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Modeling and Control of Indoor Climate Using a Heat Pump Based Floor Heating System

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***Abstract*—Modeling and control of an indoor climate using a floor heating system based on a heat pump is discussed. The thermodynamic models of the heat pump, room, floor and circulated water are theoretically developed and then identified through experimental data. Several control strategies, namely relay control, P control, PID control and PID control with pre-filtering of reference input, are developed and simulated based on the developed system model. The simulation results turn out that the PID control with pre-filtering strategy generated the best system performance in terms of the smallest overshoot, fast response, highest COP in the transient period, and the smallest deviation and the least sensitivity to disturbance in the steady period. Meanwhile, this strategy also uses least power compared with other strategies. Instead of using the standard relay control, this study reveals a huge potential to use advanced control techniques to optimize operation of this kind of thermal system from thermal comfort and energy consumption perspectives.**

I. INTRODUCTION

The purpose of Heating, Ventilating and Air Conditioning (HVAC) control systems is to keep people comfortable in terms of desired temperature, humidity, pressure and air quality etc. within an enclosed space [4], [8]. In latest decade almost all new single-family houses built in Denmark have been equipped with floor heating systems instead of conventional radiator systems. However, due to the huge heat capacity of the floor system where the circulating water pipes are often embedded inside a thick concrete layer, the floor heating system has a very slow response to any changes of desired or ambient temperatures. This can often cause large oscillation in the indoor temperature that may lead to residential discomfort [11]. Thereby some proper control strategies are required in order to keep the floor heating operation at a satisfactory level [6], [11].

The use of heat pumps as an effective indoor heating resource is gaining market as days go by due to its relatively low energy consumption and especially the potential mitigation of global warming [1], [7], [9]. In order to minimize the energy consumption and CO^2 emission of heating and refrigerating, a heat pump should operate almost at its highest efficient point, i.e., the highest Coefficient Of Performance (COP) [14]. So far there are extensive work about modeling, simulation and control of floor heating systems, heat pump systems, or other kinds of HVAC systems. Some sophisticated simulation tools are also available [8], such as HAMLab [13].

In the following we consider an indoor heating system consisting of a floor heating system and a heat pump together. A challenging question arises as to how to control this kind of HVAC system so as to ensure optimal energy use and meanwhile keep sufficient thermal comfort [4], [14]. As studied in [7], to optimize a heat pump operation is a complicated issue. It relates not only to the features of different refrigerants used in the system, the exergy efficiencies of different components and cycles, but also to thermodynamic performance parameters, such as COP, heating/cooling capacities. For our considered system and problem, besides the challenging optimization issue of the heat pump, the slow response feature of the floor heating system and the residential thermal comfort are also needed to be taken into consideration for the control design.

The objective of this work is to set up a mathematical model of an indoor climate using a heating system which consists of a floor heating and a heat pump, and then investigate the potential improvement of system performance using feedback control mechanisms w.r.t. the energy consumption and residential thermal comfort. A simple heat pump model proposed in [13] is extended as the model for our considered heat pump. The thermodynamics of the floor and room are modeled as linear first-order systems, respectively. The system coefficients are identified through experiments. A set of feedback controllers, namely relay control, P control, PID control and PID control with pre-filtering, are developed and simulated for the combined total system. From the point of views of thermal comfort and energy consumption, different control strategies are compared and discussed when the ambient temperature could change. The results turn out that the strategy using a PID control with pre-filtering consumes least energy and keeps the most comfortable temperature level, provided the same testing conditions for all kinds of controllers.

The rest of the paper is organized as: Section II introduces a mathematical model of a vapor compression heat pump after presenting its functional principle; Section III sets up thermal models of a closed room equipped with a floor heating system by combining theoretical and experimental ways; Section IV simulates and discusses system performances under different control strategies; finally we conclude the paper in Section V.

