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Multi-parameter Analysis of Influences of Air Conditioning Systems on Electric Vehicle

Hao Pan[1], Liangfei Xu[1,2,*], Jianqiu Li[1,2] and Minggao Ouyang[1]
[1]Department of Automotive Engineering, State Key Laboratory of Automotive Safety and Energy, Tsinghua University, Beijing 100084, P.R. China
[2]Collaborative Innovation Center of Electric Vehicles in Beijing, Beijing 100081, P.R. China
[*]Corresponding author

***Abstract*—Due to significant impact of energy consumption of air conditioning system on driving range of electric vehicles, study on air conditioning system of electric vehicles is of great importance. However, the research existing is not comprehensive enough, which is limited to the performance and experimental research of the electric air conditioning components. As for influences of air conditioning systems on electric vehicle, model-based multi-parameter analysis is still a lack. In this paper, a thermal and humidity model of electric vehicles is firstly proposed, followed by control strategy of the air conditioning system. Energy consumption analysis of electric vehicles under different driving and ambient conditions is conducted based on simulation results.**

Keywords-electric vehicle; thermal and humidity systems; control strategy; energy consumption analysis

I. INTRODUCTION

Regarded as a solution for the environment pollution and energy crisis, electric vehicles are drawing more and more attentions for their advantages of zero emission, not relying on fossil fuels and high energy efficiency. Nonetheless, there are still many challenges that prevent its widespread commercialization [1]. Short driving range of electric vehicle due to limited capacity of battery in current technical conditions is the most significant challenge. With the demand of comfortable temperature for both passengers and the battery pack, air conditioning system is very necessary for electric vehicle. Electric air conditioning system causes much energy consumption, thus making limited driving range of electric vehicle shorter. According to [2], 1/3 of the electric vehicle's energy is consumed at air conditioning system in winter. Therefore, designing an air conditioning system that is suitable for an electric vehicle is of great significance.

There are many researches focusing on thermal management system of electric vehicles [3-7]. However, the research existing is not comprehensive enough, which is limited to the performance and experimental research of the electric air conditioning components. As for simulation analysis for thermal and humidity systems of the whole electric vehicle including air conditioning system, it lacks more systematic and comprehensive study. Research on modeling and simulation of thermal and humidity system of the whole electric vehicle has practical application value for parameters matching and application development of the electric vehicle.

In this paper, modeling, control and simulation analysis of electric vehicle thermal and humidity system is proposed. This paper is organized as follows. Section 2 models electric vehicle thermal and humidity systems. Section 3 illustrates control strategy of the electric air conditioning system. Section 4 does some simulation and then do some energy consumption analysis based on different driving and ambient conditions. Section 5 draws conclusions of this paper.

II. MODELING

A. Basic Description

The electric vehicle in this paper is an electric bus, whose main parameters are list in Table 1.

B. Heat Model

Heat transfer methods between the vehicle and the environment are conduction, convection and radiation. As the environment changes frequently, the state of heat transfer between the vehicle and the environment is usually unstable. For simple calculation, this paper uses approximate algorithm of steady heat transfer. The thermal equilibrium equation is as bellows,

$$Q = Q_b + Q_w + Q_{pr} + Q_v + Q_p + Q_d \tag{1}$$

where Q is the total heat that is transferred into the vehicle, Q_b is the heat transferred through the body envelope structure, Q_w is the heat through windows of the vehicle, Q_{pr} is the heat through the powertrain room of the vehicle, Q_v is the leaked heat because of ventilation and poor sealing, Q_p is the heat that emitted by the passengers and Q_d is the heat emitted by the electronic devices.

TABLE I. MAIN PARAMETERS OF THE ELECTRIC BUS

Parameters	Value
Length	12 m
Width	2.5 m
Height	3.2 m
Maximum speed of motor	6200 r/min
Rated power of motor	110 kW
Maximum power of motor	180 kW
Rated capacity of battery	250 kWh
Rated voltage of bus	700 V
Weight(empty/full)	14000/16800 kg
Highest speed	80 km/h

As temperature difference exists inside and outside of the vehicle, heat from the environment transfers into the vehicle cabin through vehicle body. Considered various parts of the vehicle body varies, we can divide surface of vehicle body into three calculation areas according to different structures, thus roof area, side area and bottom area of the vehicle body. Each part is calculated as follows,

$$Q_b = (Q_r + Q_s + Q_{bo}) \cdot a$$

$$Q_r = K_r F_r (t_Z - t_{in})$$

$$Q_s = K_s F_s (t_Z - t_{in}) \tag{2}$$

$$Q_{bo} = K_{bo} F_{bo} (t_Z - t_{in})$$

where Q_r, Q_s, Q_{bo} are heat transferred through roof area, side area and bottom area of the vehicle body, a is the correction factor due to the simplification of heat transfer process.

