An engineering-oriented variable water flow air-conditioning system terminal flow rate estimation method based on component flow resistance characteristics

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Abstract
Water flow rate plays an important role in the modeling prediction, fault detection and diagnosis, and performance optimization of the variable water flow air-conditioning (VWFAC) system. However, flowmeters employed by the system's terminals have not been widely used in engineering applications for the constraints in installation space and high installation and retrofit costs. Therefore, in this study, a terminal flow rate estimation method is proposed for the VWFAC system to reduce the dependency on flowmeters. The water flow rate estimation model is developed based on the flow resistance characteristics inside the air handling unit (AHU) and was trained and verified at the Monitoring and Control Laboratory established in the Dalian University of Technology. The results indicate that the maximum root-mean-square error (RMSE) during the training and validation sessions are 0.038 m$^3$/h and 0.028 m$^3$/h, respectively, while the corresponding mean absolute percentage error (MAPE) are below 1.4% and 6.5%, which is acceptable for engineering applications. To improve the flow rate estimation accuracy of the model, water temperature, water flow rate, and pressure difference are suggested to cover a wide varied range during the training session as much as possible. Systematic error is an important index in determining the demands for sensor's accuracy class. According to the results of water flow rate, the systematic errors of the estimates are ranged between 0.51% and 0.76%, only about 1/4 to 1/3 of the measurements systematic errors. Based on the error propagation theory, the water flow rate estimates would be reliable if the systematic error of the pressure measurements does not over twice that of the water flow rate measurements. This flow rate estimation method can be further applied to other thermal engineering systems to bring considerable economic benefits for engineering applications.

Keywords: end unit, error analysis, flow rate estimation, resistance, variable water flow air-conditioning system
1 | INTRODUCTION

For many countries, high quantity of building electricity consumption has been a critical challenge to be addressed. As a major end user of electricity, heating ventilation and air-conditioning (HVAC) systems play an important role in building energy conservation.1,2 Compared with conventional constant water flow air-conditioning systems, variable water flow air-conditioning (VWFAC) systems enable the regulation of refrigerant flow rate or supply air volume to cope with the variation in cooling/heating loads, so that are much more energy efficiency under changing operating conditions.3-5 In recent years, VWFAC systems are gradually being applied to many energy-efficiency retrofit projects in China, making it meaningful to monitor the performance of the system under various operating conditions.

Meteorological conditions (both indoor and outdoor), operational and structural parameters, and the occupant's behavior are major factors that can affect the electricity consumption of the VWFAC system. By monitoring parameters that closely related to these factors can provide detailed information about the system's operating performance for further data analysis in many related application fields of system operation optimization,6,7 fault detection and diagnosis,8,9 and cold/heat metering,10 etc Without considering the impact of the occupant's behavior, major parameters that worthy concerned could be effectively identified through a rough analysis of the energy exchange process between the system and its environment. Figure 1 illustrates the typical energy exchange process among the VWFAC system, indoor environment, and outdoor environment in winter conditions, wherein $\Phi_{hl1}$ represents the heat gains of chilled water attributes to the heat source, $\Phi_{hl1}$ represents the system heat loss that attributes to the motor efficiency, $\Phi_{hl2}$ represents the heat exchange between the water side and the air side within the terminal, and $\Phi_{hl3}$ represents the radiation heat transfer between the terminal vents and indoor environment. By considering the refrigerant cycle, chilled water cycle and cooling water cycle as a whole and analyzing it in terms of the persistence of energy, it would be clear that the temperature and flow rate of chilled water, the temperature and flow rate of cooling water, the temperature, flow rate and humidity of outdoor air, and the temperature, volume flow and humidity of supply air are major meteorological and operational parameters that can directly affect the motor power consumption of the VWFAC system and thus can be meaningful to be monitored.

