DEALING WITH "BIG CIRCULATION FLOW RATE, SMALL TEMPERATURE DIFFERENCE" BASED ON VERIFIED DYNAMIC MODEL SIMULATIONS OF A HOT WATER DISTRICT HEATING SYSTEM

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ABSTRACT

Dynamic models of an indirect hot water district heating system were developed based on the first principle of thermodynamics. The ideal model was verified by using measured operational data. The ideal and verified models were applied to obtain and analyze the system characteristics such as the tendency of the issue relating to "big circulation flow rate, small temperature difference". From the simulated and analyzed results, it is realized that the electrical pumping cost could be significantly reduced due to the over circulation water mass flow rate. It is also shown that by applying proper supply water temperature set point in the control system, the zone air temperature can be automatically adjusted with enough accuracy. The major reasons and solutions for the water mass flow rate and the temperature difference problem have been given in the last section.

KEYWORDS

District heating system; dynamic models; simulations; pumping cost; reasons and suggestions;

NOMENCLATURE

c	specific heat, J/Kg℃ or a factor related to heat transfer coefficient test of heaters
C	controller or thermal capacity, J/℃
e	error
E	electricity consumption, W
f	factor
F	area, m^2
G	mass flow rate, Kg/s
HV	heating value, J/Kg
k_p	proportional gain
k_i	integral gain
KF	heat transfer coefficient, W/℃
q	heat per unit area, W/m^2
Q	heat, W
t	time, s
T	temperature, ℃
TD	temperature difference, ℃
u	control signal
$\alpha_0 - \alpha_3$	factors

Subscripts

1, 2	primary, secondary system
act	actual
arg	average
b	boiler
d	design
en	enclosure
ex	exchanger
f	fuel
h	heater
int	internal
n	number of HES
o	outside
r	return
s	supply
sp	set point
sols	solar radiation from south side
v	verify
w, w2i HES	water, water in secondary system for each HES
z	zone

1 INTRODUCTION

Due to the most share of energy consumption for district heating (DH) in construction field, the energy efficiency, energy savings and environmental protection play very important rules these days. Therefore, focusing on the current situations combining with future solutions has been attracting researchers to develop applications. For example in the existing issues in DH systems, the operation mode: “big circulation flow rate, small temperature difference” could be found in many DH systems, this phenomenon is mainly resulted from both design and operation phases. In design stage, designers wish to ensure more safety due to the responsibility; they usually try to increase safety factors from the heating load calculation, terminal selection, pipe dimension, pump selection and the equipment for the heat source as well. This in fact will cause over sizing of the overall system configuration. On the other hand, in the system operation stage, some operators and managers also have the same thinking by increasing the operational parameters to overcome the problems resulted from uncomfortable zone air temperature and hopeless system control and adjustment. Some researchers and experts have found the scenarios in system operation, and described the reasons caused it with words. In order to study the situations including the reasons, the results and the deep solutions, a dynamic model of a typical and real indirect district heating (IDH) system was chosen for this research to assist the analyses and advices for reducing the pumping cost from with quantificationally and qualitatively.

Based on this motivation, an actual IDH system with heated floor area 0.44Mm^2 was selected as an objective. The system has been renovated in 2013 by using Danfoss high-technologies and products and operated in 2013-2014 heating period.

In the IDH system, there are 4 high-temperature hot water boilers installed in the heat source, and 5 heat exchange stations (HES) in the overall system with 9 heat exchange units in total. The terminals were installed by radiator. The served buildings were built in 1992-1994 as economic buildings. The simplified system diagram is plotted in Figure 1 with aggregated boilers, heat exchange stations, heaters as three parts with control strategy configuration.

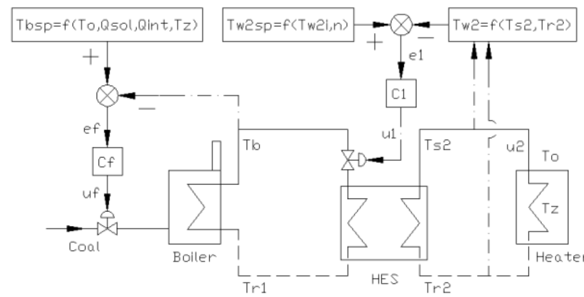

Figure 1. Simplified Diagram With Control Configuration

2 DYNAMIC MODEL DEVELOPMENT

2.1 Ideal Dynamic Model

The design parameters of indoor and outdoor air temperatures, supply and return water temperatures in the primary and secondary systems, water mass flow rates in the primary and secondary systems, and heating load index are given as 18℃, -26℃, 110℃, 70℃, 80℃, 60℃, 475T/h, 951T/h, 50W/m^2 respectively. These parameters are used for the development of the mathematical model.

