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# Hydraulic-Powered Fan Coil for Advanced Hydronic Distribution Systems, Part I: Thermal Performance and Energy Characterization of Conventional Fan Coil Units

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## ABSTRACT

*This paper reports the testing of two conventional fan-coil units of different size in a controlled environment rig to determine their thermal performance and energy requirements during cooling operation. Part II of this paper describes replacing the fan motor with a small water turbine as the power source to modify the same two units. The thermal performance and energy requirements of the modified units were then measured and compared with those of the conventional ones for the same inlet conditions and fan speed.*

*For the cooling mode tests, chilled water from the evaporator of a refrigeration unit entered the coil at 45°F (7.22°C). The air inlet conditions were 75°F and 80°F (23.89°C and 26.67°C) dry bulb and 54.9% and 50.8% relative humidity, respectively. The tests were conducted at three fan speed settings. The water flow rates were 1.0, 1.5, 2.0, 2.5, and 3.0 gpm (0.063, 0.095, 0.126, 0.158, 0.189 L/s). Steady-state measurements of the air and water inlet and outlet conditions, flow rates, and water pressure drop across the fan-coil units were recorded for each test. Wattage to the electric motor was registered at each fan speed.*

*It is found that for both units, the sensible heat ratio and outlet air temperature are reduced when the water flow rate through the coil is increased and/or the fan speed is decreased. For the same inlet air and water conditions, the sensible heat ratio and outlet air temperature of the smaller fan-coil unit are lower than their counterparts of the larger unit.*

## INTRODUCTION

The efficiency of forced-air heating and cooling systems is greatly affected by the distribution system and its location. Seventy-five percent of heating and cooling systems in U.S. households rely on ducts to supply conditioned air to the building. Present duct systems require large amounts of space, tend to be noisy, are extremely prone to leakage, and can result in maldistribution of air and large infiltration losses. Dust collection and the growth of mold and mildew inside ducts can aggravate indoor air quality problems (Kesselring 1993). Estimates of energy losses for ducts in unconditioned spaces and partially conditioned spaces are 35% and 20%, respectively (Gupta et al. 1995).

Hydronic distribution systems are a means of reducing the energy losses associated with forced-air duct systems and improving the delivered efficiency of a building's HVAC system (Sarkisian et al. 1990). An all-water system requires much less hydraulic distribution energy than a forced-air system since the working fluid has much greater thermal capacitance. Air leakage from ducts contributes to thermal energy loss and may lead to pressure differences that could cause pollutants such as radon to infiltrate the conditioned space. Hydronic systems also have individual room control with quick response to thermostat settings and freedom from recirculation of air from other conditioned spaces. The estimated potential national energy savings for all energy sources from reducing thermal and hydraulic distribution losses is 1.33 quads.

Hydronic heating and cooling is also viewed as a technology that could increase the market share for zone control and thus reduce the energy consumption of the system even further by providing conditioned air only to the spaces as needed. Present zone-control techniques for forced-air systems utilize individual dampers that are cumbersome, unreliable, and costly. With individual fan-coil units in each room, zone

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control is more easily accomplished, resulting in better air distribution and improved comfort throughout the room.

In Part I of this paper we report measurements of the thermal performance and energy requirements of two varieties of conventional fan-coil units. These measurements include outlet air and water conditions and electric motor wattage consumption at two inlet air conditions, five inlet water temperatures, and three fan speeds. Part II is concerned with determining the feasibility of using the circulating water in a hydronic system to drive the fan by replacing the fan motor with a turbine to realize savings on the fan energy requirement. A secondary benefit investigated in Part II is the ability of the fan-coil unit design to provide improved comfort by controlling the sensible heat ratio at the coil irrespective of the water flow rate and coil load.

## EXPERIMENTAL PROCEDURE

Two conventional low-profile vertical fan-coil units, models A and B, were selected for the present study. Model B is smaller in size and capacity than model A. Each of the fan-coil units is equipped with a fan driven by an electric motor and a coil fitted with water-source connections. The fan is positioned so that when it is in operation the air is forced to flow across the coil surface. The control panel of the fan-coil units allowed push-button setting of power on and off, cooling or heating operating mode, seven fan speeds, and air outlet temperature. Each unit has an air filter at the inlet and the air is delivered through an outlet grille.