Water flow rate plays a critical role not only in VWFAC systems but also in other thermal engineering systems.11-19 In the past few years, numerous types of water flowmeters have been invented, made available, and studied to meet various needs.20-23 To determine the measurement errors caused during their installation, Martim et al24 compared the flow rate as measured by an electromagnetic flowmeter under different installation conditions. Other studies25,26 indicated that gas-liquid two-phase flows could be measured by a Coriolis flowmeter or electromagnetic flowmeter. Zhang et al27 introduced a novel cross-section-shaped averaging Pitot tube that enabled a higher differential pressure than some other known tubes. Mu et al28 proposed a new design of a butterfly valve flowmeter that could be used effectively in variable air volume air-conditioning systems. However, in terms of engineering applications, premium manufacturing, installation, maintenance, and replacement costs together with the space constraints making it impractical for flowmeters to be widely applied to air-conditioning water systems in the short run.29

![FIGURE 1](image_url) Energy exchange between the VWFAC system and its environment
One promising potential solution to reduce the sensors cost is by exploring indirect measurement schemes of water flow rate, which have to be simple, reliable, and much more cost-efficient compared with flowmeters.\(^{30}\) The indirect water flow rate measurement, in brief, is to reduce the need of flowmeters by replacing its functions with other cheaper sensors. Based on the theory that the pump flow rate of the water system could be estimated through related dynamic parameters, that is, motor power and pump head,\(^{31-33}\) Liu et al\(^{34}\) developed a simplified in situ fan curve measurement procedure that uses the fan curve from the manufacturer and measurement of air flow and fan head at one point. The calculated curve was found to be not only closely matching with the measured one, but also much more time-efficient. Wang et al\(^{35}\) presented a theoretical model for pump flow stations to avoid extremely high installation and retrofit costs caused by replacing the flowmeters. Similarly, Qin et al\(^{36}\) developed a mathematical model using regression analysis to measure water pump flow rate. Despite providing acceptable measurement accuracy, this method is quite time consuming and cannot be used in energy-efficiency renovation projects. Wang et al\(^{37}\) subsequently proposed a virtual pump water flowmeter that could be implemented in building automation systems. Long-term experiments conducted by Kim et al\(^{38}\) showed that flow rates measured by the virtual water flowmeter were quite close to those detected using ultrasonic measurements. Liu et al\(^{30}\) further introduced a high precision pump flow virtual monitoring model whose input parameters can be easily obtained.

Although the pump flow rate estimation methods against the air-conditioning water system based on the dynamic characteristics of water pumps have been extensively studied, the high cost and installation matters came up with flowmeters still failed to be resolved fundamentally. This is because the pump flow rate has limited effect in reflecting the flow status of the cooling water within each terminal, while this can be critical in terms of the comfortableness improvement and energy conservation of the VWFAC system. Unfortunately, even with much more volume of demand than pump flowmeters, flowmeters equipped for the terminals of the air-conditioning water system cannot be replaced by various virtual pump flowmeters studied previously due to the significant structural differences between water pumps and terminals. To the authors’ knowledge, few studies have ever tried solving the flowmeter issues at terminals level.

To bridge this research gap, this study proposed a convenient and cost-effective terminal water flow rate estimation method for the VWFAC system. A flow rate estimation model is developed based on the flow resistance characteristics within the terminal, which enables the water flow rate across each end unit accurately estimated by relevant operating parameters. This can also be used as part of the estimation model of supply air volume, so as to further reduce the sensors cost. The rest of this paper is organized as follows. Estimation models are developed for water flow rate and supply air volume in Section 2 to clear the measurement and control demands in this study. Section 3 provides a brief introduction on the Monitoring and Control Laboratory developed for the VWFAC system. Section 4 present the training and verification results of the estimation models. Finally, conclusions are presented in Section 5.

## 2 | FLOW RATE ESTIMATION MODEL DEVELOPMENT

The terminal flow rate estimation models are developed for the water flow rate and supply air volume passing through the air handling unit (AHU) based on the component’s flow resistance property to determine operating parameters that need to be measured and controlled.

### 2.1 | Water flow rate estimation model development

For in-tube flows, the relationship among the flow rate, pressure difference, and impedance is defined by Equation (1).\(^{39}\)

\[
Q = \sqrt{\frac{\Delta P}{S_T}} \tag{1}
\]

The total impedance of an AHU is made up of the impedance of water distribution accessories, copper tubes, and water collecting accessories, as given by Equation (2).\(^{39}\)

\[
S_T = S_d + S_{ct} + S_c \tag{2}
\]

For parallel-connected copper tubes, their total impedance could be written as follows.\(^{39}\)

\[
\frac{1}{\sqrt{S_{ct}}} = \sum_{i=1}^{n} \left(\frac{1}{\sqrt{S_i}}\right) \tag{3}
\]

