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A Fast and Elaborate Simulation Method for Complex Chilled Water Distribution System#

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ABSTRACT

A fast and elaborate simulation method is always essential to enhance the energy performance of a chilled water system. This study proposed a new zonal approach to divide a chilled water distribution system into different hydraulic zones. And an improved numerical solution is employed to shorten the simulation time through multi-thread parallel computation. A real chilled water system evaluated the performance of the proposed simulation method. Results indicate that the relative errors between the hourly actual data and simulation results on water pressure, flow rate, and temperature on a typical day are 2.99%, 1.89%, and 4.16%, respectively. Also, compared to the conventional solution, the average simulation time of the proposed solution is reduced to only 4.5 %.

Keywords: Chilled water distribution system, Hydraulic behavior, Zonal method, Parallel computation, High-rise building

NONMENCLATURE

Abbreviations	
VT	Virtual terminal node
VPS	Virtual pressure source node
Symbols	
A	Associated matrix
B	Basic circuit matrix
c_p	heat capacity
d	Diameter (m)
f	Friction factor
F_r	Operation frequency of pump
g	Gravity acceleration (m/s^2)
L	Length of the pipe (m)
M	Mass flow rate (kg/s)
P	Pressure (Pa)
Q	Heat exchange (W)
R	Hydraulic resistance (Pa)
T	Temperature (℃)
v	Velocity (m/s)
$\dot{V}$	Volume flow rate (m^3/s)
W	Power consumption (W)
α	Relaxation coefficient
ρ	Density (kg/m^3)
ε	Roughness of pipe surface (m)
ζ	Local resistance coefficient
ϵ	Relative error (%)
Subscripts	
b	Branch of pipe network
CH	Chiller
CT	Cooling Tower
i	Index of pipe inlet node
j	Index of pipe outlet node
o	Outer
p	Pressure
s	Supply water
sur	Surrounding
r	Return water
T	Temperature
v	Valve
V	Volume flow rate
m	Measured data
new	Value of current iteration
old	Value of last iteration
rate	Rated value
s	Simulation result

1. INTRODUCTION

In many developed cities, urban buildings have consumed over 40% of the total power consumption [1]. The Heating, Ventilating, and Air Conditioning (HVAC) system contribute to almost 30% of the building energy consumption in the U.S [2]. The central chilled water system is one of the most popular configurations of the air-conditioning system in commercial buildings, especially in high-rise buildings [3]. Improving the energy efficiency and safety of the chilled water system has attracted considerable attention. Many studies have been conducted on its optimal design, intelligent operation, and performance evaluation [4, 5]. Although various models and simulation tools are available [6, 7], a fully practical hydraulic model for the chilled water system in a high-rise building is still rare in previous studies [8].

There are mainly two approaches to address the HVAC system modeling: the data-driven (black-box) model and the physical-based (white-box) model. As far as the physical-based models are concerned, they have been developed in many studies using the governing laws of physics and the detailed knowledge of the technical process. Wu et al. [9] presented a physics-based linear parametric model to predict room temperature in office buildings by employing the energy balance and heat transfer laws. Wemhoff et al. [10] estimated the chillers' and pumps' power consumption by lumped models considering the chilled water loop and COP of the chiller. Li et al. [11] developed an inertia model of chilled-water systems with three sub-models: chiller model, chilled water pipe model, and cooling coil model. Yuan et al. [12] reviewed the analytical models of ground source heat pumps using line-source theory and cylindrical-source theory in China. However, the physical-based models usually have many simplified assumptions which cannot be easily satisfied in most circumstances. Therefore, the physical-based models should be carefully calibrated before application.

As for the chilled water distribution system of the high-rise buildings, the physical modeling of those central air conditioning systems would be more complicated while considering the severely nonlinear hydraulic behavior of the chilled water circulation across different floors. Ma and Wang [13] presented several simplified models for components of a complex building central chilled water system to formulate an optimal control strategy. Ma et al. [14] developed simplified semi-physical chiller and cooling tower models to predict the system energy performance and environment quality. A pressure drop model for the water network for formulating an optimal pump sequence control strategy [15]. Wang et al. [16] presented adaptive optimal models for online control of complex chilled water systems. Recently, Gao et al. [17] also demonstrated an online robust control strategy to avoid the low delta-T syndrome for the chilled water system in the ultra-high-rise building. Fang et al. [18] proposed an evaluation method to design a chilled water system based on the optimal operation performance of equipment. However, in those previous studies, the models mainly focused on the thermal behavior of the chillers, terminal units, and pumps, neglecting the hydraulic or thermo-hydraulic behavior of the whole chilled water system. A hydraulic model is necessary for the energy performance evaluation of a chilled water system. Also, a fast-speed numerical solution is needed to adapt real-time operation for the complex chilled water system in a high-rise building. The hydraulic modeling approaches in the district heating area [19, 20] and the general thermal models for the complex pipe network [21, 22] can be valuable references to model the chilled water system in a high-rise building.

The main contribution of this paper is to present a fast and elaborate simulation method for the complex chilled water distribution system, which can be a tool to evaluate or enhance the energy performance of the chilled water system in a high-rise building. In this paper, the new zonal method divides a complex chilled water system into different hydraulic zones during modeling. In each zone, the hydraulic model is developed independently, which elaborately considers the water density and water flow regime variations (i.e. laminar, turbulent, and transitional regimes) in the pipes of different floors. Then a holistic physical model is developed by effectively patching those zones together. Moreover, based on the proposed zonal method, an improved numerical solution is also presented by adopting the multi-thread parallel computation technique, which can significantly shorten the computation time.

2. SYSTEM DESCRIPTION AND NEW ZONAL METHOD

The chilled water system generally consists of chillers, terminal units (such as AHU or Coil), pumps, valves, pipes, etc. A typical configuration of the chilled water system in a high-rise building is shown in Fig.1.

