18th IEEE International Conference on Control Applications
Part of 2009 IEEE Multi-conference on Systems and Control
Saint Petersburg, Russia, July 8-10, 2009

Simplified Optimal Control in HVAC Systems

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Abstract— **This paper presents simplified optimal control of a heating, ventilating, and air-conditioning (HVAC) system. This HVAC system is a typical one composed of two heat exchangers: an air-to-air heat exchanger (a rotary wheel heat recovery) and a water-to-air heat exchanger. First the optimal control strategy which was developed in [1] is adopted for implementation in a real life HVAC system. Then the bypass flow problem is addressed and a controller is introduced to deal with this problem. Finally a simplified control structure is proposed for optimal control of the HVAC system. Results of implementing the simplified optimal controller show all control objectives are met (The cost function consists of electrical and thermal energy consumption by the HVAC system).**

I. INTRODUCTION

A great part of the produced energy in the world is consumed by heating, ventilating, and air conditioning (HVAC) systems. Due to the huge energy costs and the shortage of energy supplies efficient control of HVAC systems is getting more and more attention. Optimal control of HVAC systems in Denmark (five million inhabitants) can result in saving of up to 100 GWh energy per year [2].

Maintaining thermal comfort and energy efficiency are two primary goals in the development of control modules for HVAC systems. Furthermore, control modules have to perform in such a way to guarantee that the operation of the HVAC system does not violate any building regulations. Thus, developing an optimal control strategy for HVAC systems is a constrained optimization problem. The optimal control of HVAC systems have been considered extensively in the literature e.g. [4], [5]. However, barely any of them has been applied as a real life optimal control strategy in HVAC systems. In this paper the optimal control structure which was developed in [3] is examined and analyzed for implementation in real life situations. The simplified optimal control structure that is presented as a final development in this paper can be applied as a substitute for one the most common control strategies in HVAC systems, called mixing control strategy, while it will have better performance in terms of energy consumption.

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TABLE I
NOMENCLATURE

q_a	inlet or outlet air flow (m^3/h)
$TE21$	outdoor air temperature (^oC)
$TE22$	temperature of outdoor air after heat recovery (^oC)
$TE11$	room air temperature (^oC)
$TE12$	temperature of room air after heat recovery (^oC)
q_{wt}	water flow of the tertiary circuit (l/h)
q_{ws}	water flow of the supply (primary/secondary) circuit (l/h)
$Twin$	tertiary supply water temperature (^oC)
$Twout$	tertiary return water temperature (^oC)
$Tinlet$	temperature of the supply air (^oC)
$Tpin$	primary/secondary supply water temperature (^oC)
$Tpout$	primary/secondary return water temperature (^oC)
$\dot{m}_{ws}$	supply water mass flow rate (Kg/h)
$\dot{m}_{wt}$	tertiary water mass flow rate (Kg/h)

In Section II, the HVAC system is briefly described. The practical optimal control strategy and the implementation of the optimal controller are presented in Section III. Section IV discusses the bypass problem and the way to deal with that problem. Section V presents the simplified optimal control structure and the implementation results. Finally Section VI explains the energy saving aspects.

II. THE HVAC SYSTEM EXPLANATION

The considered HVAC system is a typical HVAC system composed of two heat exchangers: an air-to-air heat exchanger and a water-to-air heat exchanger.

The air-to-air heat exchanger is a rotary enthalpy wheel which plays the heat recovery role (illustrated in Fig. 1). The rotor control comprises a gear motor with frequency converter. Two fans are installed to produce the desired inlet and outlet air flow.

Fig. 2 shows the water-to-air heat exchanger. A variable speed pump supplies hot water to the coil. The speed change of the variable speed pump provides the mean to control the tertiary flow. The primary/secondary flow is controlled by a motorized valve. Tertiary circuit is hydraulically decoupled from the primary/secondary circuit through the bypass pipe.

III. OPTIMAL MODEL-BASED CONTROL

This section first briefly reviews the optimal control strategy and the optimal dynamic control of the HVAC system which were developed in [1] and [3]. Then the result of the implementation of the optimal controller is presented.