Assuming that the water in the tubes maintains a laminar flow, the impedance of a single copper tube, water distribution accessories, and water collecting accessories could then be calculated by Equations (4)-(6).\(^{39}\)

\[
S_i = \frac{8 \cdot \rho_w \cdot \xi_i}{\pi^2 \cdot d^4} \left(\sum \xi_i + 0.11 \cdot \frac{l}{d} \cdot \left(\frac{2 \cdot 10^{-4}}{d} + \frac{68}{Re}\right)^{0.25}\right) \tag{4}
\]

\[
S_d = \frac{8 \cdot \rho_w \cdot \sum \xi_d}{\pi^2 \cdot d^4} \tag{5}
\]
\[ S_c = \frac{8 \cdot \rho_w \cdot \sum \xi_c}{\pi^2 \cdot d^3} \]  \hfill (6)

To simplify the calculation process, assume that the flow rate in the parallel loop is evenly distributed, while the water distribution accessories and collecting accessories share equal total local resistance coefficient. In this case, the total impedance of the AHU can then be calculated by Equation (7).\(^{39}\)

\[ S_T = \frac{0.88 \cdot f \cdot \left( \frac{2 \cdot 10^{-4}}{d} + \frac{68}{Re} \right)^{0.25} + \rho_w \cdot \frac{16 \cdot \sum \xi_d + \frac{8}{9}}{\pi^2 \cdot d^4}}{9 \cdot \pi^2 \cdot d^4} \]  \hfill (7)

In this study, clean water was adopted as the working fluid and it was assumed that there was no scaling in the tubes. As indicated by Equation (7), for an AHU with a defined structure, the total impedance is mainly affected by the water temperature, flow rate, and other uncertainties, which could be approximately expressed as a linear polynomial of mean water temperature across the AHU, when the flow rate is fixed, and vice versa, as indicated by Equation (8).

\[ S_T = C_0 \cdot \left( C_1 + C_2 \cdot T_{wm} \right) \cdot \left( C_3 + C_4 \cdot Q \right) \]  \hfill (8)

where the mean water temperature (represented by \( T_{wm} \)) is defined as the mean value of supply water temperature and return water temperature, \( C_0 \) to \( C_4 \) are unknown constants that remains to be estimated.

By substituting for \( S_T \) in Equation (1) that in Equation (8), the water flow rate estimation model with undetermined constants is given as follows.

\[ Q = \sqrt{\frac{\Delta P}{C_0 \cdot \left( C_1 + C_2 \cdot T_{wm} \right) \cdot \left( C_3 + C_4 \cdot Q \right)}} \]  \hfill (9)

Therefore, with the undetermined constants estimated through the experiments, the supply water temperature, return water temperature, and pressure difference across the AHU could be used to estimate the water flow rate of AHUs according to Equation (9).

### 2.2 Supply air volume estimation model development

The supply air volume estimation model against winter conditions is developed based on the heat transfer analysis between the terminals and the environment per unit time.

The heat transfer between the AHU and water is equivalent to the enthalpy difference between the outlet and inlet of water channel, as shown in Equation (10).

\[ \Phi_1 = 3600 \cdot c_w \cdot \rho_w \cdot Q \cdot (T_{ws} - T_{wr}) \]  \hfill (10)

The heat transfer between the AHU and the unconditioned air is equivalent to the enthalpy difference between the outlet and inlet of air passage, as shown in Equation (11).

\[ \Phi_2 = 3600 \cdot c_a \cdot \rho_a \cdot V \cdot (T_{as} - T_{ar}) \]  \hfill (11)

The heat dissipation of the AHU’s built-in motor is decided by fan power and fan efficiency, as shown in Equation (12).

\[ \Phi_3 = P_m \cdot (1 - \xi) \]  \hfill (12)

The radiant heat transfer between the AHU’s air diffuser and indoor environment is defined as follows.

\[ \Phi_4 = \varepsilon \cdot A \cdot \sigma \cdot \left( (T_{as} + 273.15)^4 + (T_{w} + 273.15)^4 \right) - 2 \cdot (T_{wall} + 273.15)^4 \]  \hfill (13)

Under winter conditions, heat is transferred from the built-in motor and hot water to the unconditioned air and the interior surfaces of room enclosures, and the energy balance equation against this process is given as follows.

\[ \Phi_1 + \Phi_4 = \Phi_2 + \Phi_3 \]  \hfill (14)