Heat through vehicle windows consists of two parts, thus heat by convection due to temperature difference inside and outside of the vehicle and heat by solar radiation through the windows. The equations are as follows,

$$Q_{w1} = K_w F_w (t_{wZ} - t_{in}) \tag{3}$$

$$Q_{w2} = \eta I_w F_{w2} \tag{4}$$

where K_w is heat transfer coefficient of vehicle windows, F_w is the total area of vehicle windows, t_{wZ} is the equivalent temperature of outer surface of vehicle windows due to solar radiation, η is penetration factor of solar radiation through windows, I_w is solar radiation intensity of outer surface of windows, F_{w2} is the effective area of vehicle windows in the direction of direct sunshine.

As an energy conversion machine, the motor inevitably produces losses during energy conversion process. Most of the energy losses turns into heat eventually, thus making the temperature of various parts of the motor rise. The controller is consisted of many electronic devices, which creates much heat during working and thus inevitably making the temperature of cabin room rise. Besides, the batteries also create heat during working. In this study, the rooms of the electric motor and the battery are assumed to be a united one, which is named as the powertrain room. As all above, Heat load through powertrain room is,

$$Q_e = F_e K_e (T_e - T_{in}) \tag{5}$$

where F_e is heat transfer area between powertrain room and the cabin, K_e is the heat transfer coefficient between powertrain room and the cabin, T_e is the average temperature of powertrain room.

For human healthy, some fresh air should be delivered into the vehicle from time to time. Heat load of the fresh air due to ventilation is,

$$Q_{v1} = 0.28 n V \rho (h_H - h_B) \tag{6}$$

where n is the number of passengers, V is the amount of fresh air for each person per hour according to human healthy standard with the unit of m^3/h per person, ρ is the density of air with the unit of kg/m^3, h_H and h_B are the enthalpy of air outside and inside of the vehicle with the unit of J/kg.

Leaked heat into the vehicle is through the gaps of the door and windows.

$$Q_{v2} = 0.28 V' L \rho (h_H - h_B) \tag{7}$$

where V' is the amount of leaked wind with the unit of m^3/h per person, L is the length of the gaps of the door and windows with the unit of m.

Heat load due to ventilation and poor sealing can be calculated in the following equation,

$$Q_v = Q_{v1} + Q_{v2} \tag{8}$$

Heat emitted by the passengers is related to many factors, such as labor intensity, environment temperature, gender, age, clothes and so on. For simplification, this paper will use the following equation to calculate heat load emitted by the passengers, which is，

$$Q_p = 116n \tag{9}$$

Where n is the number of passengers and the unit of Q_p is W.

Electronic devices in the vehicle emit much heat. Thus causing the temperature of the cabin rise. Heat load emitted by electronic devices can be simply calculated as follows,

$$Q_d = 1000P \tag{10}$$

where P is the power of electronic devices. The unit of Q_d is W and the unit of P is kW.

C. Energy Consumption Model

The main function of electric air conditioning is to send cold or heat stream to the cabin in order to change the temperature and humidity to comfortable ones. Electric air-conditioning is controlled by the motor directly. Cooling capacity and heat capacity of electric air conditioning are approximately linear with speed of electric air conditioning compressor. The equation is as follows,

$$Q_s = K_1 n_{ac} \tag{11}$$

where Q_s is cooling capacity or heat capacity of electric air conditioning, n_{ac} is speed of electric air conditioning compressor.

