

The Optimization of Control Parameters for VAV HVAC System

Commissioning

Katsuki Maehar
Student, Graduate
School

The University of
Kitakyushu

Kitakyushu, Japan

Noriyasu Sagara
Prof., Faculty of
Eng.

The University of
Kitakyushu

Sung-ki Song

Lecturer

Toyohashi
University of
Technology

Toyohashi, Japan

Abstract: One of the technical subjects in commissioning for HVAC system is to enhance control performance and time efficiency, while the tuning of the optimal parameters to control HVAC system takes much time and labor in particular. Therefore, we propose a kind of commissioning technique as follows. We identified the dynamic characteristics of components of an actual VAV HVAC system such as rooms, VAV dampers, two way valves and chilled water coils, and verified the behaviors of the models and choose the optimal control parameters on a personal computer. Then adopting them for the actual system, we verified the control performance. Through the procedure, we showed the possibility of off-site and off-line control parameter tuning to reduce the cost and time at on-site and in on-line using the measured data in a real HVAC system.

INTRODUCTION

The tuning of the optimal parameters efficiently and ensuring satisfactory control performance in commissioning for HVAC system contributes realizing designed performance. However, improving fully satisfactorily control performance within limited short test time at on-site is often not easy, when control parameters are tuned through a trial and error based on the past experience, specially for a control system with a large time constant of the dynamic characteristics of the components such as a room air temperature.

Therefore, we tried to reduce an on-site and on-line tuning procedure for a control system. First, we collected time series data during some cooling

period, operating an actual VAV HVAC system without special operations for a tuning or identification as much as possible in order to adopt a following method easily in commissioning for actual systems. Second, using the data we identified the dynamics of components of the system, verified the behaviors of it and found the optimal control parameters with a system identification method on a personal computer at off-site and in off-line. And then we adopted the estimated values of optimal control parameters for those of the actual system and confirmed that the system performance was fully satisfied without some repeated trouble.



Photograph 1. Appearance of the HSPTA

We report the proposed technique for a tuning of PID control parameters for a VAV system of HVAC System Performance Testing Apparatus of The University of Kitakyushu (HSPTA).

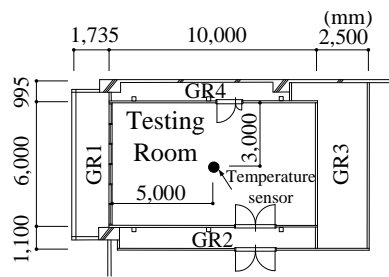


Fig. 1. Plan of HAPTA

Tab. 1 Specification of building and air-conditioning equipments

Item	Specification
of TR	Flat deck concrete 150mm thick
Southern wall of TR	Full double sliding windows, clear glass 5mm thick
Inner walls of TR	Double plasterboard 12mm thick
Raised floor of TR	Floor board: 600*600mm silica calcium board 29mm, tile carpet 6.5mm thick
Ceiling board of TR	640*640mm rock wool soundboard 15mm thick
Southern wall of GR1	Fixed window, clear glass 6mm thick+air space 5mm thick+clear glass 5mm thick
Refrigerating machine of ice thermal storage tank	Inverter refrigerating machine, refrigeration capacity 32kW
Ice thermal storage tank	Open type, 3m ³ , external melting coiled type, direct expansion system, one set
Chilled water coil of AHU	Refrigeration capacity 28kW, plate fin type, fin pitch 3.2, 6 lines, 14 rows, face area 0.37m ² , single flow type, inlet air temp 28.0(C), outlet air temp 13.2(C) inlet water temp 5(C), outlet water temp 10(C), water flow rate 81L/min, one set
AHU Fan	Total airflow rate 3,240 or 2,000m ³ /h, change-over, one set
Two-way valve	1/2 inch
VAV damper	Circular duct type, 250mm diameter, opposed-blade, two sets

