

Article

Prediction and Evaluation of Dynamic Variations of the Thermal Environment in an Air-Conditioned Room Using Collaborative Simulation Method

Lin He ¹, Shunan Zhao ¹, Guowen Xu ², Xin Wu ², Junlong Xie ² and Shanshan Cai ^{2,*}

¹ State Key Laboratory of Air-Conditioning Equipment and System Energy Conservation & GREE Electric Appliances, Inc. of Zhuhai, Zhuhai 519070, China; helin@cn.gree.com (L.H.); zhaosn@cn.gree.com (S.Z.)

² School of Energy and Power Engineering, Huazhong University of Science and Technology, Wuhan 430074, China; m201871202@hust.edu.cn (G.X.); m201971239@hust.edu.cn (X.W.); jlxie@mail.hust.edu.cn (J.X.)

* Correspondence: shanshc@hust.edu.cn

Abstract: In this study, a collaborative simulation method is proposed to predict dynamic variations of the thermal environment in an air-conditioned room. The room thermal environment was predicted and analyzed by varying the structural and control parameters of the air conditioner considering the dynamic coupling effect. Connections and regularities were established between the applicable parameters and evaluation indices of the thermal environment. The simulation results demonstrated the interactions among the system structural parameters, control parameters, and the thermal environment. Within a certain parameter range, the evaporator structure exhibited a significant effect on temperature uniformity and vertical air temperature difference, followed by predicted mean vote (PMV) and draught rate (DR). The associated evaluation indices were sensitive to fin spacing, tube spacing, and tube outer diameter, in the same order, which were structural parameters of the evaporator. The effect of the air supply angle on the vertical air temperature difference was evident; however, its influence on the PMV, DR, and temperature uniformity did not indicate consistent variations.

Keywords: air conditioning system; room thermal environment; dynamic coupling; evaluation indices; collaborative simulation



Citation: He, L.; Zhao, S.; Xu, G.; Wu, X.; Xie, J.; Cai, S. Prediction and Evaluation of Dynamic Variations of the Thermal Environment in an Air-Conditioned Room Using Collaborative Simulation Method. *Energies* **2021**, *14*, 5378. <https://doi.org/10.3390/en14175378>

Academic Editor: Boris Igor Palella

Received: 12 July 2021

Accepted: 23 August 2021

Published: 30 August 2021

Publisher's Note: MDPI stays neutral with regard to jurisdictional claims in published maps and institutional affiliations.



Copyright: © 2021 by the authors. Licensee MDPI, Basel, Switzerland. This article is an open access article distributed under the terms and conditions of the Creative Commons Attribution (CC BY) license (<https://creativecommons.org/licenses/by/4.0/>).

1. Introduction

Investigations on the thermal environment test of a room air conditioner are normally performed in accordance with the existing standards, such as ISO 7730 and ISO 17772-1 [1,2]. In the existing standards, the evaluation indices of the thermal comfort include temperature fluctuation, temperature uniformity, predicted mean vote (PMV), draught rate (DR), etc. Such indices are closely associated with the dynamic procedure of the air conditioning system and its associated effects on the temperature and velocity fields in the room. The dynamic procedure of an air conditioning system depends on the structures of the main components (such as air supply outlet and evaporator) [3], control parameters and strategies (such as setting temperature, air supply angle, proportional integral derivative control, and fuzzy control) [4,5], and ambient conditions (including outdoor temperature, air velocity, return air temperature, and positioning for installation) [6,7]. Dynamic variations of the air conditioning system directly affect the supply air conditions and further vary the distribution characteristics of the room thermal environment [8,9]. Therefore, to predict the dynamic procedure and evaluate associated indices of the thermal environment of an air-conditioned room, the models of the air conditioning system and room need to collaborate. Such prediction and evaluation methods can not only help improve the testing efficiency but also provide a theoretical reference for designing parameters and optimizing air conditioners [10].

