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Object-Oriented Modeling of an HVAC system

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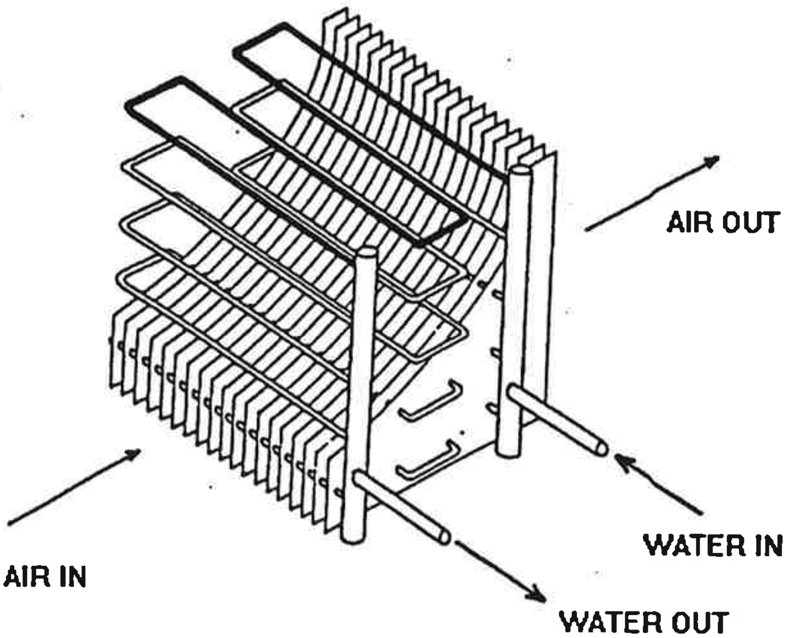

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Object-Oriented Modeling of an HVAC System (Objekt-orienterad modellering av ett HVAC system)

Abstract

This Master's thesis describes physical modelling and simulation of an HVAC System for investigating short time dynamics in order to improve control. Omola, an object oriented modelling language was used. A major effort has been done in building a beginning of a structured library for simulating short time dynamics in Omsim. Omsim is an advanced tool for object-oriented simulation where equations written in Omola are manipulated algebraically, sorted and solved. Simulations show that switch strategy between different control modes alone can cause supply temperature oscillations. Simulations also show that oscillations can be removed with new strategies and/or PID controllers. They also show that proportional control give almost as good results as PID with the new switching strategy. Experiences from building a simulation library with several components are that care has to be taken during modelling to avoid simulation problem.

Key words
heating, ventilation, air conditioning, object-oriented modelling, simulation, dynamics, control, pid, heat exchanger, omola

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Preface

About this report

This report is a Master's thesis in Engineering Physics presented at the Department of Automatic Control, Lund Institute of Technology.

Purpose of the work

The purpose of this work was to build a typical an HVAC (Heating, Ventilation and Air-Conditioning) System in the modelling language OMOLA and simulate it in OMSIM environment. This Model Data Base was said to be so well structured, that it could be the beginning of a model library for HVAC systems.

The minimum system to be modeled was a variable-air-volume air handling unit (AHU). It has several components: two variable speed fans, a steam/heated water to air heat exchanger, a chilled water to air heat exchanger and two valves.

John E. Seem at Johnson Control Inc. in Milwaukee, WI, USA proposed the project and provided test examples and valuable input. Johnson Control Inc. wanted to find a good tool for simulating short term dynamics. This tool should be able to design and simulate new control strategies to improve control performance. Then problems such as alternating behavior at dampers and preheating valve could be avoided.

Acknowledgments

I would like to thank my supervisors Karl Johan Åström, Jonas Eborn and co-supervising John E. Seem at Johnson Controls Inc. Wisconsin (WI, USA). I am also thankful for all help I have got from Sven Erik Mattsson and Tomas Schönthal with OMOLA and OMSIM. Especially I thank my supportive wife for taking care of me.

1. Introduction

An HVAC system has the purpose of maintaining certain environment characteristics such as temperature, fresh air and correct humidity. Maintaining fresh air and comfortable temperatures under varying circumstances is not easy. Especially when economical parameters are considered. The circumstances causing these variations that need to be controlled are changing weather, amount of heat generating equipment and the amount of people and their level of activity. These circumstances have very different time scales, where short term dynamics are on the scale of minutes and longer terms are on the scale of hours or days.

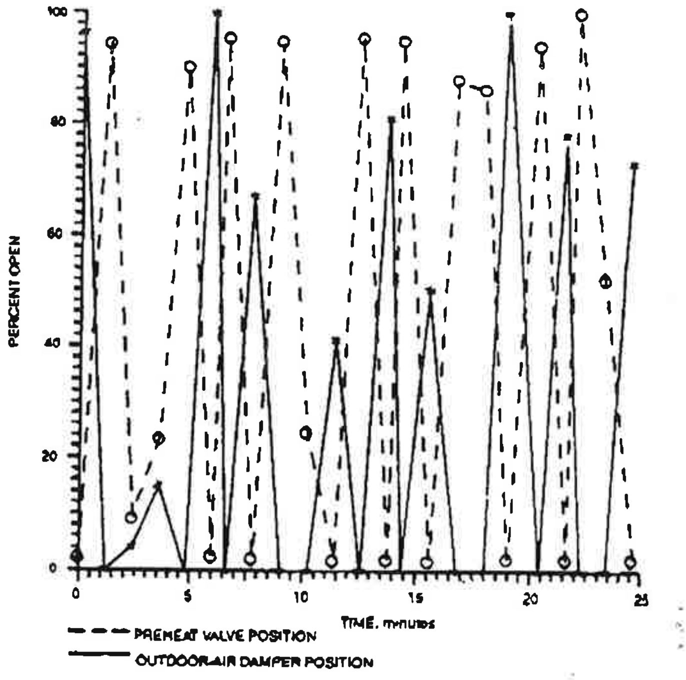

Figure 1.1 Example of bad control. Alternating control signals approximately 180 degrees out of phase.

Research that has been made give good guidelines and restrictions for how a system should perform. Research on human comfort satisfaction has been made by Wyon *et al.* (1970); Sprague and McNall (1970). Conclusions from this research tells us that periodically changing temperatures are more acceptable if the period is long and if we are active. They found that for a less active person temperatures should not vary more than $\pm$ 0.5 degrees Celsius and air speeds should not be higher than 0.2 m/s. On the contrary humidity has no direct impact. Instead we notice that dry air cause higher static electricity and irritated throats. Dry throats lose their capability to defend themselves and allergies are therefore more easily developed. To prevent this many HVAC systems in dry areas have humidifiers. A comfortable temperature sensation also depend to a high degree of the heat radiation from the walls, e.g. a room can feel cold even if the air temperature is well above normal temperature.

HVAC systems can be found in various places, e.g. offices, homes and industry laboratories and processes. Air conditioning requirements are different depending on where conditioning is done. In industrial processes like semi-conductor manufacturing temperature and low particle concentration are crucial for the process. A chemical laboratory can have high requirements on fresh air and constant temperature. In such cases the equipment must be better designed to cope with small variations.

Control problems that occur are often disastrous for economy and comfort. John E. Seem sent an example of bad control Figure 1.1. In this case the dampers who let the outside air in and mix it with the return air in order to obtain that supply air temperature have set point temperature. By this the damper opposes the action taken by the preheat valve. This action keeps on as long as the temperatures stays within a certain temperature interval. One major cause is that the system has time delays, both transport delays and in thermometers, which are not accounted for. One other reason is that controller gain is set too high. What happens is that the damper reaches its maximum open position trying to cool the supply air with fresh air. The system realizes finally that it is too cold and starts closing the damper while opening the preheater valve. The control of the valve suffers from the same problem and it starts over again. This causes wear of the damper and valve actuators and great heat losses. On top of that these air temperature fluctuations cause discomfort. By demonstrating that thecontrol problem in Figure 1.1 can be modelled in OMOLA and simulated in OMSIM it is pointing at possibilities for using this tool in industrial research.
The system mentioned was a Variable-Air-Volume VAV Air-Handling-Unit AHU with heating and cooling coil with separate opening valves with three dampers and fan. This VAV-AHU is used as a reference system for developing air conditioning control, see Figure 1.2.

In this Master's thesis an HVAC reference system was modelled in the Object-oriented MOdelling LAnguage OMOLA and simulations were carried out in the Object Modelling and SIMulation environment OMSIM, see literature Mattsson and Andersson (1992); Andersson *et al.* (1994); Andersson (1994); Mattsson and Andersson (1993); Mattsson *et al.* (1993); Mattsson *et al.* (1994); Eborn and Nilsson (1994).
The models written in OMOLA form an HVAC model database, that can be reused and added to in a convenient manner. The database consists of a number of libraries. Some contain very basic parts such as terminals, which are used to connect models with each other and libraries such as `Heat Flow`, that describe heat flows from one physical model to another. Parts in the `Terminal` and `Heat Flow` library are no simulation models, but by putting them together subunits, units and system models can be constructed. System models are the final product that will be simulated.

With the HVAC libraries built in this Master's thesis it is possible to build very different system configurations. As an example an HVAC system, see Figure 1.2 specified by Kelly (1993) with one AHU unit and with dampers, reheaters and return paths was built, see Chapter 2. Simula-

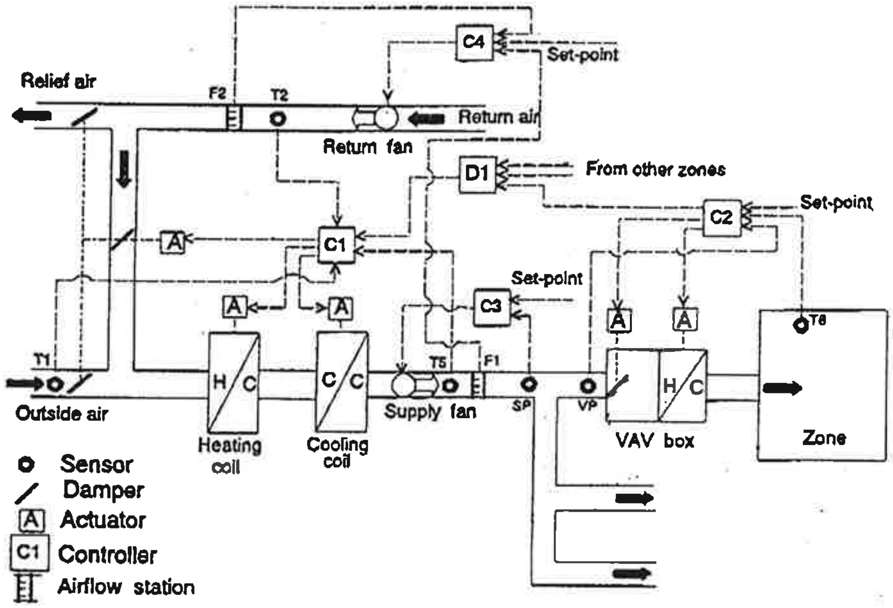

Figure 1.2 An HVAC reference system in a three zone environment.

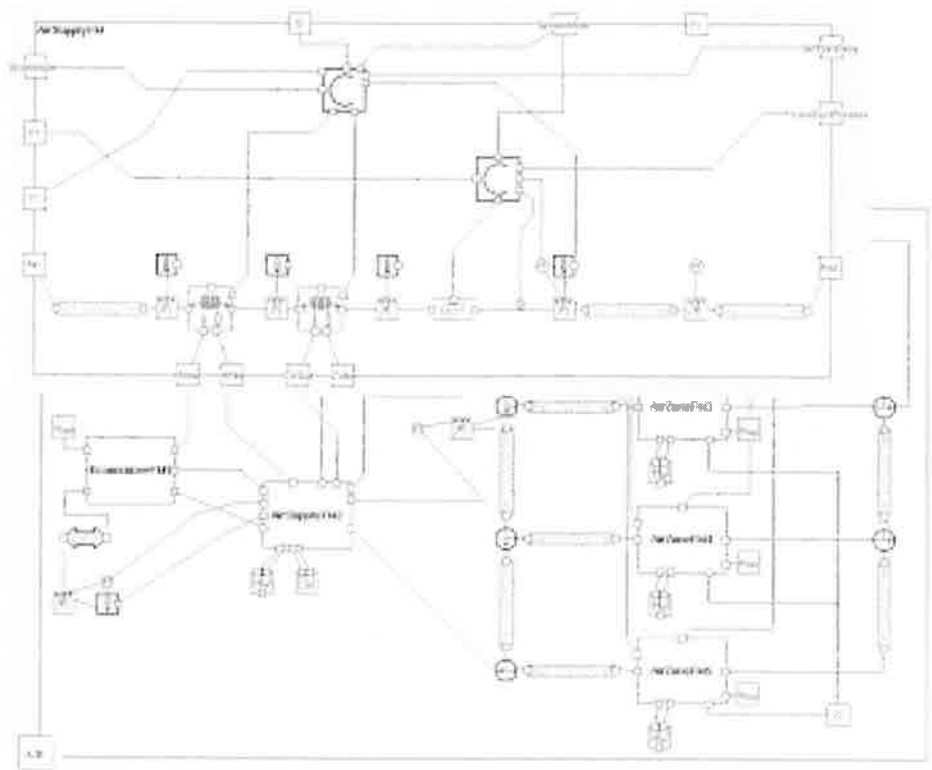

Figure 1.3 A graphical view given by the OmSim environment Model Editor of the HVAC reference system model and the "Air Supply" model on top of it. The "Air Supply" model is an AHU heating and cooling section following the economizer. A fan, control boxes, sensors and piping system are also modelled.

tions were made on the example to show that control can be improved by using PID-controllers see Chapter 3. In Figure 1.3 is a diagram over the specified system and on top of it is the air supply model which is a section of the AHU unit. The section includes preheating and cooling coil (heat exchanger), a fan and ducts (air ventilation pipes). The air flows in from the left, coming from outside or via an economizer, that usually is a part of the AHU. It passes one heat exchanger for preheating purposes followed by one for cooling and a fan. Three ducts can also be seen one to the left of the first heat exchanger and two to the right. Ducts are not considered as a part of the AHU unit, but represent delays and some dynamics in the system. The system is controlled by the fan, valves and dampers using information from flow, pressure and temperature sensors. The fan has a central role transporting the air through the system and by holding a constant over-pressure in the system ensuring that fresh air is available at all times. Valves open and close for hot and cold media used in heat exchangers. Dampers mix outside air with return air as to keep temperature as close as possible to demanded temperature with at least a certain minimum fraction of fresh air, e. g., 20%. The demanded temperature setting for the AHU unit is lower than the temperature setting for the zones. It is then possible to cool the air in the rooms if required. In case that this supply temperature drifts away from the supply temperature setting control actions are carried out for heating or cooling the air by the heat exchangers. If there is an economizer connected to the left as it is in the reference system, then out door air can be mixed with return air in such proportions that the demanded temperature is reached. On the other hand if a zone needs heating each zone activates its own reheating exchanger. Activation is controlled by temperature sensors in the zone. A damper also controls the amount of air entering the zone and thereby controlling the amount of cooling.

2. HVAC Models in Omola

In HVAC system control is realized through vents, dampers, actuators, fans and controllers that can be PID or other. The controllers get measurement data from thermometers and air flow sensors. Here in this chapter the models for these and their use will be presented. Many models in Lebrun and Bourdouxhe (1996); Clark (1985) written for HVAC are not straightforward to use. The reason is that they often calculate specific phenomena that cannot be broken down to general sub-models. Often these models rely on measurements for these phenomena without giving the parameters. Sometimes the model consists of a transfer function where parameters are given without direct relation to any model or physics and cannot be broken down to object related equations. In an object oriented language as OMOLA physical phenomena are easily modelled using equations and built together to a system model e.g. a heat exchanger. In which the calculation of heat exchange is better separated from flow calculation. The separation of phenomena makes formulas simple and straight forward to use. In order to make this library work simple models were created to model basic features in the HVAC equipment.