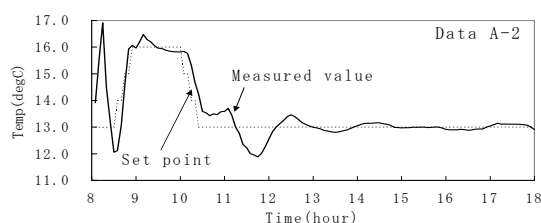


Fig. 2. Supply air temperature in Data A-2

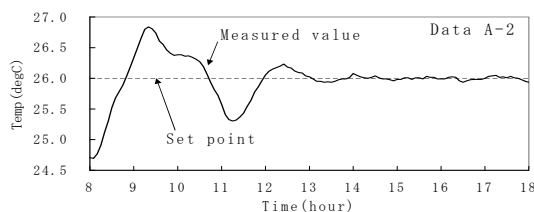


Fig. 3. Room temperature in Data A-2

HVAC SYSTEM PERFORMANCE TESTING APPARATUS OF THE UNIVERSITY OF KITAKYUSHU (HSPTA)

1. Building Outline

The HSPTA has a 6.0x10.0x2.7m testing room with 60 square meters of the floor area. The plan is shown in Figure 1 and the building and equipment specification are shown in Table 1. The testing room (TR) is surrounded by six rooms called guard rooms (GRs), the temperatures of which can be artificially maintained from 5 to 35 deg C, thus it equivalent to an occupancy space of an general office, the upper space and the lower space of which have the same thermal characteristics as those of it. It is separated from a GR (GR1) on the south side by double sliding windows. GR1 is used to be a pseudo outdoor, divided with the outdoor air by double-glazing. The testing room is surrounded by GRs (GR2 to 4), the three walls of which are made by the steel frame plasterboard except on the south side. The testing room with a ceiling space (a height of 900mm) is contiguous to an upper space GR (GR5) separated by the flat deck concrete slab (150mm thickness) and to a lower space GR (GR6) separated by the raised floor and the concrete slab with same specification.

2. HVAC System Outline

A VAV HVAC system is installed at the testing room. The supply air is cooled with the chilled water sent to the air-handling unit (AHU) form an ice thermal storage tank under the air conditioning mode. In this experiment, the outdoor air is not taken in to this system. The cooled cold air is supplied to the testing room through nine ceiling diffusers, flows to the ceiling space through the return louvers installed on the ceiling board, and returns through the return openings in the ceiling space to the AHU. The fan coil units (FCUs) installed in the perimeter zone are not used in this experiment. The evaporators of a direct expansion system and the heat exchangers with a hot water coil are installed in each GR for the room air temperature control.

3. Experiment Outline

The experiments were done on the high thermal load conditions for a midsummer term. The room temperature set point of GR1 was adopted as an hourly outdoor air temperature for an air-conditioning design in Fukuoka near Kitakyushu,

Japan and was periodically fluctuated per one day. Solar radiation to GR1 was protected by closing a

computing equation), PI-D controller (Equation 1) is used for the supply air temperature control system

Tab.2. The sign and sampling period of the experiment data

Sign	Date of experiment	Sampling period (minute)	The set point of PID control parameters					
			VAV control system			Supply air Temp control system		
			PB(%)	Ti(s)	Td(s)	PB(%)	Ti(s)	Td(s)
Data A-1	2003/10/13	5	7.5	500	0	165	100	25
Data A-2	2003/10/10	5	7.5	500	0	165	100	25

curtain on the inner side of double-glazing of GR1. The room temperatures of GR 2 to 4 are controlled to become the same temperature as that of the testing room in both an air-conditioning condition and a non air-conditioning condition. The AHU was automatically operated from 8:00 to 18:00 according to the time schedule. During air-conditioning operation, the fluorescent lights, 100Wx20 sets, and filament lamps, 32 Wx4 sets / unit, a total of nine units, lighted up to be internal sensible heat loads.

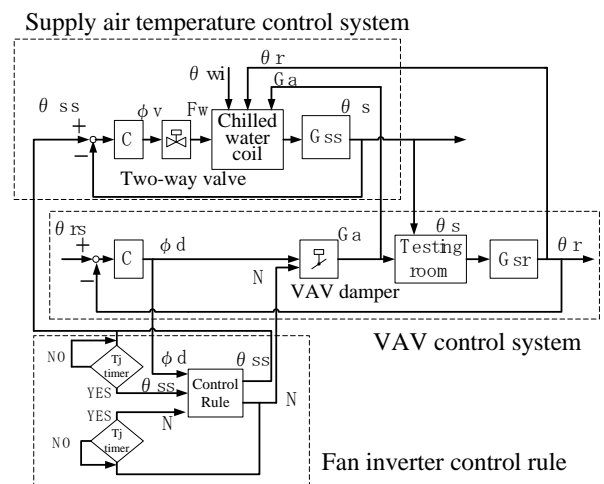
The control data collected in the experiment conducted for the research of thermal property of a HVAC system with building structure thermal storage is used for identification of the dynamic characteristics of components. The signs and sampling period of the experiment data is shown in Table2. Data A-1 and A-2 were collected in the experiments where the PID control parameter values were temporarily adopted by the trial-and-error method and were used for the identification of components for PID parameter tuning. The supply air temperature and the room temperature of Data A-2 are shown in Figure 2 and 3. Air-conditioning was started at 8:00 and these temperatures were almost equal to the set point about 6 hours later.