The dynamic model of an air conditioning system is based on the heat transfer mechanism of the air-conditioning unit and thermodynamic characteristics of each component. It is generally established using MATLAB Simulink to calculate key parameters of the air conditioning unit, such as the energy efficiency ratio and supply air conditions [11,12]. However, many studies combine air conditioning systems with the traditional room thermodynamic model by calculating room load, which cannot clearly reflect the distribution characteristics of the spatial environment. Numerical simulation tools using computational fluid dynamics methods, such as Fluent, can perform a comprehensive simulation and analysis of thermal environment variations, which can be used to moderately predict the testing procedure of air conditioning units [13,14]. However, studies on the dynamic variation characteristics of room thermal environments mainly analyze the influences of parameters such as return air location, supply air outlet size, and supply air state parameters on the flow and temperature fields in an air-conditioned room under the boundary conditions of fixed wall temperature, heat flux, fixed supply air temperature, and velocity [15,16]. Consequently, it is difficult for the existing methods to accurately explore the dynamic procedure and collaborative interaction between the air conditioning unit and room thermal environment during the actual testing and operation periods.

Co-simulation is a modelling approach full of potential. To predict the building energy consumption and indoor environment, some researchers integrated one-dimensional building energy simulation and three-dimensional computational fluid dynamics programs to provide complementary information. Results indicated that the coupling program matched with the experimental data; it behaved more accurately and realistically than the stand-alone simulation [17]. Another study proposed an integrated simulation framework for the comprehensive analysis of HVAC systems in off-highway mechanical cabins. The typical one-dimensional/three dimensional co-simulation approach is enhanced by combining an artificial neural network to simplify the overall complexity of the model [18]. The TRNSYS-FLUENT quasi-dynamic co-simulation method was provided in one study to evaluate the energy effect of a phase change material heat exchanger by considering one-dimensional energy simulation and two-dimensional CFD (computational fluid dynamics) model [19]. Co-simulations are also widely used for connecting system performance with control strategies. A co-simulation tool was reported to couple the energy system model by Dymola–Modelica and the building model by EnergyPlus. It aimed to investigate the optimal control of a HVAC system with thermal comfort and minimum energy consumption [20]. Another co-simulation method is built between EnergyPlus Functional Mockup Units (FMUs) with the Python environment to investigate the operations of the ground source heat pump. EnergyPlus output the necessary status parameters to Python and obtained the control signals from Python in the co-simulation platform [21]. Besides, multiple researches have developed co-simulations between EnergyPlus and MATLAB in the BCVTB platform to analyze the performance of HVAC (heating, ventilation, and air conditioning) units [22], energy consumptions in buildings, and the variations on the zone environment [23]. Among these studies, the energy simulation is normally the one-dimensional models with control strategies and interactions among different components to be fully considered. Some software and simulation tools are also proposed in the current literature to predict the dynamic thermal variations and energy efficiency in buildings [24–27]. However, the studies in the current literature mainly focused on the building applications without analysis of the potential connections between the structure characteristics of air-conditioner and the thermal variations, as well as thermal comfort related indices in an air-conditioned space.

This study aims to simulate the behaviors of the air conditioners with different structure and control parameters when they are in the thermal environment/comfort test. A parallel collaborative simulation model is proposed to solve the coupling issue between the operation of the air-conditioning unit and dynamic variations of the room thermal environment. The proposed model comprehensively analyzes interactions among the system structural parameters, control parameters, and the thermal environment, which

has never been reported in the open literature and it is the novelty of this work. It is true that non-uniform flow affects the thermal environment, but structural parameters and control strategies also have their impacts on the heat transfer capacity, response time, and efficiency, all of which further affect dynamic variations of the thermal environment.