Energy consumption of electric air conditioning is also approximately linear with speed of electric air conditioning compressor. It can be calculated as follows,

$$Q_{ac} = K_2 n_{ac} \tag{12}$$

D. Humidity Model

Assuming that all the air supply of the air conditioning is return air, the relative humidity of the air inside the vehicle can be calculated in the following equation,

$$\varphi = \frac{P_{H_2O}}{P_{S0}} = \varphi_0 + \left(\frac{R_g T_0}{V_0 P_{S0}} - \frac{R_g T_0}{V_0 P_0}(\varphi - \varphi_{send}(t_1, dm))\right)\int dmdt \tag{13}$$

where P_{H_2O} is the partial pressure of water vapor, φ_0 is the initial relative humidity of the air inside the vehicle, T_0 is the temperature of the air inside the vehicle, V_0 is the volume of the cabin, P_0 is the pressure of the air inside the vehicle, φ_{send} is the relative humidity of air supply of the air conditioning, t_1 is the temperature of air supply of the air conditioning, dm is the mass flow of air supply of the air conditioning, P_{S0} is the saturated vapor pressure at the current temperature, which can be calculated by an empirical formula,

$$P_{S0} = 611.2\exp\left(\frac{17.62}{243.12 + t_{in}}\right) \tag{14}$$

where t_{in} is the temperature of air inside the vehicle. The unit of P_{S0} is Pa and the unit of t_{in} is ℃. t_{in} can be calculated as follows,

$$t_{in} = \int \frac{Q - Q_{ac}}{\rho V_0 c_p} dt \tag{15}$$

III. Control Strategy of Air Conditioning System

The basic idea of control strategy is as follows, by comparing target temperature and actual temperature, target humidity and actual humidity, we can adjust speed of electric air conditioning compressor and the humidity of air supply in order to make the temperature and humidity of the cabin approach the target temperature and target humidity. Input of the control strategy is temperature of the air in the cabin, relative humidity of the air in the cabin. Refer to an actual air-conditioning system which can control speed of electric air-conditioning compressor and the humidity of air supply, the output of the control strategy is speed of electric air-conditioning compressor and the humidity of air supply.

According to(11), cooling capacity of electric air conditioning is only related to speed of electric air conditioning compressor, which can be simplified to linear relationship. According to (13), humidity of the air in the cabin is related to speed of electric air conditioning compressor and the humidity of air supply, which is non-linear relationship. Therefore, control of the temperature of the vehicle is a linear control issue, while control of the humility of the vehicle is a non-linear control issue.

As for linear control of the temperature of the vehicle, this paper will use a PI controller as the control strategy. In order to improve the system response speed and reduce the dynamic overshoot and steady-state error in the same time, this paper is going to use feedforward and feedback control. The tunable parameters of the control strategy are feed forward factor K_f and feedback factors K_p and K_i. The set speed of electric air conditioning compressor is,

$$n_{ac} = K_f Q_{ac0} + K_p(t_{in,tg} - t_{in}) + K_i \int (t_{in,tg} - t_{in}) dt \tag{16}$$

where Q_{ac0} is steady feed forward cooling capacity of electric air-conditioning. It can be calculated as follows,

$$Q_{ac0} = Q(t_{in}) - \frac{\rho V c_p (t_{in,tg} - t_{in})}{T} \tag{17}$$

where Q is the total heat that transfers into the vehicle which is same as that in (1), T is control time of temperature users expect.

As for non-linear control of the humidity of the vehicle, this paper uses bang-bang control strategy based on synovial algorithm. When the humidity of the air inside the cabin rises and then exceeds the threshold φ_{10}, set the humidity of air supply to be φ_{airin_1}. When the humidity of the air inside the cabin decreases and then exceeds the threshold φ_{10}, set the humidity of air supply to be φ_{airin_2}. The control strategy can be expressed by a set of parameters as follows,

$$\theta = \lfloor K_f, K_p, K_i, T, \varphi_{10}, \varphi_{11}, \varphi_{airin\_1}, \varphi_{airin\_2} \rfloor \tag{18}$$

And the strategy can be optimized by optimizing these parameters using numerical algorithms such as generic algorithm (GA).

IV. Simulation and Analysis

A. Simulation of Two Typical Cases

This paper is going to choose an electric bus for simulation. Main parameters of the heat and humidity models are shown in Table 2.