As known that IDH systems are very complicated systems when considering dynamic processes. To avoid the complexity, certain conditions are assumed: the heat losses and the makeup water losses from the pipe network are ignored without affecting the system responses too much due to limited amounts actually. In addition, several physical parameters are aggregated such as thermal capacity, overall heat transfer rate of HES, radiator and building enclosure. By applying for the first principle of thermodynamics, 5 state variables are taken into account for the simplified ideal dynamic model, and the dynamic equations are expressed as follows.

Boiler model:

$$C_b \frac{d(T_b)}{dt} = u_f G_{fd} HV[\alpha_0 + \alpha_1(\frac{T_b}{T_{bd}}) + \alpha_2(\frac{T_b}{T_{bd}})^2 + \alpha_3(\frac{T_b}{T_{bd}})] - c_w u_1 G_{1d}(T_b - T_{r1}) \quad (1)$$

In Equation (1), the net heat stored in the boiler body is equal to the difference of fuel combustion and the heat transported from the primary network in the heat source. Note that the boiler efficiency is stated as nonlinear property based on experience.

Heat exchanger model:

$$C_{ex1}\frac{d(T_{r1})}{dt}=c_w u_1 G_{1d}(T_b-T_{r1})-f_{ex}KF_{ex}\{[(T_b-T_{s2})-(T_{r1}-T_{r2})][\ln(\frac{T_b-T_{s2}}{T_{r1}-T_{r2}})]^{-1}\} \quad (2)$$

$$C_{ex2}\frac{d(T_{s2})}{dt}=c_w u_1 G_{1d}(T_b-T_{r1})-c_w u_2 G_{2d}(T_{s2}-T_{r2}) \quad (3)$$

In Equations (2) and (3), the net heat stored to the heat exchangers in both sides are considered as the heat transferred from the primary system, the heat exchanger itself and the secondary system. The heat exchangers are selected as plate type, and the temperature difference between the primary and secondary sides is computed by using logarithmic mean temperature difference method.

Radiator model:

$$C_h\frac{d(T_{r2})}{dt}=c_w u_2 G_{2d}(T_{s2}-T_{r2})-f_h KF_h(\frac{T_{s2}+T_{r2}}{2}-T_z)^{(1+c)} \quad (4)$$

Due to the non-linear characteristic of radiators, the heat transfer from the terminals is calculated by using exponential model. In Equation (4), the net heat stored in the heaters is the difference between the heat input from the secondary system and the heat output from the terminals.

Room model:

$$C_z\frac{d(T_z)}{dt}=c_w u_2 G_{2d}(T_{s2}-T_{r2})+q_{sols}F_{sols}+q_{\text{int}}F-f_{en}KF_{en}(T_z-T_o) \quad (5)$$

In Equations (5), the net heat stored in the zone air can be expressed by the differences between the heat obtained from heat inputs (the secondary system, solar radiation and the internal heat gains) and the heat outputs from the building enclosure. For simplicity, the solar radiation is considered by south side of the buildings only, and the internal gains are assumed by 3.6W/m^2 as maximum value.

In summary, the simplified ideal IDH system dynamic model consists of 5 dynamic equations.

2.2 Properties Analysis From The Ideal Model Simulations

Design values of the network are set to the ideal dynamic model for simulations. With the designed fuel consumption and outside air temperature as inputs, the dynamic responses are displayed in Figure (2). From this figure it can be seen that the outputs present identical as the design values, meaning correct relationships among the inputs and the outputs.