A. Optimal Control Strategy

Many industrial processes like HVAC systems work in their steady state conditions most of the operation time. In this situation the main task of the controller is to reject the disturbances which act upon the process. To optimize the

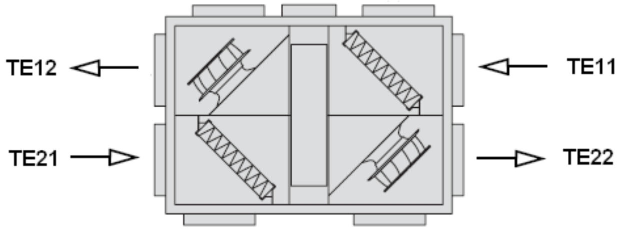

Fig. 1. The Air-to-air Heat Exchanger Scheme

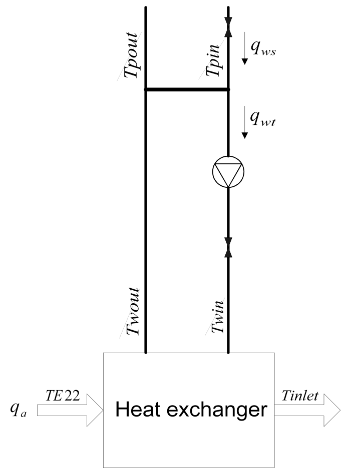

Fig. 2. The Water-to-air Heat Exchanger Scheme

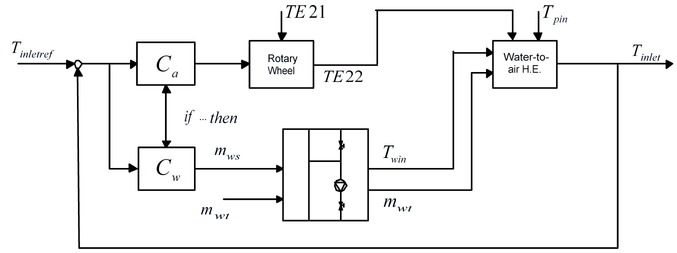

Fig. 3. The Current Control Scheme

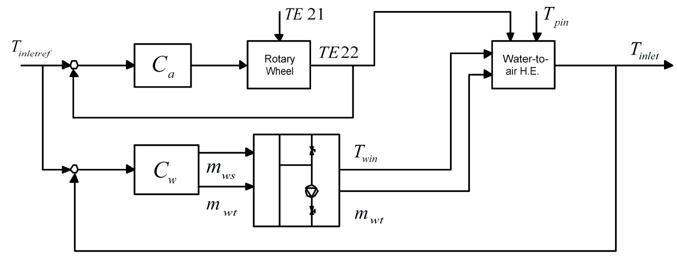

Fig. 4. The Optimal Control Scheme

performance of this kind of systems, applying the steady state model of the process in the optimality analysis is objective. Following this approach for the optimality analysis of the mentioned HVAC system results in two criteria that have to be respected by the controller to guarantee optimal performance of the system [1]:

1) The maximum possible exploitation of the air-to-air heat exchanger has to be achieved.
2) In the steady state conditions supply water flow (q_{ws}) must be equal to the tertiary water flow (q_{wt}). That is, it is optimal to make the system work in a way that no water passes through the bypass pipe. However, it is not possible to eliminate the bypass pipe because it makes the tertiary hydronic circuit hydraulically decoupled and it is necessary to keep the bypass to remove fast disturbances.

Here the term 'optimal performance' means minimum thermal and electrical energy consumption by the HVAC system while it is maintaining the thermal comfort.

B. Optimal Dynamic Control

The dynamic modeling and the controller design procedure of the HVAC system are described in [3]. Fig. 3 and 4 show the mixing control scheme and the optimal control scheme, respectively.

The current system is equipped with a constant speed pump. So, the tertiary water flow (q_{wt}) is not controllable and has to be set to its maximum value to meet the maximum heat demand. The temperature of the hot water to the coil (and as a result the inlet air temperature) is controlled by the motorized primary valve. Two controllers, the heat recovery wheel controller and the water-to-air heat exchanger controller, communicate with one another through some if-then rules to exploit the heat recovery wheel as much as possible. As a result, the first optimality criterion is fulfilled; however, there is no chance for the satisfaction of the second optimality criterion.

The proposed optimal controller in [3] meets both optimality criteria. Moreover, the optimal controller is simpler because two controllers, the heat recovery wheel controller and the water-to-air heat exchanger controller, are completely independent.

The rest of the paper is dedicated to point out some issues about controlling the water-to-air heat exchanger which are important in real life implementation.