The fan-coil units were tested for cooling operation only. Chilled water was supplied from the evaporator of a refrigeration unit. A 1 hp (0.746 kW) variable-speed pump circulated the water in a closed loop between the cooling source and the fan-coil unit. An expansion tank existed in the closed loop to provide for water volume changes due to temperature variations. Varying the pump speed manually regulated the water flow rate and/or the opening of a two-way throttling valve installed in the inlet water tubing. All tests were conducted in a controlled-environment test rig capable of maintaining a constant dry-bulb temperature within the range 0°F to 110°F (–17.78°C to 43.33°C). A variable-capacity humidifier was used to increase the humidity within the test rig as needed. The steady-state experimental measurements included:

a. Air dry-bulb temperature and relative humidity at the inlet and outlet of the fan-coil unit
b. Air flow rate through the fan-coil unit
c. Water temperature at the inlet and outlet of the coil
d. Water flow rate through the coil
e. Water pressure drop across the coil
f. Wattage through the electric motor

The instrumentation used to obtain temperature and relative humidity measurements conformed with the requirements of ANSI/ASHRAE Standards 41.1-1991, 41.3-1989, and 41.6-1982 (ASHRAE 1991, 1989, 1982). Five T-type thermocouples were placed, one each, at the air and water inlet, two at the air outlet, and one at the water outlet. The air outlet temperature was taken to be the average reading of the two thermocouple positioned at the air outlet. The relative humidity of air was measured by two electronic relative humidity sensors located, one each, at the air inlet and outlet. A digital vane-type anemometer was used to measure the average air outlet velocity at different fan speed settings. During each measurement, the anemometer was placed adjacent to the air outlet grille and moved back and forth in a direction parallel to the grille and an average reading was recorded. The airflow rate was determined from the product of the air outlet velocity and area. The water flow rate was measured by a turbine flow meter located at the inlet of the fan-coil unit. An electronic pressure transducer was employed in measuring the water pressure drop across the coil. The pressure taps were constructed and installed according to ANSI/ASHRAE Standard 37-1988 (ASHRAE 1988). A watt transducer recorded the electric power consumption of the motor. All temperature, relative humidity, water flow, pressure drop, and wattage measuring instruments were hooked up to a microcomputer via a data acquisition system. The data acquisition software was set to store data at 60-second intervals. Data display on the computer screen was updated every six seconds to facilitate monitoring and fine-tuning of instrumentation to maintain the desired conditions. For each test, the data were collected for at least 30 minutes after which they were downloaded for analysis. A schematic diagram of the fan-coil unit, including the measuring instrument positions, is shown in Figure 1, where $T$ is temperature, $f$ is relative humidity, $Q$ is volumetric flow rate, $\Delta p$ is water pressure drop, and $W$ is wattage. Subscripts $a$ and $w$ stand for air and water, respectively, and subscripts *in* and *out* refer to inlet and outlet conditions, respectively.

The fan-coil units were tested for rating in accordance with the provisions set forth in ARI Standard 440-98 (ARI 1998). The standard temperatures for cooling rating conditions are listed in Table 1. The water flow rate is determined using the standard rating water temperature conditions. The fan-speed setting should be at its highest and the outside air dampers closed. For applications ratings, selection of a range of conditions commonly encountered is permitted. Also, the selected application rating conditions for the present work are given in Table 1. Note that only three fan speeds, 1, 4, and 7, are tested out of seven possible speed settings. These settings are selected to represent low, medium, and high fan speeds. Approximate relative humidities equivalent to the specified wet-bulb temperatures calculated from the ASHRAE psychometric chart are also quoted in Table 1.

## EXPERIMENTAL RESULTS AND DISCUSSION

For the larger conventional unit (A), the cooling rating conditions referred to in Table 1 yielded, on average, an air outlet temperature and relative humidity of 63.7°F (17.61°C) and 89.66%, respectively, a water flow rate of approximately 1.0 gpm (0.063 L/s), and electric motor wattage of 15.75 W corresponding to the highest fan speed setting. The air outlet velocity and area were 549 ft/min

