MN-00-13-4

Hydraulic-Powered Fan Coil for Advanced Hydronic Distribution Systems, Part II: Thermal Performance and Energy Requirements of Modified Fan-Coil Units

Evelyn Baskin, Ph.D.
Associate Member ASHRAE

Edward A. Vineyard, P.E.
Member ASHRAE

Essam A. Ibrahim, Ph.D., P.E.

ABSTRACT

The two conventional fan-coil units described in Part I of this paper were modified by replacing the fan motor with a small water turbine as the power source. The water passing through the coil provides the energy for the turbine. Hence, both the coil capacity and the fan speed are simultaneously dependent on the coil water flow rate. The thermal performance and energy requirements of the modified units were measured in a controlled environment rig and compared with those of the conventional ones under the same inlet conditions and water flow rate. One purpose of the comparison was to determine whether the energy consumption of the turbine is less than that of the motor for the same fan speed. Another purpose was 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.

For cooling 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 speeds. The water flow rates were 2.5, 3.0, and 3.5 gpm (0.158, 0.189, 0.221 L/s). Steady-state measurements of the air outlet temperature, relative humidity, and water outlet flow rate and temperature were recorded for each test. The hydraulic power requirement of the turbine was determined from the measured water flow rate and difference in water pressure drop across the conventional and modified fan-coil units at the same fan speed.

It was found that for the larger units, an increase in the sensible heat ratio and decrease in the outlet air temperature were observed as the water and airflow rates were increased. For the smaller unit, both the sensible heat ratio and air outlet temperature were simultaneously increased as the water and airflow rates increased. At the same inlet air and water conditions, the sensible heat ratio and outlet air temperature for the smaller size fan-coil units were lower than their counterparts of the larger units.

INTRODUCTION

At present, hydronic systems are mainly used for heating. One of the major components in a hydronic system that requires research to broaden the market to include cooling applications is a high-performance, affordable fan-coil unit. Thermal performance, sound level, compactness, condensate removal, and comfort are among the developmental issues to be addressed (Gupta et al. 1995).

Fan-coil units can be installed in existing buildings at low cost and minimum interference in the use of occupied space. Each fan-coil unit is its own zone with a choice of heating or cooling at all times, and no seasonal changeover is required. Present zone-control techniques for forced-air systems utilize individual dampers that are cumbersome, unreliable, and costly. In addition, modulating the airflow in conventional forced-air systems results in a decreased efficiency for the space conditioning equipment. With a hydronic heat pump, capacity modulation is more easily accomplished with no appreciable loss in efficiency. Additional future opportunities for this technology include its application in the utilization of waste heat (water) from on-site power generation sources, such as fuel cells.

The main purpose of this study was to determine the feasibility of using the circulating water in a hydronic system to drive the fan to realize savings on the fan energy requirement. A secondary benefit that was investigated is the ability of the

Evelyn Baskin and **Edward A. Vineyard** are research engineers at Oak Ridge National Laboratory, Oak Ridge, Tenn. **Essam A. Ibrahim** is a professor in the Mechanical Engineering Department at Tuskegee University, Tuskegee, Ala.

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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. With comfort becoming an emerging issue in the marketplace, the ability to provide humidity control under part-load conditions is a major concern.

EXPERIMENTAL PLAN

The same two fan-coil units described in Part I of this paper were modified by replacing the fan motor with a small water turbine as the power source. The thermal performance and power requirements of the modified units were then evaluated at variable inlet air temperature and relative humidity, inlet water temperature and flow rate, and fan speed.

The water flowing in the fan-coil unit provides cooling at the coil as well as the energy required to operate the water turbine connected to the fan. Measurements of the steady-state air temperature, relative humidity, and water flow rate and temperature at outlet conditions were utilized to compare the thermal performances of conventional and modified units. The use of a turbine instead of an electric motor to drive the fan offers the possibility of intrinsic energy savings. This arrangement also has the potential for substantial reduction in installation cost compared to conventional technology. Each fan-coil unit will not require electrical connections; thus, only one trade is needed for the hydronic connections. Water flow at the fan-coil unit determines the coil water flow and the fan speed.