C. Controller Implementation and Results

When zero voltage is applied to the variable speed pump, the pump will keep running at a specified minimum speed and will result in non-zero (minimum) tertiary flow (q_{wt}). Therefore, to reach the tertiary flow which is less than the minimum flow the pump has to be pulsed. Pulsing of the pump will cause some problems:

- Short-circuit in the bypass pipe when the pump stops (delivering hot water to the return supply water that is not acceptable in hydronic systems)
- Possible change in supply tertiary water temperature because of pump starts and stops
- Possible oscillations around the set-point ($Tinlet = T_{ref}$) due to pump pulsing
- Adding a pulse modulating board to the pump will impose more initial cost to the system

Due to the above troubles pulsing the pump is not a good solution. Thus, a simple and practical solution is to follow the mixing control strategy when the applied voltage to the pump is zero and there is less demand for the heat. The mixing control strategy stands for the control strategy where the tertiary flow (q_{wt}) is constant and the inlet air temperature is controlled by changing the supply water flow (q_{ws}) through motorized primary valve.

The proposed controller in [3] also shows satisfactory results while following the practical optimal control strategy. Therefore, there is no need to design a new controller to follow the practical optimal control strategy. Fig. 5 and 6 illustrate the result of implementing the controller on the HVAC system. Fig. 5 shows the controller perfectly tracks the set-points. Looking at the the tertiary flow curve reveals that when the set-point changes from 23^oC to 19^oC, the controller switches from the optimal control strategy to the mixing control strategy and when the set-point changes from 19^oC to 22^oC the switching from mixing control strategy to optimal control strategy happens. The switches between the two different control strategy can be inferred by analyzing the temperature measurements shown in Fig. 6 too. The controller also shows good performance in the sense of disturbance rejection. The disturbances from outdoor air temperature (shown in Fig. 5) and the primary supply water temperature (shown in Fig. 6) are perfectly compensated.

IV. Bypass Flow Problem

In this section the bypass flow problem is defined and the implicit measuring of the bypass flow is presented. Then the way to deal with this problem is discussed.

A. Measuring The Bypass Flow

It is not reasonable to measure the bypass flow through a flow-meter in real life HVAC systems. However, we can measure the bypass flow implicitly through thermocouples. This way of measurement is acceptable due to the cheap price of thermocouples.

First the bypass flow has to be defined as a quantity. Consequently, we will always treat bypass flow as a difference between the supply water flow and the tertiary water flow ($q_{ws} - q_{wt}$). That is, when the primary flow is greater than the tertiary flow the bypass flow will have a positive sign and when the primary flow is less than the tertiary flow a negative sign will accompany the bypass flow. These two cases are considered as follows:

- Negative Bypass Flow
  When there is a negative bypass flow the return primary water temperature ($Tpout$) and the return tertiary water temperature($Twout$) are equal. However, the supply primary water temperature ($Tpin$) is always greater than the supply tertiary water temperature ($Twin$). The difference between two recent temperatures is proportional to the ratio of the primary water flow and the tertiary water flow:
  The energy balance equation for the supply water side will result in the following equation:

