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Keywords: Modeling and simulation, PID control, Heating and Ventilation.

Abstract

This paper discuss modeling and control of the inlet temperature in an **A**ir **H**andling **U**nit, **AHU**. The model is based on step response experiments made at a full scale test plant. We use gain scheduling to lower the correlation of the air flow with the process dynamic which simplify the control task. A simple way to determine the air flow with no extra equipment or experiments is suggested. Tuning of PI(D) controller based on step response identification is made using two different tuning methods. The paper describes the basic ideas, which are illustrated by simulations and plant experiments.

1 Introduction

Some of the most energy consuming installations in buildings are pumps and fans used in HVAC applications. One way of reducing the energy consumption is to use a variable speed drive to control the speed of the fans and pumps. In [3] calculation shows that it is possible to save over 32% of the energy for 30 kW pump system with a given load profile. The calculations are done for a pump motor driven by a frequency converter and one driven by a two-way valves. For a cooling tower fan driven by a 2-speed motor (2/3 and full speed) or driven by a frequency converter energy saving calculation shows, that for a typical load profile it is possible to save over 86% of the energy, see [2].

In addition we gain some reduction of mechanical wear and tear of belts, bearings, valves etc. This will provide a reduction of the overall maintenance cost of the equipment. When lowering the speed of the controlled device, we also reduce the acoustics noise level of the system.

Another important issue for the energy consumption is tuning of controllers. A lot of the controllers used in industries today runs with the factory settings, which normally gives poor performance. One reason for this is that the commissioning time often is very time consuming.

As the amount of supervisory functions increase in modern building management systems it is much easier to evaluate the performance of the control loops. Therefore we have to continuously put more effort in improvement of the control performance (e.g. response time, overshoot, damping, etc.). At the same time we must assure that the energy consumption is kept at a minimum level.

This means that more work must be put into tuning of controllers if the demands on performance must be fulfilled and in order to save more energy.

A lot of work has been put into modeling and control of HVAC system and especially in self tuning control of HVAC systems. In [12] and [11] a new self-tuning controller is described, in which a discrete time process transfer function is calculated from a relay experiment. They obtain a substantial reduction in commissioning time compared to conventional controllers. They also determine the sampling period and the desired close-loop poles from the experiments. The international ASHRAE standards states that in order to obtain an acceptable indoor air climate the temperature of the inlet air must not exceed $\pm 6°C$ of the wanted room temperature. With that in mind it is natural to split the control of the HVAC system into control of two subsystems (Cascade Control). An inner loop system which controls the temperature of the air from the AHU and a slower outer loop which controls the room temperature.

The purpose of the inner loop is to control the air temperature in the pipe just before the air enters the room and keep the temperature within the range of $\pm 6°C$. The AHU is a MIMO-system, see Figure (1), but since the air flow V_o can be controlled by adjusting the reference frequency f_{ref}, the task is only to control the outlet air temperature θ_o. In this work we only look at the inner loop.

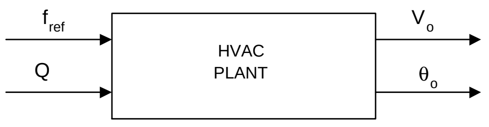

Figure 1: Inputs and outputs from the HVAC plant.

2 Physical modeling of the HVAC plant

In the past there has been put a lot of effort into modeling the different components in a HVAC system, see e.g. [1], [7], [4] and [6]. The model obtained in this section is going to be used as basis for some automatic tuning of a PID controller for controlling the outlet air temperature θ_o, see Figure (2).