2. Collaborative Simulation Workflow

A closed-loop coupling simulation model was constructed to accomplish the objective of this study. In the actual thermal environment/comfort test of an air conditioner with the ambience of the outdoor unit maintained constant, the structure of heat exchangers and the control strategy of the air-conditioning unit affect the real-time thermal state in the room. The real-time thermal state affects the return air status, which affects the real-time operation of the air-conditioning unit. In order to describe the dynamic feedback between the air conditioning system and the room thermal environment during the test period, a collaborative method between the air conditioning model and room model can reflect the real-time variations, expected to be closer to the actual application. To include the structural parameters of components in the air conditioning system, a mathematical model is required to predict the variations among main components. To indicate control strategy, control models, such as PID, fuzzy PID modules are required to reflect the responses of the air conditioning system. To predict the flow field in the air-conditioned room, CFD modeling is a commonly used simulation method. Considering the requirements in different models, as well as the characteristics in different simulation tools, a collaborative simulation method can eliminate the limitations of single tool to some extent. During simulation, the air conditioning system model passes the air supply state data to the inlet boundary of the room thermal environment model that in turn feeds back the corresponding calculation results of the room temperature field as an input to the air conditioning system model. This needs to occur in real time. Therefore, the air supply state parameters of the indoor unit are provided by the air conditioning system model, and the operation mode of the air conditioning system is adjusted according to the return air state parameters. The room thermal environment model considers the temperature fluctuations in the outdoor chamber and heat transfer in the structure of the building envelope. By continuously exchanging data, the air conditioning system can adjust its operation mode by checking on the updated return air status, and the room thermal state can vary according to the updated supply air status. Such a closed-loop coupling simulation is able to include the real-time interaction among different systems which is close to the real application of the air-conditioning unit.

This study combined the advantages of Simulink and Fluent. Simulink was used to establish a dynamic mathematical model of an air-conditioning system, and Fluent was used to build a transient model of the room thermal environment. The data interface was developed using the S-function in Simulink and user-defined functions (UDF) in Fluent, and the data exchange and update were realized by accessing the shared data through file I/O functions. Thus, a collaborative simulation platform for the prediction and evaluation of the thermal environment in an air-conditioned room was developed.

The workflow of the collaborative simulation is illustrated in Figure 1. When the collaborative simulation platform is started, Simulink and Fluent run proactively and work alternately, and the entire simulation process is fully automated. During the data transmission cycle of a certain time step, the two software packages continuously interchange data via the parallel collaborative simulation model. The computational efficiency can be improved by the time-division processing of the outdoor parameters and simplified processing of the lumped partition model of the air conditioning system.