$$(q_{wt} - q_{ws})\ Twout\ +\ q_{ws}\ Tpin\ =\ q_{wt}\ Twin \qquad (1)$$

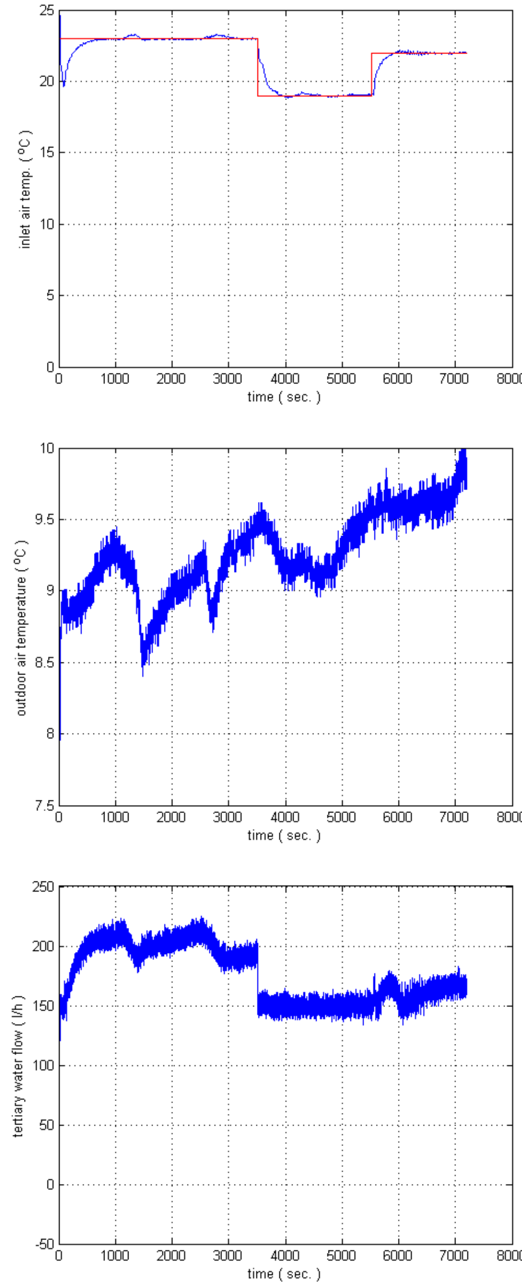

Fig. 5. The Implementation Result of The Practical Optimal Controller (a)

By rearranging the above equation we will have:

$$\frac{q_{wt}}{q_{ws}} = \frac{Tpin\ -\ Twout}{Twin\ -\ Twout} \qquad (2)$$

We subtract 1 from both side of the equation. So,

$$\frac{q_{wt}}{q_{ws}} - 1 = \frac{Tpin\ -\ Twin}{Twin\ -\ Twout} \qquad (3)$$

- Positive Bypass Flow
  The story of the negative bypass flow is similar to the one of the positive bypass flow. Thus, When there is a positive bypass flow the supply primary

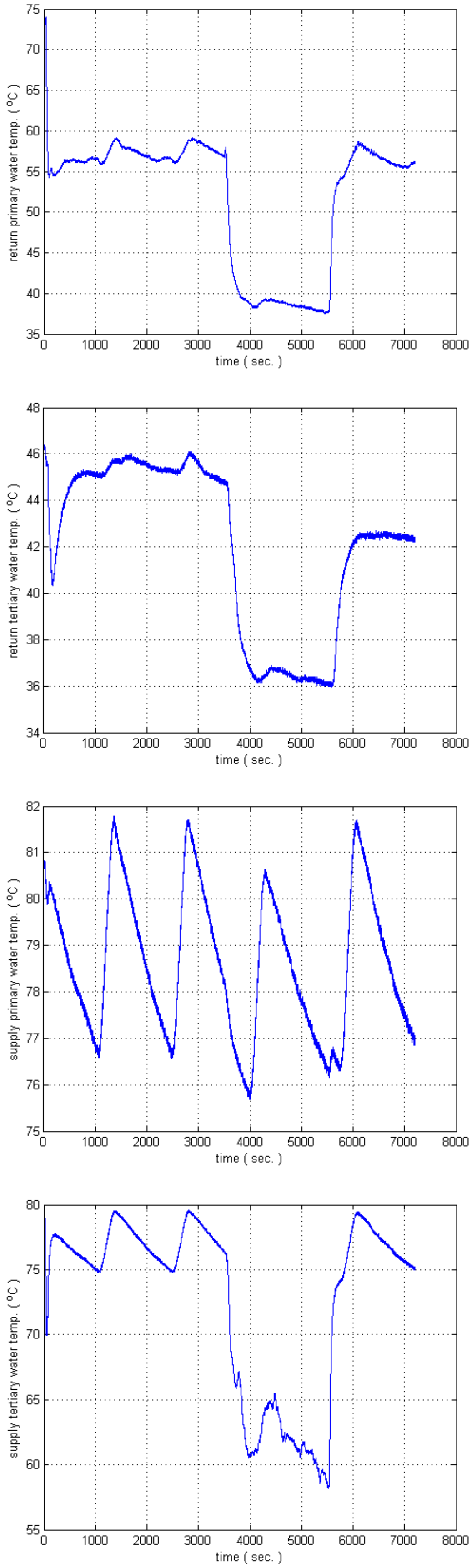

Fig. 6. The Implementation Result of The Practical Optimal Controller (b)