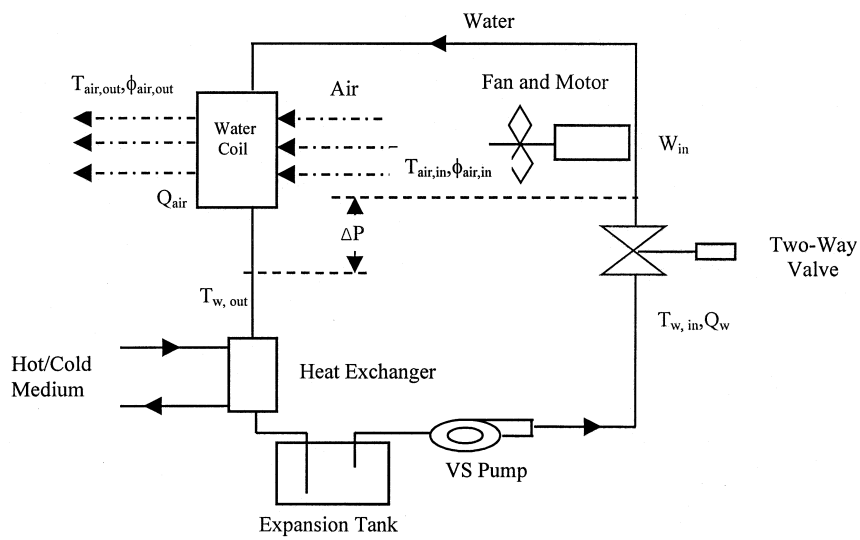


***Figure 1*** *Conventional fan-coil unit test arrangement.*

**TABLE 1**
**Cooling Rating Conditions**

| Cooling Rating Conditions | Standard | Application |
|---|---|---|
| Entering air dry-bulb temperature, °F (°C) | 80 (26.67) | 75, 80 (23.89, 26.67) |
| Entering air wet-bulb temperature, °F (°C) | | |
| (Relative humidity, ϕ) | 67 (19.44) | |
| (ϕ = 50.8%) | 64, 67 (17.78, 19.44) | |
| (ϕ = 54.9%, 50.8%) | | |
| Entering water temperature, °F (°C) | 45 (7.22) | 45 (7.22) |
| Outlet water temperature, °F (°C) | 55 (12.78) | NA |
| Water flow rate, gpm (L/s) | NA | 1.0, 1.5, 2.0, 2.5, 3.0 (0.063, 0.095, 0.126, 0.158, 0.189) |
| Fan speed | Setting 7 (highest) | Settings 1, 4, 7 |

(2.79 m/s) and 0.594 ft$^2$ (0.055 m$^2$), respectively, resulting in an air flow rate of 326.1 cfm (153.9 L/s). The average results for the cooling rating condition tests of the smaller conventional unit, model B, were an air outlet temperature and relative humidity of 61.46°F (16.37°C) and 82.69%, respectively, a water flow rate of approximately 1.0 gpm (0.063 L/s), and electric motor wattage of 8.87 W corresponding to the highest fan speed setting. The air outlet velocity and area were 447 ft/min (2.27 m/s) and 0.247 ft$^2$ (0.0229 m$^2$), respectively, resulting in an air flow rate of 109.9 cfm (51.87 L/s).

Tests for the cooling application rating were conducted on both the A and B conventional units in accordance with the conditions listed in Table 1. Since the air is being cooled below its dew point, some of its water vapor content will condense and the air will be dehumidified. Therefore, the heat losses from the air side to the water side are composed of a sensible heat loss that is manifested by a drop in air temperature and a latent heat loss associated with water vapor condensation. In equation form, the heat transfer balance may be expressed as

$$q = m_w c_{pw} (T_{Wout} - T_{Win}) = q_s + q_l = m_a c_{pa} (T_{ain} - T_{aout}) + m_a (W_{in} - W_{out}) i_{fg} \quad (1)$$

where $q$ is the rate of heat transfer, $m$ is the mass flow rate, $c_p$ is the specific heat at constant pressure, $T$ is temperature, $W$ is the humidity ratio, and $i_{fg}$ is the latent heat of condensation at the condensate temperature. Subscripts $w$, $a$, $l$, and $s$ denote water, air, latent, and sensible, respectively. Subscripts *in* and *out* designate inlet and outlet conditions, respectively. The first term on the air side of the above equation represents sensible heat and the second term represents latent heat. The ratio of the sensible heat to the sum of the sensible heat and latent heat (i.e., total heat, $q$) is known as the sensible heat ratio.

Figure 2 shows the sensible heat ratio variation with water flow rate and fan speed settings for the A unit at air inlet condi-