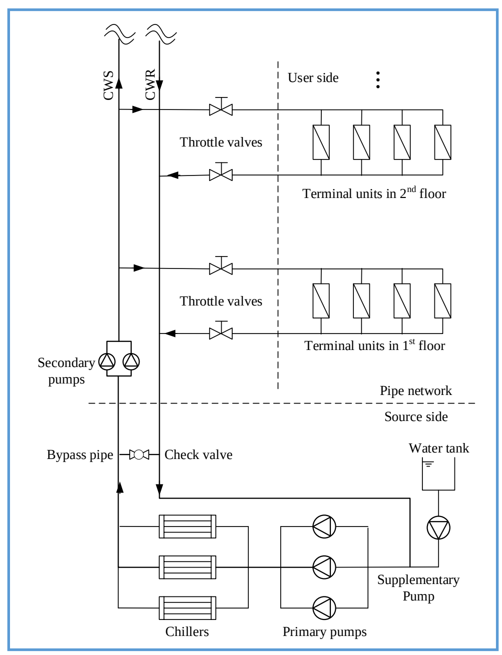

Fig. 1 The configuration of a chilled water system

This study proposed a new zonal method, which divides the holistic chilled water distribution system into several hydraulic zones, as shown in Fig.2. In zone 1, the loop consists of the chillers, primary pumps, and a Virtual Terminal (VT). The so-called VT is a virtual terminal with a known water flow rate determined by the initial value or simulation result from zone 2. Therefore, VT in zone 1 represents an aggregated terminal node from the view of the source side. In zone 2, the loop consists of the secondary pumps, throttle valves, a Virtual Pressure Source (VPS), and many Virtual Terminals. The so-called VPS in zone 2 is a virtual source with known water pressure determined by the initial value or simulation result from zone 1. So, VPS in zone 2 represents an aggregated source node from the view of the pipe network. The multiple VTs in zone 2 are similar aggregated nodes representing the terminal units of individual floors. For instance, VT1 represents all terminal units and pipes on the first floor, whose water flow rate determined by the initial value or simulation result of zone 3. In zone 3, the terminal units are connected with VPS1, a virtual pressure source from the view of the terminals' side on the first floor. So is the second floor, the third floor, and so on. The virtual source nodes (i.e. VPS, VPS1, VPS2, etc.) should be Pressure nodes; the virtual terminal nodes (i.e. VT, VT1, VT2, etc.) should be Flow Rate nodes. Otherwise, it would be hard to reach numerical convergence during simulation.

2.1 Zone 1

As shown in Fig.2.1, in zone 1, the chillers, primary pumps, and virtual terminal VT are linked as a water loop. As the block valves are often fully open in operation, the modeling of this zone is focused on the pumps and the chiller. The hydraulic characteristic of the primary pumps, which usually operate in constant speed, is given by

$$\begin{cases} \Delta \mathrm{P_m} = \rho g \left[\mathrm{k_2} (\dot{\mathrm{V}}_\mathrm{m})^2 + \mathrm{k_1} (\dot{\mathrm{V}}_\mathrm{m}) + \mathrm{k_0} \right] \\ \mathrm{W_m} = \frac{\Delta \mathrm{P_m} \cdot \dot{\mathrm{V}}_\mathrm{m}}{3600 \eta_\mathrm{m}} \end{cases} \quad (1)$$

Where $\Delta \mathrm{P_m}$ is the pressure increment of the pump m; $\dot{\mathrm{V}}_\mathrm{m}$ is the volume flow rate of the pump; $\mathrm{k_0}$, $\mathrm{k_1}$ and $\mathrm{k_2}$ are fitting coefficients which are usually provided by the pump manufactory. $\mathrm{W_m}$ is the total power consumption of the pumps indexed from 1 to M; η_m is the efficiency of the pump m. The inlet pressure value of the primary pump is set by the supplementary pump that gives the static pressure insurance for the water pipe network.

The hydraulic resistance $\mathrm{R_m}$ is related to the chiller m and its circulating flow rate $\dot{\mathrm{V}}_\mathrm{m}$. Take a typical chiller as an example, $\mathrm{R_m}$ would mainly be the hydraulic resistance of the pipes and fittings in the evaporator. When the chiller is in normal operation, the relation between $\mathrm{R_m}$ and $\dot{\mathrm{V}}_\mathrm{m}$ can be obtained by a regression function according to the measured data from onsite meters. In this model, the relation between $\mathrm{R_m}$ and $\dot{\mathrm{V}}_\mathrm{m}$ can be estimated as

$$\mathrm{R_m} = \mathrm{r_0} + \mathrm{r_1} \cdot \dot{\mathrm{V}}_\mathrm{m} + \mathrm{r_2} \cdot (\dot{\mathrm{V}}_\mathrm{m})^2 \quad (2)$$

Where $\mathrm{r_0}$, $\mathrm{r_1}$ and $\mathrm{r_2}$ are fitting coefficients which could be calibrated with a few pairs of $(\mathrm{R_m}, \dot{\mathrm{V}}_\mathrm{m})$.

In compliance with Kirchhoff's current law, the total pressure difference in a water loop would be zero. Therefore, the pressure difference of the VT can be given as,

$$\Delta \mathrm{P_{VT}} = \Delta \mathrm{P_m} - \mathrm{R_m} - \mathrm{R_{pipe}} \quad (3)$$

Where $\mathrm{R_{pipe}}$ is the total hydraulic resistance of the pipes and fittings in zone 1. The hydraulic resistance of the pipes can be calculated using pipe network hydraulic analysis, which would be introduced in the following part. The pressure difference $\Delta \mathrm{P_{VT}}$ would be utilized to determine the pressure of the node VPS in zone 2.

The terminal VT is a virtual flow rate terminal, whose flow rate value is determined by the simulation result of zone 2. It can be given as,

$$\dot{M}_{\mathrm{VT}}^{\mathrm{new}} = \alpha \cdot \dot{M}_{\mathrm{VPS}} + (1 - \alpha) \dot{M}_{\mathrm{VT}}^{\mathrm{old}} \quad (4)$$

Where $\dot{M}_{\mathrm{VPS}}$ is the new value provided by the simulation result of the zone 2; $\dot{M}_{\mathrm{VT}}$ is the flow rate of the VT; Superscript "old" represents the initial value or the last iteration value of the pressure of VT; Superscript "new" represents the current value; α is the relaxation coefficient to accelerate the simulation. The value of α should be between (0, 2.0).