2.1 The HVAC System

The reference system modelled was prepared by Kelly (1993), see Figure 1.2. There system functionality and temperature data was specified. The reference system contains all physical parts including fault detection. This system model tries to follow the specifications as close as possible. This means that the system is not optimized for control investigation of the supply air and not for building load calculations. The system models the complete specified system and is thereby allowing the user to monitor almost any possible variable.

The specified system is a pressure independent variable volume (VAV) system with hot water reheat at the VAV boxes. Kelly (1993) says that it was chosen for studying fault detection and diagnosing of operational problems. Some of the criteria used were cost, operating cost, performance level and reliability to suit the desired requirements. Previous experiences of different air conditioning systems give at hand that VAV provide the best combinations for most applications as for offices.

As can be seen in Figure 1.2 the VAV boxes are placed close to each zone that will be air conditioned. The VAV box includes a damper and a reheat heat exchanger unit. The damper is used to control the amount of cool fresh air that will enter the zone. The reheater heat the entering air when needed. Control of the VAV box is done in a local control unit. By monitoring flow and the zone temperature the damper and vent position are corrected accordingly. The control box also signals a supply air set point temperature to the supply air control unit. This means that the supply air temperature will adapt when zone temperature rises above its set point. The temperature in the supply duct (air pipe) is controlled with the economizer, preheat coil and cooling coil in sequence. Each of them is controlled by a controller. These controller are set into different modes depending on the outside temperature. In that way only one controller will be active at any time instant. The economizer consists of three dampers, see Figure 3.2. By changing their position with an actuator it will be possible to control the amount of return air mixed into the outside air. In that way air temperature is controlled. When the damper reach either endpoint other means of heating and cooling are needed. The mode changes and either the cooler or the heater become active. Their vents change position according to the measured temperature error in the supply duct. The error is the difference between duct temperature and set point given by the VAV boxes. The amount of air is controlled by a frequency controlled fan and its controller. In this system there is also a return fan that operates in close sequence with the supply fan. It removes less air and thereby allows an over-pressure to build up in the zones. By having this over-pressure air will not enter the zone except from the supply air duct ensuring even temperature in the zone.

In the following sections some models and some of their important equations will be presented and discussed. Those are the main models and they are repeatedly used with minor modifications in the whole system.

2.2 Compartment models

Compartments are the volumes of air/liquid in the system model. Here the state of the media inside the compartment is calculated, see Appendix A. The state is characterized by the media mass, temperature and humidity. Media energy could instead have been characterized by enthalpy, but by choosing temperature a less complicated model was developed.

Some volumes have interfaces with the surrounding spaces exchanging mass and/or energy. In these cases flow or heat resistor models are connected to the compartment model. In the flow/heat resistor model the rate of exchange is calculated, see FinTubeHeatResistorFM in Appendix A. These models use secondary media states like pressure, density and conductivity etc. for calculating the rates. Secondary media states therefore are calculated in media sub models of the compartment model and passed on to the exchange models.

$$
\begin{aligned}
t' &= \frac{w_{in}t_{in} - w_{out}t_{out} + \frac{q}{C_p}}{m} && (2.1)\\
m' &= w_{in} - w_{out} && (2.2)
\end{aligned}
$$

The temperature t and mass m in the compartment is defined by (2.1) and (2.2). This temperature t is an average temperature for the media and can be considered as the temperature leaving the compartment if its volume is so large that the media can be considered as well mixed. This means that when using this temperature for determining the temperature difference between two compartments e.g. in a heat exchanger it will be slightly incorrect due to the temperature gradient.

In Heat Exchangers

In heat exchangers there is a temperature gradient in flow direction. It means that an approximation for this gradient is needed. In the compartment model there is no equation for the gradient. Instead the gradient is considered as a part of the heat flow resistor model, see Section 2.4. There a temperature difference approximation is used to determine the mean temperature difference in order to calculate the heat transfer. Also by splitting the compartment in a number of smaller compartment models approximates the gradient. Each of them having its own set of state variables. Plotting these compartment temperatures will result in a gradient. Such an descretisation makes the model model more realistic and simualation error smaller. By using an efficiency factor can the same results be achieved

If the heat exchanger is a cooler where condensation takes place then the increased heat flow through the exchanger during condensation should be modelled. In these models humidity condensation phenomena has not been modelled.

2.3 Flow models

Vent model

Vents are controlling the amount of media being used for heating or cooling the air. The media is mainly steam or hot water and cold water. When steam is considered also more complicated valve characteristics have to be modelled. Therefore liquid water was chosen. Also by stating that the valve characteristics is completely linear and having a leak flow when closed simplifies the model.

$$w = \max(w_{leak}, V_{pos} w_{max}) \tag{2.3}$$

Here $V_{pos} = [0, 1]$ and w_{leak} and w_{max} are chosen according to the modelled system. Other characteristics of the valve such as authority is neglected by assuming that the incoming water is under constant pressure and that the pressure drop also is constant. Therefore valve open area controls the flow completely. If pressure dynamics is considered and flow is determined from the pressure drop the valve dynamic interaction with the surrounding system will result in a system depending authority.

Temperature of the media and position of the valve is controlled in models for hot/cold water plants and actuators respectively.

Duct model

The Duct model have four sub models. Two air compartment, one heat reistor and one flow resistor. The flow in a duct can be considered as turbulent all the time and is calculated below. //

In HVAC air flow is dynamic and vary from very low to full flow during one day. This can cause numerical problems, since you need to change from low flow to either laminar flow or turbulent flow models. To avoid these problems a model that can approximate low flows from laminar flow and turbulent flows is needed. In the equations below an geometrical surface dependent factor zv is calculated, where ϕ is a friction factor and L the length of the pipe (duct) and d_h is the hydraulic diameter. Different ϕ are used dependent if the normal flow for the compartment is laminar or turbulent. The laminar factor depends on Reynolds number and in the turbulent case ϕ is approximately constant for different flow speeds and depend instead of the surface roughness r and the hydraulic diameter d_h. Finally flow is calculated from (2.7). By then taking the minimum of the two as the actual flow the switch between normal flow and low flow is smooth numerically. Using only the root function would cause inherent numerical problems due to the infinite derivative of the sqrt function in the vicinity of zero.

$$\phi_{lam} = \frac{64}{Re + \varepsilon} \tag{2.4}$$

$$\phi_{turb} = \left(3.2 - 2.5 \ln\left(\frac{r}{d_h}\right)\right)^{-2} \tag{2.5}$$

$$zv = \frac{\phi L}{d_h} \tag{2.6}$$

$$w = C_{open} A_{cross} \sqrt{\frac{2rho}{zv}} \min\left(\Delta P, \sqrt{\Delta P}\right) \qquad [kg/s] \tag{2.7}$$

Damper model

Dampers are controlling the amount of air passing into controlled area. In economizers dampers are used for mixing return air with outside air. Here a damper is modelled as volume (compartment) and a flow resistance separately. The compartment is needed merely to make the model general. In the damper friction model the air flow resistor model is used. Control is implemented by multiplying the fraction open area to the flow equation. Here pressure dynamics cause flow to change and therefore authority is dependent of the system configuration.

Fan model

Fans are the most basic unit to HVAC except the ducts. They are transporting the air through the building,giving fresh air from the outside and returning air from inside for disposal or mixing.

Fans are mostly electrical and often frequency controlled. The electrical energy is converted both to mechanical energy and heat. Both are transferred to the flowing air with a certain efficiency causing its pressure and temperature to rise. According to the literature review in the ASHRAE

Reference Guide by Lebrun and Bourdouxhe (1996) a temperature rise in the regime of 1.5 K in steady state is common if the fan motor is situated in the duct. For transient speed changes the temperature rise is governed by a time constant of 30s.

In this fan model no heat exchange was modelled and the flow was modelled as a linear function of fan speed. The fan speed changes with a minimum rise time that limits how quickly the fan speed can accelerate.

$$w'_{fan} = min(C_{ref}, w'_{max}) \tag{2.8}$$
$$w = min(max(w_{min}, w_{fan}, w_{max}) \tag{2.9}$$

The control signal represented by C_{ref} depends on a pressure difference that is scaled to give an appropriate amplitude.

2.4 Heat flow models

Heat flow resistor

Heat exchange between two compartments depends on the surface heat resistances α_1, α_2 and the conduction λ in the wall material. The temperature difference at each boundary defines the energy rate passing the boundary. Multiplying with the temperature difference gives the the energy rate passing the wall from one media to the other (2.12) .

$$R = \alpha_1 + \frac{1}{\lambda} + \alpha_2 \tag{2.10}$$
$$\Delta T_{tot} = T_1 - T_0 \tag{2.11}$$
$$q = \Delta T_{tot} \frac{1}{R} \tag{2.12}$$

In a parallel flow heat exchanger it can be enough to take the mean value of the temperature differences at the ports. If hot H_{in}, H_{out} and cold C_{in} and C_{out} are the ports on the side of the hot and cold media respectively then the mean temperature difference is calculated by (2.15). It can also be an approximate mean temperature for a counter flow heat exchanger if the difference between $\Delta T_A c(t)$ and $\Delta T_B c(t)$ is small.

$$\Delta T_{Ap} = T_{Hin} - T_{Cin} \tag{2.13}$$
$$\Delta T_{Bp} = T_{Hout} - T_{Cout} \tag{2.14}$$
$$\Delta T_{mean} = \frac{\Delta T_A + \Delta T_B}{2} \tag{2.15}$$

$$\Delta T_{Ac} = T_{Hin} - T_{Cout} \tag{2.16}$$
$$\Delta T_{Bc} = T_{Hout} - T_{Cin} \tag{2.17}$$
$$\Delta T_{logm} = \frac{\Delta T_{Ac} - \Delta T_{Bc}}{\ln\left(\frac{\Delta T_{Ac}}{\Delta T_{Bc}}\right)} \tag{2.18}$$

For cases when the difference is large a log mean difference function is commonly used in heat exchanger calculations. The log mean difference is given by (2.18) and works well as long as the differences are not too close to zero or have opposite signs. The log mean function is ill conditioned and not defined in these cases. To avoid this other functions giving a ΔT_{mean} value are used. Another problem occur when switching between these functions. The functions cannot be matched, which result in a jump in ΔT_{mean} that causes numerical problems. These can cause the simulation to stop when trying to solve forthe new state.

The values for surface heat resistances α and wall conduction λ is defined in sub models. Resistances are calculated from Reynolds Re and Prandlt Pr number giving a Nusselt Nu number. To avoid division by zero ε is introduced in the denominator.

$$Re = \frac{Vd}{A\nu} \tag{2.19}$$

$$Pr = \frac{C_{Pfluid}\nu}{\lambda} \tag{2.20}$$

$$Nu = C Re^{n_1} Pr^{n_2} \tag{2.21}$$

$$\alpha = \frac{1}{\frac{Nu\lambda A_{heat}}{d_h} + \varepsilon} \tag{2.22}$$

Fin-tube heat exchangers, see Figure 2.1, are often used in air conditioning. For such an exchanger it is a little more laborious to calculate the total resistance. The fins increase the contact area resulting in higher efficiency (2.24). In (2.22) C, n_1 and n_2 constants are depending on the type of heat exchanger and the media used in that compartment. In the following equation we have a convection coefficient h_{conv} that depends on the flow speed in the fin compartment and ε_{fin} is the calculated efficiency because of the temperature gradient in the fins.

$$\lambda = \sqrt{\frac{2h_{conv}}{\lambda_{Al}\Delta L_{fin}}} \tag{2.23}$$

$$\varepsilon_{fin} = \frac{2\tanh(\lambda \Delta L_{fin})}{\lambda \Delta L_{fin}} \tag{2.24}$$

$$\alpha = \frac{1}{A_{tube}h_{conv}\lambda_{Cu} + A_{fin}\varepsilon_{fin}\lambda_{Al}} \tag{2.25}$$

Zone Wall Heat Resistor Model

A zone is one or more rooms in a building that have the same air-conditioning reheater exchanger and air flow control. Depending on the heat load on the zone heat has to be added or removed by the air-conditioner. Heat passes through the windows and walls and is stored in them as well. A model of a wall has been created, see (2.27). In these equations are t_{win} and t_{wout} the walls temperatures facing the outside and the inside while t_{in} and t_{out} are the outside and zone temperatures. U_1-U_3 are heat resistances defined for each side. In the model for the wall each material have been modelled as a separate conductivity λ and capacitance C, so that heat flows between the layers within the wall are calculated similarly. Q_{out} is the heat flowing into the zone from the wall. Zone temperature is given by (2.29). The temperature is calculated from the energy difference between the entering and leaving air and the added heat Q_{out}. These calculations are done for every boundary.

$$U_{surface} = \alpha_1 \qquad U_{wall} = \frac{L}{A\lambda} \tag{2.26}$$

$$Q_{out} = \frac{t_{wout} - t_{out}}{U_1 + U_{wout}/2} \tag{2.27}$$

$$Q_{in} = \frac{t_{in} - t_{win}}{U_{win}/2 + U_3} \tag{2.28}$$

$$C_{zone} t'_{wout} = w_{in} t_{in} - w_{out} t_{out} + Q_{out} \tag{2.29}$$

Wall types as windows are both conducting heat and transparent to radiation. The heat resistor must in this case take care of both phenomena. Conduction is taken care of as in wall and the amount of radiation that passes through depends on the window transmission factor. The transmitted sunlight is added to the zone indoor wall. Other energy factors can be solved in a similar fashion.

2.5 Control models

Actuator model

The actuator is in HVAC used to control the position of ventilation dampers and hot and cold media vents for the heat exchangers. In Lebrun and Bourdouxhe (1996) models for actuators can be found, both detailed examples describing the electrical behaviour and frequency response giving the actuator position by a transfer function.

$$\frac{\Delta y(s)}{\Delta E(s)} = \frac{\pi B d N}{R(\tau_{Mot} s + 1)(M_1 s^2 + B_1 s)} \tag{2.30}$$

For the purpose of improving control it is not necessary to know the detailed behaviour of the actuator. The differential equation models the dynamic well. Therefore knowledge of parameters such as maximum speed or time constant and hysteresis is enough. Then by letting the control signal decide the speed of the actuator in either direction up to the decided maximum speed the actuator position can be calculated. The speed of the actuator is then transferred to the controlled unit if $abs(A_{pos} - U_{pos}) >= A_{hyst}$, see (2.31). The time constant τ_A models the actuator response time.

$$A_{speed} + \tau_A * A'_{speed} = \min(C, A_{max}) \qquad U_{pos} \in [min, max] \tag{2.31}$$

$$U'_{pos} = A_{speed} \qquad abs(A_{pos} - U_{pos}) >= A_{hyst} \tag{2.32}$$

$$U'_{pos} = 0 \qquad abs(A_{pos} - U_{pos}) < A_{hyst} \tag{2.33}$$

According to Lebrun and Bourdouxhe (1996) common time constants for actuators are $T_A = [20s, 60s, 180s]$. They have also a model with a motor dead-band (0.01), which is not included in this model.

Other approaches involve the concept of events. Events is a way of introducing discrete changes in variables during simulation. These are mainly of two kinds: scheduled time events and events caused by a controlled variable or expression. An event is created when the variable or expression value changes its sign. The simulator then searches for the time instant when zero was passed (root). Events have the advantage that they more exactly simulate the change-over from one mode with fully developed hysteresis to modes where the unit is not moving and back. Again this accuracy is not needed. Instead it is worth noting that simulation time increases due to searches for time instants where mode changes occur. If events occur too often it can make simulation impossible. Events that occur too close can be missed.