water temperature ($Tpin$) and the supply tertiary water temperature($Twin$) are equal. Nevertheless, the return primary water temperature ($Tpout$) is always greater than the supply tertiary water temperature ($Twout$). Again, the difference between two recent temperatures is proportional to the ratio of the primary water flow and the tertiary water flow:
The energy balance equation for the return water side will result in the following equation:

$$(q_{ws} - q_{wt})\ Tpin\ +\ q_{wt}\ Twout\ =\ q_{ws}\ Tpout \quad (4)$$

By rearranging the above equation we will have:

$$\frac{q_{wt}}{q_{ws}} = \frac{Tpin\ -\ Tpout}{Tpin\ -\ Twout} \quad (5)$$

We subtract 1 from both side of the equation. So,

$$\frac{q_{wt}}{q_{ws}} - 1 = \frac{Twout\ -\ Tpout}{Tpin\ -\ Twout} \quad (6)$$

The recent equation can be rewritten as

$$\frac{q_{wt}}{q_{ws}} - 1 = \frac{Twout\ -\ Tpout}{Twin\ -\ Twout} \quad (7)$$

According to the above discussion and combining equations 3 and 7 we will have:

$$\frac{q_{wt}}{q_{ws}} - 1 = \frac{(Tpin\ -\ Twin)\ +\ (Twout\ -\ Tpout)}{Twin\ -\ Twout} \quad (8)$$

or equivalently:

$$\frac{q_{wt}}{q_{ws}} - 1 = \frac{(Tpin\ -\ Twout)\ -\ (Twin\ +\ Tpout)}{Twin\ -\ Twout} \quad (9)$$

To avoid singularity in the recent equation it should be used as the following equation:

$$\frac{q_{wt}}{q_{ws}} - 1 = \frac{(Tpin\ -\ Twout)\ -\ (Twin\ +\ Tpout)}{|Twin\ -\ Twout - 1|\ +\ 1} \quad (10)$$

So, by measuring the four temperatures ($Tpin$, $Tpout$, $Twin$, and $Twout$) and using the recent formula there will be enough information for a controller to manipulate the bypass flow.

Remark: Those four thermocouples which measure $Tpin$, $Tpout$, $Twin$, and $Twout$ should be installed far enough from the bypass pipe to avoid disturbances due to heat conductance of pipes.

B. Problem Definition

As it was mentioned, Fig. 6 illustrates the four water temperatures around the bypass pipe (supply primary water temperature ($Tpin$), supply tertiary water temperature ($Twin$), return primary water temperature ($Tpout$), and return tertiary water temperature ($Twout$)). Comparing $Tpout$ and $Twout$ reveals that apart from the time that the controller follows the mixing control strategy there is always a small short circuit (positive bypass flow). That is, the return supply water is warmed up. In this case, not only the controller stays