4. Control System Outline

The control system block diagram of the VAV HVAC system in HSPTA is shown in Figure 4. The control system consists of two feedback control systems, one is to control a room temperature by operating a VAV damper to vary the supply air volume, and another is to control a supply air temperature by operating a two-way valve to change the water flow through a chilled water coil. As for the computing equation of a Proportional-Integral-Derivative (PID) controller (following PID

and I-PD controller (Equation 2) is used for the VAV control system. The temperature difference corresponding to 100% of proportional bands is 80K for the VAV control system and 60K for the supply air temperature control system. Control performance is adjusted by the tuning of three parameters, PB, Ti, and Td.

$$MV = \frac{AG}{PB} \left(E + \frac{1}{T_i \cdot s} E + \frac{T_d \cdot s}{1 + (T_d/m_d) \cdot s} \cdot PV \right) \quad (1)$$



- θ_{ss} : Supply air temperature set point (deg C)
- θ_s : Measured Air temperature of chilled water coil outlet (deg C)
- ϕ_v : Two-way valve position control signal (%)
- θ_{wi} : Measured Water temperature of chilled water coil inlet (deg C)
- ϕ_d : VAV damper position control signal (%)
- N : Fan inverter output control signal (%)
- θ_{rs} : Room temperature set point (deg C)
- θ_r : Measured room temperature (deg C)
- G_a : Supply airflow rate (m3/h)
- F_w : Chilled water flow rate (L/min)
- G_{ss} , G_{sr} : sensor
- C : PID controller

Fig. 4. The control system block diagram of HSPTA

$$MV = \frac{AG}{PB} \left(PV + \frac{1}{T_i \cdot s} E + \frac{T_d \cdot s}{1 + (T_d/m_d) \cdot s} \cdot PV \right) \quad (2)$$

n_b =orders of a numerator polynomial. $n_b = n$
 when $m=0$, $n_b = n-1$ when $m \geq 1$.
 m =delays from input to out put time
 q^{-1} =shift operator

Tab. 3. The identified models of components

Components	n	m	A(q)			B(q)					
				a1	a2	a3	b0	b1	b2	b3	
Chilled water coil	1	0	$A_{\Delta \theta_s}(q)$	-8.483×10^{-1}	-	-	$B_{\Delta F_w}(q)$	-8.30×10^{-2}	5.23×10^{-2}	-	-
			$B_{\Delta \theta_{wi}}(q)$				2.33×10^{-1}	-1.04×10^{-1}	-	-	
			$B_{\Delta \theta_r}(q)$				-9.26×10^{-1}	-7.36×10^{-1}	-	-	
			$B_{\Delta G_a}(q)$				1.90×10^{-3}	-1.38×10^{-3}	-	-	
Testing room	3	2	$A_{\Delta \theta_r}(q)$	-9.659×10^{-1}	1.27×10^{-2}	4.507×10^{-2}	$B_{\Delta G_a}(q)$	-6.17×10^{-4}	-8.56×10^{-6}	3.54×10^{-4}	-
			$B_{\Delta \theta_s}(q)$				1.04×10^{-1}	-7.23×10^{-2}	-1.27×10^{-2}	-	
VAV danper	3	0	$A_{\Delta G_a}(q)$	-7.793×10^{-1}	-1.862×10^{-1}	2.031×10^{-1}	$B_{\Delta \phi_d}(q)$	3.41×10^0	-3.90×10^0	5.12×10^0	-3.99×10^0
			$B_{\Delta N}(q)$				1.89×10^1	-1.26×10^1	-2.08×10^0	-1.53×10^{-1}	

As shown in Table 1, HSPTA has two VAV dampers, all of which work simultaneously following the output of one VAV controller. HSPTA has one set of two-way valve for chilled water coil and one set of cold-water circulating pump.