The thermal models of the chiller have been developed in many studies. For instance, according to the sophisticated simulation tool Energy Plus® [23], the energy performance of a chiller can be estimated concerning the technical data sheet of manufactories and characteristics of the chillers. Here the chillers are simplified by aggregated model, the supply water temperature of the chillers can be estimated as,

$$T_{\mathrm{s,CH}} = f(T_{\mathrm{r,CH}}, \dot{V}_{\mathrm{CH}}, T_{\mathrm{CT}}) \quad (5)$$

Where $T_{\mathrm{s,CH}}$ is the supply water temperature of the chiller; $T_{\mathrm{r,CH}}$ is the return water temperature of the chiller; T_{CT} is the outlet water temperature from the cooling tower. $\dot{V}_{\mathrm{CH}}$ is the volume flow rate of the chiller.

2.2 Zone 2

As shown in Fig.2.2, Zone 2 is composed of one Virtual Pressure Source (VPS), secondary pumps, multiple virtual terminals (VT1, VT2, …, VTn), throttle valves, and pipes. The so-called VPS has the same pressure and flow rate as that of the VT of zone 1. All terminals on the first floor are aggregated as a virtual node VT1. The terminals on the second floor are

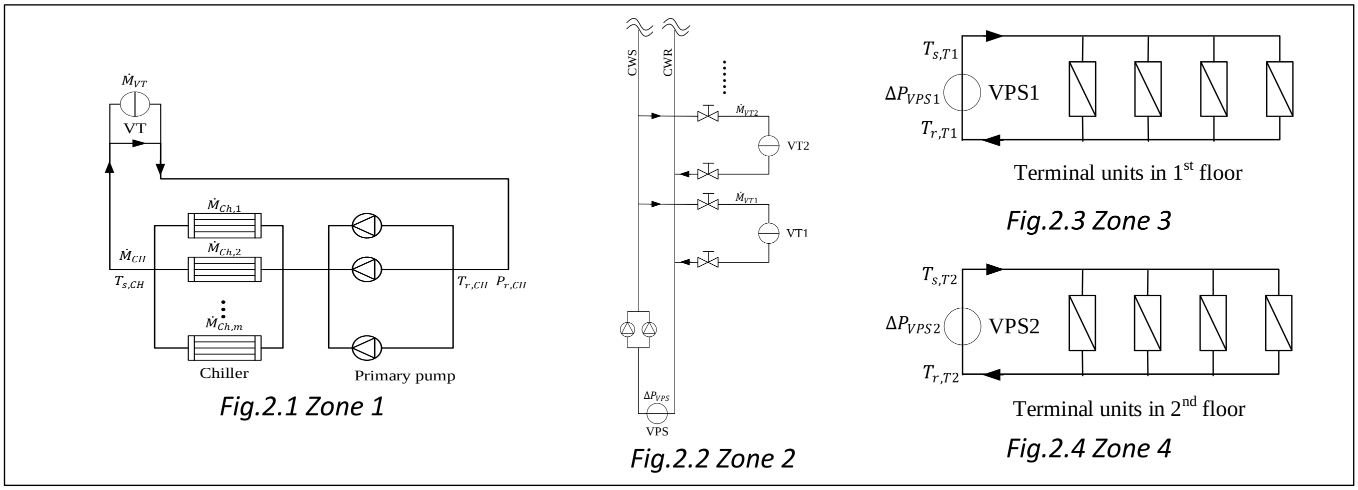

Fig. 2 The hydraulic zones of a chilled water system

aggregated as a virtual node VT1. The terminals on the second floor are aggregated as a virtual node VT2, and so on.

Firstly, the pressure of VPS is determined by the simulation result of zone 1 or the initial value. It can be written as,

$$\Delta P_{\mathrm{VPS}}^{\mathrm{new}} = \alpha \cdot \Delta P_{\mathrm{VT}} + (1-\alpha)\Delta P_{\mathrm{VPS}}^{\mathrm{old}} \quad (6)$$

Where $\Delta P_{\mathrm{VPS}}^{\mathrm{new}}$ is the new value provided by the simulation result of the zone 1; The superscript "new" represents the current value; The superscript "old" represents the initial value or the last iteration value of the pressure of VPS; α is the relaxation coefficient to accelerate the simulation. ΔP_{VT} is determined by equation (3).

The supply water temperature is also given by the simulation result of the zone 1.

$$T_{\mathrm{s,VPS}} = T_{\mathrm{s,CH}} \quad (7)$$

The terminals VT1, VT2, …, VTn are virtual flow rate terminals, whose flow rate values are determined by the simulation results of the zone 3, zone 4, …, zone n, respectively. For instance, the flow rate of the VT1 is the simulation value of the zone 3. It can be given as,

$$\dot{M}_{\mathrm{VT1}}^{\mathrm{new}} = \alpha \cdot \dot{M}_{\mathrm{VPS1}} + (1-\alpha)\dot{M}_{\mathrm{VT1}}^{\mathrm{old}} \quad (8)$$

Where $\dot{M}_{\mathrm{VT1}}^{\mathrm{new}}$ is the new value provided by the simulation result of the zone 3; The superscript "old" represents the initial value or the last iteration value of the pressure of VT1; The superscript "new" represents the current value; $\dot{M}_{\mathrm{VPS1}}$ is the flow rate calculated from zone 3; α is still the relaxation coefficient.

The return water temperature of the VT1 is determined by the simulation result of zone 3. Similarly, the return water temperatures of terminals VT1, VT2, …, VTn can be written as,

$$T_{\mathrm{r,VT1}} = T_{\mathrm{r,T1}} \quad (9)$$
$$\cdots$$
$$T_{\mathrm{r,VTn}} = T_{\mathrm{r,Tn}}$$

Where $T_{\mathrm{r,T1}}$, …, $T_{\mathrm{r,Tn}}$ are the return water temperature provided by the simulation result of the zone 3, … ,zone n, respectively;

The hydraulic characteristic of the secondary pumps, which are paralleled to operation in the same rotation speed, is given by

$$\begin{cases} \Delta P_n = \rho g \left[k_2 \left(\dot{V}_n\right)^2 + k_1 \left(\dot{V}_n\right)\left(\frac{Fr}{Fr^{rate}}\right) + k_0 \left(\frac{Fr}{Fr^{rate}}\right)^2 \right] \\ W_n = \frac{\Delta P_n \cdot \dot{V}_n}{3600 \eta_n} \end{cases}$$

(10)

Where ΔP_n is the pressure increment of the secondary pump *n*; $\dot{V}_n$ is the volume flow rate of the pump; k_0, k_1 and k_2 are fitting coefficients which are provided by the pump manufactory. Fr is the operation frequency of the pump. Fr^{rate} is the rated frequency of the pump. W_n is the total power consumption of the pump *n*; η_n is the efficiency of the pump *n*.