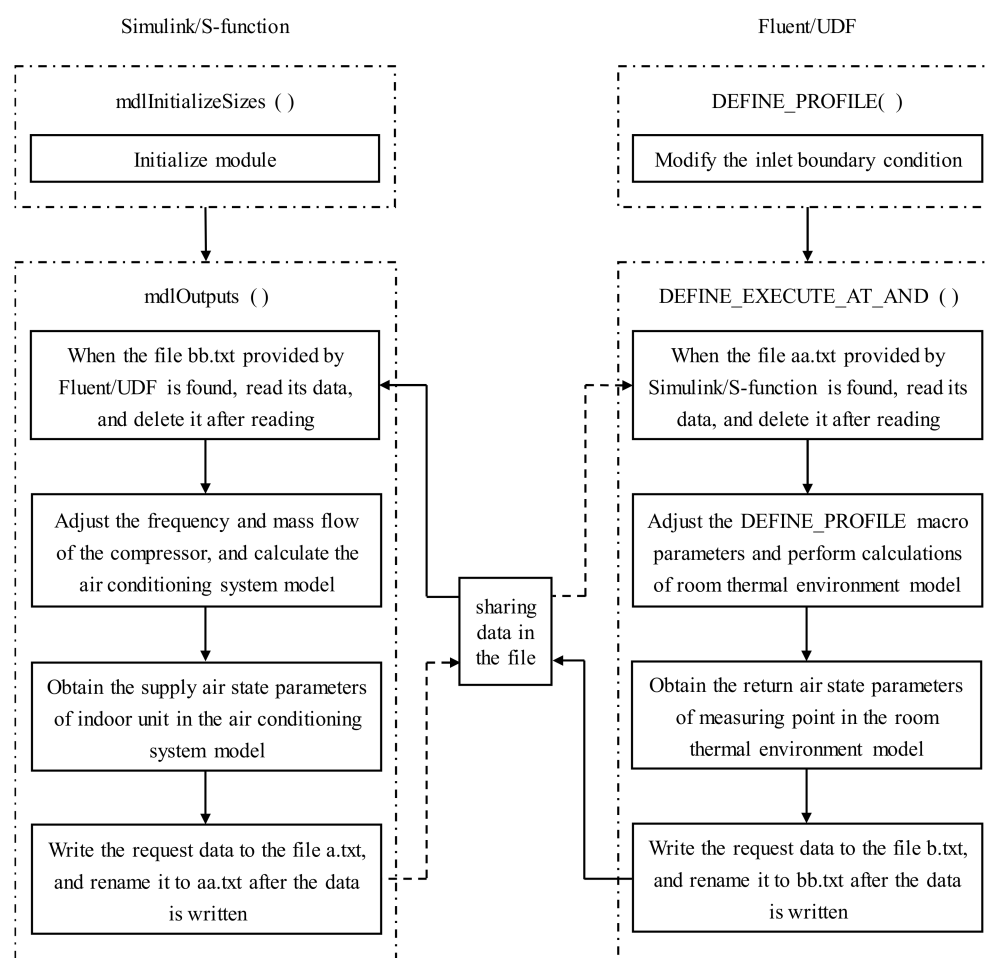


Figure 1. Workflow of collaborative simulation.

Under the premise of ensuring the correctness and rationality of the dynamic mathematical model, the proposed collaborative simulation platform can realize efficient dynamic coupling and collaborative prediction between the air conditioning system model and room thermal environment model. It includes the dynamic variation of the thermal environment in an air-conditioned room under actual working conditions. In addition, it can also be used for the controller design of the air conditioning system. Meanwhile, the parallel collaborative simulation model has a rationality that can utilize other software systems or programming languages for data coupling.

3. Collaborative Simulation/Mathematical Model

3.1. Model Assumptions

The establishment of an air conditioning system and room thermal environment models, and the collaborative simulation calculation of the overall model are based on the following assumptions:

1. The refrigerant flows in one dimension along the tube, and the pressure loss is neglected;
2. The air out of the tube flows in one dimension, and the physical properties are uniform;
3. Only the radial heat conduction of the tube wall is considered;
4. The phenomenon of air-side dehumidifying and frosting in the evaporator is not considered;
5. Indoor air is regarded as incompressible fluid, which conforms to Boussinesq hypothesis;
6. The radiant heat between solid surfaces, such as walls, ground, and roof, is ignored;
7. The indoor air flow is at a low speed, and the dissipated heat caused by the viscous force of the fluid is neglected;

8. The effect of air leakage on doors, windows, and walls was ignored, and the air tightness of the room was acceptable;
9. The influence of humidity distribution on the thermal environment is ignored.

3.2. Air Conditioning System Model

3.2.1. Mathematical Model of Air Conditioning System

In the dynamic simulation of an air-conditioning system, the compressor and expansion valve have the characteristics of fast action response and small thermal inertia; therefore, the steady-state lumped parameter method is adopted. In the compressor model, the refrigerant mass flow rate, compressor power consumption, and exhaust temperature can be calculated as follows:

$$m_{\text{com}} = \lambda V_{\text{th}} / v_{\text{suc}} \quad (1)$$

$$N_z = N_{\text{th}} / \eta_i \eta_m, \quad (2)$$

$$T_d = T_{\text{suc}} \varepsilon^{(k-1)/k} \quad (3)$$

where λ represents the gas transmission coefficient of the compressor, V_{th} is the theoretical volumetric capacity, v_{suc} is the suction specific volume of the refrigerant, N_{th} is the theoretical power consumption, η_i and η_m are the indicated efficiency and mechanical efficiency, respectively, T_{suc} is the suction temperature, ε is the pressure ratio, and k is the polytropic exponent of compression.

In the expansion valve model, the mass flow rate and outlet enthalpy of the refrigerant can be obtained as:

$$m_v = C_v A_v \sqrt{2 \rho_{\text{in},v} (P_{\text{in},v} - P_{\text{out},v})} \quad (4)$$

$$H_{\text{out},v} = H_{\text{in},v} \quad (5)$$

where C_v is the flow coefficient, A_v is the opening area, $\rho_{\text{in},v}$ is the refrigerant density at the inlet of the expansion valve, $P_{\text{in},v}$ and $P_{\text{out},v}$ are the inlet and outlet pressures, respectively, and $H_{\text{in},v}$ is the inlet enthalpy.