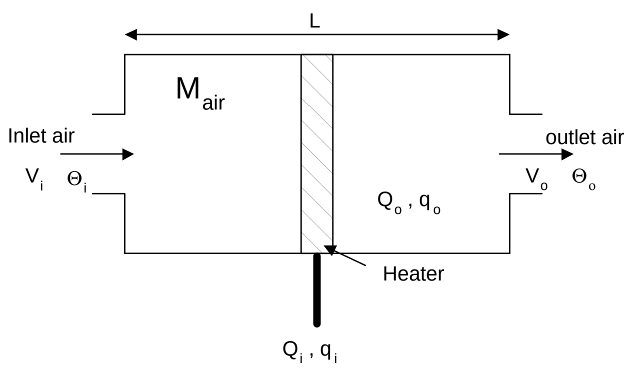

Figure 2: Thermal system

2.1 Model of an AHU

This section will describe a simple model of the AHU. A schematic drawing of the thermal system is shown in Figure (2).

2.1.1 Assumption

The chamber is well insulated to eliminate heat loss to the surrounding air, and we assume no heat storage appears in the insulation. Further it is assumed that the air is mixed so the temperature is uniform distributed which means that a single temperature measurement can be used to describe the air temperature in the chamber. It is also assumed that the flow of the inlet air is equal to the flow of the outlet air, i.e. $V_i = V_o$.

Definition of used symbols are shown in Table 1.

Table 1: Definition of symbols

Θ_i	=	Steady state value of the inlet air ^oC
Θ_o	=	Steady state value of the outlet air,^oC
V_x	=	Steady state air flow rate, $[kg/sec]$
M	=	Mass of the air in the chamber, $[kg]$
c_p	=	Specific heat of the air, $[kcal/kg^oC]$
R	=	Thermal Resistance, $[^oC sec/kcal]$
C	=	Thermal Capacistance, $[kcal/^oC]$
Q	=	Input to the electrical heater, $[kcal/\sec]$

3 Model of a simple Thermal system

The thermal resistance of the system can be calculated as

$$R = \frac{\text{CHANGE IN TEMPERATURE DIFF } [^oC]}{\text{CHANGE IN HEAT FLOW } [kcal/sec]} \tag{1}$$

and the Thermal Capacitance can be calculated

$$C = \frac{\text{CHANGE IN HEAT STORED}[kcal]}{\text{CHANGE IN TEMPERATURE } [^oC]} = M_{air} \cdot c_p \tag{2}$$

see [7], where M_{air} is the weight of the air considered in $[kg]$ and c_p is the specific heat of the air in $[kcal/sec^oC]$. θ_i is kept constant or is assumed to be constant. A step change in the input heat rate q_i is carried out. The output heat will then gradually increase by q_o. The temperature of the out flowing air will then also increase by an amount θ_o. The parameters q_o, R and C can be calculated as

$$q_o = V_o c_p \theta_o \ , \quad C = M_{air} c_p \quad R = \frac{\theta_o}{q_o} = \frac{1}{V_o c_p} \tag{3}$$

The differential equation of the system can then be written as

$$C \frac{d\theta_o}{dt} = q_i - q_o$$

Using equation (3) and writing the Laplace transfer function relating θ_o and q_i gives

$$\begin{aligned} \frac{\Theta(s)}{Q_i(s)} &= \frac{K_p}{Ts+1}, \\ \text{where } K_p &= R \quad \text{and} \quad T = RC. \end{aligned} \tag{4}$$

This shows that the model parameters are inversely proportional to the air flow V_o which means that the dynamics change with varying air flow.

When the air is transported from one point to another dead times will be present. Taking this into account the process can be modeled by a simple first order model with a time delay as shown in Equation (5)

$$G(s) = \frac{K_p}{sT+1} e^{-sL}, \tag{5}$$

where also the dead time L is inversely proportional to the air flow V_o. Measured step responses carried out with different air flows clearly indicate this dependence of the air flow, see section 4.

3.1 Determination of the air flow

As shown in the previous section, the transfer function from the air flow to the temperature is nonlinear and dependent on the air flow V_o, see Equation 3 and 4.