Measurements were made to determine if the sensible heat ratio of the fan-coil unit is fixed as a result of the dual regulation of water flow rate and fan speed. Our approach is to allow all of the water that passes through the turbine to pass through the coil. Electric and hydraulic power measurements were recorded to determine the energy penalty for the turbine in comparison with that of the motor used in the conventional fan-coil unit.

EXPERIMENTAL PROCEDURE

Two conventional low-profile vertical fan-coil units, model A and B, were selected for the present study. Model A is larger in size and capacity than model B. Each of the fan-coil units was originally equipped with a fan driven by an electric motor and a coil fitted with water-source connections. Removing the electric motor and installing a small water turbine at the water coil inlet modified the units. The turbine selected is an impulse type and is typically used to power shower/bathroom exhaust. The turbine shaft was aligned and directly coupled with the fan shaft so that the turbine and fan rotate at the same speed. The fan was 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 operation, 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. In the testing described in Part I of this paper, 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, but for the testing described here, the loop was opened due to the large pressure drop encountered when the turbine was installed. 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°F 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 the following:

a. Air dry-bulb temperature and relative humidity at the inlet and outlet of the fan-coil unit

b. Airflow 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 and turbine combination

f. Fan speed

The instrumentation used to obtain temperature and relative humidity measurements conformed with the requirements of *ANSI/ASHRAE Standards 41.1-1991 (RA 91)*, *41.3-1989*, and *41.6-1994* (ASHRAE 1991, 1989, 1994). 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 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 flowmeter located at the inlet of the fan-coil unit. An electronic pressure transducer was employed in the water pressure drop measurement across the coil. The pressure taps were constructed and installed according to *ANSI/ASHRAE Standard 37-1988* (ASHRAE 1988). The fan speed was measured by an optical stroboscope. All temperature, relative humidity, water flow, pressure drop, and wattage measuring instruments were hooked-up to a desktop computer 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 modified fan-coil unit indicating locations of

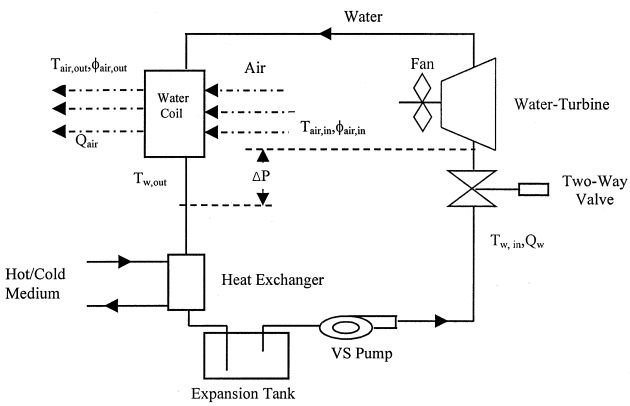

Figure 1 *Water-turbine modified fan-coil unit test arrangement.*

the measuring instruments is given in Figure 1, where T is temperature, ϕ is relative humidity, Q is volumetric flow rate, and Δp is water pressure drop. 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 applications ratings for a range of conditions that are commonly encountered. The selected application rating conditions for the present work are given in Table 1. Approximate relative humidities equivalent to the specified wet-bulb temperatures calculated from an ASHRAE psychometric chart are also listed in Table 1.

EXPERIMENTAL RESULTS AND DISCUSSION

Tests for cooling application ratings were conducted in accordance with the conditions listed in Table 1. The resulting sensible heat ratio and air outlet temperature versus water flow rate and fan speed for both the conventional and modified units are reported in Figures 2 to 9. Since the results for conventional units were presented and discussed in Part I of this paper, they are reproduced in Figures 2 to 9 of the present paper for comparison purposes only.

Measurements of the airflow rates for both models A and B of the modified fan-coil units corresponding to each water flow rate are given in Table 2. The airflow rates are computed from the product of the measured outlet air velocity and flow area. The outlet airflow area of the larger unit was found to be 0.594 ft^2 (0.055 m^2) and for the smaller unit the outlet airflow area is 0.247 ft^2 (0.0229 m^2). Note that at water flow rates of 2.5 and 3 gpm (0.158 and 0.189 L/s) the airflow rate through the smaller unit is larger than that for the larger unit. This is because the same type turbine is used to drive the fan in both fan-coil units, which produces a higher fan speed in the smaller unit due to its smaller fan size. However, the airflow rate of the larger unit becomes larger than that of the smaller one at a water flow rate of 3.5 gpm (0.221 L/s) because the increase in the fan speed of the smaller unit is not large enough to overcome the difference in fan size for the two units.