IDENTIFICATION AND MODELING OF A VAV HVAC CONTROL SYSTEM

1. Structure of Dynamic Model and Procedure of Identification

The ARX model is chosen for as the structure of dynamic model of components of a VAV HVAC system and is shown in Equations 3 to 6.

$$A(q)y(t) = B(q)u(t) + e(t) \quad (3)$$

$$A(q) = 1 + a_1q^{-1} + \dots + a_{na}q^{-na} \quad (4)$$

$$B(q) = q^{-m}(b_0 + b_1q^{-1} + \dots + b_{nb}q^{-nb}) \quad (5)$$

$$q^{-1}u(t) = u(t-1) \quad (6)$$

,where

$y(t)$ =output signal. $u(t)$ =input signal.

$e(t)$ =disturbance. t =time

n_a =orders of a denominator polynomial. $n_a = n$,

n is the orders of model.

$A(q)$ and $B(q)$ = polynomials in q
 q^{-1} =shift operator

$A(q)$ and $B(q)$ = polynomials in q

In order that the technique in this study may enable easy application to the commissioning process, we got the data from a real HVAC system normally in operation and avoided the special operation for identification as much as possible. We abandoned the data within one hour after the starting time of the system for identification, because the non-linearity of the thermal characteristics could strongly appear with the big change of amount of thermal storage of the components during its period. The control system design software MATLAB was used for the following calculation. To remove the influence of low frequency disturbance on the model, the sample means were subtracted from the input-output data. In modeling for the tuning of PID-parameters, Data A-1 was is used for the estimation and Data A-2 for the validation. $\hat{A}(q)$ and $\hat{B}(q)$ are estimated by least-squares method.,

$$\hat{y}(t) = \hat{B}(q)u(t) + [1 - \hat{A}(q)]y(t) \quad (7)$$

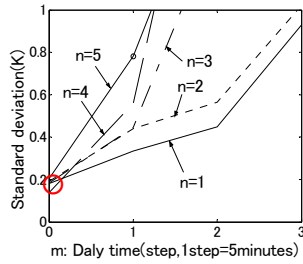


Fig. 5. Standard deviation of estimated error with each n and m

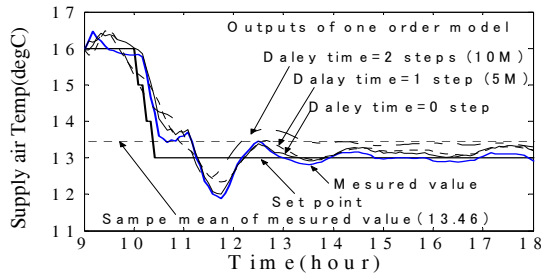


Fig. 6. The supply air temperature of chilled water coil model with model order n=1 and delay m=0 to 2 and the measured one

where $\hat{A}(q)$, $\hat{B}(q)$ =estimation values of $A(q), B(q)$

The optimal values for the order n of the model and delay time m were selected as follows. The outputs of the models with n=1 to 5 and m= 0 to 3

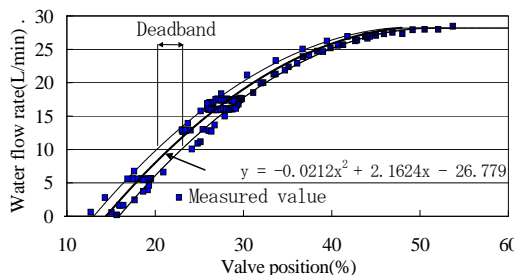


Fig.7. Relation between the two-way valve position (ϕ_v) and chilled water flow (F_w)

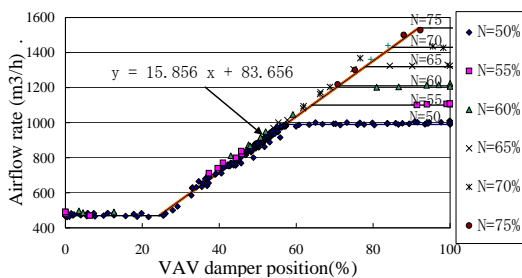


Fig.8. Relation between the VAV damper position (ϕ_d) and supply airflow (G_a)

and the measured one were calculated and those for the supply air temperature are shown in Figure 5 as example. The each standard deviation of error for the model with all of n and m is shown in Figure 6, using the data between 9:00 and 13:00 that has a big change in the measurement values. The model with m=0 and n=1 was selected as an optimal model to show the least standard deviation. The dynamic characteristics of sensors were neglected. The identified models of components are shown in Table 3.