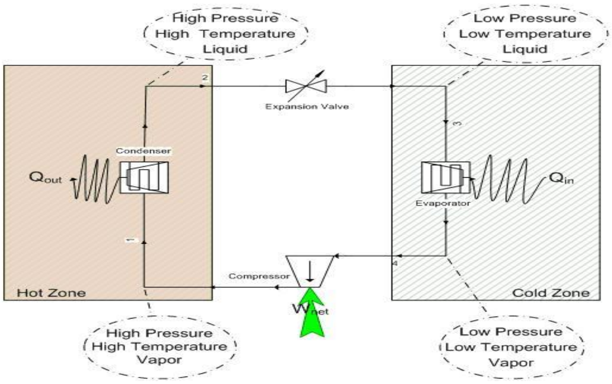

Fig. 1. Schematic diagram of a vapor compression heat pump

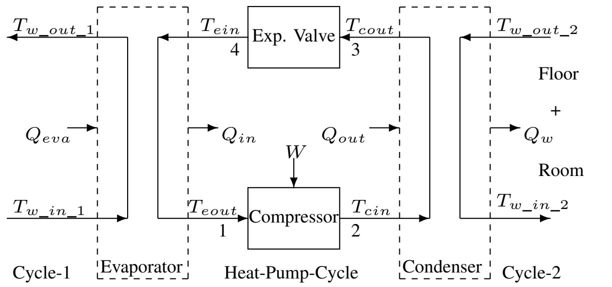

Fig. 2. Schematic diagram of a heat pump model

II. HEAT PUMP PRINCIPLE AND MODELING

A. Heat Pump Principle

A heat pump is a device that delivers an amount of heat from a lower temperature level to a higher temperature level. It reverses the natural heat flow direction. Here we consider a vapor compression of heat pump. As shown in Fig.1, a vapor compression heat pump is composed of four components:

- a compressor;
- a condenser which converts a substance from its gaseous state to its liquid state;
- an expansion valve; and
- an evaporator which converts a substance from its liquid state to its gaseous state.

The refrigerant flows through these components following a reverse Rankine cycle [5]. The low pressure and low temperature liquid refrigerant is fed into the evaporator where it gains an amount of heat (Q_{in}) and evaporates. The gaseous refrigerant is then fed into the compressor where it is compressed due to the extra work (W_{net}) and thus the temperature and pressure rise. The high pressure and high temperature gaseous refrigerant is then fed into the condenser where it release an amount of heat (Q_{out}) and then condenses into high pressure and high temperature liquid state. The high pressure and high temperature liquid refrigerant then passes through an expansion valve and changes to be low pressure and low temperature liquid. This cycle repeats. More information about heat pump can be found from [5].

The efficiency of heat pump is evaluated by its COP, which is defined as the ratio of the amount of heat transferred by a heat pump to the net work done to the compressor. In practice, the COP could be approximated using the temperatures of the cold and hot zones, i.e.,

$$COP \hat{=} \frac{Q_{out}}{W_{net}} \approx \frac{T_H}{T_H - T_C}, \tag{1}$$

where T_H and T_C are the hot and cold zone temperatures, respectively. In the following we focus on a steady state heat pump model developed in [12] at this initial stage. We would leave the exploration of a detailed dynamic model, such as the transient model used in [3]for our future work.

B. Heat Pump Modeling

All variables and coefficients used to model a heat pump are listed in Table 1.

Notation	Description
T_{cin}	Temp. of refrig. into Condenser ($^\circ$C)
T_{cout}	Temp. of refrig. out of Condenser ($^\circ$C)
T_{ein}	Temp. of refrig. into Evaporator ($^\circ$C)
T_{eout}	Temp. of refrig. out of Evaporator ($^\circ$C)
Q_{in}	Heat transferred into the HP cycle (J)
Q_{out}	Heat transferred out of the HP cycle (J)
Q_{eva}	Heat transferred into the Evaporator (J)
Q_w	Heat transferred out of the Condenser (J)
W	Work done to the HP cycle (Nm)
K	Efficiency of the Compressor
C_e	Heat capacity of the Evaporator ($\mathrm{J/kg \cdot K}$)
C_c	Heat capacity of the Condenser ($\mathrm{J/kg \cdot K}$)
C_f	Heat capacity of the refrigerant ($\mathrm{J/kg \cdot K}$)
$\dot{m}_{w1}$	Massflow rate of refrigerant in evaporator (kg/s)
$\dot{m}_{w2}$	Massflow rate of refrigerant in condenser (kg/s)
m_e	Mass of evaporator (kg)
m_c	Mass of condenser (kg)