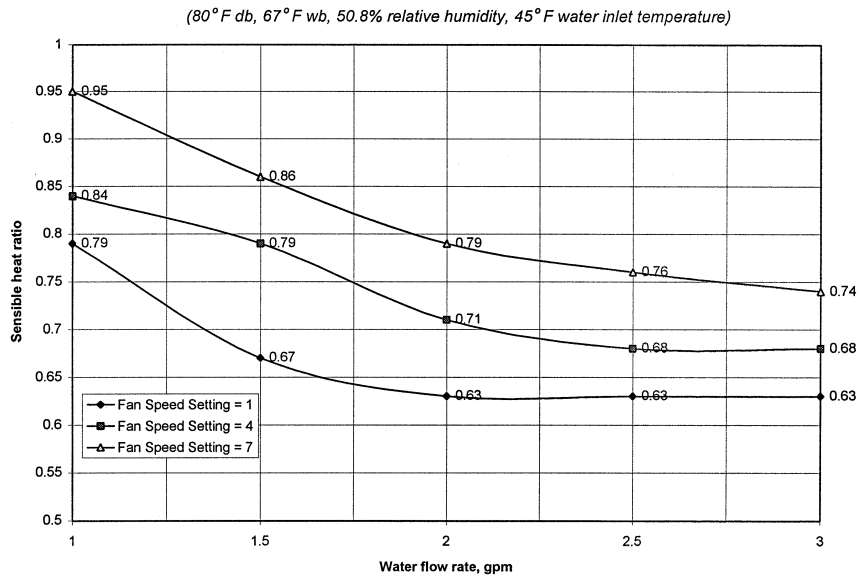


***Figure 2*** *A unit: sensible heat ratio vs. water flow.*

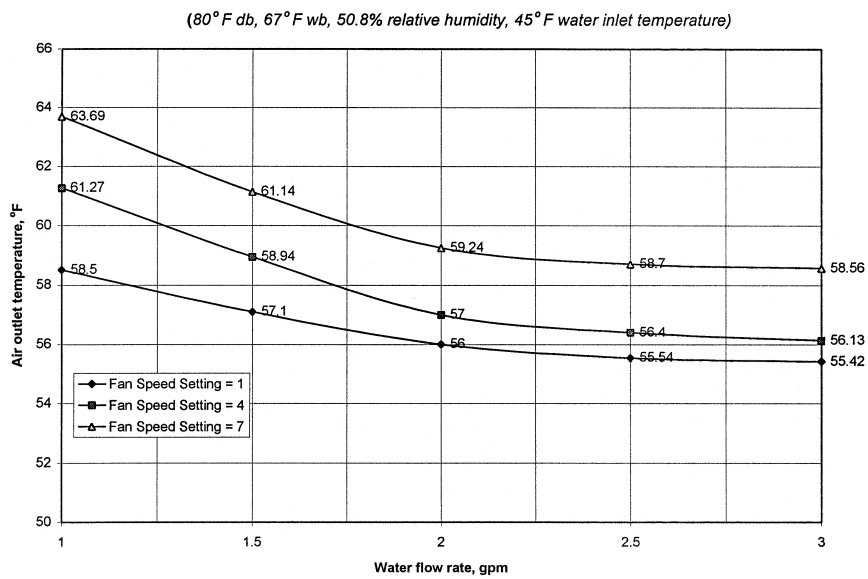


***Figure 3*** *A unit: air outlet temperature vs. water flow.*

tions of 80°F (26.67°C) dry bulb and 67°F (19.44°C) wet bulb (50.8% relative humidity) and water inlet temperature of 45°F (7.22°C). The sensible heat ratio was computed from the air inlet and outlet dry-bulb and relative humidity measurements using psychometric software. It is seen in Figure 2 that the sensible heat ratio decreases with increasing water flow rate and/or decreasing fan speed. This is expected since a higher water flow rate enhances the overall heat transfer coefficient and, hence, the heat transfer rate from the air side to the water side of the coil. Because the air and water inlet conditions are maintained constant, the air outlet temperature must drop to satisfy the heat transfer balance. A lower air outlet temperature accelerates the rate of water vapor condensation, which increases the latent heat, and, hence, a lower sensible heat ratio is achieved. On the other hand, a reduction in the fan speed, which causes the air flow rate to decrease, also produces a lower outlet air temperature and sensible heat ratio. This is because a lower air outlet temperature is necessitated by the need to compensate for the reduced air flow rate to keep the balance of heat transfer. This predicted behavior of the air outlet temperature is confirmed in the results documented in Figure 3.

The cooling application rating tests were also conducted for the model A conventional units at inlet air conditions of 75°F (23.89°C) dry bulb and 64°F (17.78°C) wet bulb (54.9% relative humidity) and water inlet tempera-

ture of 45°F (7.22°C). Figures 4 and 5 depict the resulting variations of the sensible heat ratio and outlet air temperature, respectively, versus the water flow rate and fan speed for these tests. It is evident from Figures 4 and 5 that both the sensible heat ratio and outlet air temperature decrease with an increase in the water flow rate or a reduction in the fan speed (air flow rate). These trends are in harmony with the ones observed earlier in Figures 2 and 3 and may be explained in a similar fashion. Because of the lower inlet air temperature associated with the results of Figure 5, the outlet air temperatures are reduced below their counterparts of Figure 3 at the same water flow rate and fan speed. The sensible heat ratios reported in Figure 4 are generally higher than those of Figure 2 except at higher fan speed or lower water flow rates. The higher sensible heat ratios observed in Figure 4 are directly related to the lower inlet and, hence, outlet air temperatures in comparison with the conditions of Figure 2.