The condenser and evaporator have a significant influence on the vapor compression system characteristics, and the moving boundary method was developed to establish a dynamic zoned lumped parameter model. The condenser was divided into superheated, two-phase, and supercooled zones, whereas the two-phase and superheated zones were the only two zones in the evaporator. Taking the condenser model as an example, the governing equations were established, including the conservation equation of continuity and energy for the refrigerant and conservation equation of energy for the tube wall. The generalized governing equations for every phase region were integrated along the length of the tube. Subsequently, the compact and ultimate form of the matrix equation in the condenser model could be expressed by Equation (6), and the detailed expressions were respectively represented as Equations (7)–(10).

$$\dot{\mathbf{x}}_c = \mathbf{D}_c^{-1} \mathbf{f}(\mathbf{x}_c, \mathbf{u}_c) \quad (6)$$

$$\mathbf{x}_c = [L_{1,c}, L_{2,c}, P_c, H_{\text{out},c}, T_{w1,c}, T_{w2,c}, T_{w3,c}]^T \quad (7)$$

$$\mathbf{u}_c = [m_{\text{in},c}, H_{\text{in},c}, m_{\text{out},c}, m_{a,c}]^T \quad (8)$$

$$\mathbf{D}_c = \begin{bmatrix} d_{11} & 0 & d_{13} & 0 & 0 & 0 & 0 \\ d_{21} & d_{22} & d_{23} & 0 & 0 & 0 & 0 \\ d_{31} & d_{32} & d_{33} & d_{34} & 0 & 0 & 0 \\ d_{41} & d_{42} & d_{43} & 0 & 0 & 0 & 0 \\ d_{51} & 0 & 0 & 0 & d_{55} & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & d_{66} & 0 \\ d_{71} & d_{72} & 0 & 0 & 0 & 0 & d_{77} \end{bmatrix} \quad (9)$$

$$\mathbf{f}(\mathbf{x}_c, \mathbf{u}_c) = [f_1, f_2, f_3, f_4, f_5, f_6, f_7]^T \quad (10)$$

where x_c is the vector of the state variables, u_c is the vector of the control input parameters, D_c and $f(x_c, u_c)$ are the coefficient matrix and function matrix, respectively, and the corresponding expressions of matrix elements can be found in [23,24]. $L_{1,c}$ and $L_{2,c}$ are the lengths of the superheated zone and two-phase zone of the condenser, respectively. P_c is the condensing pressure, $H_{out,c}$ and $H_{in,c}$ are the outlet and inlet enthalpy, $T_{w1,c}$, $T_{w2,c}$, and $T_{w3,c}$ represent the tube wall temperatures of each phase region, $m_{in,c}$ and $m_{out,c}$ are the refrigerant mass flow rate at the inlet and outlet, and $m_{a,c}$ is the air mass flow rate.

Combining the relationship between the import and export parameters of each component, the visual modeling and simulation of the mathematical model of the air conditioning system is realized on the Simulink platform. The steady-state model of the compressor and expansion valve are considered as the boundary conditions of the system model to provide the inlet and outlet boundary parameters for the dynamic model of the condenser and evaporator. After initialization, the variable step size algorithm was used to solve the matrix equations in the above simulation model. Consequently, the overall convergence speed was faster, and the convergence effect was better than that of the fixed step size algorithm.

Although a two-dimensional or three-dimensional air-conditioning model collaborated with three-dimensional fluent model have the ability to describe the variations and the distributions of air flow in an air conditioned room, the calculations are highly restricted by the calculation speed and stability. In most cases, the iterations cannot be converged. In the current literature, very rare studies provided multi-dimensional air-conditioning models collaborating with three-dimensional fluent model. The one-dimensional air-conditioning model with phase partition cannot only overcome the difficulties in calculations of distributing parameter model, but also reflect the heat transfer characteristics and dynamic variation trend of refrigerant side in different phase regions. Considering such characteristics, the one-dimensional air-conditioning model was selected in this study.

3.2.2. Experiments in the Psychrometric Chamber

The psychrometric chamber was used to validate the simulation results and verify the accuracy of the proposed mathematical model of the air conditioning system. As illustrated in Figure 2, the space of the test equipment is divided into indoor and outdoor chambers by an insulated partition wall. The surrounding walls, ceiling, and floor were also insulated. The two chambers were equipped with a heater/cooler, humidifiers, and other air conditioning equipment to confirm the required indoor and outdoor working conditions. The pictures for these two chambers are shown in Figure 3.