2. Static model for a Two Way Valve for Chilled Water

The relation between the two-way valve position (ϕ_v) and chilled water flow (F_w) is shown in Figure 7. The static characteristic of the valve was expressed as a secondary approximated curve with the maximum value for F_w and the backlash with nonlinear effect, by 2.8% for ϕ_v .

3. Static model for a VAV Damper

The relation between the VAV damper position (ϕ_d) and supply airflow (G_a) to the testing room is shown in Figure 8. The static characteristic of the VAV damper was expressed as a primary approximated line with the minimum value and maximum value for every fan inverter output and with the backlash by 0.8% for ϕ_d .

4. Validity Verification of Each Component Model

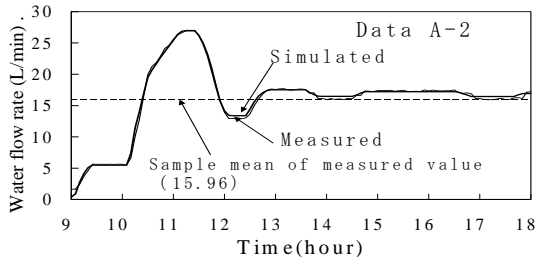


Fig.9. Measured and simulated output of chilled water flow rate (F_w)

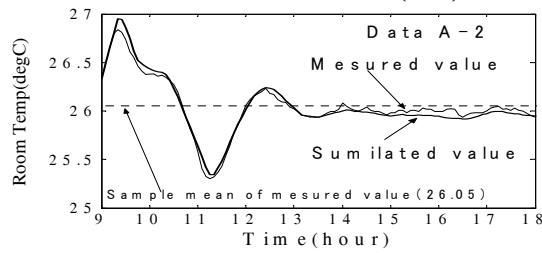


Fig.10. Measured and simulated output of testing room temperature (θ_r)

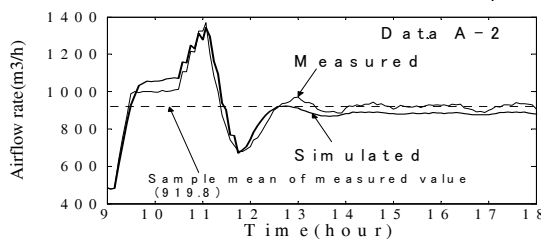


Fig.11. Measured and simulated output of supply airflow rate (G_a) for damper dynamic model

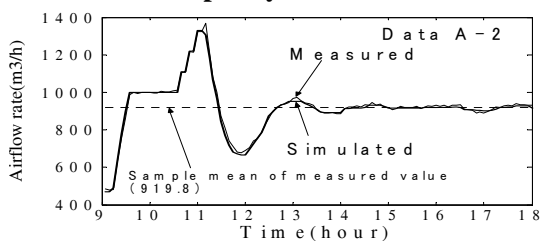


Fig.12. Measured and simulated output of supply airflow rate (G_a) for damper static model

The outputs simulated by inputting Data A-2 into the above component models are shown in Figures 9 to 12. In each component model, simulated outputs can fit the measurements in Data A-2 with a sufficient degree of accuracy.

5. Comparison of a Dynamic Model for the VAV Damper and a Static One

A VAV damper static model is more accurate than a dynamic one in comparison with Figures 11 and 12. A real commissioning requires a simple model possibly identified with the data easily