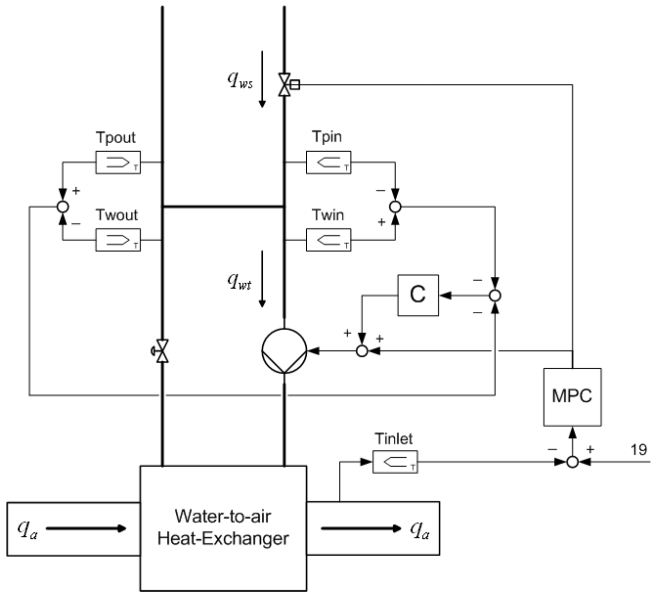

Fig. 7. The Control System along with The Bypass Compensator

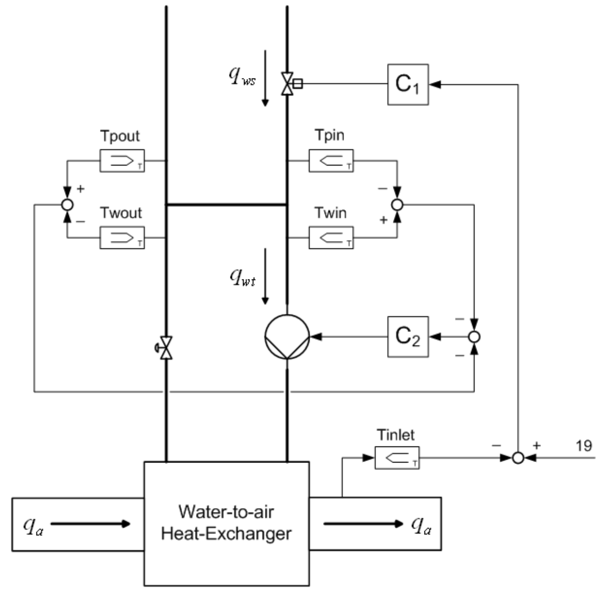

Fig. 8. Simplified Optimal Control Scheme

away from the optimal performance but also it violates the constraints (Copenhagen Building Regulations). Thus, it is vital to solve the bypass flow problem.

C. Bypass Flow Compensation

The control system structure along with the bypass flow compensator is shown in Fig. 7. In this control structure the main controller (the MPC controller) is as same as before and the bypass compensator which is slower than the MPC controller deals with the bypass flow problem. According to the equation 10 a simple PI controller can be applied as the bypass flow compensator.

Remark: The PI controller which controls the bypass flow has to be along with an anti-windup module because for some time that the main controller follows the mixing control strategy there will be accumulation of the bypass flow error.

V. Simplified Optimal Control Structure

Fig 8 shows the simplified optimal control structure. One PI controller (C_1) determines the primary water flow (q_{ws}) through the information from the inlet air temperature feedback. To design this controller the linearized model from the primary water flow to the inlet air temperature [3] has been used. Tertiary water flow (q_{wt}) is controlled by a PI controller (C_2) which tries to keep the tertiary water flow close to the primary water flow. Actually C_2 is as same as the bypass compensator but C_2 is a fast controller here. The variable speed pump acts as an actuator to control the tertiary water flow. Because the variable speed pump is much faster than the primary valve which acts as an actuator to control the primary water flow, the two controllers are decoupled in time domain again. Here also for the same reason that was mentioned before the pump compensator has to be equipped with the anti-windup module.

The results of applying the recent control system to the HVAC system is shown in Fig 9 and 10. As can be seen, the control system has perfect tracking of the set-point. The control system also shows good disturbance rejection (disturbances from outdoor air temperature and the hot water temperature have been successfully rejected). Either the tertiary water flow curve or the water temperature curves around the bypass pipe obviously reveal the switch between two control strategies.

VI. Energy Saving Aspects

The energy consumption of the pump is proportional to the third power of the flow. Mixing control strategy which requires constant tertiary water flow will impose continuous circulation of 300 l/h of water through the coil. However, the optimal control strategy plays with the tertiary water flow as the heating demand (thermal load) changes. The tertiary water flow curve in Fig. 9 illustrates this fact. Therefore, applying the optimal control strategy instead of the mixing control strategy will save up to 82% energy consumption of the tertiary pump. The recent energy saving figure is based on the Blue Angel profile (used in the German energy labeling scheme).

VII. Conclusions

The optimal model-based controller which had been developed in [3] was implemented while following the practical optimal control strategy. Going through the bypass flow problem and its solution led to a simplified optimal controller. Implementation results for the recent controller showed fulfillment of the control goals.

VIII. Acknowledgments

The authors would like to thank financial supports and contributions of Danish Energy Net, Aalborg University, Center for Embedded Software Systems (CISS), Grundfos Management A/S, Exhausto A/S, and Danish Technological Institute (DTI) to this project.