PID controller model

In HVAC PI controllers are already common as controllers. These controllers have the disadvantage that they cannot deal very well with time delays. The problem is that small gains have to be chosen in order to get a stable system. It also gives a long settling time after a disturbance. By chosing a small gain the difficulties increases to reach zero error when the disturbance has ramp shape. An integrator solve that problem, but still there can be problems with overshoots or long settling time. By adding a derivative part the controller will be more sensitive to disturbances and damped when the measured variable changes. By introducing a low pass filter quick disturbances will be excluded. The PID controller used is governed by (2.34-2.39). The controller has also anti-windup, which resets the integrator when maximum signal is reached. An Improved PID would also reset the integrator when the error $e(t)$ change sign.

$$
\begin{aligned}
e(t) &= y(t) - y_{ref}(t) && (2.34)\\
p(t) &= K * e(t) && (2.35)\\
i'(t) &= \frac{K}{T_i} * e(t) && (2.36)\\
y_f'(t) &= \frac{N}{T_d}(y(t) - y_f(t)) && (2.37)\\
d(t) &= -K * N * (y(t) - y_f(t)) && (2.38)\\
v(t) &= p(t) + i(t) + d(t) && (2.39)
\end{aligned}
$$

2.6 Sensor models

In this system there has been introduced three kinds of sensors thermometers, flow and pressure meters. Thermometers are often modelled with time constants modelling the heat resistance and capacitance in the material.

Flow and pressure sensor model

Real flow and pressure meters have varying sensitivity, but can be considered as instantaneous in these applications. While testing the HVAC system an instantaneous flow sensor involved in an algebraic system caused numerical noise, which forced the simulator to take smaller time-step and thereby increasing simulation time. A better solution was to introduce a differential equation causing the measured signal to be filtered. This does not affect the result when the time constant is small enough. The same noise problem occurred with the pressure meter model. The "measured" pressure variation introduced quick control signals resulting in even more variations in the pressure also causing slow simulation. In both cases the sensor signal is used as control signal influencing the measured variable. The equation causing the fluctuations is the flow equation (2.7). It requires a pressure change that is large to result in a smaller flow change as a result of the square root dependence.

Thermometer sensor model

A second order model can be found in Lebrun and Bourdouxhe (1996). This model consists of two differential equations describing the thermal relation between the thermometer and the surrounding wall and the wall relation to the surroundings. This can be simplified to one differential equation because of the relatively slow variations occurring in HVAC systems. The temperatures in the air-conditioned zones vary one degree on the time scale of a ten minutes or more and in the supply ducts on the time scale of tens of seconds. A fast sensor is needed only in the supply duct. If a slower sensor is used then control is affected because of the delay in measurement signal.

2.7 Heat Exchanger model

The normal heat exchanger has separate compartments for the fluids and/or gases. The wall separating them is usually conducting heat well. In the modelled fin-tube heat exchanger, see Figure 2.1, the large contact area due to the fins improves heat transfer. How much heat that will be transferred from one side to the other depends on several parameters such as conduction λ, wall surface heat resistance α, wall and fluid thermal capacitance C, fluid velocity, fluid and wall temperatures and of the walls geometrical shape.

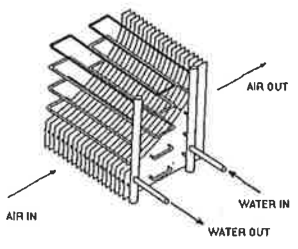

Figure 2.1 The physical shape of a fin-tube heat exchanger.

Figure 2.2 shows that the physical phenomena for heat resistance and flow resistance has been modelled separate from the model defining the state of the media in the two compartments. Observe that no flow resistance has been modelled on the water side. In the figure air is coming in from left entering the compartment. There heat interaction with the wall occurs and several state variables are calculated see listing for HHEXAirCompartmentFM in Appendix A. The air then leaves the compartment and enters the flow resistor, see Listing for AirFlowResistorFM in Appendix A. Only the flow out of the compartment is calculated here. The model should reflect the flow resistance caused by the compartment. Pressure difference $\Delta P = C_1.P - C_2.P$ and current flow velocity v is given by this and following compartment. In the flow damper model described in Section 2.3 the friction factor ϕ_{turb} a simple turbulent factor is calculated at one time. In the fin heat exchanger modelled here the flow is approximately laminar. This laminar factor depend on the flow speed and is calculated continuously.

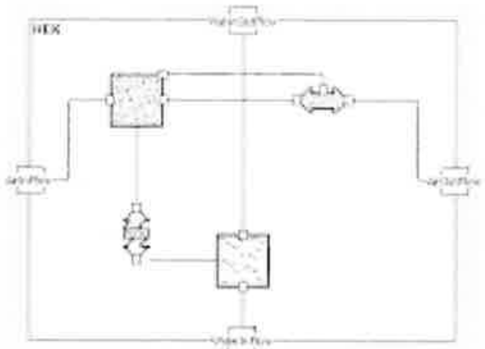

Figure 2.2 The graphical layout of a section of the heat exchanger model.

When calculating ϕ_{lam} (2.40) it can happen that flow speed becomes zero, e.g. when simulating a system that stops during night therefore a small number ε is necessary to avoid division by zero. All flow dynamics in the water side have been removed. The reason is that dynamics on the water side is approximately instantaneous, because of incompressibility of water. Flow on water side is still controlled by a vent.

$$\phi_{lam} = \frac{64}{Re + \varepsilon} \tag{2.40}$$

The wall between the two fluids is in this case a fin-tube wall with the fins welded onto the tubes, see Figure 2.1. On the air/gas side the fins increase the contact area, while the water/fluid runs in the tubes. Heat exchange is calculated in a few steps. First the Reynolds, Prandlt and Nusselt numbers (2.22) are calculated giving an air resistance and similarly on the waterside. Here a choice can be made between assuming that the wall resistance is small in comparison to the other two and neglecting its influence or incorporating it. In this model the wall resistance is calculated. The effects of air convection over the fins causing a fin efficiency are taken care of giving (2.25).

Numerically it can be difficult when having comparably high mass flow into a small compartment. In such a compartment the entering media flows through without mixing. In heat exchangers used in HVAC the volume is about $0.3m^3$ and the flow is about $1 - 3m/s$. This volume is also divided between some hundred fins see Figure 2.1 causing the media to stream through without mixing. The numerical problem lies in the fact that compartment temperature will vary almost as quick as the entering temperature. Leading to quick variations in other state variables such as pressure and indirectly the mass flow. This forces the simulator to increase the number of steps for solving the equations. A quick pressure loss could result in negative pressure difference, but usually only at low flow speeds. Negative pressures should cause negative flows, but these are set to zero. If they were not set to zero these equations had to be solved also in the opposite direction. It is possible to do this, but then double equations need to be solved for every flow equation. These variations affect surrounding object models causing the variations to migrate through the whole system. Simulation will be slowed down by this. If the entering temperature or flow have realistic variations and models are well constructed then there is no problem simulating the system quicker than real time.

3. Simulations

Simulation methods

The models written in OMOLA are simulated in the OMSIM environment. In the simulator configuration algebraic solver methods and integration methods can be chosen see Table 3.1. Some integration methods are implicit and some explicit. Dart (Dassl with root finder) has been used in these simulations. It is a multi-step, root finding implicit method. Root finder is used to find the instant of an event, e.g. $A - B > 0$. Events are avoided for the reason of the time it takes to find the roots. Especially when events occur often and close to each other. B-Euler is also an implicit and one step method that was tried with good results. Radau5 is an implicit one step method that have not been used successfully. This method handles stiff problems better. Also algebraic loops have been avoided and because of this there is no need for an algebraic solver. Earlier in the development of the models there were a few algebraic loops. Hairer was used then.

DAE solver	Algebraic Methods	other options
Beuler	**Hairer**	Analytical Jacobian
Dasrt	Hompack	Cramer's Rule for Manipulation
Dopri45	Minpack	Scalarize Equations
Euler		**Optimize Code**
Lsodar		absolute error= $1 \cdot 10^{-6}$
Radau5		relative error= $1 \cdot 10^{-4}$

Table 3.1 Simulation methods and some settings

Initialization

Before running a simulation all variables must be initialized. Initial values can be included when writing the model, but when the model is used several times a separate initialization script is convenient. If only a few variables are going to be set they can be set through variable windows available from the simulator menu. If many are going to be set it is more convenient to write scripts. In scripts commands can be written that control all other simulation settings and model instantiation including running and plotting . Initialized variables must also be consistent with the model equations. Starting the simulation invokes an algebraic solver that calculates all the variables to the relative and absolute accuracy set. Too big residuals in the first simulation step will stop the simulator.

In all models all air temperatures has been set to 20°C and water temperature to 21°C in heat exchangers and to 19°C in coolers. The air density must also be set. For these simulations the density difference was chosen equivalent to a few Pascal pressure drop between the volumes. The density multiplied with the known model volume gives the present mass. Now pressure is calculated from the known mass volume and initialized temperature. Outdoor pressure is set to a normal pressure $101325[Pa]$.

After having initialised the variables the simulation can be started. Due to fast transients is stable condition reached after $\tau = 2 \cdot 10^{-5}$ seconds. After startup the simulation is stable to run for long periods. System is not initialised to stationary state. How long time it takes depends on how far from stationary state the system is and what time constants are present in the system. It can take very different long time to reach this state. Especially if zones with long time constants are used.

3.1 General

Simulation has been carried out with step temperature variations and artificial weather temperature variation, see (3.1). Weather data have a smooth temperature variation to show that the control acts as supposed to and if there are control problems. With the weather data it can also be seen how well reasonable disturbances are taken care of. The step variation can be used to determine the settling time etc.

$$T(t) = 18 + 8\sin\left(\frac{2\pi t}{24 \cdot 3600}\right) + \sin\left(\frac{2\pi t}{3600}\right) \tag{3.1}$$

Outside Pressure and humidity is set to be constant. The temperature step variation tests the performance of the PID controllers and can be compared with the proportional control simulations. The problematic action in the case with proportional control was presented in Figure 1.1 will be observed in the proportional controller simulations.

Outside temp. [°C]	$T < 12.8$	$T < 18.3$	$18.3 < T < 22.2$	$22.2 < T$
Economizer	Min[1]	Ctrl[12]	Max	Min
Preheater	Heating[3]	Heating[3]	Off	Off
Cooler	Off	Off	Cooling	Cooling

Table 3.2 The controller modes in the temperature intervals.

In all simulations with temperature step variation the simulation time is 20.000s and in the artificial weather temperature variation the simulation time is 86.400s= 24h. In all cases the controllers will cooperate in

[1]There is no switch between minimum mode and the control mode. The damper closes for decreasing temperatures and reaches minimum position at about 12.8 degrees.

[2]In one simulation a rule equivalent to the rule for the preheater has been added. The rule says that the economizer damper will not open until the preheater has closed its vent.

[3]This mode has an additional requirement. The economizer damper have to be in minimum position otherwise will no heating occur and if the damper leave the minimum position then will the preheater mode switch to Off.

trying to maintain the *supply temperature* at a *set point temperature* of 18.3°C see also Table 3.2.

3.2 To Simulate an HVAC System

Simulation of a complete system, see Figure 3.1, as it is done in the following sections shows that it is possible to simulate large systems in real time or quicker. The Reference system involve parts with long time constants on the order of hours to days while other parts have time constants on the order of tens of seconds. To simulate a system like this efficiently needs simplification of parts that are not in focus.

The Reference System

The whole system will be simulated as it is, but only the results from how the AHU system and its controller performs will be presented, see Figure 3.1. Two reference models have been made. One model with proportional control and one with PID controllers. Every unit in both systems such as a damper, vent and fan has its own controller. It means that they can be tuned separately. Tuning has not been carried out fully.

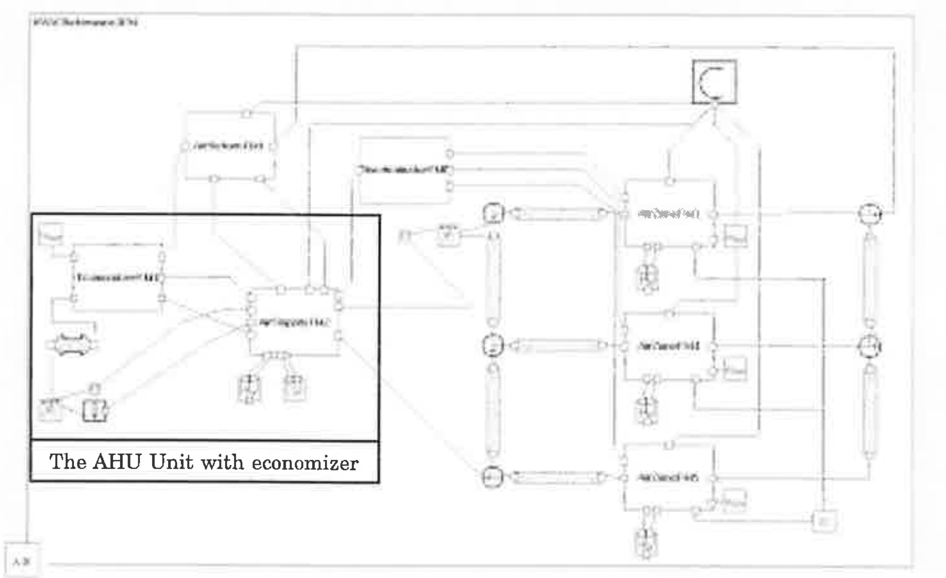

Figure 3.1 The complete Reference System and the AHU sub-system that will be simulated separately.

The performance of simulating a large system shows the capability of the OmSim environment. The difference between having the Reference system and the AHU unit is mainly that the variation in return temperature is fed back to the AHU and that supply mass flow varies . These are constant when only the AHU unit is simulated. Simulation of the whole system when only interested in a part of it is a waste of computing power. Many unnecessary state variables are calculated. If parts not in focus can be simplified e.g., set to be constant etc., then the system can become much smaller and computing more efficient. Therefore a simplified version will be simulated as well. This version consists of an AHU unit including an economizer. Here it will be called the AHU unit. It was created from the Reference System from which unnecessary parts were deleted. These parts such as zones, ducts and air return system etc. have slow dynamics and were replaced with constants. Comparison of results and use of computing power will be made.

The Air Handling Unit

From the reference system, see Figure 3.1, unnecessary models were simply deleted in the Model EDitor (MED) and replaced with relevant constants. This operation was done in a few minutes and resulted in a working sub model. It is almost as easy to build a new model having a library with unit models.

Consider that control of the Air Handling Unit AHU Figure 3.2, Figure 3.3 is to be simulated. In that case most parts have short time constants. Other parts having long time constants can be considered as constant. The parts that have long time constants are especially the walls in the zones τ = 24-72h. Because of the walls the zone also will have long time constants, but on the order of an hour. This leads to that the return air temperature also can be considered as constant. The return air is used by the economizer to mix with air from the outside, see Figure 3.2, and thereby controlling the supply air temperature.

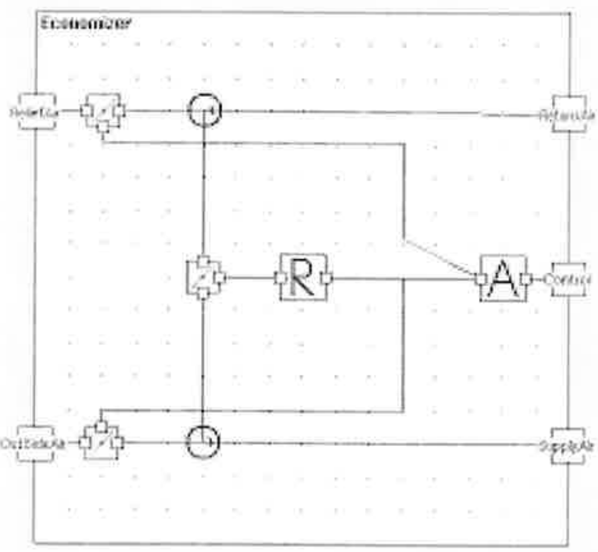

Figure 3.2 The economizer model. One actuator moves the three dampers in unison. One of the dampers is reversed. Return air is split and one portion is mixed with the outside air.