Table 1 Parameter list for heat pump modeling

As shown in Fig.2, besides the functional heat pump cycle which circulates refrigerant, two external cycles - "water" cycle 1[1] and water cycle 2 are also attached to the evaporator and the condenser, respectively. Regarding the "water"-cycle relevant to the condenser, we assume that the temperature of the refrigerant that goes out (into) of the condenser at point 3 (2) in the heat pump cycle, denoted as T_{cout} (T_{cin}) is the same as the temperature of the water going out (into) of the condenser denoted as $T_{w\_in\_2}$ ($T_{w\_out\_2}$) in the water-cycle 2. By looking at the parts relevant to the evaporator, we assumed the temperature of the refrigerant that goes into the evaporator at point 4 in the heat pump cycle, denoted as T_{ein} is the same as the temperature of the water coming into the evaporator in the water-cycle 1, denoted as $T_{w\_in\_1}$. Based on these assumptions, there are

$$T_{w\_in\_1} = T_{ein}, \quad T_{w\_in\_2} = T_{cout}, \quad T_{w\_out\_2} = T_{cin}. \tag{2}$$

With respect to the approximation in (1), the COP of the heat pump cycle can be further approximated by [12]

$$COP = K \frac{0.5(T_{cin} + T_{cout}) + 273.15}{0.5(T_{cin} + T_{cout}) - 0.5(T_{ein} + T_{eout})}. \tag{3}$$

where K is the efficiency coefficient of the compressor. Notation $\dot{X}$ is used to represent $\frac{dX}{dt}$ for a time-dependent variable X in the

[1] It is not necessarily water since it depends on how the heat pump interacts with cold zone.

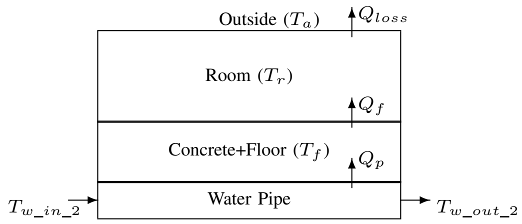

Fig. 3. Schematic diagram of a floor heating room

following. From the fundamental definition of (1), there is

$$\dot{Q}_{out} = COP\dot{W}. \tag{4}$$

Considering the energy balance $\dot{Q}_{out} = \dot{Q}_{in} + \dot{W}$, there is

$$\dot{Q}_{in} = (COP - 1)\dot{W}. \tag{5}$$

According to the thermodynamic theory [5], the heat rate $\dot{Q}_{eva}$ gained by the evaporator from Water Cycle 1 is described by

$$\dot{Q}_{eva} = \dot{m}_{w1}C_f(T_{ein} - T_{eout}), \tag{6}$$

and the thermodynamics of the evaporator is

$$C_e m_e \dot{T}_{eout} = \dot{Q}_{eva} - \dot{Q}_{in} = \dot{Q}_{eva} - (COP - 1)\dot{W}. \tag{7}$$

Similarly, the heat rate transferred from the condenser to the Water Cycle 2 is

$$\dot{Q}_w = \dot{m}_{w2}C_f(T_{cout} - T_{cin}), \tag{8}$$

and the dynamic of the condenser is

$$C_c m_c \dot{T}_{cout} = \dot{Q}_{out} - \dot{Q}_w = COP\dot{W} - \dot{Q}_w. \tag{9}$$

Finally we have

$$\begin{cases} \dot{T}_{eout} = -\frac{\dot{m}_{w1}C_f}{C_e m_e}T_{eout} + \frac{\dot{m}_{w1}C_f}{C_e m_e}T_{ein} - \frac{(COP-1)}{C_e m_e}\dot{W}, \\ \dot{T}_{cout} = -\frac{\dot{m}_{w2}C_f}{C_c m_c}T_{cout} + \frac{\dot{m}_{w2}C_f}{C_c m_c}T_{cin} + \frac{COP}{C_c m_c}\dot{W}. \end{cases} \tag{10}$$

Equation (3) and (10) consist of a simple mathematical model of a heat pump. These equations are implemented by both m-function and S-function in Matlab/Simulink [10].