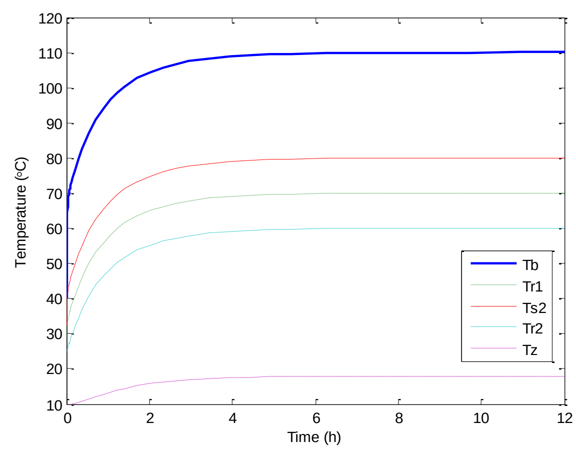

Figure 2. Dynamic Responses In Ideal Conditions

The system properties could be obtained by simulating with the ideal dynamic model, which the method is entitled as open loop tests. When the solar radiation and the internal heat gains are set to ~~be~~ zero, and the zone air temperature maintains 18℃, the simulation results are addressed in Figure 3 associated with the changes of water mass flow rates in the primary and secondary systems and different outside air temperature. Only one parameter changes while the other was considered as ideal condition in the simulation. As shown in Figure 3(a), it can be observed that, when the water mass flow rate in the primary system varies from lower to over design value, the temperature difference (TD) in the primary side will change from higher to lower values, and the TDs in the secondary system are almost identical; by observing the situation in the secondary side plotted in Figure 3(b), it has the same tendency compared with the Figure 3(a). In addition, the varying rate in the lower outside air temperature is faster than that in the higher outside air temperature.

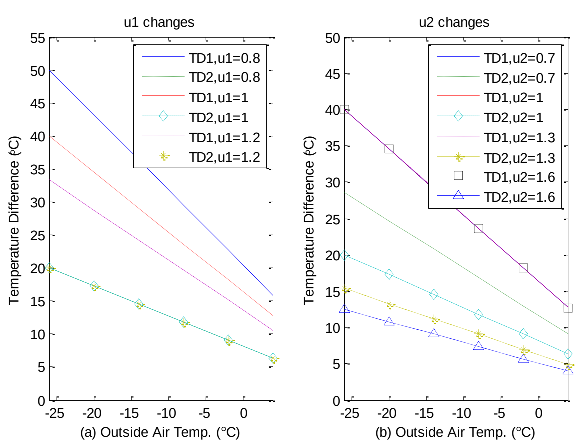

Figure 3. System Properties With The Changes Of Different Parameters (TD and u refer to the temperature difference and water mass flow rate control signal respectively)

2.3 Verified Dynamic Model

Because the developed dynamic model is suitable for ideal conditions only, it should be verified to reach enough accuracy for simulations with actual conditions. A "trial and error" method is applied for obtaining the factors based on actual data from the IDH operation in 2013-2014 and represented by $[f_{ex}, f_h, f_{en}, u_1, u_2] = [1.1, 1.25, 1, 1.35, 1.4]$. The responses of the water temperatures and the fuel consumption with different outside air temperature from the verified model are shown in Figure 4. Comparing the results (lines) from the verified model with the measured data (dots), this verified model could be utilized for system simulations and predictions. Due to the heating value changes of the fuel and manually control of the fuel feed to the boiler(s), the actual fuel consumption is revealed a bit de-centrality. Note that the heat transfer areas of the HESs and the radiators are oversized and the water flow rates in the primary and secondary systems are running with excess of design values in the real system operation.

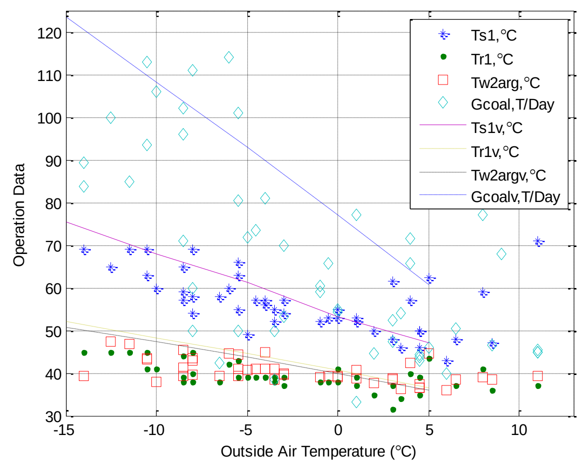

Figure 4. Verified Model Responses With Operational Data

2.4 Properties Analysis From The Verified Model Simulations

Based on the verified model, the factors $[f_{ex}, f_h, f_{en}] = [1.1, 1.25, 1]$ of the IDH system are fixed when the system was implemented. Due to the fact of over circulation flow in system operation, the water mass flow rate control signals of the primary and secondary systems are set between 1 and 1.4, and the simulated data is stated in Figure 5.