For investigating control it is not necessary that the physical quantities give exactly correct results. It is more important that system time constants are close to real systems. It means that time constants for actuators, vents, fans and heat exchange in heaters and coolers have to be well known. In these investigations actuators have time constants of 60 seconds. It means that large changes can occur within a minute in this system. A simulation should then have a time-scale resolution of a few seconds.

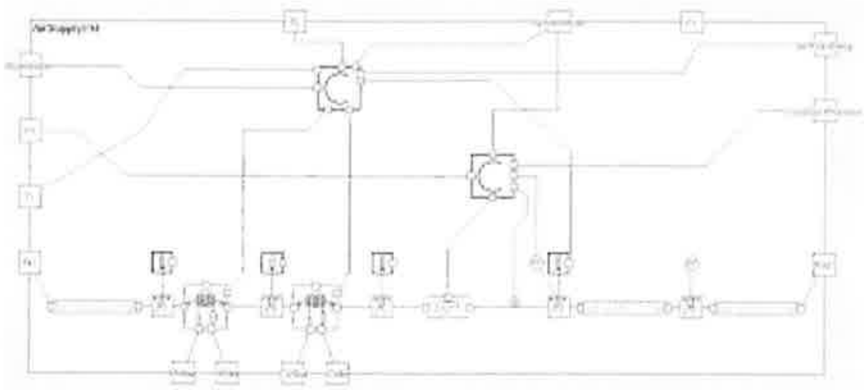

Figure 3.3 The Air Supply model.

The second part of the AHU, the `AirSupply` model was already presented in the introduction. Here you see the ducts, one to the left and two to the right. They inflict transport delays and heat dynamic on the system. Transport delay depends on the flow speed in the ducts and is not more than a few seconds for every ten meters. By considering that the time constants for actuators etc. usually have one of these values $[20, 60, 180]$ seconds, is it necessary to include these delays in the model. Chosen rise-times for actuators, dampers and vents was 60 seconds. Heat dynamics in the ducts depend largely on if the inside of the duct is insulated. In insulated ducts heat interaction can be set to zero. In the AHU there are two heat exchangers. They have time constants on the order of one minute. Even the heat exchanger that is not used, have heat dynamics on the same order. It means that control changes take relatively long time to take effect considering the time constants in the AHU. The fan introduce another kind of dynamics. It changes the temperature indirectly by changing the heat transfer efficiency mainly in the heat exchangers. So an increase in flow causes larger heat flow, but less per unit media, causes the supply temperature to be too low. A decrease in flow give the opposite result. The fan also acts fast as its time constant is only $20-30$ seconds.

Control will be too quick when the delays are not considered, see Figure 1.1. A too quick system will wear out the actuators, wasted energy and cause discomfort to the people. On the other hand will a too slow system at least cause discomfort. This can be solved with a PID controller.

3.3 Simulation with Proportional Controllers on the Reference System

The controller gain settings for proportional control were set high enough resulting in the control problem mentioned in the introduction, see Figure 1.1. Table 3.3 show the gains set on each controller. Some older systems built when electronics was more expensive have only one controller for all dampers and vents in the AHU unit (economizer dampers, preheater and cooler valve). In the simulated system all units have their own controller and they have been given a slightly different gain. For slow loops with long delays the gain was chosen to be smaller.

Unit	Economizer	Preheater	Cooler	Reheater	Damper
Gain K_P [°C^{-1}]	0.5	0.6	0.6	0.006	0.02

Table 3.3 The controller gains in the proportional control case. economizer, preheater and cooler belong to the AHU Unit and reheater and Damper belong to the VAV Unit in each zone.

Simulation with a step temperature variation

It is obvious from Figure 3.5 that this system has control problems when outside temperature is below 18.3°C. In Figure 3.4 a small mixed air temperature variation can be discerned as well as a large supply air temperature variation. For other temperatures there are no problems of that kind. Only when the economizer switches its mode between minimum and maximum open position. Switches during cooler control give a great task for the cooler to adapt. It is clear that if outside temperature would oscillate with a short period in the neighbourhood of these switching temperatures then the cooler would not be able to stabilize the supply temperature, see spikes in Figure 3.5. Figure 3.6 and Figure 3.7 show that the amplitude of the oscillations are highest when the preheater causes the supply temperature to swing the most over the set point temperature. It occurs when the outside temperature is around −2°C. Below that temperature the economizer damper is in minimum position almost all the time and the preheater oscillations are not so large. In other simulations preheater oscillations stopped below −14°C.

In Figure 3.8 the simulation time interval 6000—7000s has been zoomed. According to the Table 3.2 a mode switch will occur if the economizer is more open than the minimum open position. As can be seen in the top figure the economizer opens when the supply temperature exceeds the set point temperature. The preheater mode changes and the preheater valve close. supply temperature drops and passes the set point temperature causing the economizer damper to close again. When closed another preheater mode switch back occur and heating restarts. The cycle is completed and starts over again.

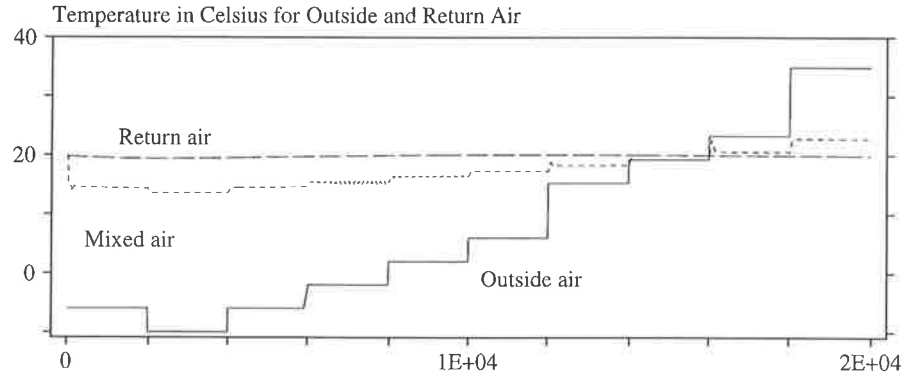

Figure 3.4 Temperature variation that enter the economizer Unit and the resulting mixed air.

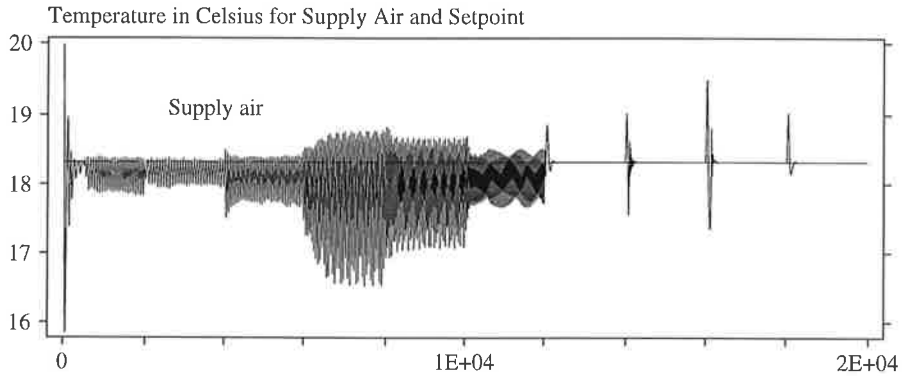

Figure 3.5 Resulting temperature after mixing and heating.

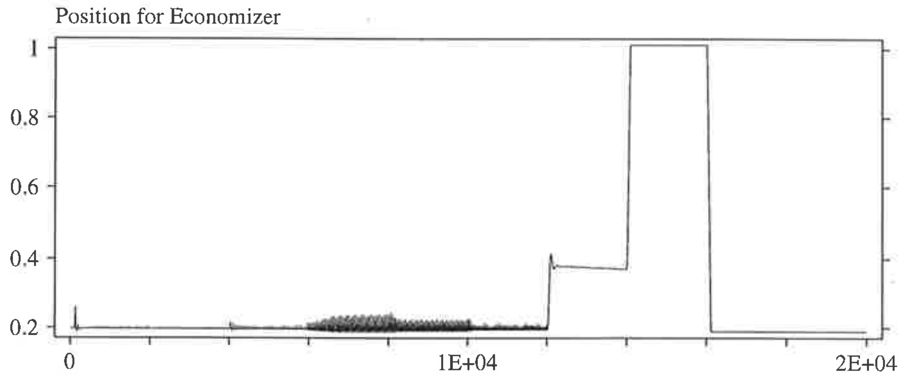

Figure 3.6 Economizer position.

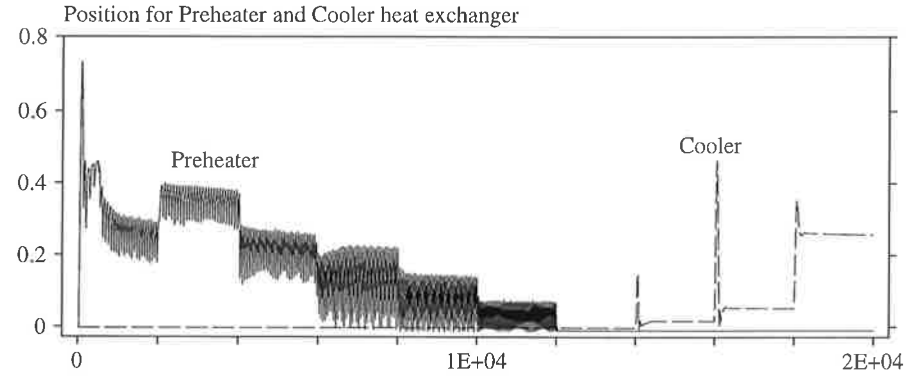

Figure 3.7 Preheater and cooler valve percentage open position.

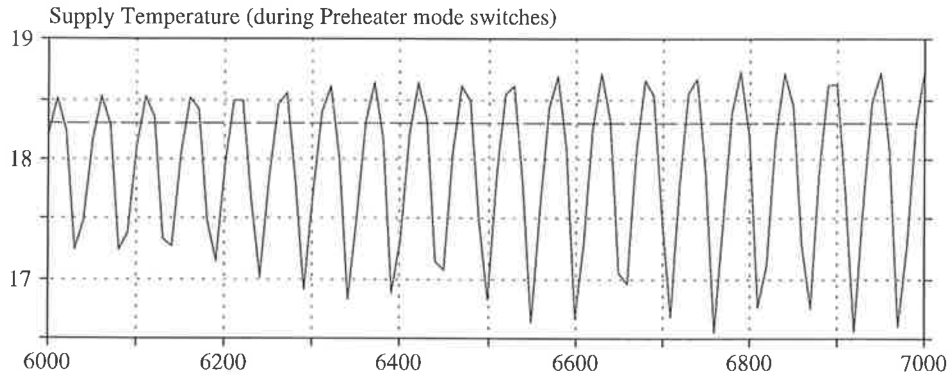

Figure 3.8 Supply air temperature result during preheater mode switches.

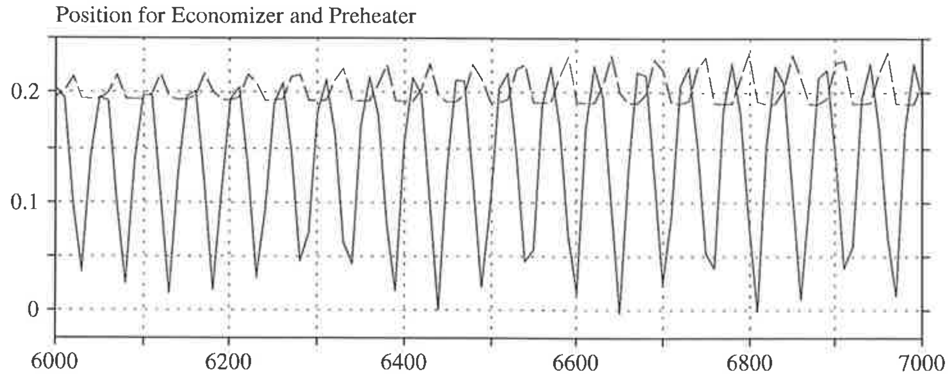

Figure 3.9 Alternating control action by economizer damper and preheater heat exchanger valve.

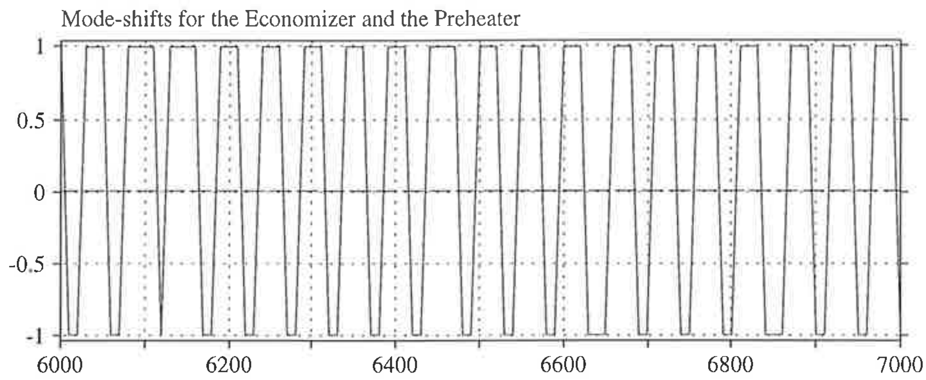

Figure 3.10 Mode switches for the heat exchanger valves.

This was the example of bad control, that was sent by John Seem. Probably the problem would disappear if gain is set lower. Such low gain would cause too slow control. Therefore other methods are needed. One method is to use PID controllers, but there are other Automatic Control methods that can be used.

There are three reasons why the alternation occur; high gain and time delays (transport and heat). High gain causes the supply air temperature to pass the set point too quick and delays result in that the measurement feedback comes too late causing the controller to continue for a longer time than necessary. The third also very important reason is that economizer is allowed to be open before the preheater valve has closed completely. Adding this switch rule that says that preheater and economizer can not be simultaneously open resulted in a behaviour very similar to the results of using an PID controller.

Simulation with an artificial weather temperature

As in the previous simulation, it is seen in Figure 3.12 that economizer and preheater alternation occur at low temperatures and that the cooler has to change valve position quickly at economizer mode switching temperatures. For other temperatures the supply temperature is close to the set point temperature.

The economizer and cooler manage well to keep the temperature close to the set point, which is an effect of that they operate without interaction from each other, see Figure 3.13 and Figure 3.14. The only interaction that occurs is during economizer mode switching as mentioned before. When no mode switching occur their control is smooth.

The economizer damper, see Figure 3.13, is fully open in the beginning until the outside temperature reaches switching temperature. Economizer switching occurs three more times. Here after the cooler finally closes completely and switches its mode to Off. At this point the supply temperature is controlled by the economizer. As the outside temperature continue to drop the economizer closes and reaches minimum open position. At eonomizer minimum position the preheater continues to control supply temperature.

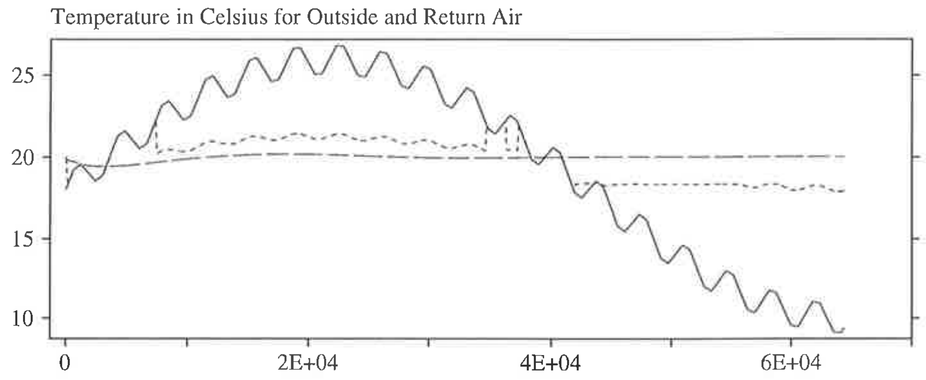

Figure 3.11 Temperature variation that enter the economizer unit and the resulting mixed air.