TABLE 1
Application Rating Conditions

Entering Air Dry-Bulb Temperature, °F (°C)	75, 80 (23.89, 26.67)
Entering Air Wet-Bulb Temperature, °F (°C)	64, 67 (17.78, 19.44) (RH = 54.9%, 50.8%)
Entering Water Temperature, °F (°C)	45 (7.22)
Water Flow Rate, gpm (L/s)	2.5, 3, 3.5 (0.158, 0.189, 0.221)

TABLE 2
Variation of Airflow Rate with Water Flow Rate for Modified Fan-Coil Units

Water Flow Rate, gpm (L/s)	Airflow Rate, cfm (L/s) Unit A	Airflow Rate, cfm (L/s) Unit B
2.5 (0.158)	20 (9.43)	37 (17.45)
3.0 (0.189)	60 (28.3)	64.4 (30.38)
3.5 (0.221)	113.3 (53.44)	81.4 (38.4)

Figures 2 and 3 show, respectively, the sensible heat ratio and air outlet temperature variations with water flow rate and fan speed (airflow rate) for the model A conventional and modified units at air inlet conditions 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 increases as the air and water flow rates are simultaneously increased (since the water flow rate dictates the fan speed). Figure 3 indicates that the air outlet temperature is decreased with increased air and water flow rates.

Similar data were obtained for the cooling application rating tests conducted for the same modified unit 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 temperature 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. Because of the lower inlet air temperature associated with the results of Figures 4 and 5, the sensible heat ratio and outlet air temperatures are generally reduced below their counterparts of Figures 2 and 3, respectively, at the same water flow rate and fan speed. The trends of variations of sensible heat ratio and air outlet temperature with air and water flow rates are in harmony with those observed in Figures 2 and 3. An exception is the measurement of the air outlet temperature at the water flow rate of 3.5 gpm (0.221 L/s) in Figure 5.

Experimental measurements for the smaller conventional and modified fan-coil unit, model B, are reported in Figures 6, 7, 8, and 9 for the same cooling application rating conditions as the larger model A unit. Figures 6 and 7 show the behavior