Substituting \( \Phi_1-\Phi_4 \) from Equations (10)-(13) into Equation (14) yields

\[ V = \frac{c_w \cdot \rho_w \cdot (T_{ws} - T_{wr})}{c_a \cdot \rho_a \cdot (T_{as} - T_{ar})} \cdot Q - \frac{P_m \cdot (1 - \xi)}{3600 \cdot c_w \cdot \rho_w \cdot (T_{as} - T_{ar})} + \frac{3600 \cdot c_w \cdot \rho_w \cdot (T_{as} + 273.15)^4 \cdot (T_{as} + 273.15)^4 - 2 \cdot (T_{wall} + 273.15)^4}{3600 \cdot c_a \cdot \rho_a \cdot (T_{as} - T_{ar})} \]  \hfill (15)

Compared with the enthalpy increment of the unconditioned air and the enthalpy drop of hot water, the heat dissipation of the AHU’s built-in motors and the radiant heat transfer between the AHU and indoor environment are of smaller orders of magnitude and have limited effect on the accuracy of the supply air volume estimations. To reduce the computation load, only the first term of Equation (15) is employed to estimate the supply air volume, and the model can be rewritten as follows.

\[ V = \frac{c_w \cdot \rho_w \cdot (T_{ws} - T_{wr})}{c_a \cdot \rho_a \cdot (T_{as} - T_{ar})} \cdot Q \]  \hfill (16)
In summary, the temperature and volume flow of supply air, supply water temperature, return water temperature, return air temperature, water flow rate, and pressure difference across the AHU are operating parameters that required be monitoring and controlling during the training and validation sessions.

3 | DEVELOPMENT OF MONITORING AND CONTROL LABORATORY

The Monitoring and Control Laboratory was established at the Institute of Building Energy, Dalian University of Technology (Dalian, China, North latitude 39°) to enable the training and validation experiments of the developed terminal flow rate estimation models. The VWFAC system is composed of two variable frequency drive (VFD) water pumps, sub-catchment devices, and three different models of AHUs. Detailed parameters of the above components are listed in Table 1, where AHUs with variable water supply temperature were running in winter in this study.

Figure 2 illustrates important operating parameters of the AHU that are required to be monitored in this study, wherein M1 means the air flowmeter for measuring the supply air volume of the AHU; M2 and M7 denote the sensors for measuring the supply air temperature and return air temperature, respectively; M3 and M5 represent the sensors for measuring outlet water temperature and inlet water temperature; M4 is the differential pressure sensor for measuring the pressure difference across the AHU; M6 means the water flowmeter equipped for the AHU. In addition, to realize the immediate transformation among various operating conditions, the number of water pumps in operation, the operating frequency of the water pumps, as well as the opening of electric control valves need to be controlled.

To meet the aforementioned measurement and control requirements, the corresponding control system was developed for the VWFAC system. A schematic of the sensor’s calibration and application in this study is shown in Table 2, wherein the sensor’s accuracy class is defined as follows:

\[
K = 100 \cdot \frac{e_{\text{max}}}{m_r}
\]

Figure 3 illustrates the framework and communication principle of the control system. After compiling and downloading the corresponding control programs through Honeywell CARE software, the XCL8010A and XL100C controllers that connected to the host computer by a C-Bus can be used to control the actuators and regulate the sensors.

4 | ESTIMATION RESULTS AND DISCUSSION

Training and validation experiments were conducted at the Monitoring and Control Laboratory to estimate the

| Device name | Model | Quantity | Rated parameters | Manufacturer |
|-------------|-------|----------|------------------|--------------|
| Sub-catchment devices | FJW215 | 2 | DN 250; Length: 1.2 m | In-house |
| Water pump | 2HMS3T 380V/3PH/50Hz | 2 | Flow rate: 4.2 m³/h; Head: 20.5 m; Power: 0.37 kW; Cos φ: 0.82; Rev: 2760 rpm | ITT |
| AHU | HFCF03L2 220V/1PH/50Hz/12Pa | 1 | Cooling capacity: 2.8 kW; Heating capacity: 4.93 kW; Flow rate: 0.14 L/s; Tube row number: 2; Air volume: 520/410/270 m³/h | Trane |
| AHU | HFCF03L3 220V/1PH/50Hz/12Pa | 1 | Cooling capacity: 3.16 kW; Heating capacity: 5.2 kW; Flow rate: 0.15 L/s; Tube row number: 3; Air volume: 510/410/270 m³/h | Trane |
| AHU | HFCF02R3 220V/1PH/50Hz/12Pa | 1 | Cooling capacity: 2.21 kW; Heating capacity: 3.5 kW; Flow rate: 0.11 L/s; Tube row number: 3; Air volume: 340/280/180 m³/h | Trane |
undetermined constants and validate the accuracy of the developed estimation models. The mean absolute percentage error (MAPE) and root-mean-square error (RMSE) defined by Equations (18) and (19) were employed to quantify the deviations of the estimated data from the measured data.\(^\text{41}\)