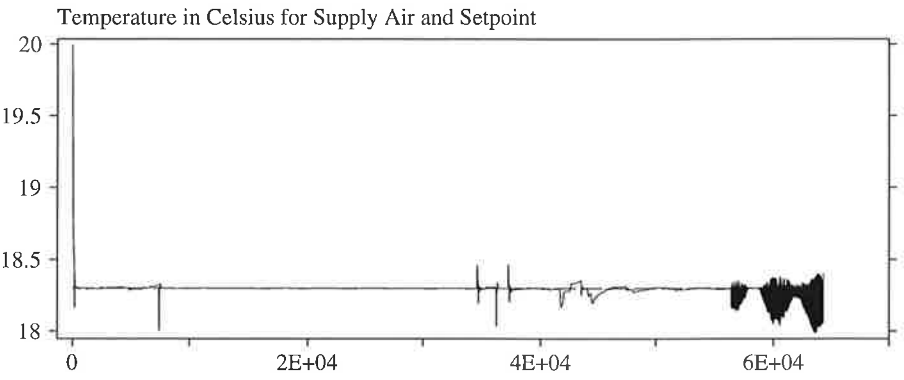

Figure 3.12 Resulting temperature after mixing and heating.

Finally we notice that return air varies so slowly that it has virtually no effect on the control of the supply set point temperature, see Figure 3.11. It also is noticed that mixed air temperature is more dependent of the outside temperature variation than the return air temperature. However, a slightly quicker variation would be expected if the zones would include interaction with the weather and have internal-heat-load-disturbances. The reason why such disturbances will not cause high amplitude and quick rise time is that the zones act as low pass filters. They accumulate heat and their rise times are long compared to those of the AHU unit and economizer.

The largest control problems occur as a result of the switch-over between economizer control mode and preheater control mode. Therefore, another simulation was made with an added switch-over rule. A rule that prevents the damper economizer to open before the preheater valve is completely closed. In the following set of plots it can be seen that control improves and comes close to the result from the simulations with PID controller, see Figures 3.15-3.18. All oscillations have vanished except those

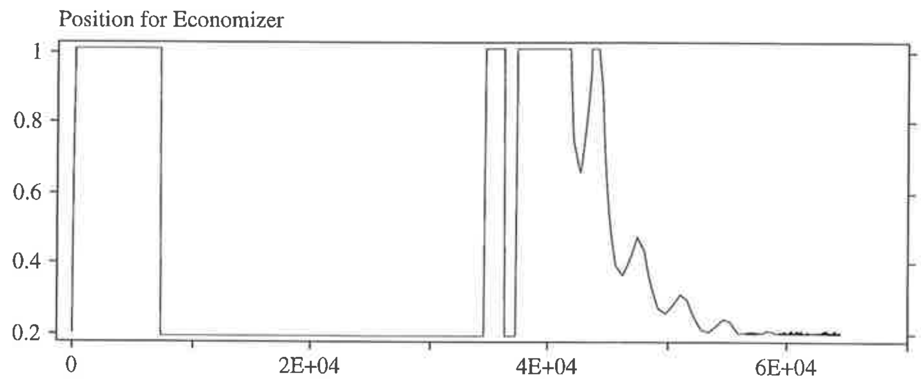

Figure 3.13 Economizer position.

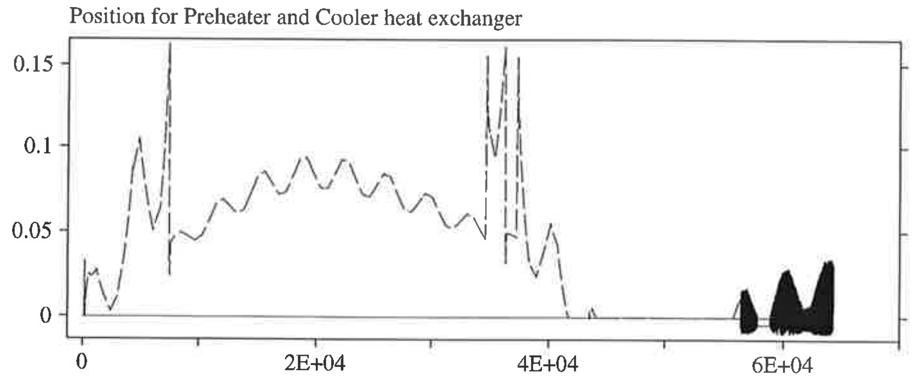

Figure 3.14 Preheater and cooler valve percentage open position.

depending on oscillating outside temperature. Comparison with PID controller simulations will be done in Section 3.5.

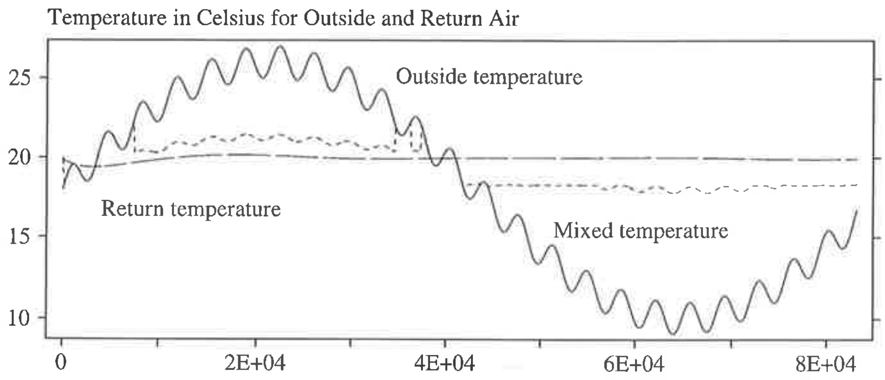

Figure 3.15 Temperature variation that enter the economizer unit and the resulting mixed air.

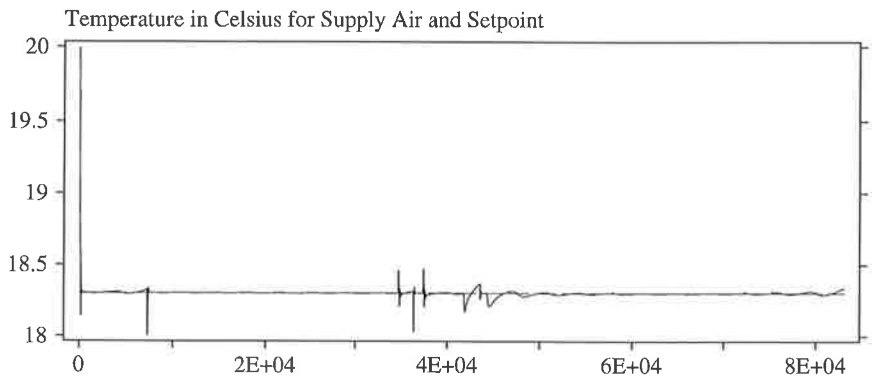

Figure 3.16 Resulting temperature after mixing and heating.

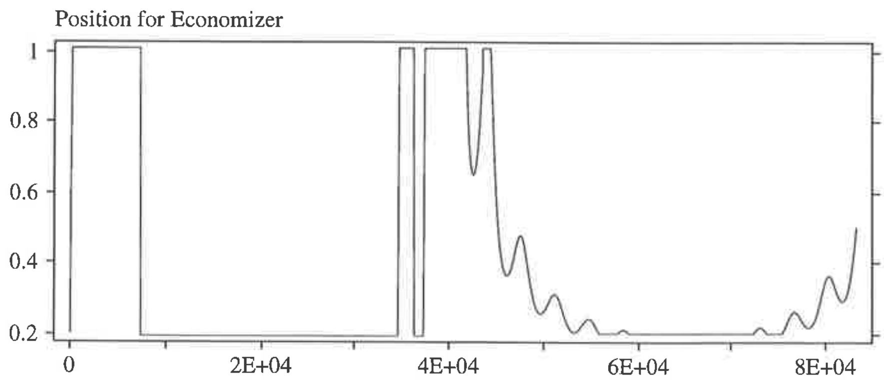

Figure 3.17 Economizer position.

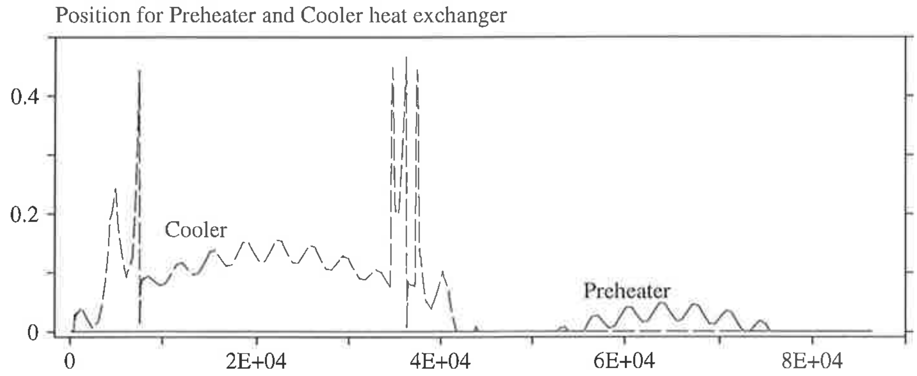

Figure 3.18 Preheater and cooler valve percentage open position.

3.4 Simulation with PID on the Reference System and the AHU Unit

Simulations with the complete reference system and the AHU unit is presented here. The AHU unit is as mentioned before a simplified version of the Reference System.

Unit	K_P [°C^{-1}]	T_I [s]	T_D [s]	N
Economizer (AHU)[4]	0.12	300	30	10
Preheater (AHU)[4]	0.18	200	30	10
Cooler (AHU)[4]	0.24	150	30	10
Reheater (VAV)[5]	0.012	600	30	10
Damper (VAV)[5]	1	100	10	1

Table 3.4 The PID controller settings for both the Reference System and the AHU Unit. Damper derivative is set with high filtering because of that insensitivity to small disturbances stabilizes simulation.

Controller settings in Table 3.4 were obtained by simulation. A step disturbance were sent through one heat exchanger and rise time and delay were measured. Repeating a few times with different gains K_P until a the system was on the border of being unstable. With curve data and the previously found K_P were gain, integration time and derivative time were calculated. PID values were continuously manipulated between simulations resulting in the finally used values in Table 3.4. Little effort has been made to find the best setting. Least effort has been made on the VAV reheater and damper settings. The variations between the units have been found after some test simulations.

[4]This Unit is modelled in the AHU and is also part of the Reference System.

[5]This Unit is modelled as a part of a VAV box controlling the air into a zone. The model is part of the Reference System, but not the AHU System.

Simulation on Reference System with a step temperature variation

In Figure 3.19 an outside step temperature variation can be observed. Controllers manage well to maintain the set point temperature. Influences from the return air is visible. Upsets in supply temperature occur when the economizer damper changes its position at the same time as the heat exchanger vents open/close. Especially when the economizer is switching mode at the same time, see Figure 3.23. A 12°C step in outdoor temperature occurs at $\tau = 18 \cdot 10^3$ seconds, while the cooler mode is in cooling mode. The cooler control action is zoomed in Figure 3.24.

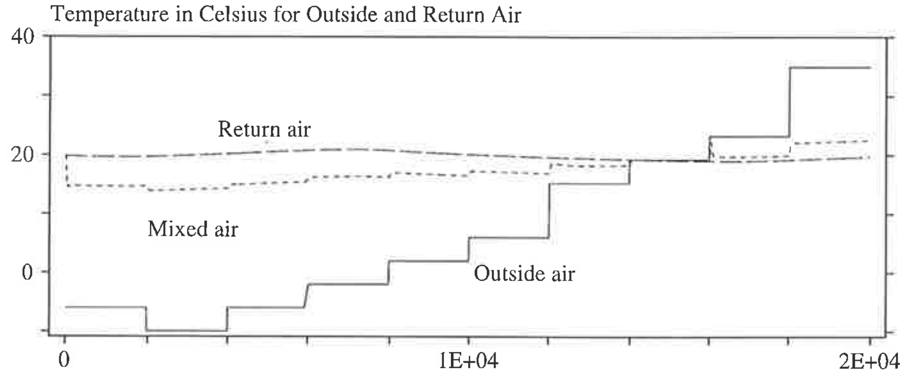

Figure 3.19 Temperature variation that enter the reference system and the resulting mixed air.

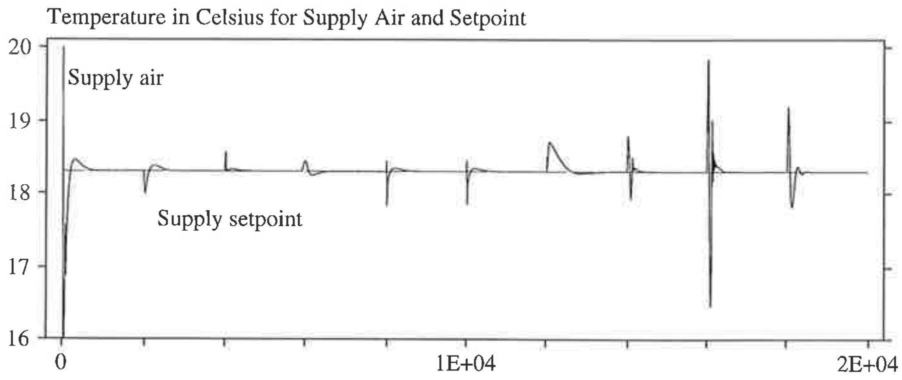

Figure 3.20 Resulting supply temperature after mixing and heating.

At all temperatures no control oscillations can be seen, see Figures 3.21-3.22. In the second Figure it can be seen that control action is done during the whole step interval. This is not due to insufficient control. The control is needed due to the variation in return air temperature, see Figure 3.19. There also the influence on the mixed air can be seen.

The outside step variation in Figure 3.19 cause disturbances on the supply temperature in Figure 3.20. The supply temperature stays close

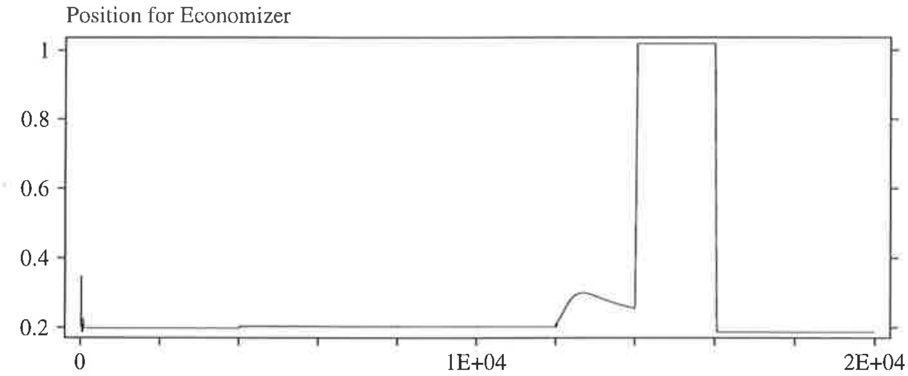

Figure 3.21 Economizer position.