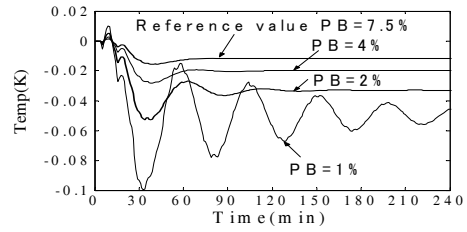


Fig.13. Step response for P control

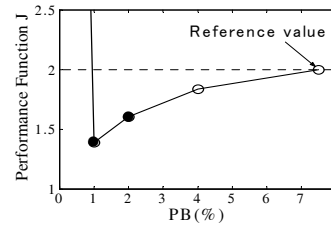


Fig.14. Performance function J for P control

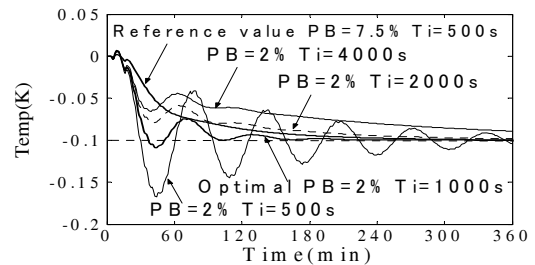


Fig. 15. Step response for PI control

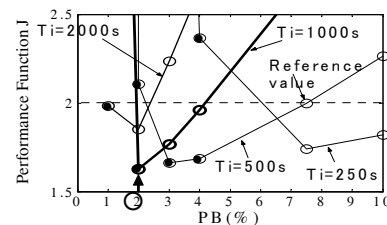


Fig. 16. Performance function J for PI control

measured. So we analyzed the properties of the both models to check the possibility to use the dynamical model that can be identified by the data collected easily in operation in spite of being inferior to the static model in a degree of accuracy.

PID TUNING THROUGH SIMULATION

The models of the both supply air temperature control system (Model A-S) and VAV control system (shown in Figure 4) were made by using identified models of components. For the model of the VAV control system, two models, Model A-RD for a VAV

damper dynamic model and Model A-RS for a VAV damper static model, were built. The optimal PID-parameters were tuned through the simulation of a step response for each of two feedback control systems with the models of the components, which were identified for a discrete-time system to be transferred to a continuous time system. The PID parameters for the Data A-2 are defined as the reference PID parameters. The Performance Function J (Equations 8 and 9) shows the magnitude of a both control error and control

$$\text{input. } J = \alpha \sum \varepsilon^2 + \beta \sum U \quad (8)$$

$$\alpha \sum \varepsilon^2 = 1, \beta \sum U = 1 \quad (9)$$

,where:

ε =The error between a set point and simulated output of a model

U=Chilled water flow rate, or supply airflow rate to the testing room.

α and β =Values are determined that a Equation 9 are realized in a reference PID parameters.

Therefore, the value of Performance Function J in the reference PID parameters is 2. The step responses of a controlled variable to the step inputs of a set point were obtained through the simulation, with PID control parameters first selected as a proportional (Ti=infinity, Td=0) and second as proportional and integral. Through the simulation, the optimal parameters for both Model A-RD and Model A-RS were chosen.

The tuning process of PID control parameters with the model A-RD for the VAV control system, is described as an example. With PB values of 4%, 2% and 1% of the only P-control adopted for the PID controller, each response of room temperature to a step input of -0.1K for the set point obtained through the simulation are shown in Figure 13. The relation between PB and the performance function J is

shown in Figure 14. The black dots in the figure show an oscillating response for reference. With PB 1%, the value of the function J is the minimum, but a step response has oscillation. Next, When PI-control is given to the PID controller, the step responses of room temperatures is shown in figure 15. The relation between PB, Ti value and Performance Function J is shown in Figure.16. Performance Function J is the minimum for the set of PB value of 2% and Ti value of 1000s. The step response shows that settling time and error are small compared with the response for reference PID parameters. D, derivative, control is not given to a VAV control system as an easy way here. The set of PB value of 2%, Ti value of 1000s, and Td of 0s value (Optimal-PID Parameter A-RD) were chosen as optimal PID parameters for Model A-RD, a VAV control system model with dynamic VAV damper model.

VERIFICATION OF THE VALIDITY FOR THE OPTIMAL PID PARAMETERS THROUGH EXPERIMENTS

We adopted the optimal control parameters estimated on personal computer for the actual system and confirmed that a system performance is fully satisfied. The optimal PID parameters and the results of verification experiments are shown in Table 4. The settling time is the time (minutes) it takes the response to converge within ± 0.3 degree C of the steady state or final value of the response (shown in Table 4). The supply air temperature is shown in Figure 17 and the room temperature is shown in Figure 18. In the experiment, where the optimal PID parameters were adopted, the settling time is getting shorter and the maximum overshoot is getting smaller than in the one, where the reference PID parameters were adopted.

