Application Study of the Pump Water Flow Station for Building Energy Consumption Monitoring and Control Optimization

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ABSTRACT: This paper presents a new building energy monitoring and pump speed control method. The pump speed is controlled to maintain the system resistance at an optimized value to approach the best pump efficiency and save pump power. The system resistance can be obtained by the pump head and the water flow rate calculated by the pump water-flow station (PWS), which was recently developed. The PWS measures the water flow rate using the pump head, pump speed, and pump performance curve. This method has been experimentally proved in real HVAC systems. A case study was demonstrated in this paper for application of this new method in a Continuous Commissioning (CC) practice. The case study shows that the PWS can control the pump speed to maintain the optimized system operating point. It can also measure the water flow rate and monitor energy consumption continuously with low installation and almost no maintenance cost. The results show that the new technology can save pump power and increase pump efficiency significantly.

Key words：Pump control; water flow; measurement.

INTRODUCTION

It is critical to accurately measure the chilled and hot water flow rates for building energy monitoring. The most common uses of flow sensors (meters) in hydronic systems are for monitoring energy usage (e.g., for sub-metering tenants served by a common hot or chilled water system) and for control of large, variable flow, central chilled water and hot water plants. [ASHRAE handbook 2000]

In a typical variable speed pumping arrangement, constant flow pump(s) recalculate the chiller or boiler source in a primary source loop, and a variable speed distribution pump located at the source plant draws flow from the source loop and distributes it to the load terminals. The speed of the distribution pump (secondary pump) is determined by a controller measuring differential pressure across the supply-return mains or across the selected critical zones. Two-way control valves are installed in the load terminal return branch to vary the flow required in the load.

In practically all variable-speed chilled water pumping applications, the pump speed is controlled to maintain a constant pressure differential between the main chilled water supply and return lines. However, this approach is not optimal. In order to maintain a constant pressure differential with changing flow, the control valves for the air-handling units must close as the load (i.e., flow) is reduced, resulting in an increase in the flow resistance. The set point of the differential pressure (DP) is always set at a very high value in normal systems. The two-way valves for the terminals are forced to close because of the high static pressure at partial load conditions. The systems are operating at partial load most of the time. The pump is forced to run at high speed and consumes extra power to maintain the high DP. The operation points may fall into the unstable zone of pump curve with very low pump efficiency.

To solve this problem, another control method was developed for Direct Digital control (DDC) systems that can send the two-way valve position signal for the terminals to energy management control systems (EMCS). A good strategy for a given chilled water set point would be to reset the DP set point in order to maintain all discharge air temperatures with at least one control valve in a saturated (fully-open) condition. This results in a relatively constant flow resistance and greater pump savings at low loads. This method can ensure that the pump is running at the design system resistance and meet the load requirement. Unfortunately, this method can not be applied to the systems using the pneumatic controller. Another difficulty of this control approach is that the valve position data is often unreliable. The valve could be stuck open or the saturation indicator could be faulty. [ASHRAE handbook 1999]

Therefore, a new control method was developed in this paper to optimize the pump variable frequency drive (VFD) speed control. The pump speed is controlled to maintain an optimized system resistance (S value). The S value can be calculated by Equation 1:

$S = H/Q^2$ (1)

where H is the pump head and Q is the water flow rate. The pump head can be obtained by the differential pressure transducer. Therefore it is very critical to measure the chilled or hot water flow rate accurately for the optimal system control. Typical water flow measurement methods are described below:

1). Differential pressure flow meter: This correlates differential pressure to flow using Bernoulli's equation. For example, the Venturi meter is commonly used for steam flow measurement, but it is less commonly used for water flow measurement because of the poor accuracy at low flow rates and high installation cost.

2) Displacement flow meter: The meter works by using the fluid to rotate or displace a device inserted into the flow stream, e.g., a turbine flow meter, tangential paddlewheel meter, etc. It causes extra pressure drop. The bearing wears out and calibration is often needed to ensure accuracy. This meter has a high cost for installation and calibration.