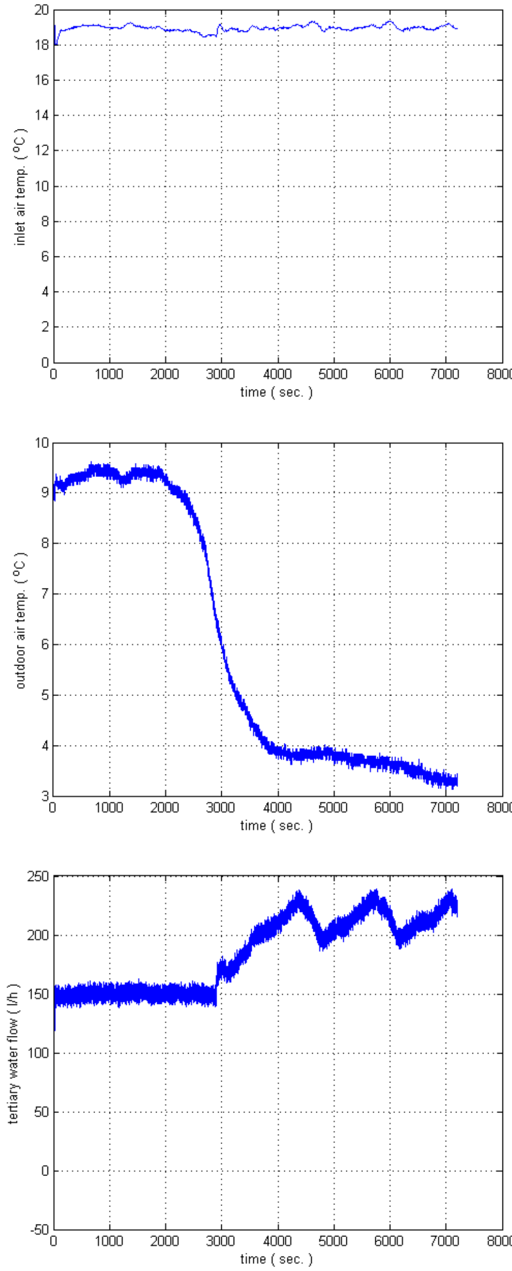

Fig. 9. Results of Applying Simplified Optimal Control System(a)

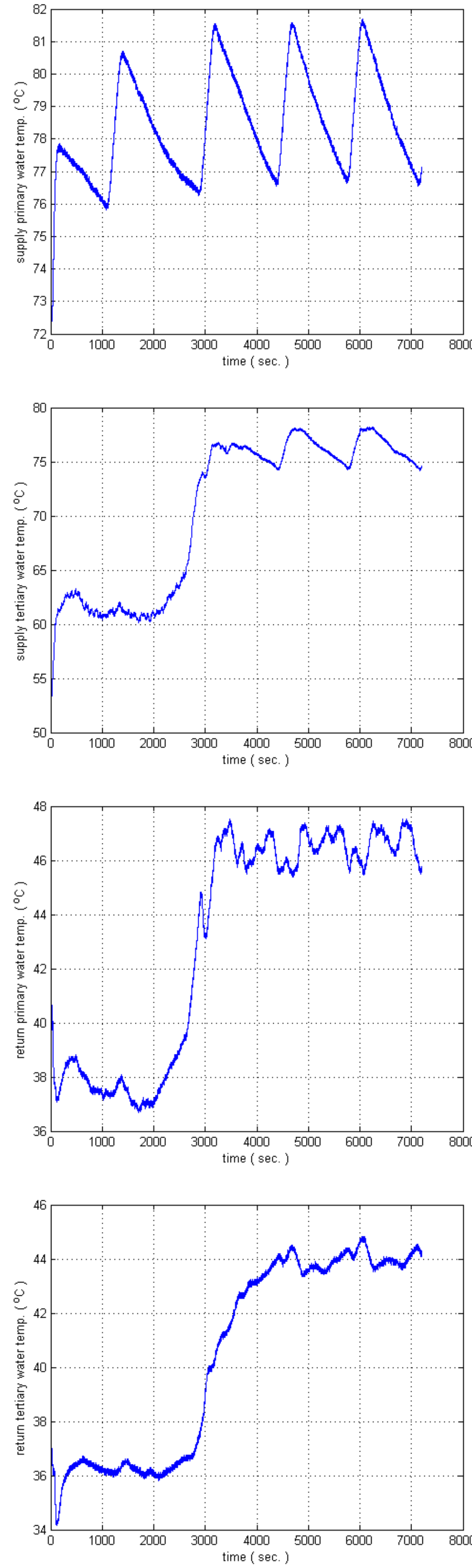

Fig. 10. Results of Applying Simplified Optimal Control System(b)

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