C. Refrigerant

The experimental studies of different refrigerants such as R22, R407C and R407A in [7] revealed that R407C offer highest exergy efficiency (higher COP) provided the exergy efficiency of the compressor is same for different refrigerant tests. The refrigerant R407C is selected.

III. INDOOR CLIMATE AND FLOOR HEATING MODELING

A. Floor and Room Thermal Dynamics

All system variables and parameters for floor and room modeling are listed in the following Table 2.

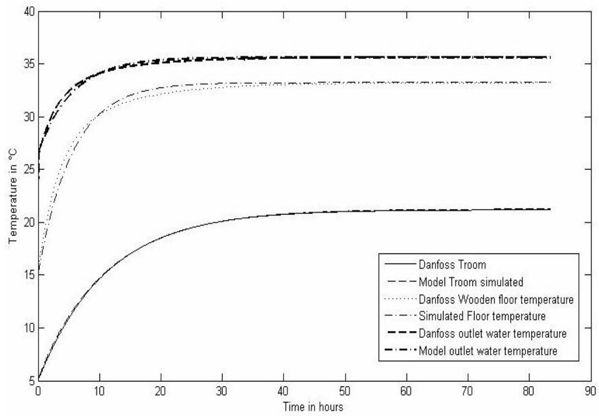

Fig. 4. Comparison of simulation models and experimental data

Notation	Description
M_f	Mass of the floor (kg)
M_a	Mass of the air (inside) (kg)
C_f	Heat capacity of floor $(\mathrm{J/kg \cdot K})$
C_a	Heat capacity of the air $(\mathrm{J/kg \cdot K})$
A_f	Surface area of the floor (m^2)
A_r	Surface area of the room (m^2)
U_f	Heat conductivity of the floor $(\mathrm{W/m \cdot K})$
U_r	Heat conductivity of the wall $(\mathrm{W/m \cdot K})$
T_f	Temperature of the floor $(^\circ\mathrm{C})$
T_r	Temperature of the air inside $(^\circ\mathrm{C})$
T_a	Ambient Temperature $(^\circ\mathrm{C})$
Q_p	Heat transferred from water to floor (J)
Q_f	Heat transferred from floor to room air (J)
Q_{loss}	Heat dissipated from room (J)

Table 2 Parameter list for floor heating modeling

As shown in Fig3, the heat is transferred from the hot water inside the embedded pipe to the concrete floor through pipe surfaces. Thereby the thermal dynamic of the floor can be described as

$$M_f C_f \dot{T}_f = \dot{Q}_p - \dot{Q}_f, \tag{11}$$

where the convected heat rate $\dot{Q}_p$ can be determined by

$$\dot{Q}_p = \dot{m}_{w2}C_w(T_{w\_in\_2} - T_{w\_out\_2}). \tag{12}$$

The heat transferred from the floor to the air inside the room, denoted as Q_f, can be estimated by

$$\dot{Q}_f = U_f A_f(T_f - T_r). \tag{13}$$

Similarly, the thermodynamics of the room can be described as

$$M_a C_a \dot{T}_r = \dot{Q}_f - \dot{Q}_{loss}, \tag{14}$$

where the dissipated heat from the room can be determined by

$$\dot{Q}_{loss} = U_r A_r(T_r - T_a). \tag{15}$$

By using the experiment data provided by Danfoss A/S, two linear first-order models are obtained for the floor and room dynamics, respectively. The developed models are validated via experimental data as shown in Fig.4.

$$\begin{cases} \dot{T}_f = \frac{1}{M_f C_f}[\dot{m}_{w2}C_w(T_{w\_in\_2} - T_{w\_out\_2}) - U_f A_f(T_f - T_r)], \\ \dot{T}_r = \frac{1}{M_a C_a}[U_f A_f(T_f - T_r) - U_r A_r(T_r - T_a)]. \end{cases} \tag{16}$$

B. Thermodynamic of Circulated Water

From (16) it can be observed that two measurements of the circulated water in water cycle 2 - $T_{w\_in\_2}$ and $T_{w\_out\_2}$ - are required in order to run simulation. $T_{w\_in\_2}$ can be provided from the heat pump model regarding (2). Therefore, a model to estimate $T_{w\_out\_2}$ is derived in the following, where the relevant parameters are listed in Table 3.