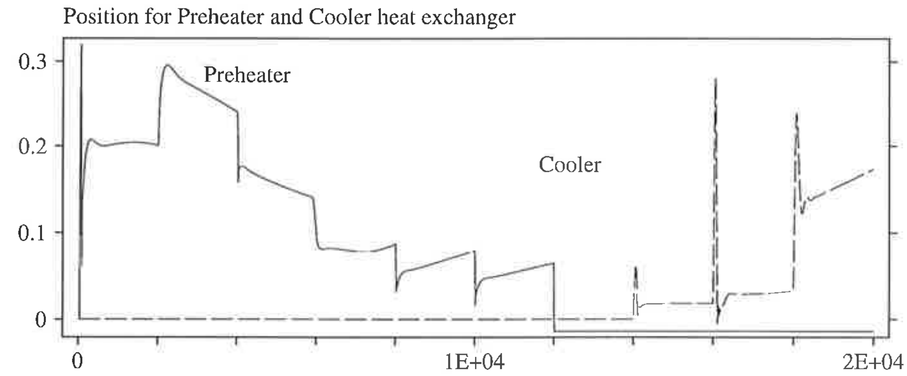

Figure 3.22 Preheater and cooler position.

to the set point temperature except during steps. Most times the settling time is short. It takes less than 60 seconds to reach an error well below 0.1°C. In cases when the economizer also changes its position in the same temperature step is the settling time longer. There are three cases when economizer switches its mode showed in Figures 3.21-3.22; preheater closes at the same time as economizer opens; cooler opens as economizer opens; cooler opens as economizer closes. The fourth case were preheater opens as economizer closes can not happen because of the rule for mode switches given in Table 3.2. The rule prevents the preheater to open while the economizer is still open.

As can be seen in Figure 3.22 there are three peaks on cooler position curve. The rightmost at $\tau = 18 \cdot 10^3$ is due to a 12°C large outside temperature step. Settling time is 500 seconds long for reaching an error less than 0.1°C, see Figure 3.23. Despite the extreme step in outside temperature settles the supply temperature within a comparably short time. Next peak at $\tau = 16 \cdot 10^3$ is the highest and is caused by the simultaneous economizer mode switch from maximum open to minimum open position and the cooler open action, see Figures 3.21-3.22 and Figures 3.25-3.26. The action taken is higher here, but settling time $\Delta\tau = 250$ seconds is shorter. Here

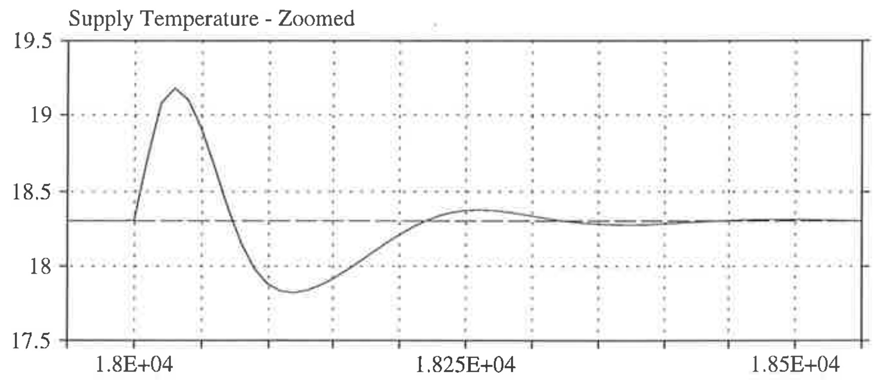

Figure 3.23 Supply temperature as a result of cooler control action.

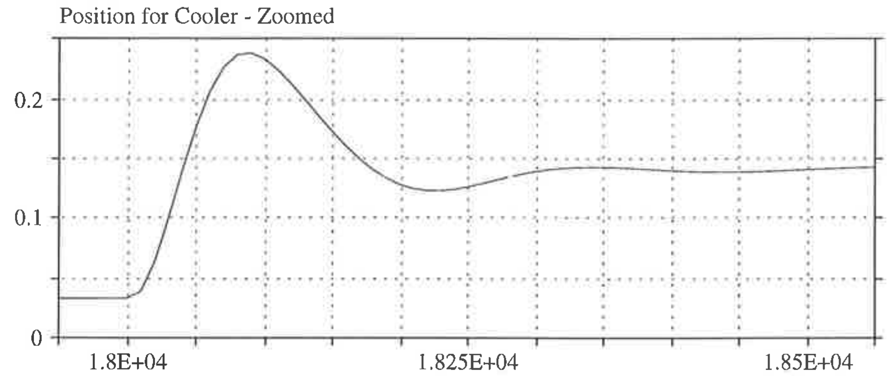

Figure 3.24 Control action taken by the cooler.

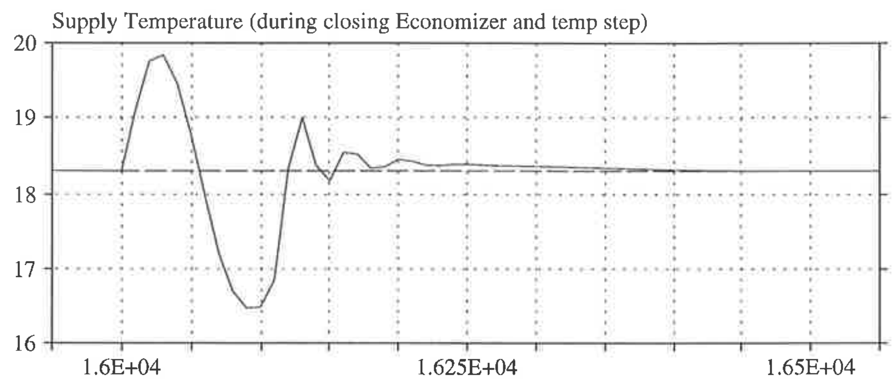

Figure 3.25 Resulting supply temperature during a step in outside temperature and a simultaneous mode change for the economizer. Mode change from max open to min open.

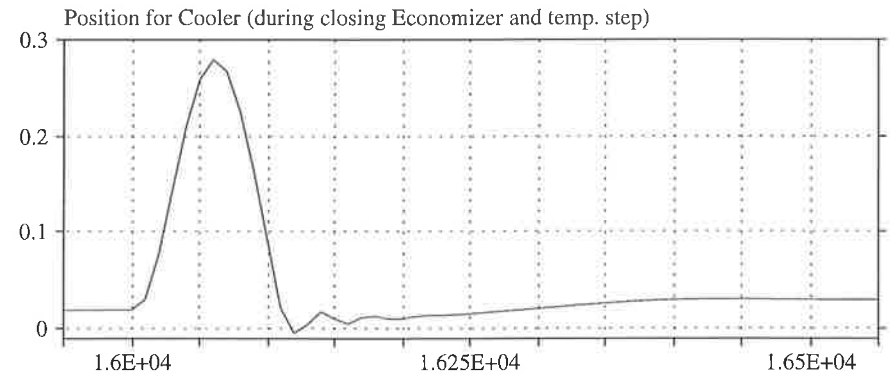

Figure 3.26 Control action taken by the cooler.

the combined control of the economizer and cooler makes the settling time short. At the third peak at $\tau = 14 \cdot 10^3$ both the economizer and the cooler opens making the settling time even shorter. The leftmost case is with closing preheater and opening economizer. Despite cooperative control action the settling time is long. Dynamics in the ducts and the heat exchanger prevent the colder air from outside to influence the supply temperature quickly. Heat exchange from and to the mixed air cause the new air to keep the previous temperature longer time. Transport dynamics influence most on the economizer, because of its placement ten meters from the AHU.

Simulation on Reference System with artificial weather temperature

A PID controller improves control. By studying Figures 3.27-3.30 it can be seen that supply air temperature is close to set point temperature except at a eight instances $\tau \approx (0, 0.73, 34.6, 36.2, 37.2, 41.8, 43.6, 44.2) \cdot 10^3$. At time $\tau \approx 36.5 \cdot 10^3$ the temperature error in supply temperature is a little more than 1°C. Again we see in Figure 3.29 that the economizer mode switches from maximum to minimum at this instance as discussed above. In this case the temperature steps are ruled out from being the cause of the temperature error. Conclusion is then that a better switch-over strategy is needed. Present switch-over strategy gives maximum control signal to the switching unit, that moves with actuator maximum speed until it reaches the new position. Actuator rise time is set to 60 seconds and suggested control would be to have a lower actuator switch speed. It would then be easier for the cooler to maintain set point temperature.

At the five instances $\tau \approx (0, 0.73, 34.6, 36.2, 37.2) \cdot 10^3$ the economizer switches its mode while the cooler maintains the supply temperature. The supply temperature and cooler control at $\tau \approx (34.6, 36.2) \cdot 10^3$ has been zoomed in Figures 3.31-3.32. There it can be seen that economizer switches inflict a disturbance in the supply temperature that takes about 250 seconds to settle below 0.1°C. The disturbance on supply temperature is in the worst moment more than 0.8°C, but only for a short period. In case that outside temperature would oscillate close to switching temperature then it would cause the supply temperature to oscillate with a possible amplitude

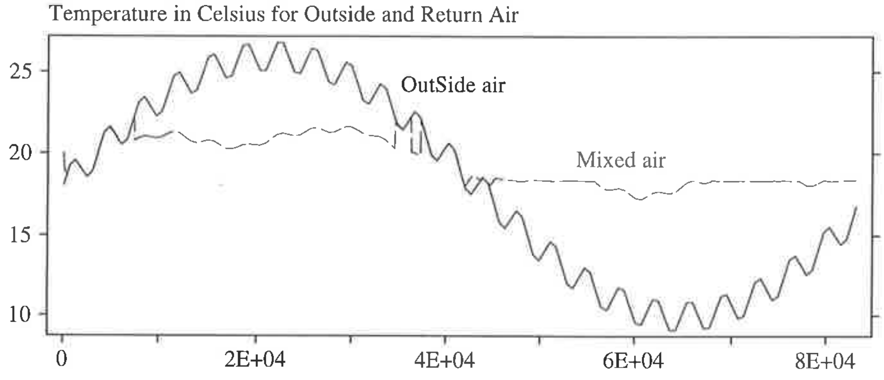

Figure 3.27 Temperature variation enter the economizer Unit and the resulting mixed air.

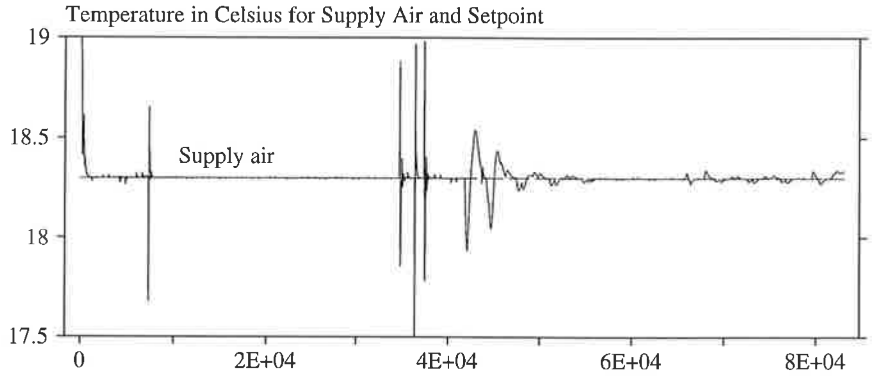

Figure 3.28 Resulting supply air temperature after mixing and heating.

of 1 degree. Such an oscillation with this amplitude would be uncomfortable because of the relatively short period of about 1 minute. If the economizer switch-over time is increased so that the disturbance amplitude in supply temperature $T < 0.1$°C still many switches can occur within a short period of time. By setting a temperature switch-over dead-zone can switch-overs with short time separation be avoided.

In the time interval $[44.1, 55.7] \cdot 10^3$ supply temperature only is controlled by the economizer. It can be seen on the supply temperature in Figure 3.29, that control is slow. A decrease in integration time to $T_{\mathrm{I}} \approx 200$ and/or increase in gain to $K_{\mathrm{P}} \approx 0.18$ would improve control speed.

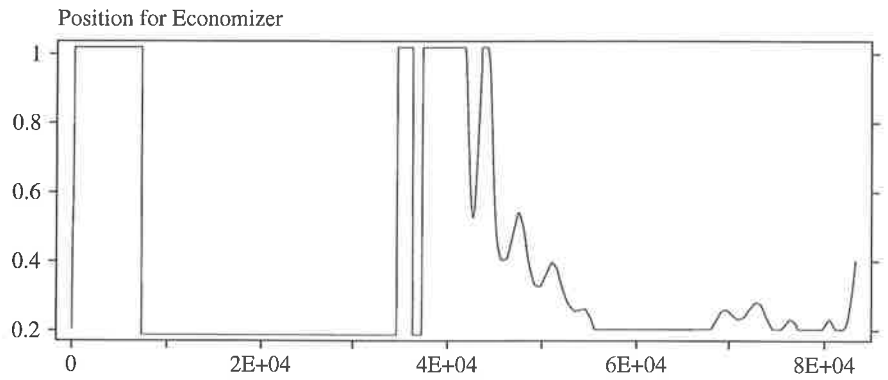

Figure 3.29 Economizer position.

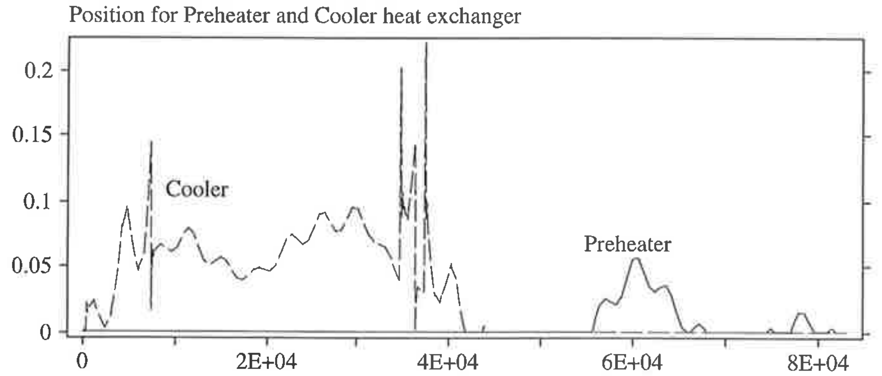

Figure 3.30 Preheater and cooler valve percentage open position.

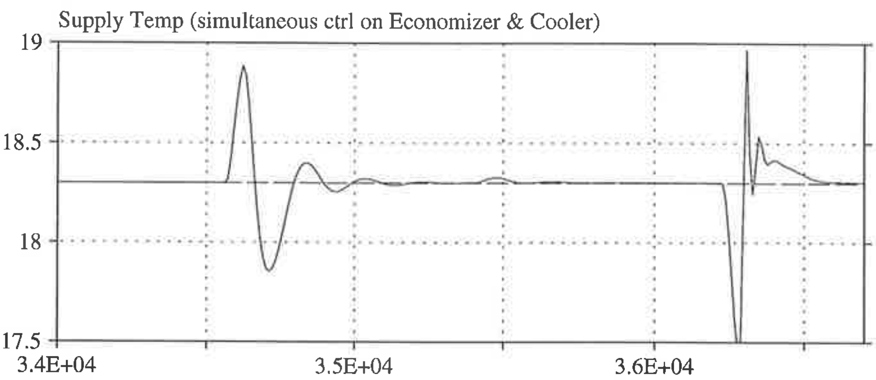

Figure 3.31 Resulting supply air temperature during a step in outside temperature and a simultaneous mode change for the economizer.

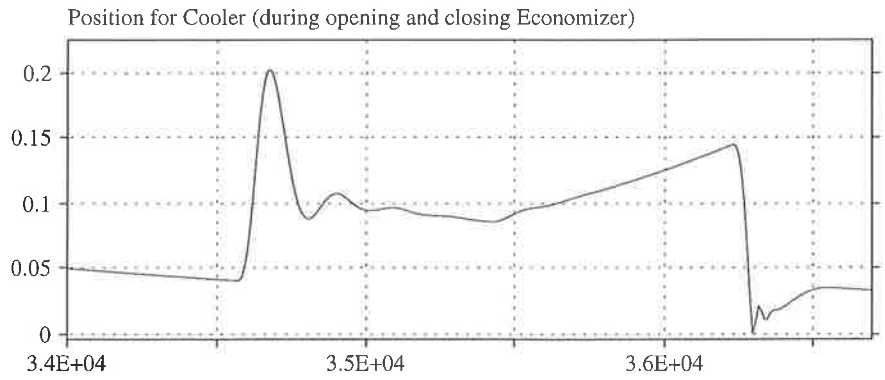

Figure 3.32 Control action taken by the cooler.