\[
\text{MAPE} = \frac{100\%}{n} \cdot \sum_{i=1}^{n} \left| \frac{X_{\text{model},i} - X_{\text{sample},i}}{X_{\text{sample},i}} \right|
\]

\[
\text{RMSE} = \sqrt{\frac{1}{n} \cdot \sum_{i=1}^{n} (X_{\text{model},i} - X_{\text{sample},i})^2}
\]

4.1 Water flow rate estimation results

The operating conditions during the training and validation sessions are listed in Table 3, and the estimation results of undetermined constants in Equation (9) are as follows. For HFCF03L2 type AHU, \(C_0 = 0.772, C_1 = 58.813, C_2 = -0.266, C_3 = 3.605, \) and \(C_4 = -2.048.\) For HFCF03L3 type AHU, \(C_0 = 0.570, C_1 = 65.403, C_2 = -0.375, C_3 = 3.120, \) and \(C_4 = -1.553.\) For HFCF02R3 type AHU, \(C_0 = 2.362,\)

**TABLE 2** Calibration and application notes of sensors

| Sensor category      | Accuracy class | Range               | Maximum indicated absolute error | Application notes                  |
|----------------------|----------------|---------------------|----------------------------------|-----------------------------------|
| Flowmeter            | 1.0            | 0.2 to 1.2 m\(^3\)/h| 0.01 m\(^3\)/h                  | Water flow rate of the AHU        |
| Differential pressure sensor | 0.5          | 0 to 45 kPa         | 0.225 kPa                        | Pressure difference across the AHU |
| Temperature sensor   | 0.05           | −70 to 450°C        | 0.2°C                            | Supply and return water temperature |
| Temperature sensor   | 0.2            | −40 to 100°C        | 0.3°C                            | Supply and return air temperature  |

**TABLE 3** Operating conditions during training and validation sessions

| Parameter          | Model         | HFCF03L2         | HFCF03L3         | HFCF02R3         |
|--------------------|---------------|------------------|------------------|------------------|
| \(Q_t\) (m\(^3\)/h) | 0.188-0.526   | 0.249-0.587      | 0.210-0.402      |                  |
| \(\Delta P_t\) (kPa) | 4.71-28.81    | 5.70-25.62       | 6.07-20.15       |                  |
| \(T_{w,\text{m,t}}\) (°C) | 14.08-35.65  | 14.57-34.61      | 19.41-36.22      |                  |
| \(Q_v\) (m\(^3\)/h) | 0.311-0.513   | 0.247-0.584      | 0.283-0.488      |                  |
| \(\Delta P_v\) (kPa) | 10.85-24.91   | 5.51-22.65       | 8.53-20.76       |                  |
| \(T_{w,\text{m,v}}\) (°C) | 37.28-40.61  | 38.05-45.25      | 37.28-39.62      |                  |
The trained terminal water flow rate estimation models are as follows:

\[ \Delta P = 0.772 \cdot (58.813 - 0.266 \cdot T_{wm}) \cdot (3.605 - 2.048 \cdot Q) \] (20)

\[ \Delta P = 0.57 \cdot (65.403 - 0.375 \cdot T_{wm}) \cdot (3.12 - 1.553 \cdot Q) \] (21)

\[ \Delta P = 2.362 \cdot (19.794 - 0.129 \cdot T_{wm}) \cdot (4.003 - 2.5 \cdot Q) \] (22)

Figures 4-6 show the experimental results from both training and validation sessions, wherein the water flow rate estimated by models agreed well with that measured by the flowmeters. For the HFCF03L2 type AHU, the MAPE and RMSE during the validation session (ie, 0.4% and 0.002 m³/h) are relatively smaller compared with that during the training session (ie, 1.3% and 0.038 m³/h). While for the HFCF03L3 and HFCF02R3 types of AHUs, opposite trends can be observed. This can be interpreted in terms of the variation ranges of the measured parameters. It is clear that the water temperature and water flow rate need to be varied in a wide range during the training session as much as possible to cover any potential observation during the validation session. By improving this coverage rate, the deviation of the flow rate estimates from the flow rate measurements during the validation session can be effectively reduced, and vice versa.