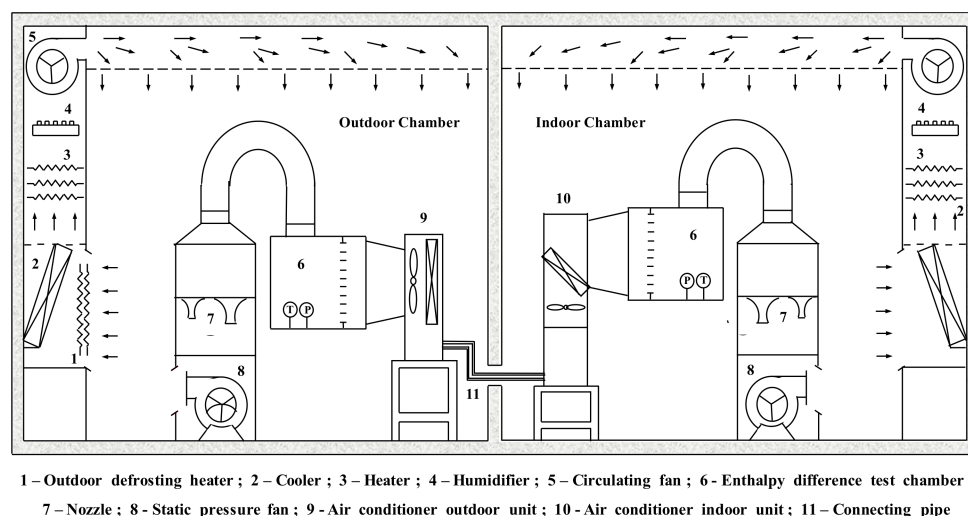


Figure 2. System diagram of the test equipment in enthalpy difference laboratory (“outdoor chamber” represents “chamber for testing outdoor unit”, and “indoor chamber” represents “chamber for testing indoor unit”).

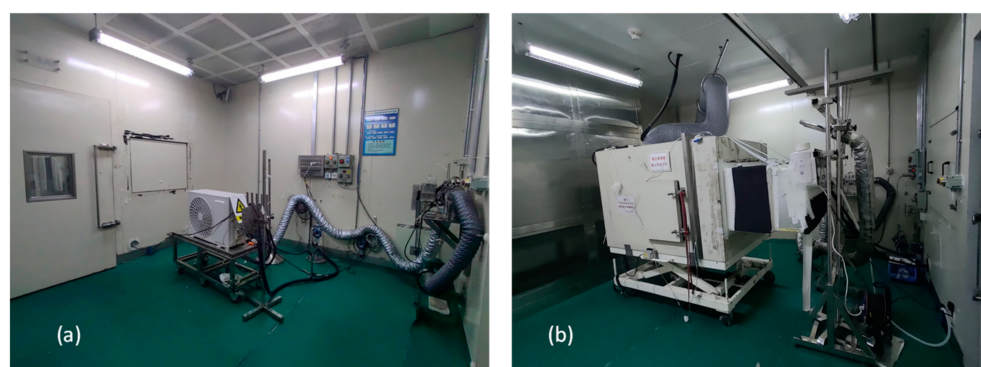


Figure 3. Pictures of the chamber: (a) outdoor chamber and (b) indoor chamber.

3.2.3. Verification of the Air Conditioning System Model

Table 1 lists the main structural parameters of a KFR-35GW type air conditioner. The indoor and outdoor conditions were standard refrigeration conditions, indoor air flow rate was $773.5 \text{ m}^3 \cdot \text{h}^{-1}$, fixed compressor operating frequency was 52 Hz, and refrigerant circulating mass flow rate in the system was $0.017 \text{ kg} \cdot \text{s}^{-1}$. Table 2 shows that when the air conditioning system is stabilized, the relative error between the simulation data and experimental results is less than 3%.

Table 1. Main structural parameters of the air conditioner.