Notation	Description
m_{wc}	Massflow of circulated water (kg/s)
L_p	Length of circulation pipe (m)
dz	small slot of the circulation pipe (m)
T_{sc}	Temp. of floor (=T_f) (°C)
$T_w(z)$	Temp of water at z position (°C)
D_{ic}	Inner diameter of pipe (m)
D_{oc}	Outer diameter of pipe (m)
$d\dot{Q}_{wsc}$	Heat transferred out within dz (J)
α_{wsc}	Heat transfer coeff. from water to floor

Table 3 Parameter list for circulated water modeling

As shown in Fig.5, consider a small slot dz of the circulation pipe. Assume the thickness of the pipe is small, i.e., $D_{ic} \approx D_{oc}$, which means that the heat convection between the concrete floor and circulated water only needs to be considered. Further, assume the temperatures of the water and the floor are uniformly distributed within dz, then, from energy conservation there is

$$d\dot{Q}_{wsc} = \alpha_{wsc}\pi D_{ic}(T_{sc} - T_w(z))dz. \quad (17)$$

From the point of view of energy balance of the circulated water, there is

$$d\dot{Q}_{wsc} = \dot{m}_{wc}C_w(T_w(z) - T_w(z + dz)),$$

i.e.,

$$d\dot{Q}_{wsc} = \dot{m}_{wc}C_w dT_w(z). \quad (18)$$

Combination of (17) and (18) leads to

$$\frac{\alpha_{wsc}\pi D_{ic}}{\dot{m}_{wc}C_w}dz = \frac{1}{(T_{sc} - T_w(z))}dT_w(z). \quad (19)$$

By taking the integration of (19) along the z-axis and considering the boundary conditions as

$$T_w(0) = T_{w\_in\_2},\ T_w(L_p) = T_{w\_out\_2},$$

besides $T_{sc} = T_f$, there is

$$T_{w\_out\_2} = T_f - (T_f - T_{w\_in\_2})exp(-\frac{\alpha_{wsc}\pi D_{ic}}{\dot{m}_{wc}C_w}L_p). \quad (20)$$

Equation (20) is used to simulated/estimate the thermal dynamic of the circulated water. The developed model is validated via experimental data as shown in Fig.6.