3). Passive flow meter: The transit time ultrasonic meter, for example, is very accurate. The reported accuracy is 1% high (HVAC control fundamentals). However, the application may be influenced by the pipe dimensions, the surface conditions, fluid properties and other practical limitations. Many failures were encountered during the measurement because of the diameter and the surface paint. Magnetic flow meter meters are also accurate, but they are too expensive to use in HVAC applications.

Therefore, we need to find a method that can measure the pump water flow accurately and reliably, and is less expensive to install.

METHODOLOGY

Performance of a centrifugal pump is commonly shown by a manufacturer's performance curve. The curves are generated from a set of standard tests developed by the Hydraulic Institute (1994). The tests are performed by the manufacturer for a given pump volute or casing and several impeller diameters, normally from the maximum to the minimum allowable in that volute. The tests are conducted at a constant impeller speed for various flows. Pump curves represent the average results from testing several pumps of identical design under the same conditions. The curve is sometimes called the head-capacity curve (H-Q) for the pump.

The pump water-flow station (PWS) was developed by Liu et al. (2006) recently. The PWS can measure the water flow through the pumps using the pump speed, pump head and the in-situ pump curve. The theoretical model has been experimentally tested and excellent agreement between the model and the experimental values was found (Liu et al. 2006). The total pump head can be regressed as a function of the water flow rate using the design pump performance curve. Equation (2) is a typical polynomial regression curve,

$$H_d = \sum_{i=1}^{n} a_i \cdot Q_d^i \quad (2)$$

where H_d and Q_d are the pump head and water flow rates for the pump at 100% speed.

The centrifugal pump, which imparts a velocity to a fluid and converts the velocity energy to pressure energy, can be categorized by a set of relationships called affinity laws. The affinity laws are useful for estimating pump performance at different rotating speeds or impeller diameters D based on a pump with known characteristics. The laws can be described as similar processes that follow these rules (ASHRAE handbook 2000):

1. Flow (capacity) varies with rotating speed N (i.e., the peripheral velocity of the impeller).

2. Head varies as the square of the rotating speed.

3. Brake horsepower varies as the cube of the rotating speed.

When the pump is operating at reduced speed, the pump curve can be expressed using Equation (3) based on the affinity laws:

$$H = \omega^2 \sum_{i=1}^{n} a_i \cdot \left(\frac{Q}{\omega}\right)^i \quad (3)$$

where $\omega = N/N_d$, $H = \omega^2 H_d$, and $Q = \omega Q_d$

Typically, the pump curve can be expressed using second-order polynomial equations. When i=2,

$$H = a_0 \overline{\omega}^2 + a_1 \cdot Q\overline{\omega} + a_2 \cdot Q^2 \quad (4)$$

Using a second order polynomial regression of the design curve, the pump water flow can be predicted using the measured pump head and pump speed with Equation (5):

$$Q = \frac{-a_1 \cdot \overline{\omega} - \sqrt{a_1^2 \cdot \overline{\omega}^2 - 4 \cdot a_2 \cdot \left(a_0 \overline{\omega}^2 - H\right)}}{2 \cdot a_2} \quad (5)$$

The pump water flow station includes a differential pressure transducer and pump speed transducer. The pump speed transducer can be replaced by the calibrated control system's command to the VFD (Liu 2005). The pump water flow station can be implemented using typical EMCS.

Then the pump head and the water flow rate calculated by PWS can be used in the pump speed control. The pump manufacturer usually plots the efficiencies for a given volute and impeller size on the pump curve to help the designer select the proper pump. The best efficiency point (BEP) is the optimum efficiency for this pump; the operation above and below the point is less efficient. The ASHRAE Handbook (2000) gives the optimum ranges to use when selecting a centrifugal pump. A selection limit of 66% to 115% of flow at the BEP is suggested. Where possible, pumps should be chosen to operate to the left of the BEP because the pressure in the actual system may be less than design due to overstated data for pipe friction and other equipment (ASHRAE Handbook 2000, 29.10).