Components	Structural Parameters
Compressor	Rotary compressor, nominal working volume = 10.3 cm^3 , and
Valve	valve size = 2.2 mm
Evaporator	The process is two in and two out, two rows, a single tube with a length of 634 mm, 16 U-shaped tubes, tube outer diameter = 5 mm, tube spacing = 19.05 mm, row spacing = 11.4 mm, louvered fin thickness = 0.095 mm, fin spacing = 1.4 mm, a total of 422 pieces
Condenser	The process is four in and four out, two rows, a single tube with a length of 760 mm, 24 U-shaped tubes, tube outer diameter = 7 mm, tube spacing = 22 mm, row spacing = 19.05 mm; corrugated fin thickness = 0.095 mm, fin spacing = 1.4 mm, a total of 1115 pieces

Table 2. Relative error of main parameters in air conditioning model.

Parameters	Experiment	Simulation	Error (%)
Condensing pressure (kPa)	2608.0	2662.1	2.11
Evaporating pressure (kPa)	1018.0	1003.5	1.42
Refrigerating capacity (W)	3361.6	3396.6	1.01
Compressor power consumption (W)	769.7	773.1	0.44
Indoor air drying bulb temperature ($^{\circ}\text{C}$)	15.53	15.41	−0.77

3.3. Thermal Environment Model of the Room

3.3.1. Mathematical Model of the Room Thermal Environment

To numerically evaluate the thermal environment of an air-conditioned room, parameters such as supply and return air in the room were set, and air current and temperature distribution were calculated. Figure 4 shows that the air-conditioned room is simplified into a 3D geometric model with the dimensions of $5.2 \text{ m} \times 3.5 \text{ m} \times 2.67 \text{ m}$ in a ratio of 1:1, which mainly includes walls, air supply outlet, return air inlet, and windows. The origin of coordinates was the center of the bottom floor. The air conditioner indoor unit was installed at the center of the front wall in the X direction, at a height of 2.3 m from the

floor. The air supply outlet was in the lower front of the air conditioner, and the return air inlet was placed above the air conditioner.

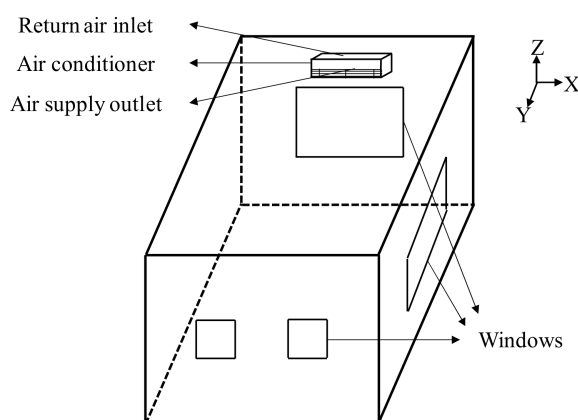


Figure 4. Geometric model of an air-conditioned room.

A hexahedral structured grid was adopted for mesh generation, and the grids at the air supply outlet and return air inlet were properly encrypted. The mathematical model applied the standard $k-\varepsilon$ 3D turbulence model, in which the governing equations comprised the conservation equations for mass, momentum, and energy. The general form of the equations can be expressed as follows:

$$\frac{\partial(\rho\phi)}{\partial t} + \text{div}(\rho u\phi) = \text{div}(\Gamma \cdot \text{grad}\phi) + S, \quad (11)$$

where ρ is the density of the fluid, ϕ is a generic variable that represents the velocity, temperature, and other solving variables, t is the time, u is the velocity, Γ is the generalized diffusion coefficient, and S is the generalized source term.

The boundary condition settings of the model are listed in Table 3. The air supply velocity direction was perpendicular to the boundary surface of the air supply outlet. The controllable temperature of the exterior wall was constant in the laboratory, resulting in variations in the interior wall temperature. The governing equations were used to load every boundary condition, energy transfer of air flow and wall surface was calculated, and temperature and velocity fields of the room were determined.

Table 3. Settings of the boundary conditions.