In a high-rise building, two aspects need more consideration than the pipe network on flat ground. (1) Due to most pipe diameters of chilled water distribution systems in a building generally being less than DN 50, the water flow regime variations (i.e. laminar, turbulent, or transitional regimes) ought to be considered during hydraulic modeling. (2) Due to the water temperature difference between the supply and return vertical pipes across the floors, the water density fluctuation cannot be neglected in a water loop.

Considering the flow regime variations in the pipes, the friction factor f should be different along with the flow regime, which can be given as [23],

$$\begin{cases} f = \frac{64}{Re} & Re \le 2320, laminar \\ \frac{1}{\sqrt{f}} = -2\log_{10} \frac{2.825}{Re\sqrt{f}} & 2320 < Re \le 3100, transition \\ \frac{1}{\sqrt{f}} = -2\log_{10}\left(\frac{2.51}{Re\sqrt{f}} + \frac{\varepsilon}{3.7 d_n}\right) & Re > 3100, fully\ turbulent \end{cases}$$

(11)

Where ε is the absolute roughness of the inner pipe surface; d_n is the inner diameter of the pipe.

According to the Darcy & Weisbach equation [24], the hydraulic resistance of a straight pipe R_{ij} can be given as,

$$R_{ij} = f\rho \frac{l_{ij}}{d_n}\frac{\left(v_{ij}\right)^2}{2} \qquad (12)$$

Where v_{ij} is water flow velocity in the pipe; l_{ij} is the length of the pipe. The subscript i and j are indexes of the pipe inlet node and outlet node, respectively. ρ is the density of the water, which mainly depends on the temperature in the chilled water distribution system, which can be written as,

$$\rho = \rho\left(T_{ij}\right) \qquad (13)$$

Where T_{ij} is the average water temperature in the pipe from node i to j;

With the water density changes, the water temperature variations in different pipes should be calculated according to thermal dynamic laws. The chilled water temperature would increase due to the thermal losses between the pipe and surroundings. As there are always thick insulation layers outside the pipe wall, the temperature increment is regularly quite small. The outlet water temperature can be estimated in a simplified equation as [25],

$$T_j = T_i - (T_i - T_{\mathrm{sur}})\left(1 - e^{-[(\pi d_o k_{ij} l_{ij})/(c\dot{M}_{ij})]}\right) \quad (14)$$

Where $\dot{M}_{ij}$ is the mass flow rate of the pipe; d_o is the outer diameter of the pipe; k_{ij} is the heat transmission coefficient for the pipe in terms of the outer pipe surface; T_{sur} is the surrounding temperature; c is the heat capacity of water.

The resistance of a throttle valve can be simplified as a local hydraulic resistor,

$$R_{\mathrm{v}} = k_{\mathrm{v}} \frac{(\dot{V}_{\mathrm{v}}/A_{\mathrm{v}})^2}{2} \qquad (15)$$

Where R_{v} is the resistance of a throttle valve; k_{v} is characteristic coefficient of the valve; $\dot{V}_{\mathrm{v}}$ is the volume flow rate through the valve; A_{v} is the opening area of the valve.

2.3 Zone 3 and Zone 4

As shown in Fig.2.3, Zone 3 consists of one Virtual Pressure Source 1 (VPS1), cooling terminals on the first floor, and pipes. VPS1 has the same pressure and flow rate as VT1 of zone 2. The flow rate of VT1 is determined by that of VPS1. The pressure value of VPS1 is determined by that of VT1, which can be given by,

$$\Delta P_{\mathrm{VPS1}}^{\mathrm{new}} = \alpha \cdot \Delta P_{\mathrm{VT1}} + (1-\alpha)\Delta P_{\mathrm{VPS1}}^{\mathrm{old}} \quad (16)$$

Where $\Delta P_{\mathrm{VPS1}}^{\mathrm{new}}$ is the new value provided by the simulation result of the zone 2; The superscript "new" represents the current value of the pressure of VPS1; The superscript "old" represents the initial value or the last iteration value.

The hydraulic resistances of the cooling terminals are related to their specific types, such as coils, Air handling unit, etc. Nevertheless, the hydraulic resistance of a terminal can usually be aggregated as a local resistor. The hydraulic resistance of a terminal R_i can be written as,

$$R_i = \zeta_i \cdot \left(\dot{V}_i\right)^2 \qquad (17)$$

Where ζ_i is the local resistance coefficient which could be calibrated with a few pairs of $\left(R_i, \dot{V}_i\right)$.

The return water temperature of the terminal i is determined by its thermal characteristic, which can be written as,

$$T_{\mathrm{r},i} = T_{\mathrm{s},i} + \frac{Q_i}{\rho_i \dot{V}_i C_p} \qquad (18)$$

Where $T_{\mathrm{r},i}$ is the return water temperature of the terminal i; $T_{\mathrm{s},i}$ is the supply water temperature of the terminal i. Q_i is the heat exchange between water stream and outside air of the terminal i. C_p is the heat capacity of water.

As shown in Fig.2.4, Zone 4 consists of one Virtual Pressure Source 2 (VPS2), cooling terminals on the second floor, and pipes. Similar to Zone 3, the pressure of VPS2 is determined by that of VT2, which can be given by,

$$\Delta P_{\mathrm{VPS2}}^{\mathrm{new}} = \alpha \cdot \Delta P_{\mathrm{VT2}} + (1-\alpha)\Delta P_{\mathrm{VPS2}}^{\mathrm{old}} \quad (19)$$

The hydraulic and thermal behavior of the cooling terminals in Zone 4 is also developed as that in Equations (17) and (18). There are usually many floors in a high-rise building, then there are as many zones as its floors. The modeling of each zone can be performed as that of Zone 3.