Using the PWS, the pump speed can then be controlled to maintain the same system resistance as the design operation point, which should be close to the BEP if the pump was properly selected. In the meantime, the building cooling and heating energy consumption can be minored and recorded using the PWS and water temperature difference. The application of PWS will be introduced in the case study.

CASE STUDY

A case study was conducted on a full-scale hot water system to validate the theory and to optimize the pump speed control. This building is located in Omaha, Nebraska. It was originally built in 1960. This office building includes a 16-story tower and a two-story additional building to the east. The total conditioned area is 216,000 square feet. The central plant is equipped with two chillers and five boilers. The boilers have a total heating capacity of 11,700 MBH. The boilers produce hot water for a heating coil of AHUs and perimeter water heating systems in the east additional section. The hot water system also provides the heat to the glycol outside air preheat coil of two AHUs through a heat exchanger. Pressure transducers were installed on both the discharge and suction sides. The accuracy is ±0.3 psi (± 0.15% of F.S.). A schematic diagram of hot water system, existing and added sensors and devices are shown in Figure 1.

1) Existing Control

The boiler is enabled when the outside air temperature is lower than 57ºF.

- There are two secondary hot water pumps. The leading pump speed is modulated to maintain the hot water loop differential pressure at its set point of 12 psi (83kPa). The backup pump will be enabled when the first pump reaches its 90% speed. The DP sensor is in the 16th floor penthouse.
- The east wing fin tube heating system is enabled when outside air temperature is lower than 50ºF

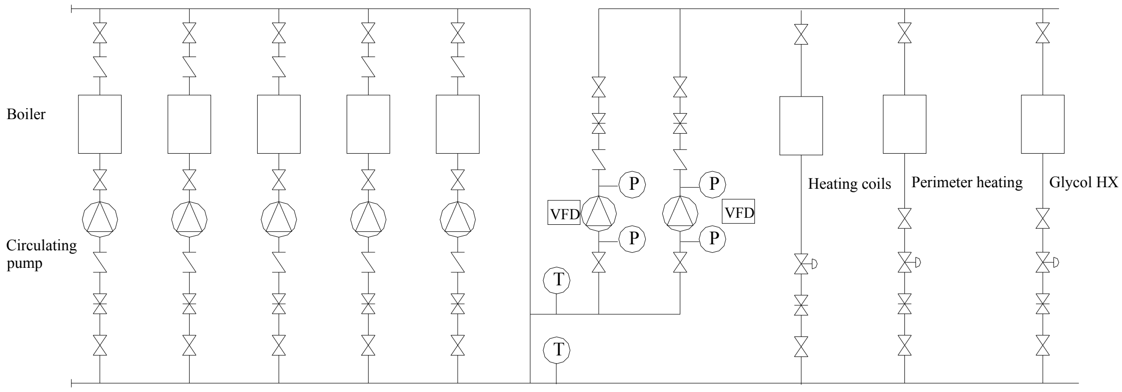

Figure 1: Hot water system diagram

Continues Commission (CC) measures have been implemented in this system. The major CC measure for the boiler system is installing the PWS.

2) Pump Flow Station

Pressure transducers are already installed on both the discharge and suction side of the two secondary hot water pumps. And they were calibrated before implementing our new control sequence. The VFD speed command and pump head can be obtained in EMCS. The hot water flow through the secondary hot water pumps can be calculated based on Equation (4) if the calibrated pump curve is used. Building heating energy consumption can be monitored using the calculated hot water flow and the difference of the supply and return hot water temperature.

I. Record the Pump Information.

Model: Horizontal Base Mounted Pumps, Standard construction
Motor: 25 HP, 1800 RPM, 460V
Impeller: 10.71"
Capacity: 560 GPM @ 100 Feet of head

II. Obtain the pump curve for the manufacturer

Figure 2 shows the manufacturer’s pump curve. The impeller diameters were shown for each curve.

III. Pump Curve Calibration

The pump curve manufacturer given is the total head curve. The pump speed is at 1800rpm with a 10.50” impeller diameter, therefore in-situ calibration is needed.