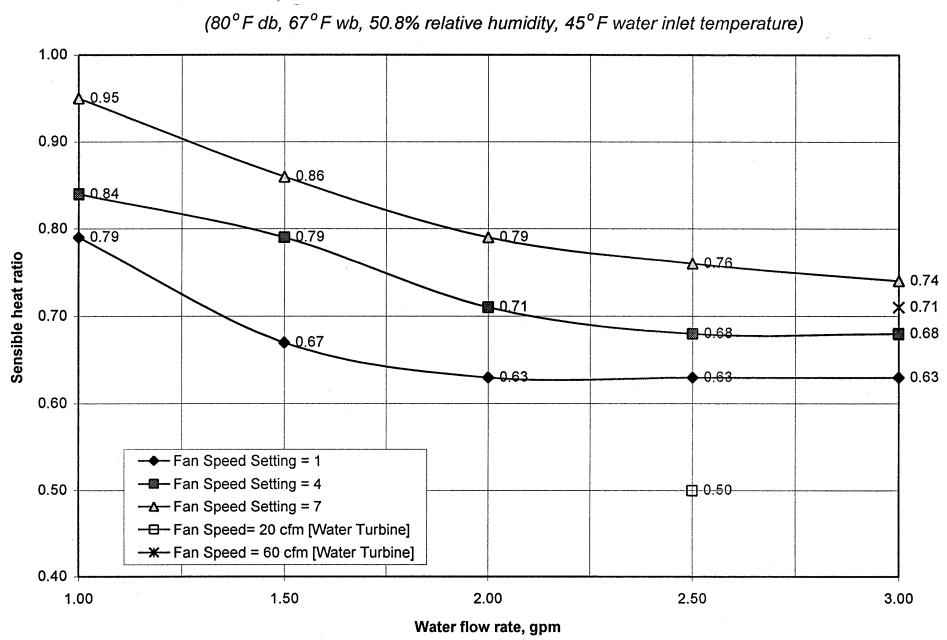

Figure 2 *A unit: sensible heat ratio vs. water flow.*

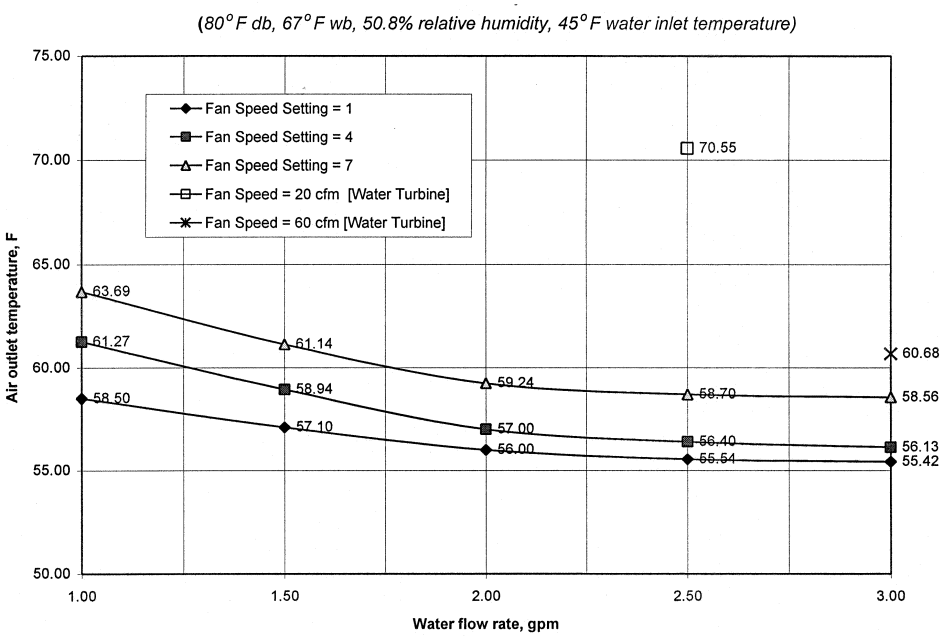

Figure 3 *A unit: air outlet temperature vs. water flow.*

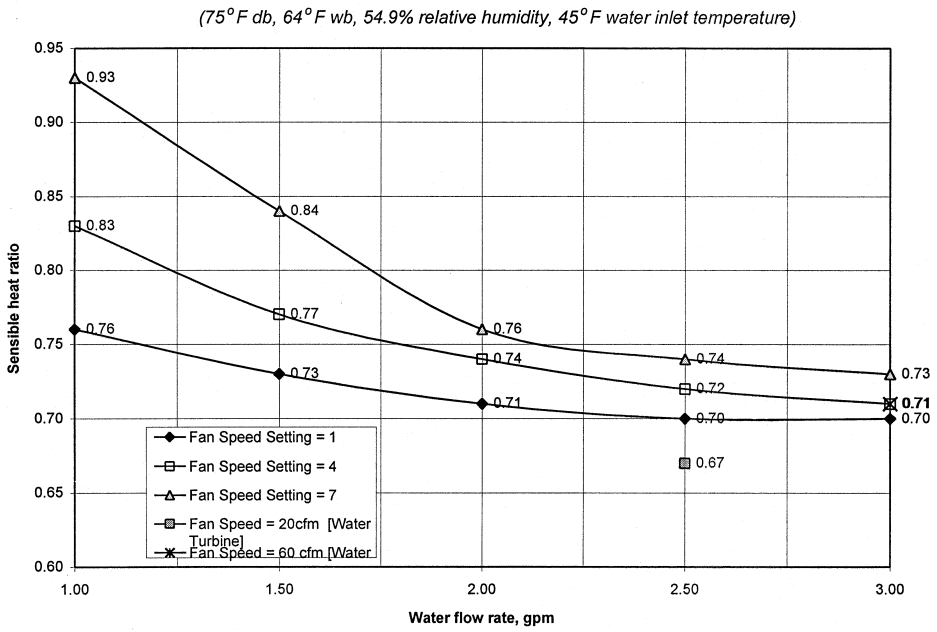

Figure 4 *A unit: sensible heat ratio vs. water flow.*

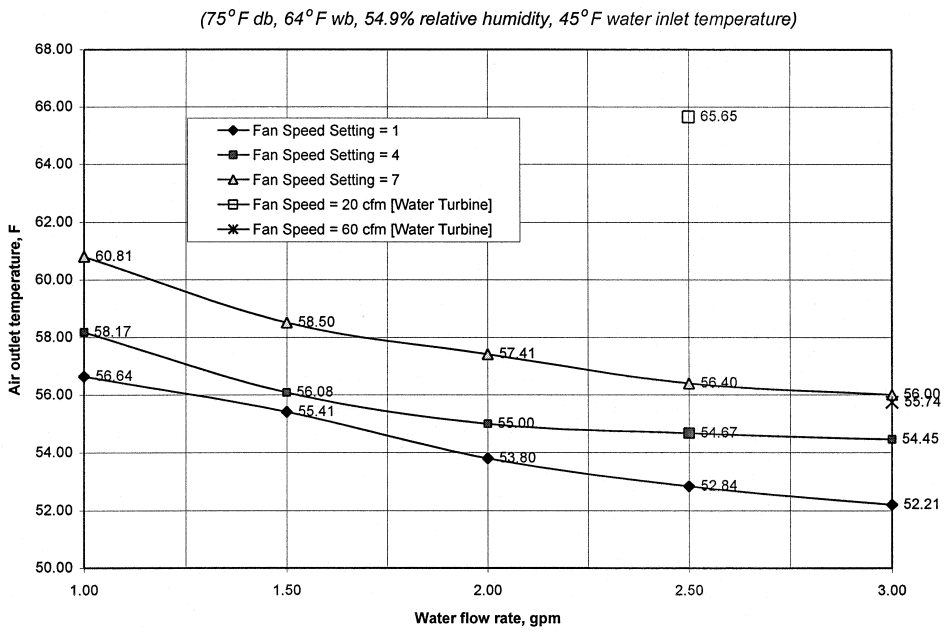

Figure 5 *A unit: air outlet temperature vs. water flow.*

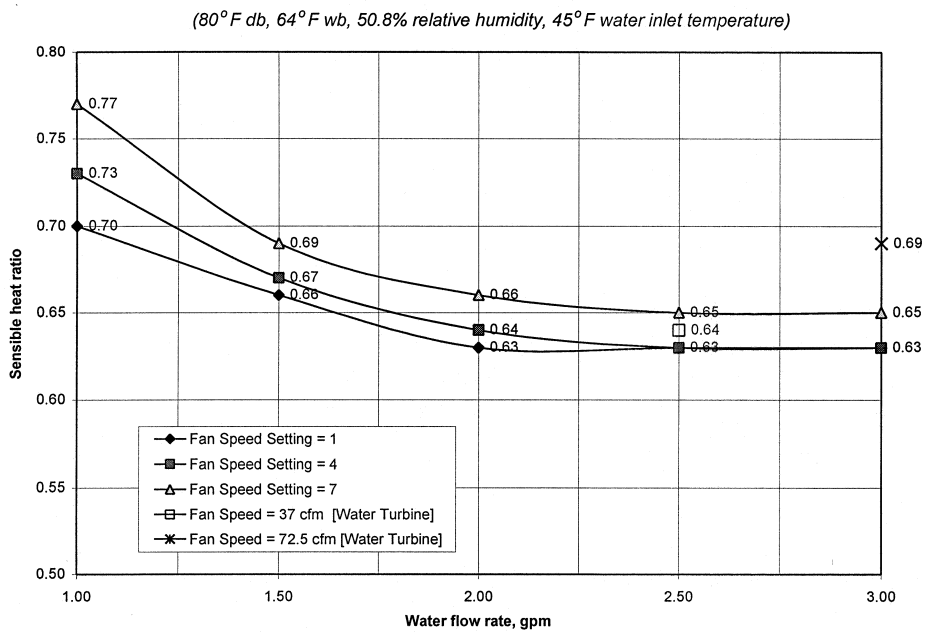

Figure 6 *B unit: sensible heat ratio vs. water flow.*

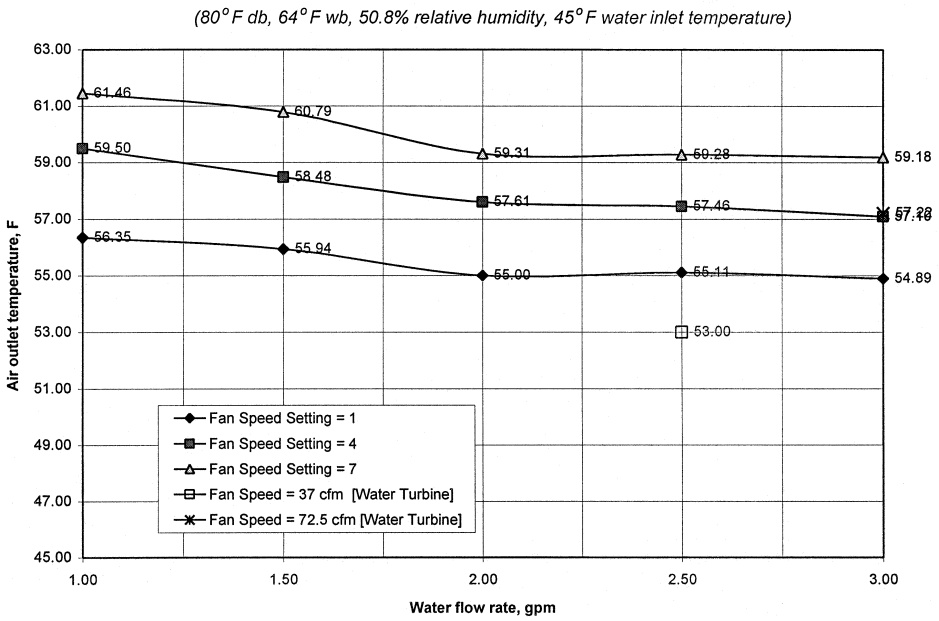

Figure 7 *B unit: air outlet temperature vs. water flow.*

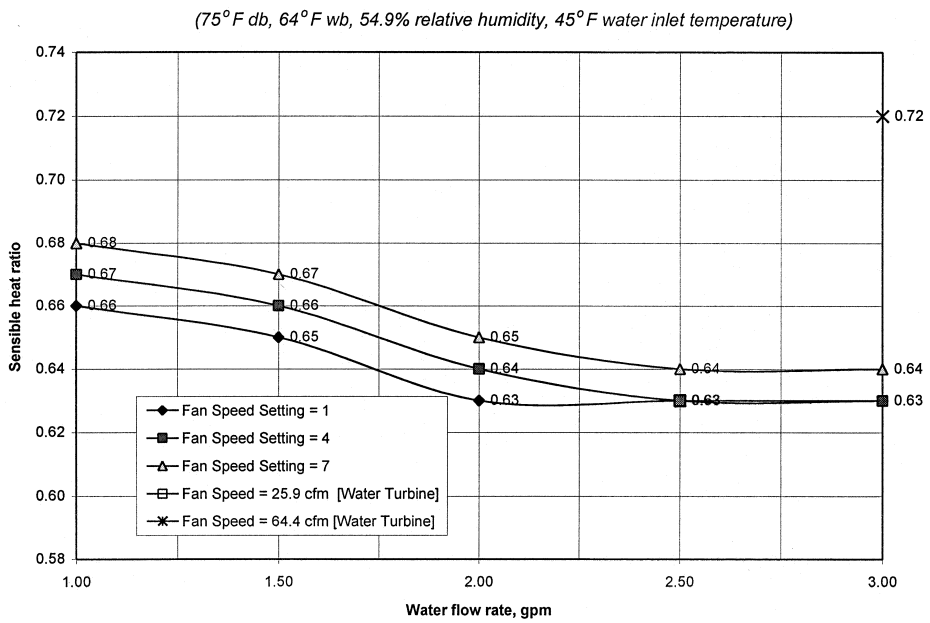

Figure 8 *B unit: sensible heat ratio vs. water flow.*

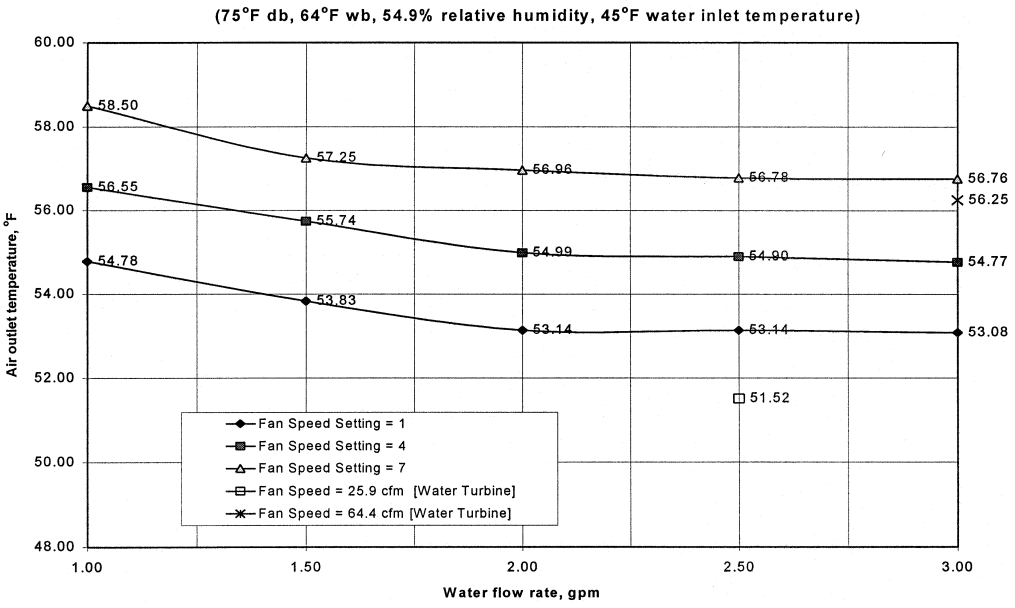

Figure 9 *B unit: air outlet temperature vs. water flow.*