Simulation on an AHU Unit with step temperature variation

The simulation of the AHU unit gave as expected almost the same results as the PID controlled Reference System. As all return air, air flow and supply air setpoint dynamics have been replaced with constant values control is expected to be smoother. In Figure 3.33 the mixed air is only dependent of the outside air temperature changes and the economizer control actions.

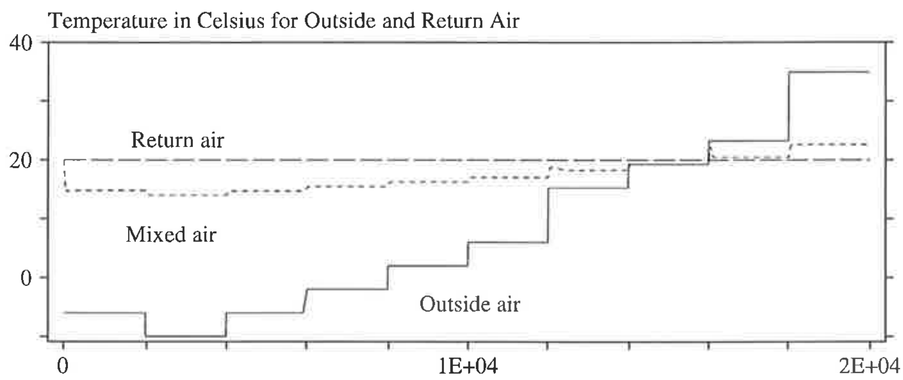

Figure 3.33 Temperature variation that enter the AHU Unit and the resulting mixed air.

If a closer look is taken on Figures 3.33-3.36 and compared with Figures 3.19-3.22 from the PID controlled reference system it can be seen that in the AHU case the supply temperature settles somewhat quicker and closer to the set point temperature. Economizer position in the next figure is somewhat flatter in the time interval $\tau = [12, 14] \cdot 10^3$. In this interval is economizer mode in control mode. The constant return temperature allows the system to settle and therefore no control action is needed after some time. The next plot for preheater and cooler the curves are flatter of the same reason, but otherwise are the figures a close match to the figures from the Reference System. Therefore is the AHU unit as good as the Reference System for investigating control.

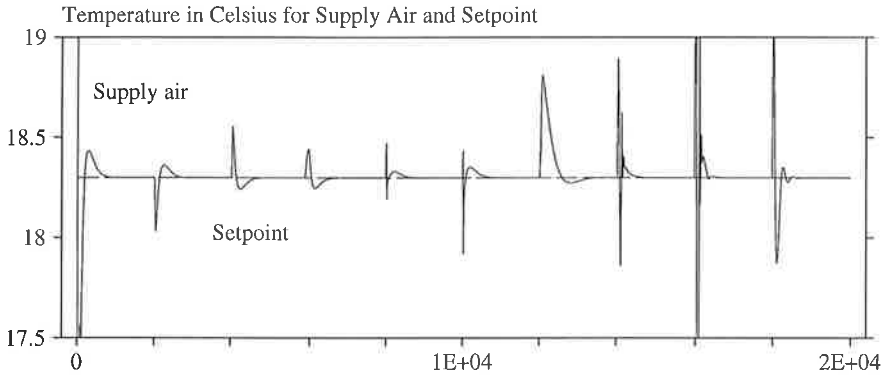

Figure 3.34 Resulting supply air temperature after mixing and heating.

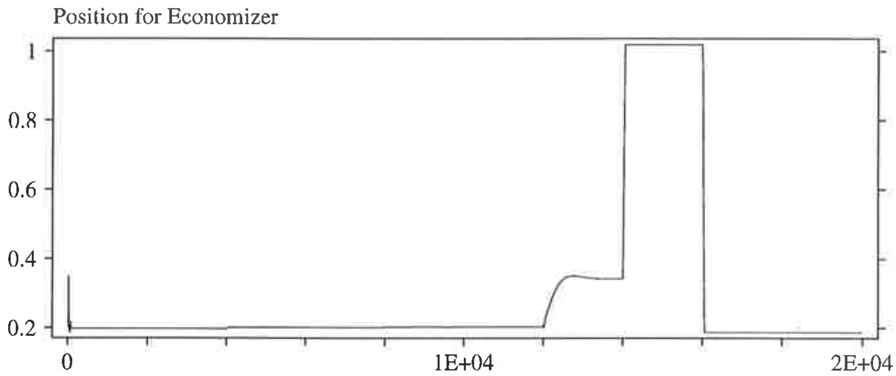

Figure 3.35 Economizer position.

The other discussions about the curves in previous paragraphs apply here too. Only small differences exist.

Simulation on an AHU Unit with artificial weather temperature

A closer look on the Figures 3.37-3.40 and compared with Figures 3.27-3.30 reveal that there are only small differences between the configurations when investigating these variables.

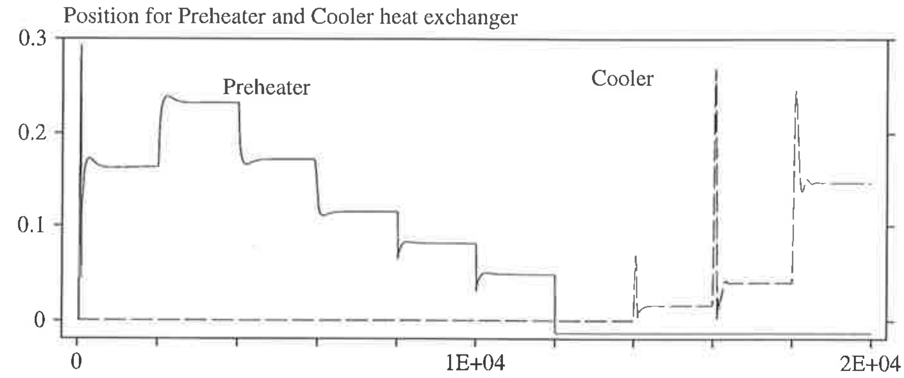

Figure 3.36 Preheater and cooler valve percentage open position.

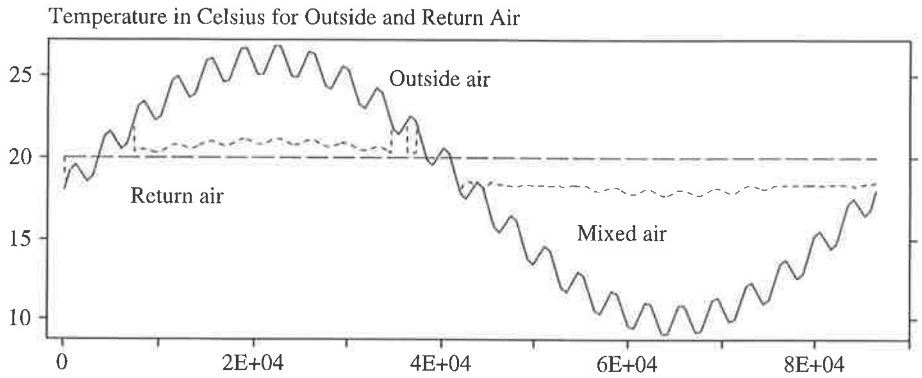

Figure 3.37 Temperature variation that enter the economizer Unit and the resulting mixed air.

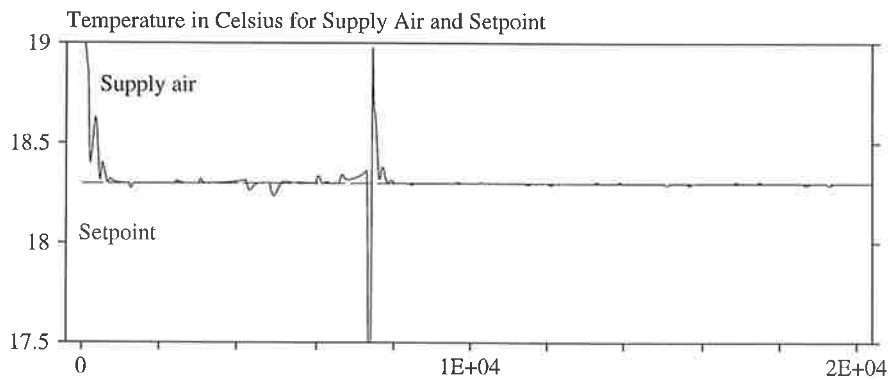

Figure 3.38 Resulting supply air temperature after mixing and heating.

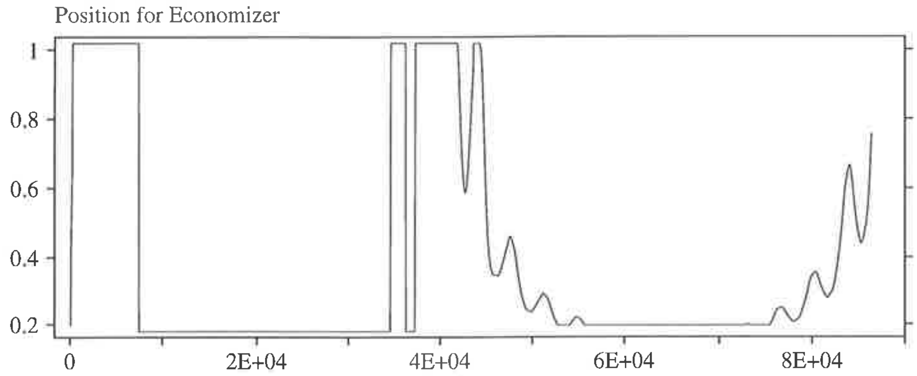

Figure 3.39 Economizer position.

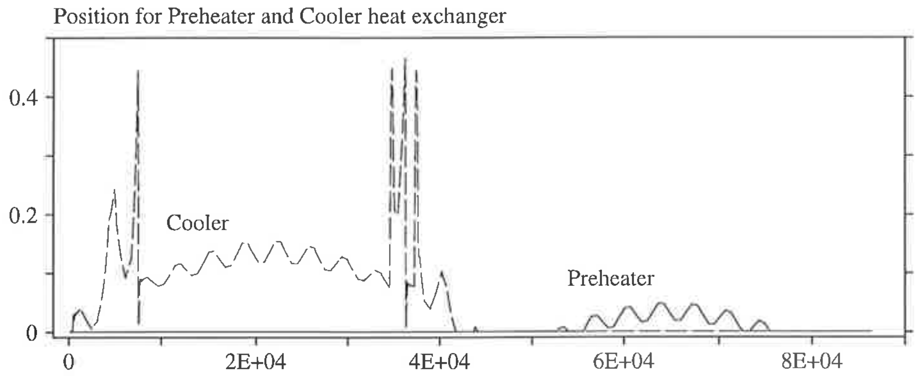

Figure 3.40 Preheater and cooler valve percentage open position.

3.5 Comparisons

Comparison between the simulations from the original system with proportional controller and the same system with PID controller is carried out here. The difference between the simplified PID controlled AHU unit and the full PID controlled system is discussed as well.

Step simulation

Oscillations were the major problem with the proportional system, see Figures 3.4-3.10. They occured because of that control was switching over between economizer and the preheater and the switches were allowed before the preheater valve was closed. These oscillations were successfully removed from the system with PID controllers, see Figures 3.19-3.23.
Other switch-over problems were the same for both controller systems. These problems were caused by the economizer when moving in either direction from minimum to maximum open. During these switches the cooler controller is active and try to compensate the change in mixed air temperature.

In temperature intervals where no switches occur and in the proportional controller case with no low temperature switches both systems manage to keep the temperature within an error $\varepsilon = 0.1$°C.

Artificial weather simulation

Both the proportional and PID case manage well to keep the supply temperature close to set point temperature. Only the proportional controller has the low switching problem as mentioned before. It can also be seen that the non-optimally set PID controller take longer time to settle after high temperature switches (at 22.2°C), see Figures 3.24-3.26.
Comparing the two PID controller systems it can be said that they substantially are giving the same result. The control needed throughout the simulation is the same except for the difference in mixed air temperature, which is an effect of the varying return air in the full system.
Finally there was made an extra simulation with a changed low temperature switch-over rule. It prevents the economizer from opening before the preheater is fully closed. The result shows no oscillations and supply temperature is kept within an error $\varepsilon = 0.1$°C, see Figures 3.15-3.18. Compared with Figures 3.27-3.30 it can be seen that there are only small differences. The largest differences depend on the fact that the controller for the reheaters manage very differently. In the proportional case the return temperature is almost constant $T = 20$°C, while in the PID controller case there is an oscillation with a period of about 6 hours. In Figures 3.41-3.42 the supply air temperature can be compared. In Figure 3.43 it can be seen that a small oscillation is forced onto both supply temperatures. Outside temperature variation cause these oscillations. Important new PID control parameters were used, see Table 3.5. New parameters were chosen when it was seen that better results could be obtained even with only P-control.

Unit	K_P [°C^{-1}]	T_I [s]	T_D [s]	N
Economizer (AHU)	0.12	200	10	10
Preheater (AHU)	0.18	200	10	10
Cooler (AHU)	0.20	100	10	10
Reheater (VAV)	0.006	600	300	10
Damper (VAV)	1	100	10	1

Table 3.5 New PID controller settings for both the Reference System.

The separate AHU unit take much shorter time to simulate. The major reason is that it has less than half the number of dynamic states of the full system. It is also easier to simulate, because of less feedback, e. g., fixed return air temperature. In Appendix B simulation statistics is shown. Among these statistics it says that one simulation time spanned over 86.400s. By checking with the computer time it was found that simulation on the separate AHU unit took 102min = 6120s of real time. Another comparison that can be made is the number of dynamic states.

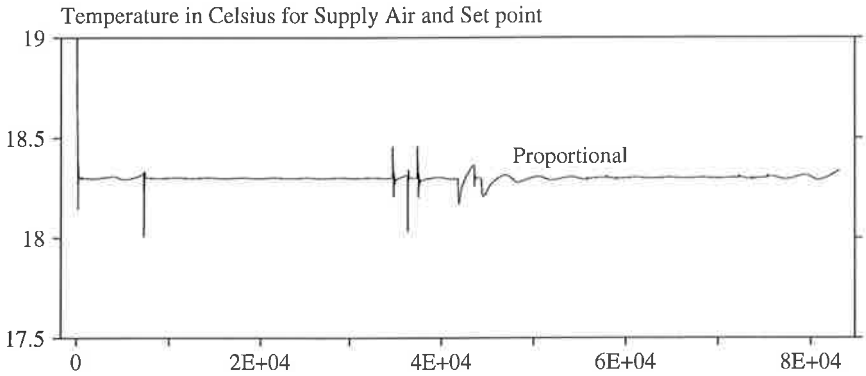

Figure 3.41 Supply temperature in Proportional System.

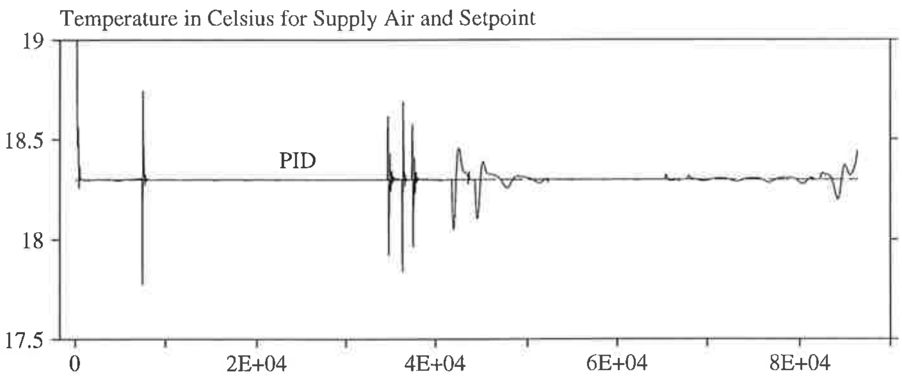

Figure 3.42 Supply temperature in PID Controller System.