- Speed and impeller diameter adjustment

First, the pump speed was measured at full speed command using the tachometer. The actual measured speed is 1778 rpm. The pump curve at partial speed can be drawn using Equation (3). Assuming the pump curve in 10.50” was used, the pump curve can be deduced based on the affinity law (ASHRAE Handbook) as follows:

$$H = a_0\left(\overline{\omega}\overline{D}\right)^2 + a_1 \cdot Q\overline{\omega}\overline{D} + a_2 \cdot Q^2 \qquad (6)$$

where $\overline{\omega}$ is the speed ratio (1778/1800) and $\overline{D}$ is the diameter ratio (10.7/10.5).

- In-situ curve calibration

Liu et al. (2005) developed a procedure to generate the in-situ fan curve. The same algorithm works for the pump performance curve. The in-situ pump curve was measured following the procedure developed by Liu et al.(2005). The in-situ curve can be expressed as H = $-0.6769Q^2 + 0.3329Q + 1.0363$

at full speed (1778rpm), where a_0=1.0363,a_1=0.3329, and a_2=-0.67691 for Equation (4). Then the actual water flow rate can be easily calculated using these coefficients and Equation (4) at partial speed conditions.

3) Application I: Heat energy consumption monitoring

All the parameters including OAT, pump speed, pump head and pump water flow calculated by the PWS were recorded every 15 minutes. Figure 4 shows the measured GPM and hot water supply and return temperature. Figure 5 shows the calculated heating energy consumption. (note that two charts at different time range)

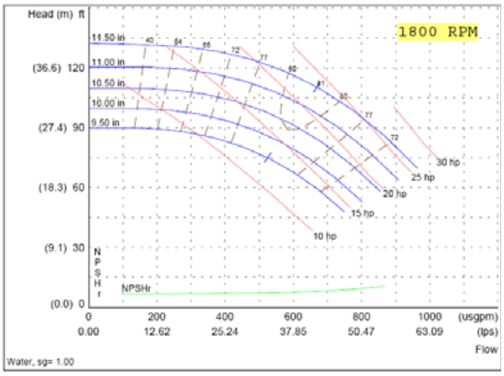

Figure 2: Manufacturer's pump performance curve

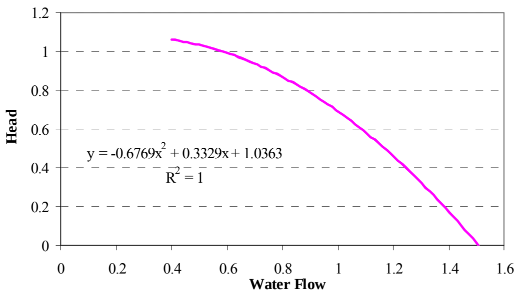

Figure 3: In-situ pump curve after calibration (dimensionless)

Figure 4 and 5 show that the pump water flow rate and the heating energy consumption can be measured and calculated continuously and reliably.

4) Application II: Pumps speed optimization control

The design water is 560 GPM at 100 Feet of Head. The dimensionless S value is 1 with the highest pump efficiency at 81% according to the pump performance curve in Figure 3. Therefore the pump speed can be controlled to maintain S=1.2 at partial speed with ±0.25 dead band.

The secondary pump was operated using the old control sequence (before CC) from 5/14 to 5/15 with a total of 490 sets of data. It was operated using the new control sequence (after CC) from 5/12 to 5/13 with 513 sets of data. The outdoor air temperature is in similar range from 45ºF (7.2ºC) to 65ºF (18.3ºF) for these two periods. All the parameters including OAT, pump speed, pump head and pump water flow calculated by the PWS were recorded every 15 minutes. Figure 6 compares the measurement results for these two control sequences. It shows that the pump was working in the unstable zone with the pump efficiency less than 60% most of the time before CC. The pump was running between 45% to 50% speed before CC. After implementing the new control sequence, the pump efficiency increased to the range of 77% to 81% by controlling the system resistance at design conditions. The pump speed decreased to the range of 16.7% to 25%.