The smaller fan-coil conventional unit, model B, was tested under the same cooling application rating conditions as the larger model A unit. Figures 6 and 7 exhibit the behavior of the sensible heat ratio and air outlet temperature, respec-

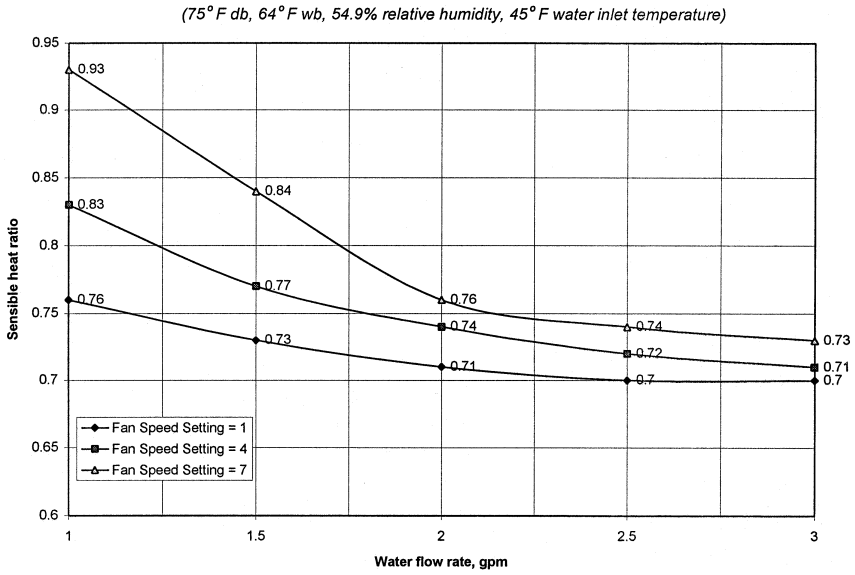


***Figure 4*** *A unit: sensible heat ratio vs. water flow.*

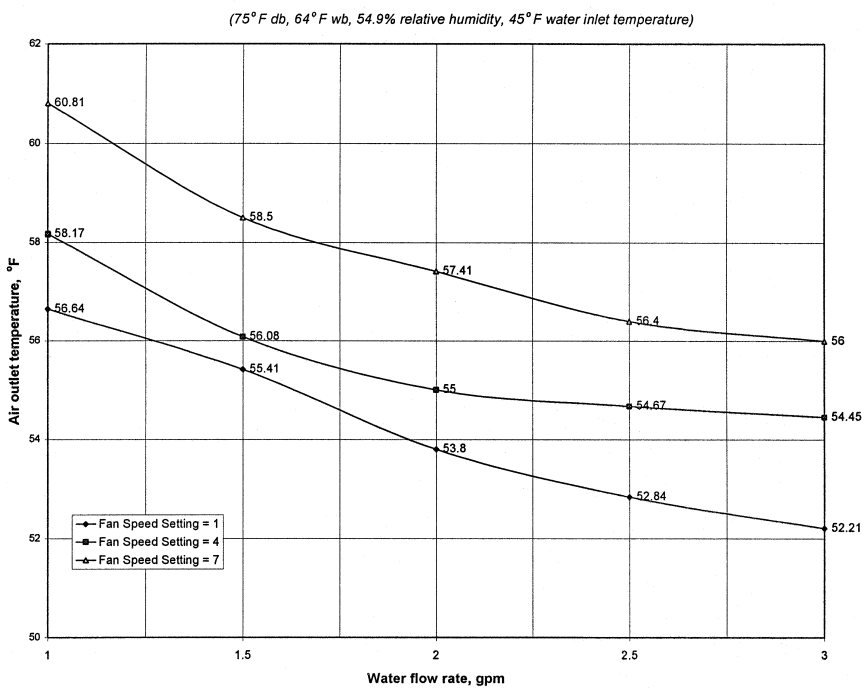


***Figure 5*** *A unit: air outlet temperature vs. water flow.*

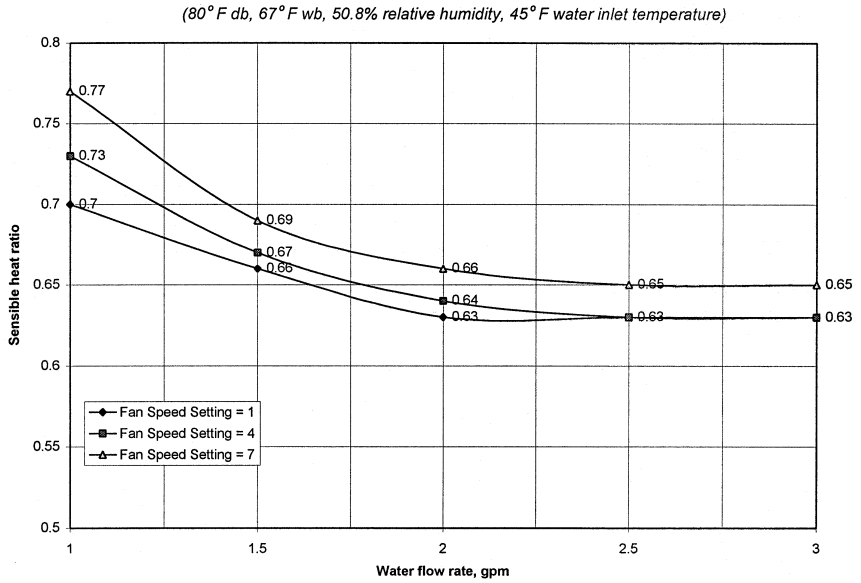


***Figure 6*** *B unit: sensible heat ratio vs. water flow.*

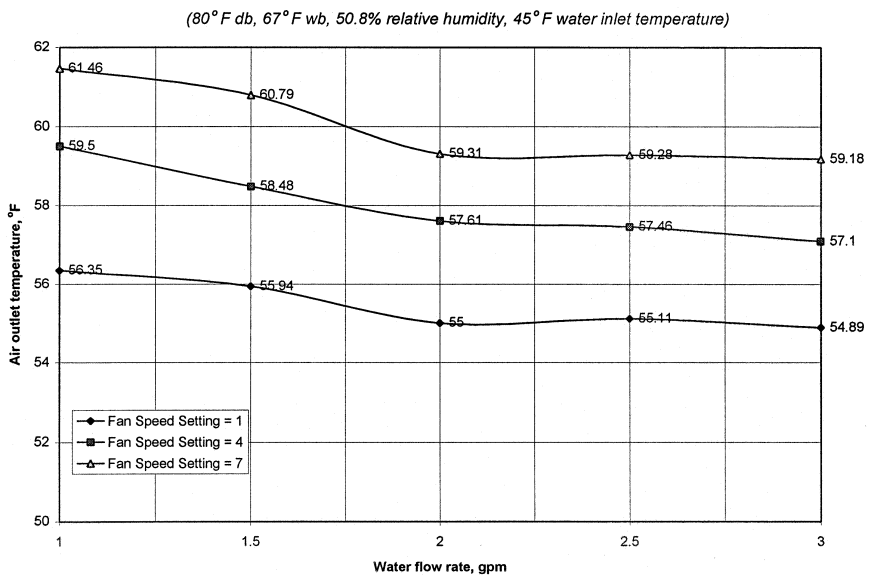


***Figure 7*** *B unit: air outlet temperature vs. water flow.*

tively, as a function of water flow rate and fan speed for the same air and water inlet conditions tested in Figures 2 and 3. It is concluded from Figures 6 and 7 that for the smaller unit, both the sensible heat ratio and outlet air temperature are generally reduced as the water flow rate is increased and/or the fan speed is decreased, as was the case for the larger unit. However, comparison of Figures 2 and 6 reveals that the sensible heat ratios for the smaller unit are lower than those of the larger unit. The lower sensible heat ratio of the smaller unit compared to the larger unit points to the fact that a larger portion of the heat removed form the air in the unit is latent heat associated with a higher rate of water vapor condensation. The enhanced latent heat removal rate in the smaller unit is due to the lower volume flow rate of air and, hence, the reduced air outlet temperature compared to the larger unit as an examination of Figures 3 and 7 verifies.