2.4 Pipe network

The hydraulic analysis of a pipe network is usually based on Graph Theory. The pipes are the branches of a network. The topology structure of the pipe network in each Zone can be summarized as the associated matrix A and basic circuit matrix B. Considering the topology of the pipe network, k is its total branch number, and (n+1) is its total node number. A is an n × k order matrix, and B is a (k-n) × k order matrix. Analogous to that of an electrical circuit, the water flow rate and pressure drop along the pipe in the pipe network complies with Kirchhoff's current and voltage law as well. Then the water flow rate at a node can be written as,

$$\boldsymbol{A} \cdot \dot{\boldsymbol{M}}_{\mathrm{b}} = 0 \quad (20)$$

Where $\dot{\boldsymbol{M}}_{\mathrm{b}}$ is the mass flow rate column vector for each branch in the pipe network, $\left[\dot{M}_{\mathrm{b,1}}, \dot{M}_{\mathrm{b,2}}, \cdots, \dot{M}_{\mathrm{b,k}}\right]$.

The water pressure drop in a basic loop can be written as,

$$\boldsymbol{B} \cdot \left(\boldsymbol{R}_{\mathrm{b}} - \rho g \boldsymbol{H}_{\mathrm{p}} + \rho g \boldsymbol{Z}\right) = 0 \quad (21)$$

Where $\boldsymbol{R}_{\mathrm{b}}$ is the resistance column vector for each branch in the pipe network, $\left[R_{\mathrm{b},1}, R_{\mathrm{b},2}, \cdots, R_{\mathrm{b},k}\right]^{T}$. The $\boldsymbol{H}_{\mathrm{p}}$ is the pump head column vector for each branch in the pipe network, $\left[H_{\mathrm{p},1}, H_{\mathrm{p},2}, \cdots, H_{\mathrm{p},k}\right]^{T}$. The term in $\boldsymbol{H}_{\mathrm{p}}$ is zero unless there is a pump installed in the corresponding branch. $\boldsymbol{Z}$ is the height difference column vector for each branch.

According to the node method for a basic loop, the Eq. (21) can also be rearranged as,

$$\boldsymbol{A}^{T} \cdot \boldsymbol{P}_{\mathrm{b}} = \boldsymbol{R}_{\mathrm{b}} - \rho g \boldsymbol{H}_{\mathrm{p}} + \rho g \boldsymbol{Z} \quad (22)$$

Where $\boldsymbol{P}_{\mathrm{b}}$ is the node pressure. The resistance of a branch is the total resistances of the pipe, valve or virtual pressure source (VPS) on its path, which can be arranged as,

$$\boldsymbol{R}_{\mathbf{b}} = \boldsymbol{S}\left|\dot{\boldsymbol{M}}_{\mathrm{b}}\right|\dot{\boldsymbol{M}}_{\mathrm{b}} \quad (23)$$

Then, from Eq. (20), (22) and (23), the following equation to obtain the node pressure $\boldsymbol{P}_{\mathrm{b}}$ can be given as,

$$\left(\boldsymbol{A}\boldsymbol{C}^{-\mathbf{1}}\boldsymbol{A}^{\boldsymbol{T}}\right)\boldsymbol{P}_{\mathrm{b}} = \boldsymbol{A}\boldsymbol{C}^{-\mathbf{1}}\rho g\left(\boldsymbol{Z} - \boldsymbol{H}_{\mathrm{p}}\right) \quad (24)$$

Where C is the linearized column vector, $\boldsymbol{C} = \boldsymbol{S}\left|\dot{\boldsymbol{M}}_{\mathrm{b}}\right|$.

3. NUMERICAL SOLUTION

An effective numerical solution to matrix equations such as Eq. (24) can use the Newton-Raphson method. For its sophisticated mechanism [26], the Newton-Raphson method is not elaborated here. Traditionally, the matrix A represents the whole pipe network, then Eq. (24) is usually too complicated to be solved quickly. However, by the proposed zonal method, matrix A has been divided into a few smaller matrices that represent different small zones. As a relaxation coefficient α is already used in Eq. (4), (6), (8) …, the numerical computation for each zone can be processed synchronously with the multi-thread parallel computation technique.

As shown in Fig.3, the numerical solution of the water pipe network model performs in multiple threads synchronously. The computation in each zone is packaged into one individual thread. After each iteration, the new value of the virtual node between two zones gets exchanged. In Fig.3, those dotted lines show the data exchange among different zones. All computations are completed when each zone has reached its numerical convergence on the virtual nodes, such as VPS, VPS1, VPS2, …, etc.

4. CASE STUDY

The chilled water distribution system in a high-rise building (a hotel in Shanghai, China) with 22 floors is employed to validate the proposed simulation method. There are four identical chillers and primary pumps on the underground floor of the building. The rated cooling capacity of the chiller is 865.6 kW and the rated power consumption is 245.2 kW. The designed supply/return water temperature is 7/12 ℃. The primary pump’s rated volume flow rate is 164.0 m3/h and the rated water head is 15.3 H2O m. There are eight identical secondary pumps with rated volume flow rates of 82.0 m3/h and a rated water head of 16.6 H2O m. Usually, there are three primary pumps and six secondary pumps in operation. The pressure at the outlet of the supplementary pump is set to 0.9 MPa. There are also 48 water pressure meters, 22 flow rate meters, and 48 temperature sensors installed at the inlets and outlets of chillers and valves. The resolutions of water pressure meters, flow rate meters, and temperature sensors are 0.1 kPa, 0.1 m3/h, and 0.1, respectively.

In this case, there are 46 valves, 202 terminal units, and 906 pipes in the simulation model. Considering that the chilled water pipe network contains 22 floors, it is divided into 24 zones by the proposed zonal method in simulation. Therefore, there are VPS and VT between Zone 1 and Zone 2. The VPS1 and VT1 are between Zone 2 and Zone 3. The VPS2 and VT2 are between Zone 2 and Zone 4, and so on. When the virtual pressure loss of each floor (VPS 1 to VPS 24) was approximately equal to the corresponding pressure loss of each virtual terminal (VT 1 to VT 24), the simulation process was converged. The metering data of inlet pressure of the primary pumps (P1) and outlet temperature of the Chillers (T3) are regarded as “known value” during simulation.

The simulation results of the chilled water distribution system are validated by the hourly measured data on a typical day of 8th August 2018. Then there are 24 (from 0 Am to 11 Pm) scenarios for comparison between simulation and measured results on water pressure, flow rate, and temperature in the pipe network, respectively. For simplicity, only the simulation results at 12 o’clock are shown here in Fig.4.1 (water pressure), Fig.4.2 (water flow rate), and Fig.4.3 (water temperature), respectively. The simulation program performs on the Microsoft platform Visual Studio 2017.