of the sensible heat ratio and outlet air temperature at conditions similar to those of Figures 2 and 3, while Figures 8 and 9 show conditions similar to those in Figures 4 and 5. It is evident from Figures 6, 7, 8, and 9 that both the sensible heat ratio and the air outlet temperature are increased as the air and water flow rates are increased. These results may be explained in light of a heat transfer balance. The heat transfer rate is proportional to the product of the mass flow rate, specific heat, and the difference between the inlet and outlet temperatures for each fluid. Therefore, if the inlet temperatures are maintained constant, the rise in the air outlet temperature as the airflow rate increases is expected since a smaller air inlet-outlet temperature difference is needed to keep the heat balance. Although the water flow rate is also increased, which results in a higher heat transfer rate, the effect of the increased airflow rate predominates the heat balance since its increase is much larger than that of the water flow rate, as may be confirmed from Figures 6 through 9. The higher air outlet temperature induces an increase in the sensible heat ratio, which is defined as the ratio of sensible heat to the sum of sensible and latent heat removed from the air. This is because the rise in air outlet temperature reduces the rate of latent heat removal from the air through water vapor condensation. Note also that the outlet air temperatures of Figure 9 are reduced below their counterparts of Figure 7 at the same water flow rate due to the lower inlet air temperature associated with the results of Figure 9.

An important objective of the present study is to compare the power consumption of the turbine in the modified fan-coil units with that of the electric motor of the conventional units. In order to do so, the hydraulic power requirement of the turbine was calculated from the measurements of the water flow rate and the water pressure drop across the conventional and modified units according to

$$P_t = Q_w\,(\Delta p_{mod} - \Delta p_{conv}) \qquad (1)$$

where P_t is the input power to the turbine, Q_w is water volume flow rate, Δp_{mod} and Δp_{conv} are the water pressure drops across the modified and conventional units, respectively. Table 3 summarizes the water pressure drop measurements across the two varieties of conventional and modified units against water flow rate. Note that no pressure drop data are available for the conventional units at a water flow rate of 3.5 gpm (0.221 L/s) since these units were not tested at that water flow rate, as reported in Part I of this paper. It is evident from Table 3 that the water pressure drop across the coil of the smaller unit is larger than that of the larger one. This is because the coil diameter of the smaller unit is narrower, causing the water velocity and, hence, the associated friction head loss through the coil to be higher.