We will use the environment of July in Beijing and January in Beijing for simulation, which can simulate both conditions of cooling and heating. The speed condition of the vehicle uses China City Bus typical Cycle (CCBC). According to the data of website of the weather of China on-line [8], environment parameters of July in Beijing and January in Beijing are list in Table 3.

The simulation is conducted in MATLAB/ SIMULINK. Figure 1 shows the variety of cabin temperature and humidity under the control strategy proposed. In July, the temperature in the cabin can reach the target in 550 s, thus staying near 25℃ with fluctuation range less than 0.5℃. The relative humidity can stay near 50% with fluctuation range less than 0.5%. In January, the temperature in the cabin can reach the target in360 s, staying near 25℃ with fluctuation range less than 0.5 ℃. The relative humidity can stay near 50% with fluctuation range less than 1%. According to the simulation results, the control strategy can control the temperature and humidity of the vehicle well for both cooling and heating cases, thus providing the passengers a comfortable travel environment.

B. Thermal Load Distribution

TABLE II. MAIN PARAMETERS OF MODEL

Parameters	Value
Heat transfer coefficient through roof, K_r	4.07 W.m^2/K
Heat transfer area through roof, F_r	30 m^2
Heat transfer coefficient through side, K_s	4.07 W.m^2/K
Heat transfer area through side, F_s	46.4 m^2
Heat transfer coefficient through bottom, K_{bo}	4.26 W.m^2/K
Heat transfer area through bottom, F_{bo}	30 m^2
Heat transfer coefficient through windows, K_w	6.38 W.m^2/K
Heat transfer area through windows, F_w	23.2 m^2
Penetration factor of solar radiation through windows, η	0.85
Heat transfer coefficient between engine room and the cabin, K_e	4.26 W.m^2/K
Heat transfer area between engine room and the cabin, F_e	1 m^2

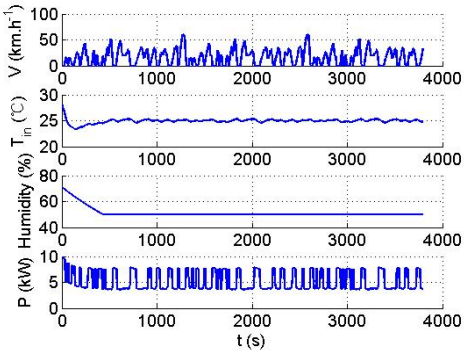
TABLE III. ENVIRONMENT PARAMETERS OF BEIJING

	July	January
Temperature	28℃	0℃
Relative humidity	71%	61%
Solar radiation intensity	223 W.m^2	111.5 W.m^2

This section mainly analyzes the distribution of thermal load of electric bus. Taking Beijing, July, CCBC condition for example, thermal load distribution is shown in Figure 2. According to Figure 2, the percentage of thermal energy that transferred through vehicle body is 43.0%, the percentage of thermal energy that transferred through windows is 20.3%, the percentage of thermal energy that emitted by the passengers is 28.3%, the percentage of thermal energy that emitted by the electronic devices is 6.1%, the percentage of thermal energy that leaked because of ventilation and poor sealing is 1.3%, the percentage of thermal energy that transferred through power room is 1.0%. As all above, thermal energy mainly distributed on thermal energy that transferred through vehicle body, through windows and emitted by the passengers, with the percentage of 43.0%, 20.3% and 28.3%.

C. Multi-Parameters Energy Consumption Analysis

This section mainly analyzes energy consumption of the air conditioning system under different driving and ambient conditions. Simulation conditions consist four information, thus solar radiation intensity, environment temperature, environment humidity and driving cycles. The five driving cycles that we use for simulation are as follows: China City



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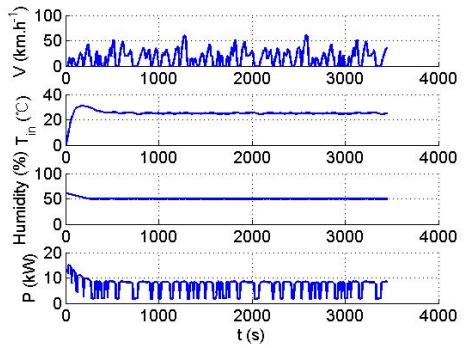


(b)