By creating a relation between the reference frequency $f_{ref}[Hz]$ given to the VLT frequency converter and the air flow $[m^3/h]$ it is possible to use this relation as gain scheduling in order to compensate for the nonlinearity, see Equation 6. Many methods can be used to determine the proportional factor,e.g. using pressure difference measurements or temperature measurements at different f_{ref} combined with the first law of thermodynamics see [5]. These methods requires either more equipment or some additional experiments compared to the following methods.

Using nominal data from HVAC plant. It is assumed that the air flow is proportional to the motor speed, i.e.e proportional to the reference frequency given to the frequency converter. This gives the following equation for the air flow

$$V_o = K_f \cdot f_{ref}. \tag{6}$$

From the specifications of the HVAC plant the fans supply the room with V_{sup}=1000 $[m^3/h]$ when the motor is running at 1500 $[RPM]$, i.e. at nominal speed, for further information about the HVAC plant see appendix A. With a slip of 5% the speed at $50[Hz]$ will be approximately 1425 $[RPM]$ at nominal load. The proportional factor K_f is then given by

$$\begin{aligned} K_f &= \left(\frac{f_{nom} - f_{slip}}{f_{nom}}\right)\left(\frac{V_{sup}}{f_{nom}}\right) \\ &= \left(\frac{50-2.5}{50}\right)\left(\frac{1000}{50}\right) = 19.0 \left[\frac{\frac{m^3}{h}}{Hz}\right]. \end{aligned} \tag{7}$$

This calculation is a very simple way of finding an approximation for the proportional constant K_f.

4 Step response identification

As shown in Equation 5 the dynamic of the system can be approximated by a first order model with the parameters K_p, L and T. These parameters can be determined from open loop experiments at different air flows. For further details on determination of such a three-parameter model see [10].

Measurements are carried out with four different air flows (20[Hz], 30[Hz], 40[Hz], 50[Hz]), i.e. with different references to the AC-motors. The power input to the heater is 5kW in each experiment. The air flow is fixed and a step in the input power to the electrical heater is made. The temperature just before the air enters the room is measured. The step response with $f_{ref} = 30[Hz]$ is shown in Figure (3) together with the simulated step response obtained with the identified model. The figure shows that the approximation with a first order model is acceptable.

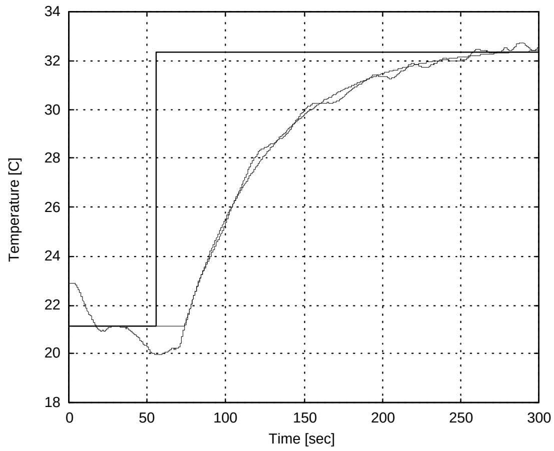

Figure 3: Measured and simulated step response at 30[Hz] reference frequency.

The parameters K_p, L and T shown in Table 2 are calculated using a Matlab function TRA() [1], in a straight forward manner.

Hz	K	T	L
20	3.95	90.1	16.98
30	2.51	61.9	14.50
40	1.27	47.5	16.14
50	1.39	32.5	8.24

Table 2: System parameters.

If we look at Figure (4) we see that the system parameters are inverse proportional with the reference frequency. From Equation 6 we see that this is also true for air flow. When using this information to adjust the controller parameters we only have to

[1]This function is made by Anders Wallén, Department of Automatic Control, Lund Institute of Technology. It makes a Least Square Fit to the step response data and calculate the parameters K_p,L and T

tune the controller parameters in one reference point and the adjust according to f_{ref}.