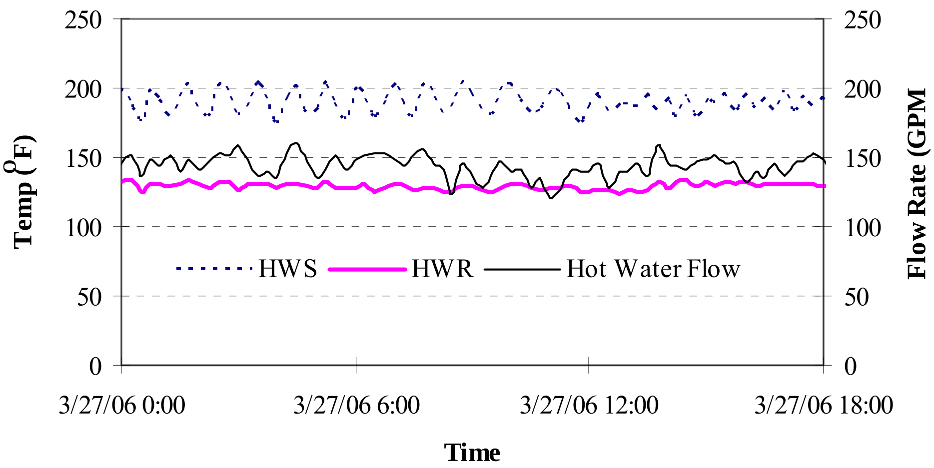

Figure 4: Hot water flow measurement

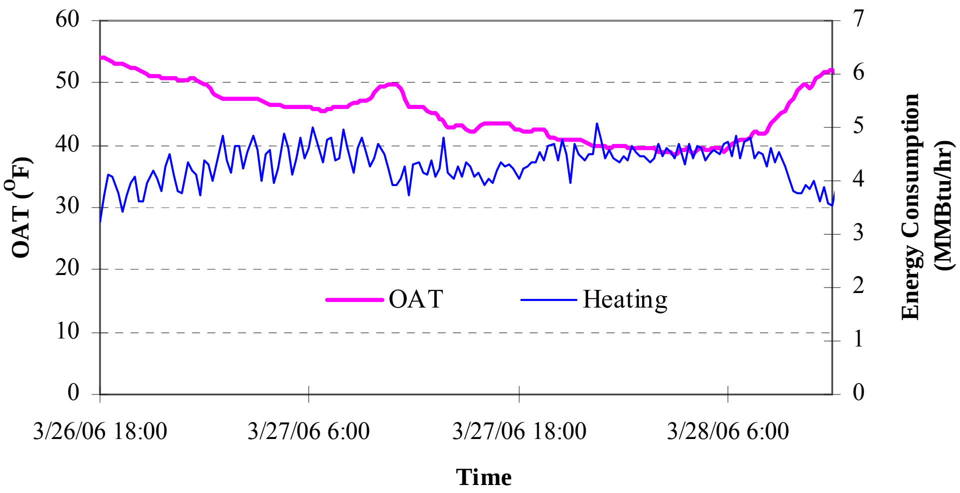

Figure 5: Measured heating energy consumption

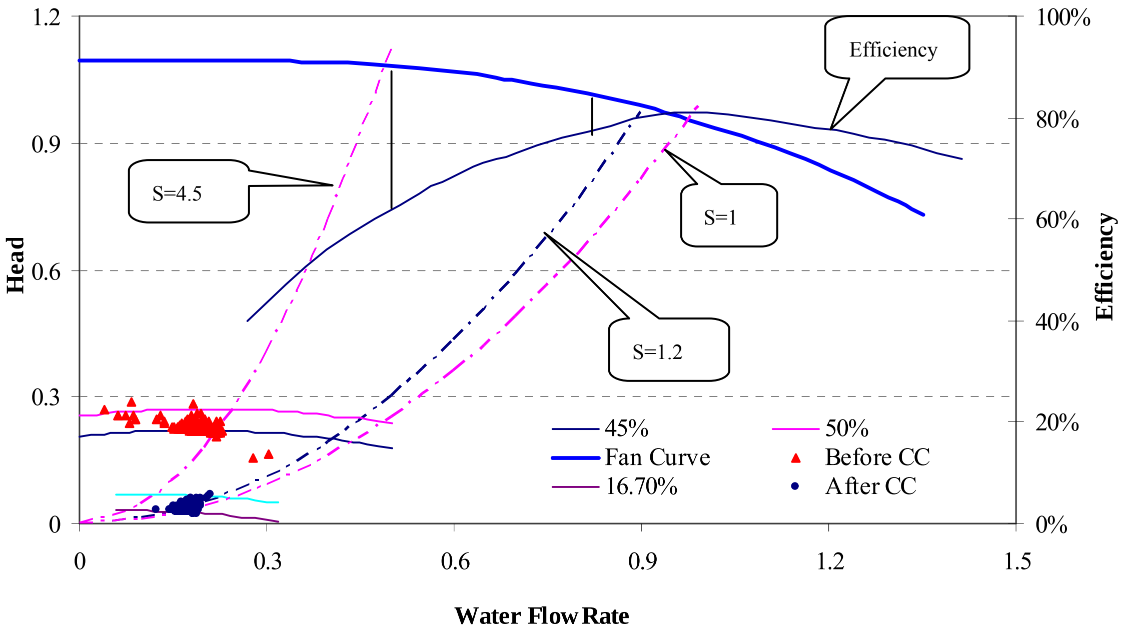

Figure 6: Comparison of the control sequence before and after CC

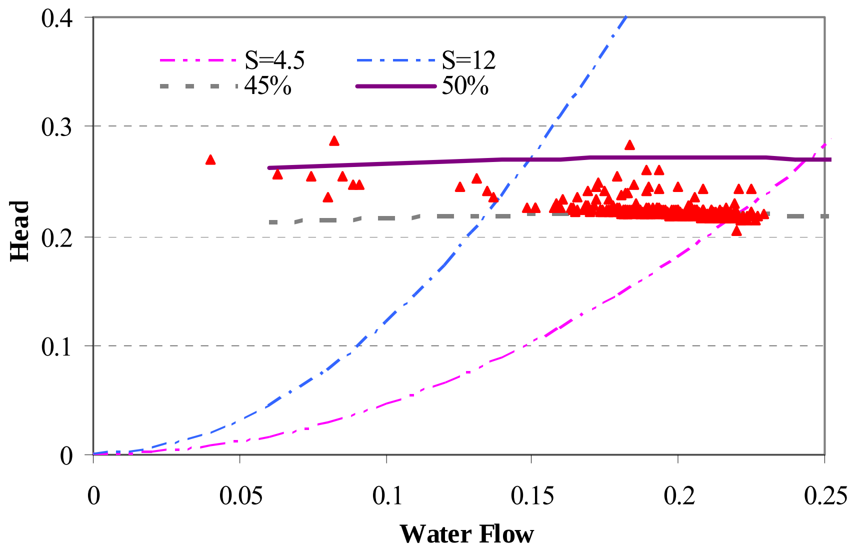

Figure 7: Measurement results before CC

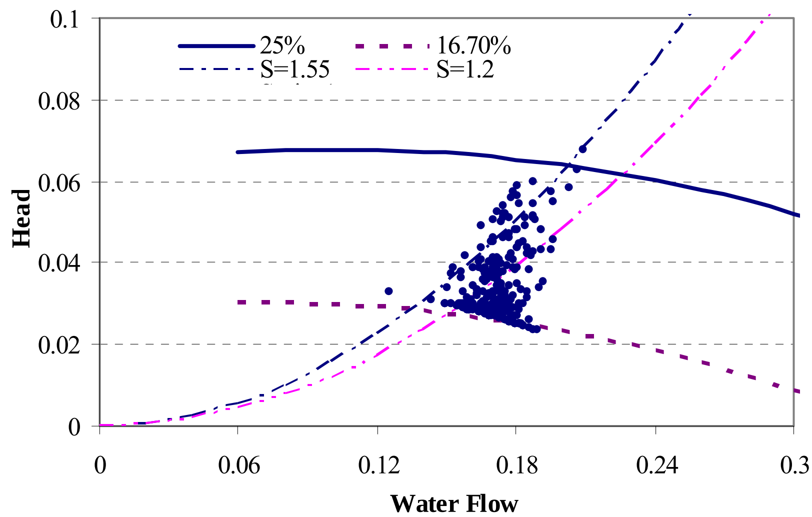

Figure 8: Measurement results after CC

Figure 7 and 8 show the zoomed measurement results before and after the new control sequence were implemented. Figures 6 to 8 show that the new control sequence, controlling the pump speed to maintain the design system resistance (or BEP) using PWS, can increase the pump efficiency and decrease the pump speed significantly. It can save pump power eventually.

Before the CC measurement, the maximum cooling valve position never exceeds 55% open. Figure 9 shows the maximum cooling valve position after the new control sequence is implemented. It shows that the maximum cooling valve position was about 80% open (average of 82.3%). The chilled water valves were fully open on the night of 5/12 because the boiler was off at that time.