FIGURE I. SIMULATION RESULTS IN BEIJING FOR (A) JULY AND (B) JANUARY

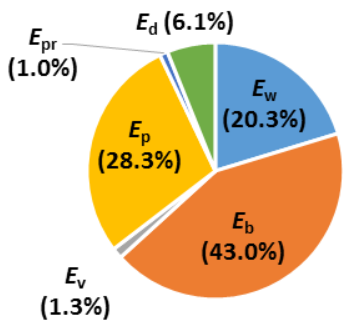
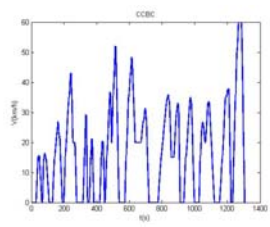


FIGURE II. THERMAL LOAD DISTRIBUTION FOR BEIJING, JULY, CCBC CONDITION

Bus typical Cycle (CCBC), Highway Fuel Economy Driving Schedule (HFEDS), SC03 and Urban Dynamometer Driving, Schedule (UDDS) and Zhengzhou Line 9 (ZZL9). The driving cycles is shown in Figure 3. As for environment temperature, environment humidity and solar radiation intensity, we choose values of twelve months in Beijing, Shanghai and Guangzhou for the simulation conditions. According to the data of website of the weather of China on-line [8], the detailed information is shown in Table 4. In Table 4 SRI is solar radiation intensity, BJ is Beijing, SH is Shanghai and GZ is Guangzhou.

Figure 4 shows energy consumption per 100 kilometers of air conditioning in different cities. In general, energy consumption per 100 kilometers in summer or winter is higher than that in spring or autumn. In particular, energy consumption when the driving cycle is ZZL9 is much higher than other cycles because a large part of the driving cycle is in



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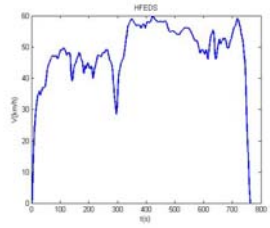

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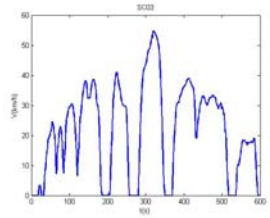

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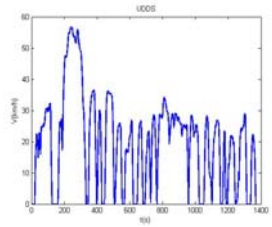
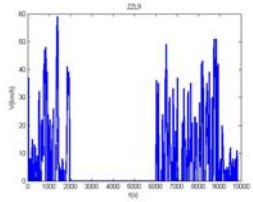


(d)



(e)

FIGURE III. DRIVING CYCLES (A) CCBC (B) HFEDS (C) SC03 (D) UDDS (E) ZZL9

TABLE IV. AVERAGE AMBIENT PARAMETERS IN DIFFERENT CITIES

Month	Temperature (℃)			Humidity (%)			SRI(W.m^2)		
	BJ	SH	GZ	BJ	SH	GZ	BJ	SH	GZ
Jan	0	8	17	61	70	73	111.5	91.5	87
Feb	0	9	15	51	75	80	111.5	91.5	87
Mar	10	12	20	45	65	85	167.25	137.25	130.5
Apr	16	16	23	39	57	90	167.25	137.25	130.5
May	21	23	26	47	71	89	167.25	137.25	130.5
Jun	26	25	30	71	77	84	223	183	174
Jul	28	28	30	71	58	86	223	183	174
Aug	26	28	29	69	63	87	223	183	174
Sep	20	25	28	68	74	83	167.25	137.25	130.5
Oct	13	21	26	60	72	72	167.25	137.25	130.5
Nov	8	16	23	42	67	75	167.25	137.25	130.5
Dec	-1	7	16	40	69	85	111.5	91.5	87

the state of parking, which causes much energy consumption of air conditioning while not driving.