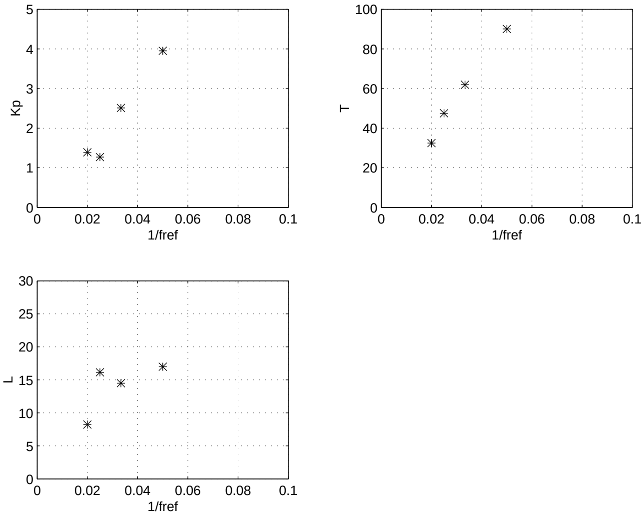

Figure 4: Change in system parameters as a function of the inverse reference frequency

5 Controller design

As seen in the previous section, the air flow has significant influence on the dynamic of the AHU. This clearly indicates that we should use a controller with varying parameters, see [9]. As controller we choose a ordinary PI (D) controller with set point weighting to handle the problem, see [10].

A comparison between two methods of tuning the controller parameters are made. The two methods are the $\kappa\tau$ method, see [10], and a method based on optimization of load disturbance rejection with constrains on the sensitivity, see [8].

These method are based upon simple characterization of the process dynamics. In the $\kappa\tau$ method the two dimension free parameters κ and τ, the relative gain and relative dead time respectively are used. The $\kappa\tau$ method is an empirical method. Controller parameters are computed for a large test batch of processes using dominant pole design. A relation between the normalized process parameters and the normalized controller parameters are found. This is done by finding expressions that gives the normalized controller parameters as function of the normalized dead time when using step response methods. In [10] the function have been approximated by

$$f(\tau) = a_0 e^{a_1\tau + a_2\tau^2} \tag{8}$$

but many other functions can be used instead. The design procedure for the optimal tuning method is to determine the controller parameters by optimizing load disturbance rejection subject to constrains on the sensitivity M_s. The set point weighting is determined so that the complementary sensitivity, M_{sp} is close to one.

For both methods the maximum sensitivity M_s is used as tuning parameter, and is defined as the inverse of the shortest distance from the Nyquist curve to the critical point -1. For convenience the definition of the sensitivity is repeated here

$$M_s = \max_{0 \le \omega < \infty} \left| \frac{1}{1 + L(i\omega)} \right| \tag{9}$$

where $L(i\omega)$ is the loop transfer function. For more details about the maximum sensitivity M_s see [10].

At the moment the optimized method can only handle tuning of PI controllers with set point weighting. Parameters of PI and PID controllers tuned with the two methods are given in Table 3. We use data from the experiment with $f_{ref} = 30Hz$ to calculate the parameters.

		M_s	K_c	T_i	T_d	b
$\kappa\tau$	PI	1.4	0.25	37.7		1.00
	PI	2.0	0.54	37.7		0.52
OM	PI	1.4	0.40	37.7		0.92
	PI	2.0	0.69	30.0		0.43
$\kappa\tau$	PID	1.4	1.03	44.0	10.0	0.49
	PID	2.0	2.00	35.0	9.0	0.26

Table 3: PI(D) controller parameters

Simulated step response for the different controllers are shown in Figures (5) to (9). As seen from the responses the PI controllers perform very well for both methods.