As shown in Figure 4.1, the taller the pipe locate, the lower the water pressure. The pipe with the highest water pressure positions the outlet of the secondary pump. As shown in Fig.4.2, the water flow of the near-terminal units on each floor is higher than that of the far-terminal units. The water streams in some pipes with much lower flow rates are in a laminar regime. The water streams in most pipes are in a turbulent regime. As shown in Fig.4.3, water temperature increases about 4-6 ℃ while passing through the terminal units. The water density would change a little with its temperature.

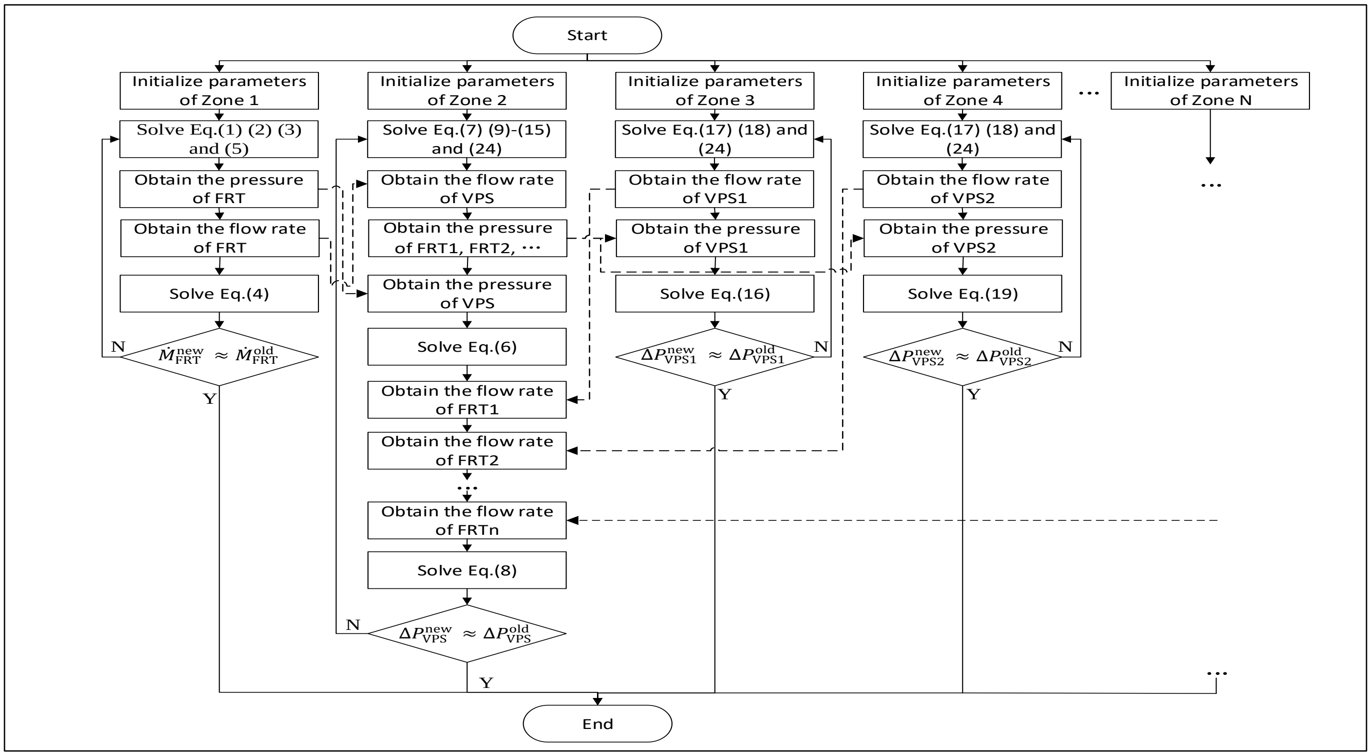

Fig. 3 Multi-thread parallel computation by the proposed zonal method

Therefore, the chilled water temperature variation in those vertical pipes should be considered during hydraulic simulation.

The comparison between the measured data and the imitation results at 12 o'clock is shown in Fig.5.1 (pressure), Fig.5.2 (flow rate), and Fig.5.3 (temperature), respectively. The simulated results with or without considering chilled water temperature variation are also compared in Fig.5.1. The measured data in P1 is recognized as a "known" value for simulation. The maximum pressure error between the measured data and the simulated results with or without considering temperature variation is 4587 Pa and 5364 Pa at the meter P48, respectively. It indicates that the simulated results considering temperature variation are in better agreement with the measured data. In Fig.5.2, the flow rate of V1 is the sum of the flow rates from V3 to V24. As there is no water flow in the bypass pipe, the flow rate of V1 is equal to that of V2. The maximum error between the measured data and simulation results from V1 to V24 is 9.97 m3/h at V1. The average absolute error between measured data and simulation results is only 1.30 m3/h. In Fig.5.3, the measured data of T3 is used as a "known" value for simulation. The maximum error between the measured data and simulation results from T1 to T42 is 0.51 at T44. The average absolute error on all temperature values is 0.32 ℃. Therefore, the simulation results are quite conformed with the measured data at 12 o'clock.

As regards the 24 scenarios (from 0 Am to 11 Pm), the maximum relative error between the simulated results and the measured data is defined as,

$$\epsilon_{\mathrm{p}} = max\left\{\frac{|p_i^{\mathrm{m}} - p_i^{\mathrm{s}}|}{p_i^{\mathrm{m}}}\right\}, i = 2, \cdots 48 \qquad (25)$$

$$\epsilon_{\mathrm{V}} = max\left\{\frac{|\mathrm{V}_i^{\mathrm{m}} - \mathrm{V}_i^{\mathrm{s}}|}{\mathrm{V}_i^{\mathrm{m}}}\right\}, i = 1, \cdots 24 \qquad (26)$$

$$\epsilon_{\mathrm{T}} = max\left\{\frac{|\mathrm{T}_i^{\mathrm{m}} - \mathrm{T}_i^{\mathrm{s}}|}{\mathrm{T}_i^{\mathrm{m}}}\right\}, i = 1,2,4 \cdots 48 \quad (27)$$

Where p_i^{m} is the measured pressure at meter Pi; p_i^s is the simulation pressure at meter Pi; $\mathrm{V}_i^{\mathrm{m}}$ is the measured flow rate at meter Vi; V_i^s is the simulation flow rate at meter Vi; $\mathrm{T}_i^{\mathrm{m}}$ is the measured temperature at meter Ti; T_i^s is the simulation temperature at meter Ti.