CONCLUSIONS

Tab. 4 Adopted PID parameters, settle time and maximum overshoot in the verification experiment

Sign of experiment	Target system	Parameter type	Adopted PID parameters			Results of experiment	
			PB (%)	Ti(s)	Td(s)	Settle time (min)	Maximum overshoot(K)
Ex1	Supply air temperature control system	Reference parameters	165	100	25	270	1.3
		Optimal parameters	25	100	25	120	0.9
Ex2	VAV control system	Reference parameters	7.5	500	0	103	0.6
		Optimal parameters A-RD	2	1000	0	56	0
		Optimal parameters A-RS	3	1500	0	80	0

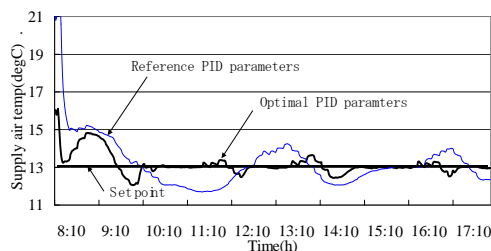


Fig.17. Supply air temperature of verification experiment

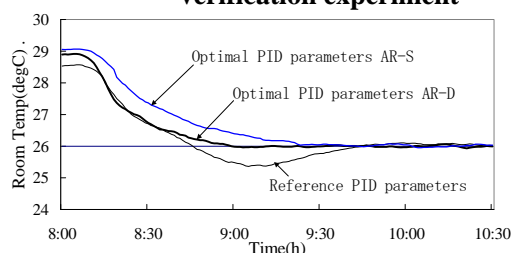


Fig.18. Room Temperature of verification experiment

One of the requirements to control parameter tuning in commissioning is to get satisfactory control performance within a limited short test period. This paper describes a series of processes from the modeling of the components of a HVAC system to the estimating of optimal control parameters and validating of the performance in a real system. The conclusions were summarized as follows.

- 1) The identification of the components such as a testing room, a VAV damper and a chilled water coil was conducted on a personal computer at off-site and in off-line with a system identification method, using the measured data without special operations for identification. Comparing the output of the identified models with the measured one, we showed the models were able to well present dynamic characteristics of a VAV control system.
- 2) The optimal PID parameters were obtained

through the simulations for a step response using a performance function J.

- 3) We adopted the optimal control parameters for an actual HVAC system and confirmed that the control performance was fully satisfied.
- 4) We verified the validity of two damper models: one is a dynamic model and another is a static model. The dynamic model showed to be superior to the static model in modeling of a damper using the easily measured data, but showed to be inferior in accuracy.
- 5) We showed the possibility of off-site and off-line control parameter tuning to reduce the cost and time at on-site and in on-line using the measured data in a real HVAC system.

REFERENCES

- [1] Katsuki MAEHARA, Noriyasu SAGARA, Sung-ki SONG, 2006, "The Optimization of Control Parameters for a VAV HVAC System Commissioning - An identification of Components and Settings of Control Parameters for a Chilled Water coils Outlet Air Temperature and Room Temperature in a Cooling Mode", J. Environ. Eng., AIJ, No.603, pp63-70, May, 2006, Architectural Institute of Japan, Tokyo, Japan (In Japanese)
- [2] SAGARA Noriyasu, RYU Yuji, OZAKI Akihito, Gao Weijun, SHIRAIISHI Yasuyuki, 2002. "The Outline of The HVAC System Performance Testing Apparatus of The University of Kitakyushu," Summaries of Technical Papers of Annual Meeting Architectural Institute of Japan, Environmental Engineering 2, pp.1209-1210, August, 2002, Hokuriku, Japan. (In Japanese)
- [3] Shuichi Adachi, 1996. "System Identification for

-
- Control with MATLAB,” Tokyo Denki University Press, Tokyo, Japan. (In Japanese)
- [4] Shuichi Adachi, 1999. “Control Engineering with MATLAB,” Tokyo Denki University Press, Tokyo, Japan. (In Japanese)
- [5] Gene F. Franklin et al., 1998. Digital Control of Dynamic Systems 3rd ed, Addison Wesley, Menlo Park, CA.
- [6] L. Ljung 1995, “System Identification Toolbox – For Use with MATLAB 3rd ed, The MATH WORKS Inc, Natick, MA.
- [7] SIMULINK-Dynamic System Simulation for MATLAB-Using Simulink Ver.4, 2000, The MATH WORKS Inc, Natick, MA.