A proper comparison of the hydraulic power requirement of the turbine in the modified units and the electric power needed for the motor of the conventional units must be made at the same water flow rate and fan speed. The water flow rate through the conventional and modified units should be the same so that the difference in water pressure drop across the units would be solely due to the turbine. This difference in water pressure drop across the modified and conventional units and the water volumetric flow rate are used to calculate the input hydraulic power to the turbine in accordance with Equation 1. The turbine must produce the same fan speed and, hence, the same airflow rate to ensure that the hydraulic output power of the turbine in the modified unit is equivalent to the electric power of the motor in the conventional unit.

Because of the restrictions of equivalent water and airflow rates, comparison of the power requirements of the turbine and electric motor are only possible for model B at a water flow rate of 3.0 gpm (0.189 L/s) where the airflow rate is 64.4 cfm (30.38 L/s) for the modified unit, as shown in Table 2. This airflow rate is approximately equal to that of 58.3 cfm (27.51 L/s) for model B of the conventional unit produced at fan speed setting 1 (597.5 rpm) as reported in Table 2 of Part I of this paper. Note that the airflow rate in the conventional unit is independent of the water flow rate. Since at a water flow rate of 3.0 gpm (0.189 L/s), the pressure drops across model B of the modified unit and conventional unit are 18.19 psi (125.33 kPa) and 8.525 psi (58.79 kPa), respectively, the hydraulic input power to the turbine is 12.61 W in accordance with Equation 1. The power requirement of the electric motor of model B of the conventional unit at fan speed setting 1 (equivalent airflow rate) is 4.13 W as documented in Table 2 of Part I of this study.

The turbine required 2.5 to 3 gpm of water and 25 to 30 psi of pressure to function, but the conventional fan-coil units A and B tested had seven fan speed settings and operated with water flow rates varying from 1 to 3 gpm depending on setting. Therefore, the flow rate requirement of the turbine restricts comparative fan-coil tests to the highest two fan speed settings. To perform comparative tests at lower settings, the turbine needs to be reconfigured.

TABLE 3
Water Pressure Drop Across Conventional and Modified Units vs. Water Flow Rate

Water Flow Rate, gpm (L/s)	Pressure Drop (Conv. Unit A), Δp_{conv}, psi (kPa)	Pressure Drop (Conv. Unit B), Δp_{conv}, psi (kPa)	Pressure Drop (Mod. Unit A), Δp_{mod}, psi (kPa)	Pressure Drop (Mod. Unit B), Δp_{mod}, psi (kPa)
2.5 (0.158)	2.23 (15.3)	7.02 (48.41)	8.06 (55.53)	13.21 (91)
3.0 (0.189)	3.53 (24.34)	8.525 (58.79)	11.82 (81.44)	18.19 (125.33)
3.5 (0.221)			19.86 (136.84)	23.95 (165)

CONCLUSIONS

The thermal performance and energy requirements for the cooling mode operation of two sizes of hydraulic-powered 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 hydraulic power consumption of the turbine-driven fan was estimated via pressure drop measurements across the turbine.

For both fan-coil units tested in the present work, the sensible heat ratio is increased as the coupled water and airflow rates are increased. The effect of increasing the water and airflow rate is to reduce the air outlet temperature for the larger unit, while the opposite is true for the smaller unit. The sensible heat ratio and outlet air temperature of the smaller unit were found to be lower than those of the larger unit.

The turbine used in the test was originally designed to be installed in the water line to operate a shower exhaust fan, which does not produce a back pressure (free flowing—no constraints). Test results for the turbine-powered fan-coil unit indicated that a higher water flow rate is required to operate the turbine, as the result of the turbine experiencing significant resistance caused by the back pressure. Therefore, design modifications are required to make the turbine less sensitive to the back pressure and capable of operating at a flowrate as low as 1 gpm. Modifications to the turbine design and further testing are required to substantiate the water turbine concept.

ACKNOWLEDGMENTS

The authors would like to thank John Tomlinson, research engineer at Oak Ridge National Laboratory (ORNL), for helpful suggestions and discussions, and Randy Linkous, technologist at ORNL, for his invaluable technical assistance. E.A. Ibrahim would like to acknowledge the support of the Department of Energy, Oak Ridge National Laboratory (ORNL, through Lockheed Martin Energy Research Corp. subcontract # 4500008725). E.A. Ibrahim also thanks Abdi Zaltash, research engineer at ORNL, for helpful discussions.

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