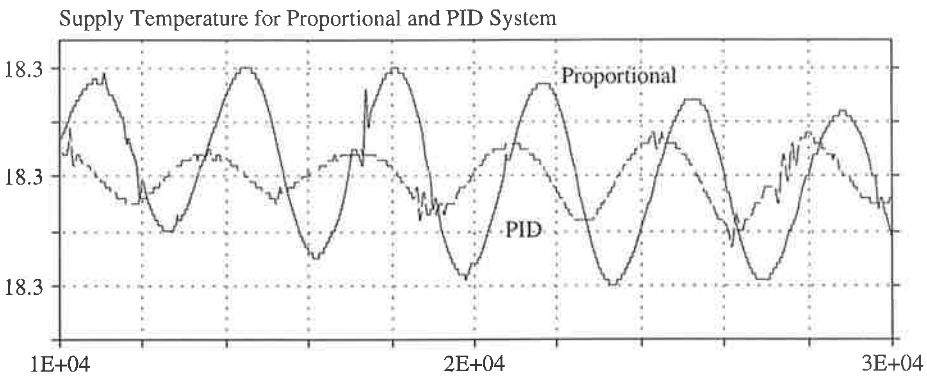

Figure 3.43 Enlargement of Figures 3.41-3.42.

4. Conclusions

In this report the modelling of an HVAC System has been described. Most effort has been concentrated in making OMOLA model libraries for HVAC system, so that dynamic flows could be simulated. The models have been developed using OMOLA - An Object-oriented MOdelling LAnguage. With the newly built library an Heating Ventilation and Air-Conditioning System was modelled from a Reference System specification. This Reference System models a typical HVAC System.

4.1 Simulation results

Simulations show that a dynamic HVAC system can be simulated many times faster than real time, (> 10). Simulations made on the system in the OMSIM working environment gave several results. By simulating the proportionally controlled system the observed oscillating behaviour was reproduced. It was found that the oscillations were generated by the low temperature switch-over rule, see Table 3.2. New simulations were made After changing the control strategy to PID control. Still the switch-over strategy at high temperature disturbed the system. It was solved by slowing down the economizer damper switch-over speed. The short high amplitude oscillation could then be decreased or even completely removed. It also became clear that the economizer would oscillate between minimum and maximum if outside temperature oscillated around the critical switch-over temperature. To avoid these oscillations a dead-band should be introduced. Through these simulations and comparisons it was found that the full Reference system could be simplified to an AHU unit (with economizer). The system could also be used for other HVAC investigations, such as building load calculation.

4.2 Simulation experiences

Experiences from modelling in OMOLA are positive. The language provides a strong mathematical base where equations can be written in their natural form. Other features such as event handling are powerful. Included in the environment are several solving methods and strategies. By having these options available in one simulation tool the time to try out different methods decrease.

During writing the models no consideration of the order of the equations have to be taken. Instead equations are sorted by the OMSIM environment during instantiation of a simulation. The advantage is that models become more flexible. It also takes less time to put a new system together provided that a library already exists. The disadvantage is that OMSIM can not trace back errors to equations written in the models. The reason is that there exist no method to trace back to the written equation after the equations have been manipulated by OMSIM. The flow equation can serve as an example. In the flow equation air mass flow is calculated from the root of the pressure difference. This cause numerical problems when the difference is close to zero (infinitive derivative). Error message tell that too many steps were taken before end of calculation step was reached. It also displays an equation, but no indication on how the error occurred. To avoid simulation problems care has to be taken with problems involving $\sqrt{}$ and ln , see Section 2.3 and 2.4

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A. Listed Omola models

```
HHEXAirCompartmentFM ISA AirCompartmentIC WITH
  %% A control volume model of a gas medium based
  %%   on dynamic energy and mass balances.
  %%
  %% Assumptions: constant volume,
  %%              static gas composition
  %%              homogenous mixed,
  %%              no work interaction.
  %% Model Use:   given mass flow directions,
  %% States:      airmass (m),
  %%              temperature (t).
  %% Medium:      Approximate ideal gas
  %% Model type : full model.
icon:
  Graphic ISA super::Graphic WITH
    bitmap TYPE String := "GasCompartment";
    % AirCompartment
  END;
parameters:
  v ISA Parameter WITH default := 1; END;
  initM ISA Parameter;
  initT ISA Parameter;
  initAh ISA Parameter;
variables:
  mass ISA Variable;% [kg] Air Mass
  t ISA Variable;% [K] Air Temperature
  ah ISA Variable WITH initial := 0; END;
eventvariables:
  init ISAN Event;
terminals:
  Fin ISA super::Fin WITH
    Graphic ISA super::Graphic WITH
      x_pos := 1;
      y_pos := 151;
    END;
  END;
  Fout ISA super::Fout WITH
    Graphic ISA super::Graphic WITH
      x_pos := 400;
      y_pos := 151;
    END;
  END;
events:
  WHEN init DO
    new(mass) := initM;
    new(t) := initT;
  END;
submodels:
```

```
  M ISA AirCompMM;% compartment medium model
  MH ISA AirHeatMM; % heat medium model
  MF ISA AirFlowMM; % flow medium model
parameterpropagation:
  M.modelVolume := v; % compartment volume
equations:
  % ------------------------------------- section pressure
  Fin.p := M.Mout.p;
  Fout.p := M.Mout.p;
  % ------------------------------------- mass balance
  mass' = Fin.w - Fout.w; % Air mass change
  % Change in humidity affects the massbalance
  ah := Fin.ah; % Here only propagation of humidity
  Fout.ah := ah;
  % ------------------------------------- energy balance
  t' = (Fin.w*Fin.t - Fout.w*Fout.t + Qin.R.q/MH.Mout.cp)/mass;
  % [kgK]
  Fout.t := t;
  % Temperature change due to flowrate of heat q and mass dm
  % ------------------------------------- pressure
  % Pressuredrop modeled outside
medium_connections:
  %------------------------- medium state to medium model
  M.min.m := mass;
  M.min.tk := t;
  MH.min.tk := t;
  MH.min.ah := ah;
  MH.min.rair := M.Mout.rair;
  MF.min.tk := t;
  Rho.rho := M.Mout.rho;
  Rho.t := t;
  %------------------------- medium temp to heat transfer
  Qin.R.Tin := Fin.t;
  Qin.R.Tout := Fout.t;
heat_medium_connections:
  Qin.M.p := M.Mout.p;
  Qin.M.w := Fin.w;
  Qin.M.cp := MH.Mout.cp;
  Qin.M.mu := MF.Mout.mu;
  Qin.M.lambda := MH.Mout.lambda;
  Qin.M.ah := ah;
END;

FinTubeHeatResistorFM ISA FinTubeHeatResistorIC WITH
  %%
  %%
  %%
  %%
  %%
icon:
  Graphic ISA super::Graphic;
parameters:
```

```
  Height, TubeLength, TubeDiameter, NoOfFins ISA Parameter;
  CrossSectionPerimeter, CrossArea, FinWetArea, TubeWetArea,
    WaterInnerArea ISA Parameter; % TubeDistance
  TubeWallMass, TubeCapacitance, FinPlateMass, FinCapacitance,
    TubeThickness, FinThickness ISA Parameter;
  WallMass ISA Parameter WITH
    value := TubeWallMass+FinPlateMass;
  END;
  WallCapacitance ISA Parameter WITH
    value := (TubeWallMass*TubeCapacitance+
      FinPlateMass*FinCapacitance)/WallMass;
  END;
  Ef ISA Parameter WITH default := 0.55;END;
    % Overall efficiency factor;
  InitT ISA Parameter WITH
    default := 293.5;
  END;
variables:
  WallTemp ISA Variable;
eventvariables:
  init ISAN EVENT;
events:
  WHEN init DO
    new(WallTemp) := initT;
  END;
submodels:
  U1 ISA HVACBasicLib::AirResistance WITH
    Graphic ISA super::Graphic WITH
      x_pos := 200.0;
      y_pos := 2255.0;
    END;
    d := 4*outer::CrossArea/CrossSectionPerimeter;
    Across := outer::CrossArea*NoOfFins;
    Aheat := outer::FinWetArea + outer::TubeWetArea;
  END;
  U2 ISA HVACBasicLib::FinTubeWallResistance WITH
    Graphic ISA super::Graphic WITH
      x_pos := 200.0;
      y_pos := 150.0;
    END;
    deltaTube := outer::TubeThickness;
    deltaFin := outer::FinThickness;
    areafin := outer::FinWetArea;
    areatube := outer::TubeWetArea;
%    finLength := outer::TubeDistance;
  END;
  U3 ISA HVACBasicLib::WaterResistance WITH
    Graphic ISA super::Graphic WITH
      x_pos := 200.0;
      y_pos := 75.0;
    END;
    d := outer::TubeDiameter - 2*outer::TubeThickness;
```

```
    l := outer::TubeLength;
    Aheat := outer::WaterInnerArea;
  END;
equations:
  % ------------------------------------
  U2.AirConvHeatTrans := 1/(abs(U1));
  % ------------------------------------ Fluid average temp
  % ---------- between inflow and outflow
%  Tm1 := (Q1out.R.Tin + Q1out.R.Tout)/2; % fluid 1 average temp
%  Tm2 := (Q2out.R.Tin + Q2out.R.Tout)/2; % fluid 2
  % ------------------------------------ Temp diff fluid - wall
  dtm ISA LogMean WITH
    x := Q2out.R.Tout - Q1out.R.Tin;
    y := Q2out.R.Tin - Q1out.R.Tout;
  END;
  % ------------------------------------
  % Heat flow out from wall (negative)
  Q1out.R.q := Ef*dtm/(U1 + U2 + U3);
  Q2out.R.q := - Q1out.R.q;
%  Q2out.R.q := dtm2/(U2/2 + U3);
  % ------------------------------------ Wall energy balance
  WallTemp' = (- Q1out.R.q - Q2out.R.q)
              /(WallCapacitance*WallMass);
% Wall temperature is not used -
  % ------------------------------------
  Q1out.M AT U1.Rin;
  Q2out.M AT U3.Rin;
END;

AirFlowResistorFM ISA AirFlowResistorIC WITH
  %% A resistor model of air based on
  %%   static energy and mass balances.
  %%
  %% Assumptions: constant temperature
  %%              no heat interaction,
  %%              no work interaction.
  %% Medium:      air
  %% Model type:  full -
  %%
icon:
  Graphic ISA super::Graphic;
parameters:
  CrossArea ISA Parameter WITH default := 1.0; END;
  InnerArea ISA Parameter WITH default := 1.0; END;
  Volume ISA Parameter WITH default := 1.0; END;
  Length ISA Parameter WITH value := Volume/CrossArea; END;
  diameter ISA Parameter WITH value := Volume/InnerArea; END;
  rho ISA Parameter WITH
    default := HVACBasicLib::AirRho;
  END;
  DuctFrictionFactor ISA Parameter WITH
    default := 1;
```

```
  END;
variables:
  w ISA Variable WITH initial := 0.0; END;
  DeltaP ISA Variable; % Piezometric pressuredrop
terminals:
  Fin ISA super::Fin;
  Fout ISA super::Fout;
submodels:
  zv ISA TubeLossFactor WITH
    Fi ISA TurbulentFrictionFactor;
    diameter := outer::diameter;
    Length := outer::Length;
  END;
equations:
  % --------------------------------------- mass (flow) balance
  Fin.w = w; % mass flow in [kg/s]
  Fout.w = w;
  % --------------------------------------- temperature balance
  Fout.t = Fin.t; % [K]
  % --------------------------------------- humidity balance
  Fin.ah = Fout.ah; % No changes in Resistor
  % --------------------------------------- mechanical energy
  w = CrossArea*sqrt(2*rho/zv)*min(DeltaP,sqrt(DeltaP));
  % min function ensures that sqrt will not
  % influence simulation at low flows
  % --------------------------------------- auxiliary variable
  DeltaP := max(0,Fin.p - Fout.p); % no vertical pressuredrop
                                  % rho*HVACBasicLib::g*DeltaZ;
END;
```

B. Simulation Data

B.1 Simulation Input Data

Artificial weather data file. This file contains only time, a mode variable and temperature. Such a data file can be created in matlab and exported. Any data converted to the OmSim Read File Format can be used.

```
=================================================================
0 0 3
V1
V2
V3
 0 1 18
 600 1 1.921498e+01
 1200 1 1.956327e+01
 1800 1 1.904421e+01
 2400 1 1.852316e+01
 3000 1 1.886549e+01
 3600 1 2.007055e+01
 4200 1 2.127167e+01
 4800 1

         1.592945e+01
 83400 1 1.713451e+01
 84000 1 1.747684e+01
 84600 1 1.695579e+01
 85200 1 1.643673e+01
 85800 1 1.678502e+01
 86400 1 1.800000e+01
=================================================================
```

B.2 Simulation Output Data

The following statistics can be printed from the OmSim simulation environment. Computer time is approximately equal to real time.

```
=================================================================
Model and Simulation Statistics at Simulation Time 86400.0
Date: Sat Dec 19 19:11:28 1998
Simulator_1

Model Complexity:
  Components:    534
  Input Objects: 12
  Terminals:     2671
```

```
  Equations:     2073
  Assignments:   1500

  Variables:     8774
    Parameters:  489
    Dyn. States: 176
    Alg. States: 0
    Discretes:   7
    Globals:     20
    Constants:   4929
    Auxiliaries: 974

Computational Effort:
  Dynamic function evaluations: 484305
  Output function evaluations:  13104
  Stop condition evaluations:   14
  Event stops:                  0
  Rejected initial steps:       0
===============================================================

Computer time 35 minutes
Simulation time 24 hours

===============================================================
Model and Simulation Statistics at Simulation Time 86400.0
Date: Sat Dec 12 22:46:41 1998
Simulator_1

Model Complexity:
  Components:    534
  Input Objects: 12
  Terminals:     2671

  Equations:     2073
  Assignments:   1500

  Variables:     8769
    Parameters:  487
    Dyn. States: 176
    Alg. States: 0
    Discretes:   7
    Globals:     20
    Constants:   4926
    Auxiliaries: 974

Computational Effort:
  Dynamic function evaluations: 833298
  Output function evaluations:  13104
  Stop condition evaluations:   14
  Event stops:                  0
  Rejected initial steps:       0
===============================================================
```

```
Reference model with Proportional Control and no simultaneous
control of the economizer and preheater

===============================================================
Model and Simulation Statistics at Simulation Time 20000.0
Date: Tue Dec 15 03:56:55 1998
Simulator_1

Model Complexity:
  Components:     1178
  Input Objects: 13
  Terminals:      6005

  Equations:      4631
  Assignments:    3549

  Variables:      19189
    Parameters:   1101
    Dyn. States: 360
    Alg. States: 0
    Discretes:    34
    Globals:      20
    Constants:    10435
    Auxiliaries: 2306

Computational Effort:
  Dynamic function evaluations: 6561089
  Output function evaluations:  5602
  Stop condition evaluations:   6
  Event stops:                  0
  Rejected initial steps:       0
===============================================================
Computer time 734 min
Simulation time 333 min


===============================================================
Model and Simulation Statistics at Simulation Time 86400.0
Date: Wed Dec 16 23:41:23 1998
Simulator_1

Model Complexity:
  Components:     1178
  Input Objects: 13
  Terminals:      6005

  Equations:      4631
  Assignments:    3549

  Variables:      19191
    Parameters:   1102
    Dyn. States: 360
```

```
    Alg. States: 0
    Discretes:   34
    Globals:     20
    Constants:   10436
    Auxiliaries: 2306

Computational Effort:
  Dynamic function evaluations: 981773
  Output function evaluations:  17606
  Stop condition evaluations:   19
  Event stops:                  0
  Rejected initial steps:       0
==============================================================
computer time 125 min = 7500 seconds
simulation time 86400 seconds
```