All in all, the proposed model has preferable precision of water flow rate measurement under the research conditions.

In addition, a systematic error was also introduced to evaluate the impact of the sensor’s precision on both the estimation and measurement results. The systematic error of direct measurements is defined as follows:

\[ \gamma = \frac{K \cdot m_c}{100 \cdot m_c} \cdot 100\% \] (23)

The impedance of the AHUs could be expressed as a function of water flow rate, supply water temperature and return water temperature as follows:

\[ S = C_0 \cdot C_1 \cdot (C_3 + C_4 \cdot Q) \cdot (T_{ws} + T_{wr}) \] (24)
Based on the error propagation theory, by substituting $S$ from Equation (24) in Equation (9), the systematic error of the estimated water flow rate could be calculated by the following equation.

$$\gamma_{Q_{\text{model}}} = \frac{S \cdot \gamma_{\Delta P} - 2 \cdot (\gamma_{T_{\text{ws}}} + \gamma_{T_{\text{wr}}})}{2 \cdot S + 3} \quad (25)$$

As for the measurements of water flow rate, pressure difference, supply water temperature, return water temperature, and the estimates of water flow rate, their systematic errors are shown in Figures 7-9. In terms of the water flow rate of AHUs, the mean systematic errors of the estimates (ie, 0.51%, 0.71%, and 0.76%) are found, respectively, smaller than that of the measurements (ie, 2.18%, 2.18%, and 2.72%). This indicates that the proposed indirect method theoretically have a higher precision of water flow measurement than the direct method under this research condition.

Figures 10-12 illustrate the deviation of the impedance estimates (ie, the impedance calculated by Equation (9), represented by $S_{\text{model}}$) from the impedance measurements (ie, the impedance derived from Equation (1), represented by $S_{\text{sample}}$). In most cases, this deviation is negligible. Compared with the water temperature, the systematic errors of the pressure difference measurements have a much more significant effect on the water flow rate estimations (Figure 9). Therefore, Equation (25) could be simplified as follows.

As is indicated by Equation (23), to ensure the reliability of the water flow rate estimations, the systematic error of the pressure difference measurements should be limited within twice that of the water flow rate estimations.

### 4.2 Supply air volume estimation

The supply air volume of the AHU under various operating conditions can be obtained in four ways as follows:

1. According to Equations (9) and (16), it could be estimated by the supply water temperature, return water temperature, supply air temperature, return air temperature, and pressure difference across the AHU (represented by $V_1$).
2. According to Equation (16), it could be estimated by the supply water temperature, return water temperature, supply air temperature, return air temperature, and water flow rate (represented by $V_2$).
3. It could be measured directly by the air flowmeter (represented by $V_3$).
4. The value given by manufacturers can be referred to (represented by $V_4$).

![Figure 7](image_url)  Systematic errors of HFCF03L2 type AHU

![Figure 8](image_url)  Systematic errors of HFCF03L3 type AHU

![Figure 9](image_url)  Systematic errors of HFCF02R3 type AHU
The HFCF03L3 type AHU was selected to participate in the validation experiments, during which it had to run under 12 different operating conditions. The estimations and measurements of supply air volume and corresponding operating conditions are listed in Tables 4 and 5.

In Table 4, $Q_1$ denotes the water flow rate estimated by the measurement of pressure difference and water temperature, and $Q_2$ is the water flow rate measured by flowmeters. The three capital letters in the “Conditions” column indicate the status of the supply air volume, supply water temperature, and water flow rate in that sequence. For example, “LHL” represents an operating condition with a low supply air volume, high supply water temperature, and low water flow rate.

As is indicated by Table 5, the values of $V_1$, $V_2$, and $V_3$ were very close to $V_4$. Assuming that $V_3$ is the actual supply air volume, the MAPE of $V_1$ and $V_2$ are 2.87% and 2.79%, respectively, while the RMSE of $V_1$ and $V_2$ were 11.22 and 10.44 m³/h, respectively. Thus, the supply air volume of the AHU can be estimated by air temperature, water temperature, and pressure difference with acceptable accuracy.