Figures 8 and 9 show the respective test results of the sensible heat ratio and outlet air temperature variations against the water flow rate and fan speed for the model B unit at inlet conditions identical to those shown in Figures 4 and 5 associated with model A unit. It is seen in Figures 8 and 9 that the sensible heat ratio and air outlet temperature continue to drop as the water flow rate or fan speed is raised, with the sensible heat ratio and outlet air temperature becoming essentially constant at higher water flow rate and lower fan speeds as noted earlier with respect to Figures 6 and 7. Respective

comparisons of Figures 8 and 9 with Figures 4 and 5 reveal that the sensible heat ratio and outlet air temperature for the smaller unit are less than their counterparts for the larger unit. These findings are in accord with earlier observations and may be interpreted in a similar manner.

It is important to note from Figures 6 to 9 that for the smaller unit the sensible heat ratio and outlet air temperature remain nearly constant at higher water flow rates and lower fan speed. This feature is unique to the smaller unit and was not observed in the larger unit. A constant sensible heat ratio allows for the removal of the same proportions of latent and sensible heats under design and part-load conditions. If the outlet air temperature is also constant, then the outlet humidity will be fixed. Therefore, the problem of part-load over dehumidification that is generally experienced with fan-coil units would be alleviated if the sensible heat ratio and air outlet temperature are invariant.