As shown in Fig.6, the values of ε_{p}, ε_{V}, and ε_{T} in all 24 scenarios are lower than 2.99%, 1.89%, and 4.16%, respectively. It indicates that the simulation results by the proposed model and numerical solution are in good agreement with that of the measured data of the chilled water pipe network.

Also, the proposed numerical solution can significantly shorten the computation time by using the multi-thread parallel computation technique. In this case, the computation for the 24 zones can be parallelized into 24 threads by the proposed method. However, in the conventional approach, the model of the whole pipe network would be solved using the Newton-Raphson algorithm. The comparisons of the computation time of the 24 scenarios between the simulation using

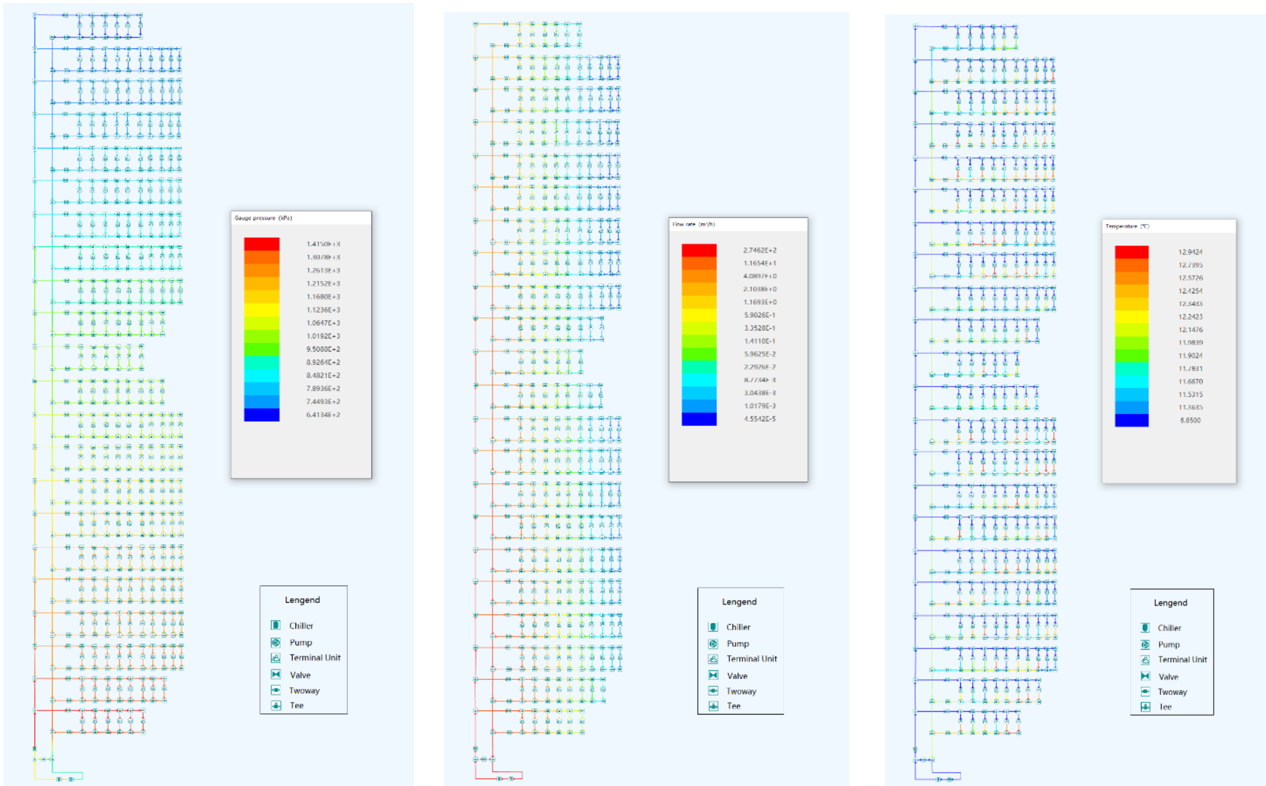

Fig.4.1 Water pressure *Fig.4.2 Water flow rate* *Fig.4.3 Water temperature*

Fig. 4 Simulation results of chilled water pressure, flow rate and temperature in pipe network at 12 o'clock

the conventional method and that of the proposed zonal method are shown in Fig.7.

The average computation time of the proposed method is 22.35 s, while that of the conventional approach is 501.91 s. The average computation time of the proposed solution in the case study is reduced to only 4.5 %. In other words, the average speedup ratio reaches as high as 22.5. It indicates that the computation time of the proposed multi-thread parallel solution has an obvious advantage over that of the conventional non-parallel computation solution.

5. CONCLUSIONS

Due to the complex vertical structure of the pipe network and the numerous types of components involved, it is rather challenging to elaborately simulate the chilled water distribution system in a high-rise building. This paper develops a fast and elaborate simulation method for the complex chilled water distribution system and validates it in a real-life case study. The main conclusions are summarized as follows.

(1) This study proposes a new zonal method to divide a central chilled water distribution system into different hydraulic zones during the modeling process. Then the hydraulic and thermal model of each zone is developed independently, which elaborately considers the water density and water flow regime variations (i.e. laminar, fully turbulent, and transition regimes) in the pipes. A holistic model is also developed by patching those models together.

(2) Based on the Newton-Raphson algorithm, this study puts forward an effective numerical solution using the multi-thread parallel computation technique. With the proposed numerical solution, the computation process for each zone performs synchronously, significantly shortening the simulation time. The pressure error or flow rate error of the virtual nodes can be used to determine whether the numerical convergence is achieved.

(3) The proposed model and solution are validated in a real-life case study. Based on the hourly measured data from 0:00 am to 23:00 on a typical day in summer, the hydraulic and thermal behaviors of the chilled water distribution system in a high-rise building are simulated by the proposed simulation method. The maximum relative errors on water pressure, flow rate, and temperature in all 24 scenarios are lower than 2.99%, 1.89%, and 4.16%, respectively.