Figure 5 shows energy consumption of air conditioning in different cities under same driving cycle of CCBC. In Beijing,

highest energy consumption of air conditioning in winter is 46.1055 kWh per 100 kilometers and highest energy consumption in summer is 31.8725 kWh per 100 kilometers. It means that in Beijing energy consumption of air conditioning in winter is higher than that in summer. But in Guangzhou, highest energy consumption of air conditioning in summer is

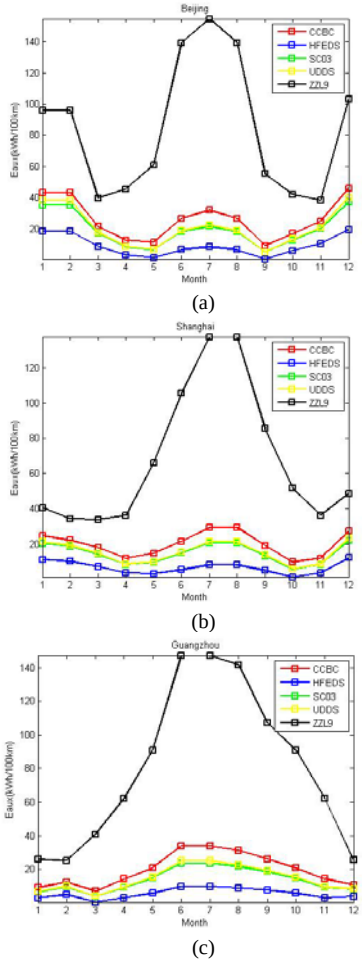


FIGURE IV. ENERGY CONSUMPTION PER 100 KILOMETERS OF AIR CONDITIONING UNDER DIFFERENT DRIVING CYCLES IN (A) BEIJING (B) SHANGHAI AND (C) GUANGZHOU

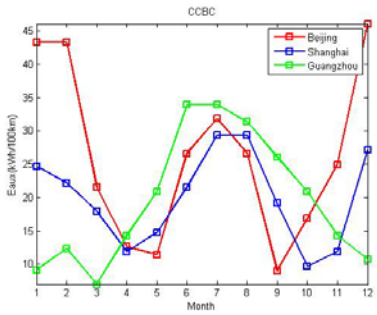


FIGURE V. ENERGY CONSUMPTION OF AIR CONDITIONING IN DIFFERENT CITIES

33.9588 kWh per 100 kilometers, highest energy consumption of in summer is 12.3460 kWh per 100 kilometers. It means that in Guangzhou energy consumption of air conditioning in summer is much higher than that in winter. That is to say, of all the four seasons in Beijing, heating in winter consumes most energy while cooling in summer consumes most energy in Guangzhou and the energy consumption of air conditioning in winter and in summer are about the same in Shanghai. It indicates that cooling or heating demand for the air conditioning is different in different latitudes. Heating effect should be mainly considered in north while cooling performance should be considered more in south.

V. CONCLUSION

This paper uses an electric bus as the simulation object, whose length is 12 meters, capacity of the battery is 250 kWh and full and empty weight are 14000 and 16800 kilogram. A thermal and humidity model of the electric bus is firstly proposed, followed by control strategy of the air conditioning system. According to the simulation results, the control strategy proposed can control the temperature and humidity of the vehicle well for both cooling and heating cases, thus providing the passengers a comfortable travel environment. The simulations show that difference between the cabin temperature and target temperature is less than 0.5 degree in both summer and winter using the proposed control strategy, and difference between the cabin humidity and target humidity is less than 1%.

As for thermal load distribution, the percentage of thermal energy that transferred through vehicle body is 43.0%, the percentage of thermal energy that transferred through windows is 20.3%, the percentage of thermal energy that emitted by the passengers is 28.3%, the percentage of thermal energy that emitted by the electronic devices is 6.1%, the percentage of thermal energy that leaked because of ventilation and poor sealing is 1.3%, the percentage of thermal energy that transferred through the powertrain room is 1.0%. Thermal is mainly distributed on thermal that transferred through vehicle body, through windows and emitted by the passengers. Energy consumption of air conditioning in different cities is very different. In Beijing, highest energy consumption of air conditioning in winter is 46.1055 kWh per 100 kilometers. It is higher than that in summer, which is 31.8725 kWh per 100 kilometers. But in Guangzhou, highest energy consumption of air conditioning in summer is 33.9588 kWh per 100 kilometers. It is much higher than that in winter, which is 12.3460 kWh per 100 kilometers. It indicates that cooling or heating demand for the air conditioning is different in different latitudes. Heating effect should be mainly considered in north while cooling performance should be considered more in south.

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