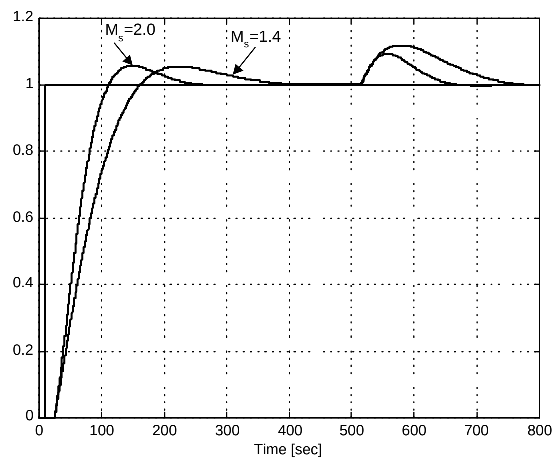

Figure 5: Response to step changes in set point and load with PI controller tuned with $\kappa\tau$ methods for $M_s = 1.4$ and $M_s = 2.0$.

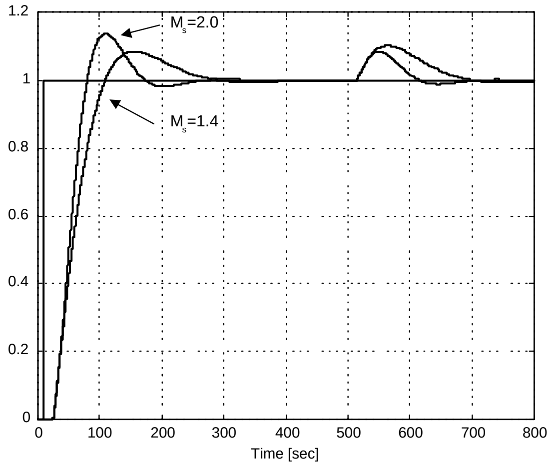

Figure 6: Response to step changes in set point and load with PI controller tuned with the optimal method $M_s = 1.4$ and $M_s = 2.0$.

For the PID controller the step response for $M_s = 2.0$ show some oscillation, see figure (7). From $M_s = 1.0$ to $M_s = 1.4$ the PID controller shows very good performance.

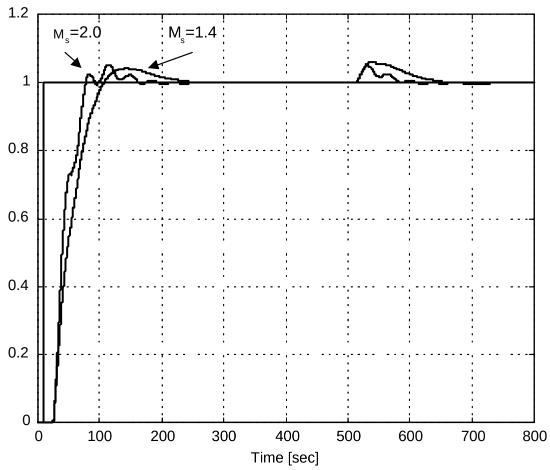

Figure 7: Response to step changes in set point and load with PID controller tuned with $\kappa\tau$ step response method for $M_s = 1.4$ and $M_s = 2.0$

The response for $M_s = 1.4$ for both methods are shown in Figure (8). Here we see that the response obtained with the optimal method is faster than the response obtained with $\kappa\tau$ method. But the $\kappa\tau$ method shows less overshoot than the optimal method.

The response for $M_s = 2.0$ for both methods are shown in Figure (9). Here the $\kappa\tau$ method again shows less overshoot than the optimal method, for the step input but they deal with load disturbances

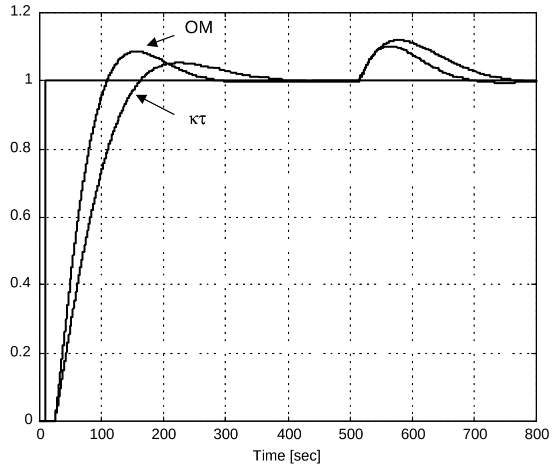

Figure 8: Response to step changes in set point and load with PI controller tuned with the optimal method and $\kappa\tau$ method for $M_s = 1.4$.