4.3 | Water flow rate estimation process and interpretation

Figure 13 illustrates the water flow rate estimation process of the AHUs. Firstly, the related factors that could affect the water flow rate are required to be assessed based on the flow resistance property of the AHU. The estimation model can then be derived from the relationship between the water flow rate and these factors. Secondly, the range and accuracy class of the selected sensors must satisfy Equation (23); that is, the systematic error of the pressure difference measured by differential pressure sensors should not larger than twice that of the water flow rate measured by the flowmeters. Subsequently, the undetermined constants in the developed model can be estimated through a series of training experiments. Then, in the validation session, the water flow rate estimation results would be compared with the measurements to verify the accuracy of the model. With the estimation accuracy acceptable for engineering applications, this model eventually could be applied to energy-efficiency renovation projects. Otherwise, the varied ranges of water temperature, water flow rate, and pressure difference during the training session need to be further enlarged to re-estimate the undetermined constants in the model.

5 | CONCLUSIONS

Terminal water flow rate is closely related to the electricity consumption of the VWFAC system, and monitoring such a key operational parameter can be meaningful in terms of comfortableness improvement energy conservation. However, the constraints in installation space and the high replacement and maintenance costs have restrict the flowmeter’s application in the air-conditioning water system. To solve the problems came up with flowmeters, this study has proposed a terminal water flow rate estimation method for
The conclusions of this paper are made as follows.

1. Based on the flow resistance characteristics within the terminals, flow rate estimation models are developed for the water flow rate and supply air volume of AHUs, which indicates that the function of flowmeters could be replaced by differential pressure sensors together with water temperature sensors theoretically.

2. For the water flow rate of AHUs, the systematic error of the estimation results ranged from 0.51% to 0.76%, which is relatively smaller than that of the measurements (ranging from 2.18% to 2.72%). To ensure the reliability of the estimation results in engineering applications, the systematic error of the pressure difference measurements should not over twice that of the water flow rate measurements.

3. RMSE and MAPE are employed to evaluate the deviation of estimations to measurements. In terms of water flow rate, the maximum of RMSE and MAPE are 0.038 m³/h and 6.5%, respectively. As for supply air volume, the RMSE and MAPE do not exceed 11.22 m³/h and 2.9%, respectively. To improve the accuracy of the flow rate estimation model, it is suggested to have the variation ranges of water temperature, water flow rate, and pressure difference during the training session enlarged as much as possible.

4. This method is expected to further benefit the flow rate measurement in other types of terminals in thermal engineering fields.

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NOMENCLATURE

| Symbol | Description                                      |
|--------|--------------------------------------------------|
| A      | Area of AHU’s air diffuser (m²)                 |
| c      | Specific heat capacity (J/(kg °C))              |
| d      | Copper tube diameter (m)                        |
| ε      | Surface emissivity of AHU’s air diffuser        |
| η      | Systematic error                                |
| Φ      | Heat transfer per unit time (J)                 |
| Q      | Water flow rate (m³/h)                          |
| Re     | Reynolds number                                  |
| S      | Impedance (g/cm³)                               |
| Si     | Impedance of single copper tube (g/cm³)         |
| T      | Temperature (°C)                                |
| V      | Supply air volume (m³/h)                        |
| T_wms  | Mean water temperature (°C)                     |
| X      | Water flow rate or supply air volume (m³/h)     |
| ΔP     | Pressure difference across the AHU (kPa)        |
| ρ      | Density (kg/m³)                                 |
| ζ      | Local resistance coefficient                     |
| ζi     | Local resistance coefficient of single copper tube |
| Φ      | Heat transfer per unit time (J)                 |
| γ      | Systematic error                                |
| ε      | Surface emissivity of AHU’s air diffuser        |
| η      | AHU’s operating efficiency                      |
| σ      | Boltzmann’s constant (W/(m² K⁴))               |

Subscripts

| Subscript | Description                                      |
|-----------|--------------------------------------------------|
| a         | Air                                              |
| ar        | Return air                                       |
| as        | Supply air                                       |
| c         | Water collecting accessories                      |
| ct        | Cooper tube                                      |
| d         | Water distribution accessories                    |
| model     | Data estimated by the model                      |
| sample    | Data measured by sensor/meter                    |
| t         | Training experiments                              |
| T         | Total                                            |
| v         | Validation experiments                            |
| w         | Water                                            |
| wall      | Interior surfaces of room enclosures             |
| wr        | Return water                                     |
| ws        | Supply water                                     |

Greek

| Symbol | Description                                      |
|--------|--------------------------------------------------|
| P      | Pressure difference across the AHU (kPa)        |

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