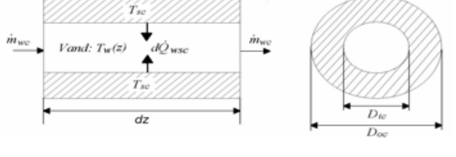

Fig. 5. Schematic diagram of circulated pipe modeling

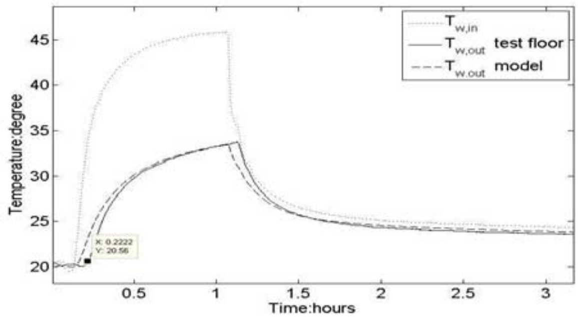

Fig. 6. Comparison of circulated water model and experimental data

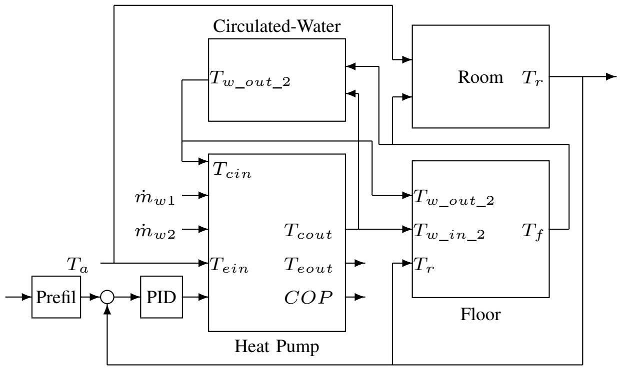

Fig. 7. Block diagram of entire system with PID+Prefilter

IV. Control and Simulation

A. Total Thermal System Model

By combining the head pump model (10), the floor and room model (16) and the circulated water model (20) together through relationships (2), a model of the total considered system is achieved and implemented in Matlab/Simulink as shown in Fig.7. The input variables are massflow rate of refrigerant in evaporator $\dot{m}_{w1}$, massflow rate of refrigerant in condenser $\dot{m}_{w2}$, net work to the compressor W, ambient temperature T_a and the compressor efficiency K.

B. Control Mechanism

The control mechanism of heat pump is to adjust the pressures inside the heat pump so as to obtain a desired mass flow and temperature in the output of the compressor, and also control the expansion valve to optimize the energy gain on the evaporator side. The control at this stage can be complicated. Therefore, we first assume the refrigerant is already superheated when it reaches the compressor. This superheating can be monitored by a superheating sensor attached to the evaporator output line [14]. We do not worry about the valve control in this stage. So in the simplest case we only consider the control of amount of work W going into compressor.

In order to study the influence of control to the total system performance in terms of tracking desired temperature and energy consumption, several control strategies are developed for the simulation tests. These control strategies are

- A relay controller. This on-off controller decides either to use full power to drive the compressor or to let it idle, which depends on predefined thresholds;
- A P controller designed through the root locus method;
- A PID controller developed through Ziegler-Nichols quarter-decay Method;