From Figure 5(a), the TD changes depending on the outside air temperature and the ratio of the water mass flow rate control signal in the primary system. Increasing water mass flow rate in the primary system by 10%, the TD will be decreased by 2.5~3.5℃ for the design outside air temperature, and it reduces faster in lower outside air temperature than that in higher outside temperature. On the other hand, in order to maintain the design zone air temperature, the water mass flow rates in the secondary (Figure 5(a)) and primary (Figure 5(b)) systems should be the same according to the simulation results depending on which side of water mass flow rate changes.

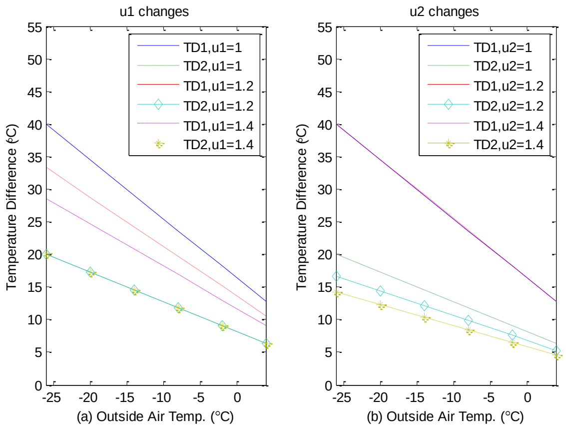

Figure 5. Water Temperature Difference With The Changes Of Outside Air Temperature And Circulation Flow Rate

To this end, no matter which side of water mass flow rate changes, the TD will be reduced when the water mass flow rate is increased, and the other TD in the IDH system does not change too much for keeping the same indoor air temperature.

3 PUMPING COST BASED ON THE CHANGE OF WATER MASS FLOW RATE

The pump affinity law has described the relationship between the water mass flow rate and the electrical power consumption of the circulation pump for the overall pipe network in DH systems. To compare with the pumping cost in the design case, the relationship can be expressed in Equation (6) as below:

$$E_{act} = E_d \left(\frac{u_{act}}{u_d}\right)^3 \qquad (6)$$

According to this equation, the power consumed will vary rapidly. In another word, for instance, if the circulation water mass flow rate is decreased by 10%, the electrical pumping cost will be decreased by 27% approximately.

4 TWO DAYS OPERATION WITH DIFFERENT WATER MASS FLOW RATE SETTINGS

According to the verified massage from the verified dynamic model, the control signals of the water mass flow rate in the primary and secondary systems are u_1=1.35 and u_2=1.4 respectively. With these situations, a control system configuration (Figure 1) is suggested to maintain the zone air temperature by setting point temperature as 20℃.

In the selected IDH system operation, the supply water temperature from the heat source has been monitored manually (fuel controller Cf) relating to the outside and inside air temperatures, solar radiation and wind speed; the heat balance has been regulated based on the average water temperature in the secondary system by adjusting the water mass flow rate (u1) of each HES in the primary system; and the water mass flow rate into the buildings has been controlled by using self-active flow control valves with certain settings (u2). However, in this paper for simplicity, the control signals of u1 and u2 are set to be 1.35 and 1.4 respectively, meaning over mass flow rates of circulation water comparing with the design values (u1=1, u2=1), and only Cf is taken into account for this simulation.

A typical PI controller C_f is used to regulate the supply water temperature from the heat source, and the fuel control signal u_f is formulated and computed in the following algorithm:

$$u_f = k_p(T_{bsp} - T_b) + k_i \int_0^t (T_{bsp} - T_b)dt \qquad (7)$$

The set point of the supply water temperature from the boiler is given as a function of outside and inside air temperatures, solar radiation and internal heat gains, and expressed in Equation (8) below:

$$T_{bsp} = f(T_o, T_z, Q_{sol}, Q_{int}) \qquad (8)$$

In order to obtain the system dynamic responses, the outside air temperature, the solar radiation, the internal heat gains used in the simulations are presented in Figures 6 (c) and (d). A design system (u_1=1 and u_2=1) and an actual system (u_1=1.35 and u_2=1.4) are simulated for comparison. As shown in Figures 6(a) and 6(b), the supply water temperatures from the heat source and the HES are almost identical because of the same supply water temperature set points from the heat source. The return water temperatures from the HES and the terminals in actual case are higher than those in the design case. The reason behind is that the over mass flow rate are operated by the primary and secondary systems; and also by doing it, the TDs in the primary and secondary systems are lower by 6-8℃ and 3-4℃ respectively than those in the design case, which means that more electrical power is required to overcome the circulating water resistance. As seen in Figure 6(c), by regulating the supply water temperature from the boiler with the suggested control strategy, the zone air temperature ranges within 20±0.1℃ in the actual case compared with that from 18.9℃ to 19.3℃ in the design system. The difference between the zone air temperatures is resulted from its set point of the supply water temperature from the heat source, and it should be set and monitored according to actual operation strategy. The accuracy achieved is obtained from the compensation of the additional heat gains and the feedback of the measured zone air temperature used to the control system.

In addition, by comparing actual system with the design system, the pumping cost in actual system has consumed about 40% more electricity in operation.

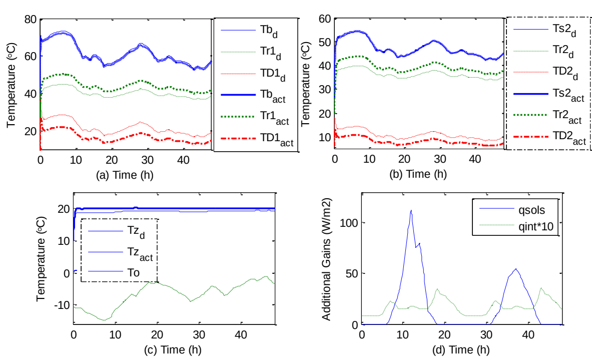

Figure 6. Two Days Operation With C_f Controller

5 SUGGESTIONS OF SOLVING THE ISSUE OF "BIG CIRCULATION FLOW RATE, SMALL TEMPERATURE DIFFERENCE"

The major reasons caused by the issue of "big circulation flow rate, small temperature difference" could be summarized as:

(1) The plan: final planning has been implemented in one step;

(2) The design: the safety factor is needed to be taken into account, but it does not mean that each heat transfer process and equipment selection is required to multiply a factor more than 1;

(3) The operation: the settings of the water mass flow rate in both pipe network sides should be calculated scientifically instead of experience only for each heat season; on the other hand, in order to improve thermal comfort level and cover up the issue of adjustment and control, the water mass flow rate is increased artificially;

(4) Suitable set points: wrong setting points of DH systems not only results in heat and hydraulic unbalance, but also leads to increasing the water mass flow rate easily to conceal the contradictory.

Due to the reasons described above, the solutions to solve this issue could be taken into account accordingly:

(1) The implementation of DH systems should track and follow the overall plan and the actual development in certain time span;

(2) To be careful of considering safety factors in design process; otherwise, it will increase the investment and the operational expense of DH systems greatly;

(3) From the simulation results, it is not necessary to increase the water mass flow rates to reach the design values of zone air temperature set points; on the opposite, it is bigger potential way for operational energy saving by reducing the flow rates;

(4) To reduce the pumping cost in operation, the water mass flow rate in both primary and secondary systems should be decreased according to the operation strategy adjusted for each heating season, and combined with suitable set points of the control system and equipment;

(5) To fulfill the heat and hydraulic balances, the set points of the overall DH systems including the heat amount and the supply water temperature from the heat source and the water mass flow rates in both hydraulic networks need to be calculated based on prediction of heating load requirement and symmetrically optimal process for each heat period.

6 CONCLUSIONS

By applying for the first principle of thermodynamics, ideal and verified dynamic models have been developed for an actual IDH system, and used for obtaining system characteristics and control strategy investigation. From the simulation results, it can be seen that the TD has been decreased associated with the increase of water mass flow rate. In the actual case, the pumping cost is consumed more than 40% comparing with that in the design condition. Also, the simulation has shown that the zone air temperature could be regulated more accuracy with suitable set point of the control system. The issue of the circulation flow rate and temperature difference has been discussed, analyzed and the solutions are suggested considering how to deal with it.

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