Figure 10 compares the pump speed versus the outdoor air temperature before and after CC measure. It shows that the new control sequence can decrease the pump speed more than 30% when OAT is higher than 40ºF. We don't have the data for pump speed when OAT is below 40ºF after CC.

The average temperature difference of hot water supply and return was 33ºF using the old control method. The temperature difference between supply and return hot water is more than 45ºF after the new control sequence was implemented.

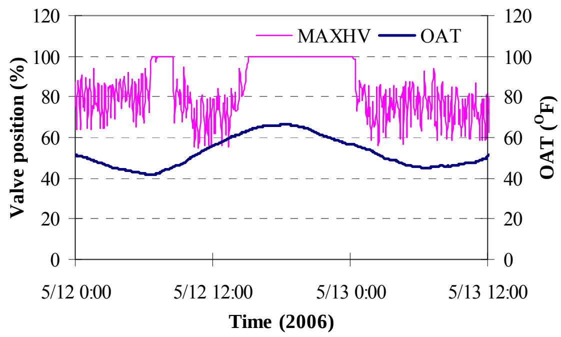

Figure 9: Maximum valve position after CC

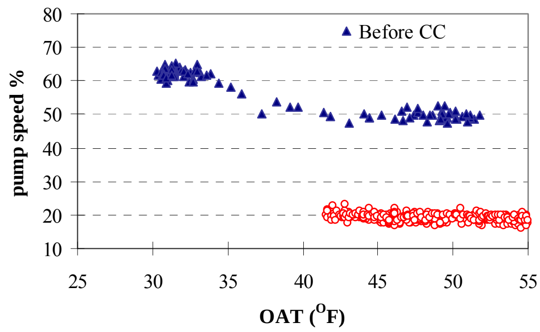

Figure 10: Comparison of pump speed

CONCLUSIONS

This paper presents a new building energy monitoring and pump speed control method. The pump speed is controlled to maintain the system resistance at an optimized value to approach the best pump efficiency and save pump power. The system resistance can be obtained by the pump head and the water flow rate calculated by a new water flow measurement method called a pump water flow station. It has been recently developed to monitor building energy consumption and optimize the secondary pump speed control. This method calculates the water flow rate based on the measured pump head, pump speed and manufacturer's pump performance curve. The pump speed is controlled to maintain the same system resistance value as the design condition using the PWS and measured pump head. This method is accurate, reliable and less expensive to install and requires no maintenance costs.

The case study shows that the PWS can help evaluate and optimize the system to operate at the best efficiency region and reduce the pump speed significantly, compared with the typical pump speed control method using the DP control. The results show that the PWS can measure the water flow rate and monitor energy consumption continuously. And this measurement method is very simple to implement and almost no cost for maintenance and can save a lot of money, time and labors. Furthermore, the pump efficiency was increased and pump speed was reduced significantly while the hot water system can works fine.

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