(4) Compared to the conventional non-parallel computation solution, the average simulation time of the proposed solution in the case study can be reduced to only 4.5%. The average speedup ratio has been reached

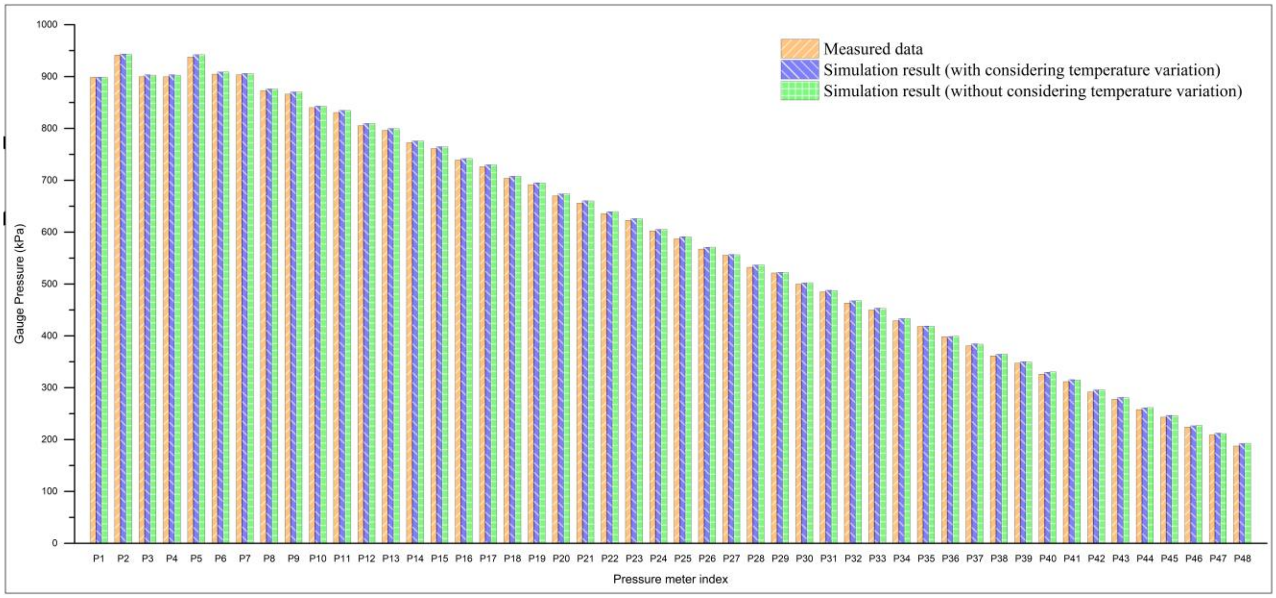

Fig.5.1 Pressure value of the measured data and simulation results

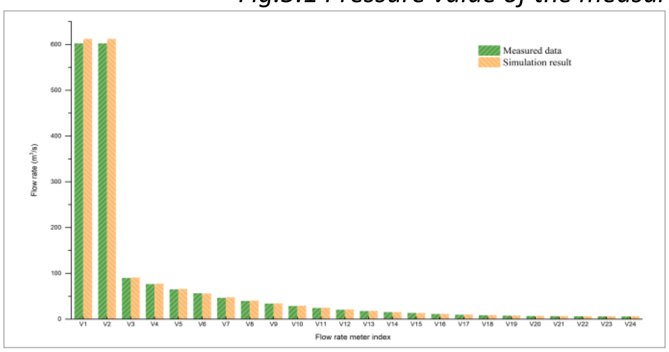

Fig.5.2 Flow rate value of the measured data and simulation results

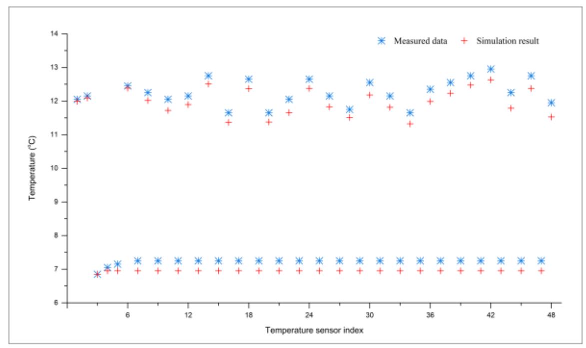

Fig.5.3 Temperature value of the measured data and simulation results

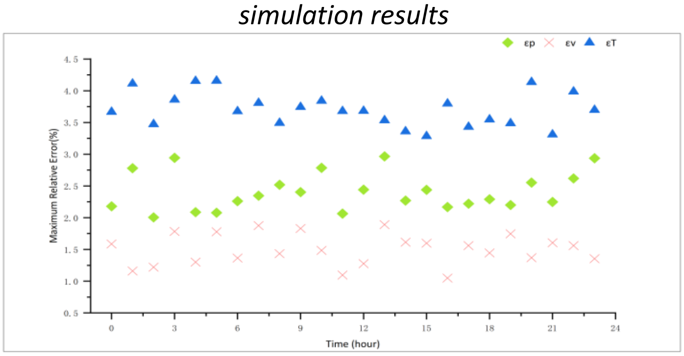

Fig.6 Maximum relative error of the 24 scenarios

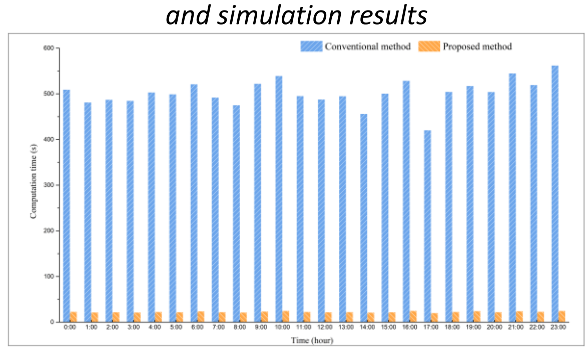

Fig.7 The computation time of the 24 scenarios using the conventional and proposed method

as high as 22.5. Results indicate that the proposed method is a fast solution to simulate the complex chilled water system.

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