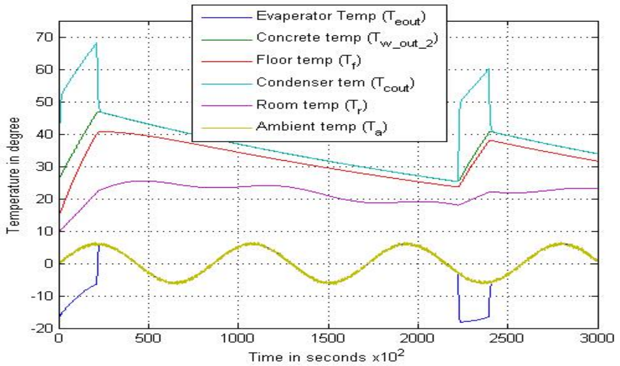

Fig. 8. System responses with a relay controller

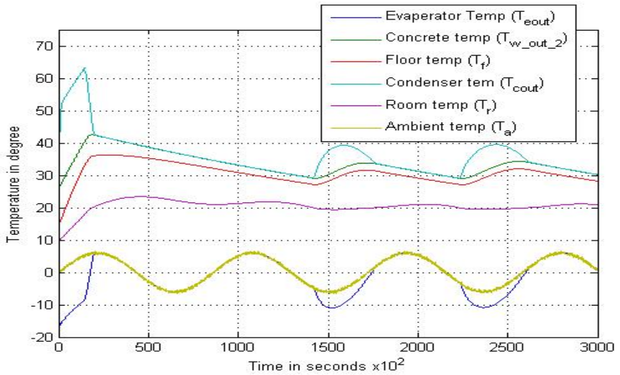

Fig. 9. System responses with a P- controller

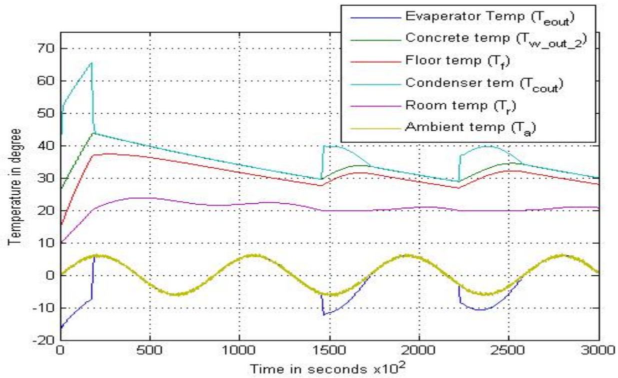

Fig. 10. System responses with a PID controller

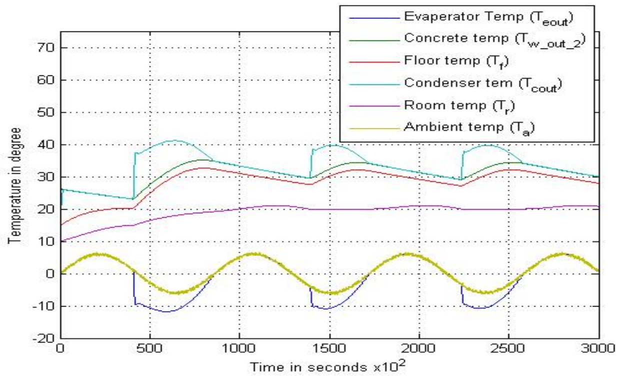

Fig. 11. System responses with a PID controller and pre-filter

- A PID controller with pre-filtering of set-points developed through Integral of Time multiplied by Absolute Error (ITAE) method [2].

C. ITAE Tuning

In the case of PID tuning using ITAE method, the nonlinear dynamics of the heat pump is neglected w.r.t. the fact that the dynamics of the floor and room are much slower than that of the heat pump. According to (16) and (20), the open-loop transfer function from $T_{w\_in\_2}$ to T_r is shown in the following.

$$G_{fr}(s) = \frac{6.97e^{-10}}{s^2 + 7.5e^{-5}s + 1.25e^{-9}}. \tag{21}$$

Define the PID controller as $G_c(s) = \frac{K_D s^2 + K_P s + K_I}{s}$, it is obvious that the closed-loop system $\frac{G_c(s)G_{fr}(s)}{1+G_c(s)G_{fr}(s)}$ is a 3rd order system [10].

By selecting the settling time as 24 hours, the natural frequency of the expected closed-loop system can be determined as $\omega_n = 5.79e^{-5}$. Then, according to the empirical ITAE optimum coefficient formula [2], three PID coefficients can be determined as $K_P = 8.5476$, $K_I = 2.784e^{-4}$ and $K_D = 3.869e^4$. In order to further improve the controlled system performance, a prefilter is designed to get ride of all zeros in the transfer function of the PID controlled system. For our case, the pre-filter for the set-point is obtained as

$$G_{fil}(s) = \frac{7.18e^{-9}}{s^2 + 2.2e^{-4} + 7.18e^{-9}}. \tag{22}$$

The complete system controlled by developed PID controller with pre-filtering is shown in Fig.7.

D. Simulation Results and Discussions

The following conditions are set up for all different control strategies:

- The initial evaporator temp. (T_{eout}), concrete temp. ($T_{w\_out\_2}$), floor temp (T_f), condenser temp. (T_{cout}), room temp. (T_r) and ambient temp. (T_a) are 0, 26.5, 15, 10, 10, 0 degrees, respectively.
- Massflow rates are set up as constants, i.e., $\dot{m}_{w1} = \dot{m}_{w2} = 0.05kg/s$.
- The working power for compressor is within the range $0 \leq W \leq 2000$ Watt, and the efficiency of the compressor is set up as 0.4.
- The desired temperature is set up as 20 degree.
- The ambient (outside) temperature is modeled by a dominant sine-wave plus some small random noise which simulates a 24 hours weather cycle.