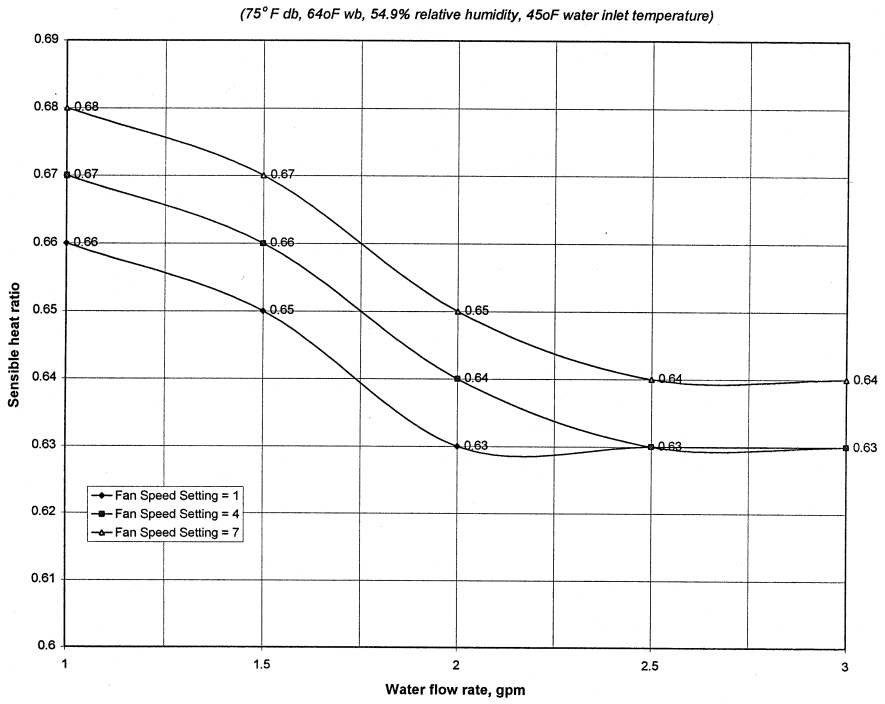


***Figure 8*** *B unit: sensible heat ratio vs. water flow.*

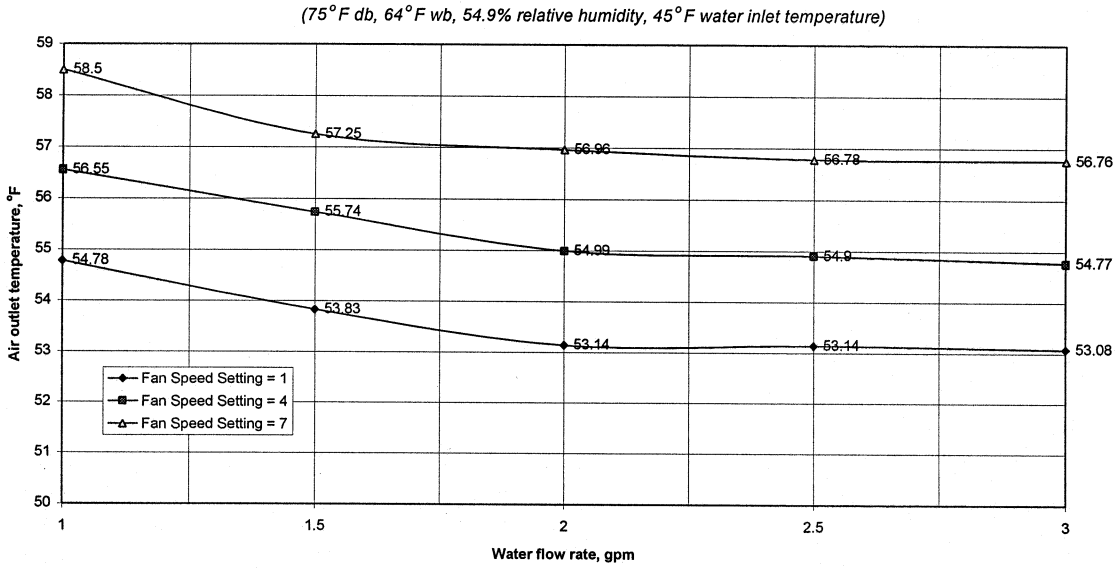


***Figure 9*** *B unit: air outlet temperature vs. water flow.*

**TABLE 2**
**Variation of Airflow Rates and Fan Power Demands with Fan Speed**

| Fan Speed Setting | Electric Power to Fan Motor, W, Model A | Electric Power to Fan Motor, W, Model B | Air Flow Rate, cfm (L/s), Model A | Air Flow Rate, cfm (L/s), Model B |
|---|---|---|---|---|
| 1 = 597.5 rpm | 5.58 | 4.13 | 166.9 (78.8) | 58.3 (27.51) |
| 2 = 667.5 rpm | 6.31 | 4.35 | 190.1 (89.7) | 67.05 (31.64) |
| 3 = 750 rpm | 7.66 | 4.85 | 218.6 (103.2) | 76.2 (35.96) |
| 4 = 835 rpm | 8.63 | 5.76 | 243.5 (114.9) | 86.45 (40.8) |
| 5 = 900 rpm | 10.91 | 6.4 | 267.3 (126.1) | 93.05 (43.91) |
| 6 = 985 rpm | 13.8 | 7.2 | 298.2 (140.73) | 101.3 (47.81) |
| 7 = 1042 rpm | 15.75 | 8.87 | 326.1 (153.9) | 109.90 (51.87) |

**TABLE 3**
**Water Pressure Drop Across the Fan-Coil Units vs. Water Flow Rate**

| Water Flow Rate, gpm (L/s) | Water Pressure Drop, psi (kPa), Model A | Water Pressure Drop, psi (kPa), Model B |
|---|---|---|
| 1.0 (0.063) | ≈ 0 (0) | 1.98 (13.66) |
| 1.5 (0.095) | 0.60 (4.13 | 2.98 (20.52) |
| 2.0 (0.126) | 1.23 (8.46) | 5.00(34.48) |
| 2.5 (0.158) | 2.23 (15.30) | 7.02 (48.41) |
| 3.0 (0.189) | 3.53 (24.34) | 8.53 (58.79) |

Note that increasing the water flow rate or decreasing the fan speed (air flow rate) also results in a slight increase of the water outlet temperature. However, the effect of increasing the water flow rate or decreasing the air flow rate outweighed the variations in the water outlet temperature. Changes in water outlet temperature are usually much smaller than those for air outlet temperature because water has a much larger heat capacity than air. Therefore, significant inlet water temperature variations would be needed to effect noticeable differences in the outlet air conditions. It is for this reason that the water inlet temperature was kept constant at its standard rating condition of 45°F (7.22°C) throughout the tests.

Measurements of the fan power requirements are given in Table 2 for both models A and B for each fan speed. The fan speed settings considered in the present study are settings 1, 4, and 7 and, hence, are thought to be representative of low, medium, and high fan speeds. The average air flow rate corresponding to each fan speed is also listed in Table 2 for reference. The air flow rates are computed from the product of the measured outlet air velocity and flow area. The outlet air flow area of the larger unit was found to be 0.594 $ft^2$ (0.055 $m^2$) and for the smaller unit the outlet air flow area is 0.247 $ft^2$ (0.0229 $m^2$).

