI is a parameter that describes the status of the water for mineral scaling. It ranges from 0.5 to 3.5; 3.5 means that severe scaling formation conditions are expected and 0.5 means little or no scale formation is expected in the water stream flowing inside the heat exchangers. LSI can assume values in between these two extremes. LSI is defined as the algebraic difference between actual (measured) pH of a water sample and its corresponding pH calculated assuming saturated conditions (pH_s). The pH_s is the computed pH at which the calcium concentration in given water sample is in equilibrium with the total alkalinity. In our work, $pH_s$was computed based on the total dissolved solid, water temperature, Ca concentration and water alkalinity. 2) According to the definition of total resistance, $R_{tot} = 1 / U$ and it is calculated as follows: $R_{tot} = R_w + R_{ref} + R_f + R_{conduction}$. When asymptotic fouling resistance is reached, the fouling resistance R_f was up to 68% of the total heat transfer resistance.

James Schaefer, Project Engineer, HTRI: Did adiabatic point (85°F) have fouling?

Cremaschi: 85°F was the water inlet temperature in our heat exchangers during the fouling tests. If the refrigerant condensation temperature is also at 85°F, we obviously have an adiabatic condition and no heat transfer across the plates. In this case, the definition of fouling resistance is meaningless. However, if a small heat transfer occurs between the water side and the refrigerant side, say because T_{sat} refrigerant is 85.2°F, we expect some fouling to occur in the water stream. We decided to adopt the fouling resistance at 85°F refrigerant saturation temperature as the minimum fouling thermal resistance that can be detected with our instrumentation sensitivity. This value was very small that is practically considered as zero in actual applications.

CH-12-034 (RP-1589)

Measurements of Frost Growth on Louvered Folded Fins of Microchannel Heat Exchangers, Part 1: Experimental Methodology

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Noma Park, Chief Research Engineer, LG Electronics: In your presentation, it appears that both weight (mass) and thickness of frost grow linearly in time. This implies that the frost density remains the same through the frosting event. However, frost density is the nonlinear function of surface temperature. Can you explain this?

Ehsan Moallem: 1) The thickness and mass grew almost linearly, not exactly linear. At the end of the test, value of frost thickness is 20% less than if it were linear. 2) Even if the thickness and mass of frost were growing exactly linear, it does not mean that frost density remains the same. Frost density is division of mass over volume and not mass over thickness. If thickness grows linearly in time (Thickness = $m \cdot t$), volume will grow as a function of the third order in time (Volume = $(m \cdot t) \cdot (m \cdot t) \cdot (m \cdot t) = m^3 \cdot t^3$) and this shows that volume is a polynomial of the third order in time (t^3). So the volume is a curve, not a straight line. Thus the fraction of mass to volume will not remain constant or even linear at all. 3) In the present study, frost thickness on the leading edge has been measured and no information is available about how the thickness of frost volume inside the microchannel changes. So we can not have detailed and accurate scientific measurements of frost volume in the present study. As mentioned before, even if we assume that all the aspects of thickness in every point of heat exchanger grows exactly similar to leading edge thickness, still volume is a third order polynomial and density varies nonlinearly. 4) In the present study, temperature of the fin bases have been kept constant; however, temperature of the *frost surface* could not be measured or controlled during the frost test. Thus frost grows in a different temperatures during a frost test and might have various densities due to formation temperature. Also, as time passes, frost density increases due to diffusion of water vapor into frost layers apart from the vapor that increases the frost thickness at the top of frost layer. Although we expect the frost density to increase during the test, because of lack of concise measurements of frost volume, talking about frost density in the present study is more speculative rather than evidential.

James Schaefer, Project Engineer, HTRI: 1) How was air side controlled? 2) Why was constant wall temperature used instead of constant heat flux?

Moallem: 1) Air side was controlled within AHRI 210/240 required tolerances, which was +–1°F in air dry-bulb an d+–0.5°F in air wet-bulb temperature as mentioned the body text of the paper and in Table 2 of the manuscript. The control of dry-bulb temperature was controlled by wind tunnel cooling coil to near setpoint and then was accurately controlled to the exact setpoint by an electrical heater as shown in Figure 2 of the manuscript. The humidity level of air (wet-bulb temperature) was controlled by turning the appropriate number of humidifiers on and off to maintain the humidity level within desired tolerances as explained in the body text. The fan RPM was constant during the test as mentioned in the manuscript. 2) The second part of the question was about constant temperature vs. heat flux. Since these coils are evaporators in real application, the two-phase flow inside the microchannel keeps a constant surface temperature during frosting working period. When frost forms on the surface, capacity of the evaporator drops (not constant) during frost test due to insulation and flow blockage effect of the frost. Therefore, constant surface temperature and decreasing heat flux are much more similar to what happens in real working condition than lowering the surface temperature and maintaining the same heat flux.

CH-12-036

Laboratory Testing of Saddle-Tap Tees to Determine Loss Coefficients

A.N. Nalla

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Patrick Brooks, Director, United McGill Corp.: Were both the downstream and upstream pressure tap distances varied to determine the effect or were they maintained at the SPC 120 recommended length of 11 diameters?

Stephen Idem: Neither the downstream nor upstream pressure tap distances were varied to determine the effect on the fitting loss coefficients; in every instance the dimensions of the experimental setups strictly adhered to the lengths required by Standard 120.

Scott Hobbs, Technical Services, McGill Airflow, LLC: Does branch/main ratio affect the Δp of the tap? Does ASHRAE have any recommendations regarding the ratio of branch/main size in regards to optimal pressure drop through the tap and when the tap size can adversely affect pressure drop vs full body fittings?

Idem: The branch-to-common area ratio does affect the pressure drop through the tap. At any given branch-to-common flow rate ratio, the magnitude of the branch loss coefficient increases proportionately to the area ratio. ASHRAE does not have any recommendations regarding the ratio of branch-to-common size in regards to optimal pressure drop through the tap, because that is dictated by the particular design requirements of the system.