in the same way. Because of the more complex design

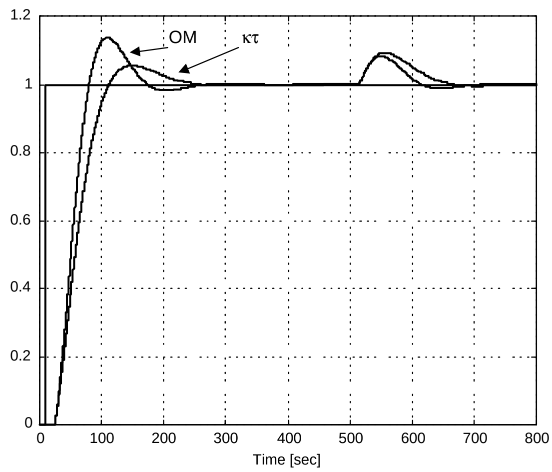

Figure 9: Response to step changes in set point and load with PI controller tuned with the optimal method and $\kappa\tau$ method for $M_s = 2.0$.

method of the optimal method compared to the $\kappa\tau$ method and because no significant improvement in the response of the close loop system the $\kappa\tau$ method is preferred as tuning rule for the controller. Because of the more sensitive response for the PID controller we choose to use a PI controller for this application.

6 Conclusion

A new way of lowering the influence of the air flow in temperature control of an AHU has been suggested. This means that we only have to tune the controller at one working point and the use gain scheduling.

Measurements on a full scale AHU shows that an approximation with a first order system for the AHU is sufficient. Further the model parameters dependence with the air flow was verified. The tuning rules used shows very good performance to step and load disturbances except for the PID controller tuned with maximum sensitivity $M_s = 2.0$ which was too sensitive.

7 Acknowledgments

The authors will like to thank Anders Wallén for his support and discussion with Matlab functions. Also thanks to Hélène Panagopoulos for calculation of PI controller parameters based on optimization of load disturbance rejection with constrains on the sensitivity.

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A System description

The HVAC system is constructed by Servex Ventilation A/S Denmark and is only for use in the laboratory at Aalborg University and Danfoss Drives A/S in Gråsten. A schematic drawing of the HVAC plant is shown in Figure (10).

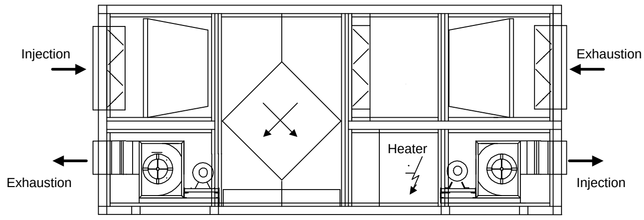

Figure 10: Schematic drawing of the HVAC plant

A brief discussion of the components in the AHU will be given in the following.

Components at the inlet air flow path

- Inlet damper controlled by a servo motor.
- Dust filter.
- Cross heat exchanger.
- Cooler (not operational at the moment).
- Electric heating unit (3x6[kW] elements).
- Fan driven by 0.75[kW] AC motor with encoder. It supplies approximately 1000[m^3/h] at a pressure at 713[Pa] when running at 1500[RPM].

Components at the inlet air flow path

- Outlet damper controlled by a servo motor.
- Dust filter.
- By-pass damper controlled by a servo motor.
- Cross heat exchanger.
- Fan driven by a 0.75[kW] AC motor. It supplies approximately 1000[m^3/h] at a pressure of 713[Pa] when running at 1500[RPM].

The two 0.75[kW] AC-motors are driven by two VLT frequency converters from Danfoss Drives A/S.