The controlled system performances with a relay, P, PID and PID with pre-filter controllers are shown in Fig.8, Fig.9, Fig.10 and Fig.11, respectively. Specially, by comparing the controlled room temperatures under different scenario as shown in Fig.12, it can be observed that the PID with pre-filter strategy generated the most comfortable thermal dynamic in terms of the smallest overshoot in the transient period and the smallest deviation in the steady period. Meanwhile, the controlled system using the PID with pre-filter also has the fast response to settle down around the desired temperature. On the contrary, the relay controlled system has the worst performance both in the transient period and in the steady period. It can be observed that the changes in ambient temperature affected system performance, and PID with pre-filter is the least sensitive strategy to this kind of disturbance. By comparing the power consumptions under different strategies as shown in Fig.14, it can

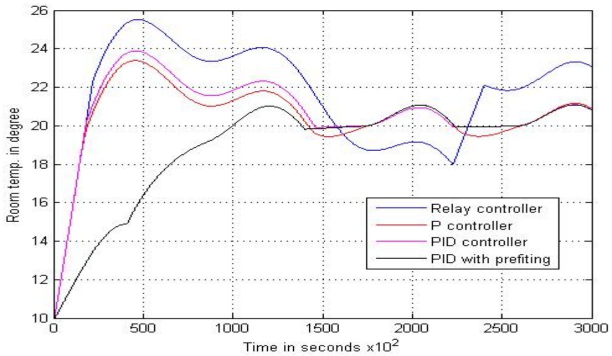

Fig. 12. Comparison of room temperature sunder different control strategies

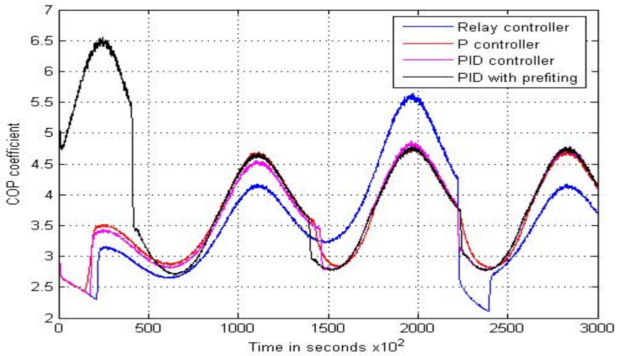

Fig. 13. Comparison of COPs under different control strategies

be observed that the PID with pre-filter strategy consumed least power while keeping the best performance. On the contrary, the relay strategy consumed most power while having the worst performance. The COPs under different strategies are shown in Fig.13. It is clear that all COPs are heavily influenced by the ambient temperature. PID with pre-filter strategy has the highest COP while the relay strategy has the smallest COP in the transient period. Within the steady period P, PID and PID with pre-filter strategies have the almost same COP performance, while the relay strategy has the largest oscillation caused by the dynamic ambient temperature.

V. Conclusions

Modeling and control of an indoor climate using floor heating system based on a heat pump is discussed. A simple heat pump model is developed based on an approximation of COP using the temperatures of the evaporator and condenser. The indoor thermodynamics including the room, floor and circulated water are modeled theoretically and then identified through experimental data. Several control strategies are investigated for the developed system, namely relay control, P control, PID control and PID control with pre-filtering of reference input. The simulation results turn out that comparing with other strategies, the PID control with pre-filter strategy generated the best system performance in terms of the smallest overshoot, the fast response, the highest COP in the transient period, the smallest deviation and the least sensitivity to disturbance in the steady period. Meanwhile, this strategy also consumed the least power comparing with others. This study reveals a huge potential to optimize the operation of this kind of thermal system by using control techniques in terms of thermal comfort and energy consumption.

Employing a detailed heat pump model, such as the model used in [3], and extending the control mechanism to the expansion valve, pressures inside the evaporator and condenser, as well as investigating advanced control techniques, will be left for future work.

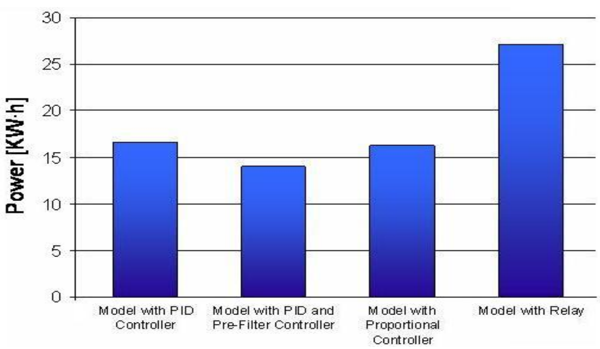

Fig. 14. Comparison of power consumption of different strategies

VI. Acknowledgments

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