Table 3 documents the measurements of the water pressure drop across each of the fan-coil units versus water flow rate through the coil. These measurements will be used to determine the hydraulic power required to drive the small turbine that will replace the fan motor in the proposed modified design investigated in part II of this paper. The power consumption of the turbine may be calculated from measurements of the water flow rate and pressure drop across the modified and conventional fan-coil units according to Equation 2:

$$P_t = Q_w(\Delta p_{mod} - \Delta p_{conv}) \qquad (2)$$

where $P_t$ is the input power to the turbine, $Q_w$ is the water volume flow rate, and $\Delta p_{mod}$ and $\Delta p_{conv}$ are the water pressure drops across the modified and conventional units, respectively. Note that the water pressure drop across the coil of the smaller unit is higher than that of the larger unit. This is due to the fact that the smaller unit's coil diameter is smaller and, hence, the water velocity is higher, resulting in a larger friction head loss.

## CONCLUSIONS

The thermal performance and energy requirements for the cooling mode operation of two sizes of conventional fan-coil units are documented in the present work. The thermal performance was evaluated through steady-state measurements of the inlet and outlet air and water conditions and flow rates. The electric power consumption of the fan motor was measured at the different fan speeds.

For both units, the sensible heat ratio and outlet air temperature decreased as the water flow rate increased or the fan speed and, hence, the air flow rate, decreased. However, the sensible heat ratio of the smaller unit was essentially constant at higher water flow rates and lower fan speeds. The outlet air temperature and sensible heat ratio of the smaller unit were found to be lower than those of the larger unit due to the reduced airflow rate delivered by the smaller unit.

In part II of this paper, the conventional units are modified by replacing the fan motor with a small water turbine as the power source. The water flowing through the coil is used to cool the air and to provide the energy for the turbine. The thermal performance and energy requirements of the modified units are reported and compared with the conventional ones.

One purpose of the comparison will be to determine if the energy consumption of the turbine is less than that of the motor for the same fan speed. Another purpose is to test the hypothesis that the synchronized modulation of the coil capacity and fan speed in the modified design would result in an essentially constant sensible heat ratio.

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