Boundary Name	Type	Setting Project	Unit	Settings
Air flow	Fluid	Material	/	Air
Wall	Polyurethane board	Constant wall temperature	K	308
Air supply outlet	Velocity-inlet	Speed	$\text{m}\cdot\text{s}^{-1}$	3
		Temperature	K	UDF
Return air inlet	Outflow	Gauge pressure	Pa	0

3.3.2. Experimental Work in the Thermal Environment Laboratory

To verify the accuracy of the mathematical model of the room thermal environment, an experimental test was carried out in an air-conditioned testing laboratory with standard test conditions. Under the refrigeration condition, the dry bulb temperature of the outdoor air is 35 ± 0.5 °C, the wet bulb temperature of the outdoor air is 24 ± 0.5 °C, and the setting temperature of the air conditioner is 27 °C. Under the heating condition, the corresponding environmental parameters are 7 ± 0.5 °C, 6 ± 0.5 °C, and 20 °C, respectively. The thermal

load of the laboratory is 70% of its rated capacity. Under the test conditions, when the indoor thermal environment is finally stabilized, the average room temperature is 26 °C, with a deviation of 1 °C from the set temperature, while the average relative humidity of the room is ultimately maintained at about 40%. Figure 5 illustrates that the laboratory was composed of an indoor chamber and an outdoor chamber. The indoor unit was placed in the indoor room to test the thermal comfort of the environment created by the air conditioner. The outdoor and environmental control units were placed in the outdoor room. The outdoor dry and wet bulb temperatures and other parameters could be adjusted by the environmental control unit. A total of 147 temperature measurement points were arranged in the room space, as per the pictures shown in Figure 6. Data collection began after the air conditioner was tested, with a data collection interval of 1 min, and the test duration was 3 h as required by the standard: that is, 2 h after the test air conditioner starts or enters the specified stable state, plus 1 h of continuous data recording.

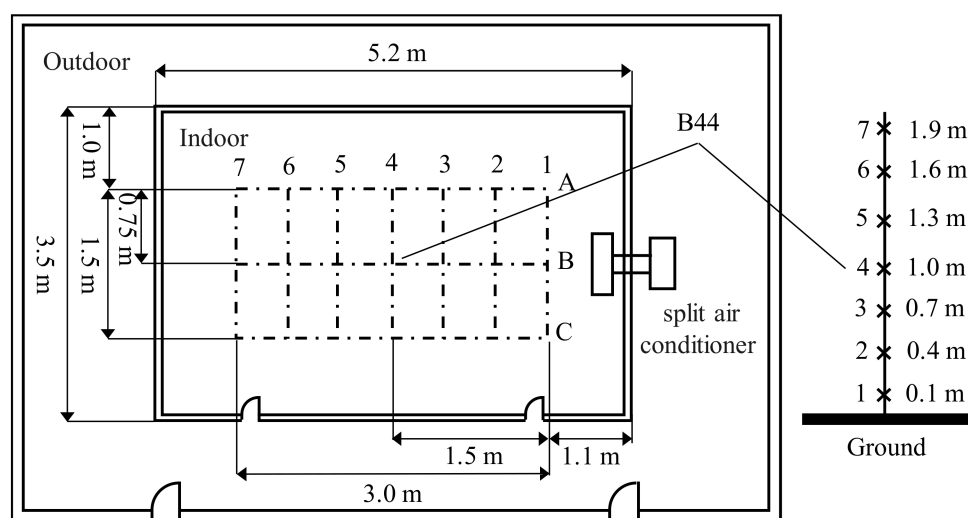


Figure 5. Schematic of an air-conditioned testing laboratory with a comfortable thermal environment.



Figure 6. Pictures of the air-conditioned testing laboratory used in this study: (a) the appearance of the inner chamber; (b) the interior of inner chamber from the door side; (c) the interior of inner chamber from the window side.

3.3.3. Verification of the Thermal Environment Model

During the model verification, the dynamic air supply outlet parameters of the air conditioner were defined. The air supply temperature was fitted into a function varying with time, followed by compiling and loading the UDF of the air supply temperature into the Fluent model. Considering the fact that during thermal environment test, there is no humidity source placed in the test chamber, the humidity effect gradually weakens. Besides, based on the experimental results of different groups, the sensible heat ratio is basically the same (around 92%). Therefore, the variations of psychrometrics are not included in the current study. Figure 7a,b, respectively, present a comparison of the numerical simulation results of room temperature under steady state and transient processes with the experimen-

tal results. When the room thermal environment reaches the steady state, the relative error between the simulated temperatures of 147 measuring points and experimental results is less than 5%. In addition, the hourly relative error between the mean value of the simulated temperatures in the transient process and those obtained experimentally is less than 10%. The other error indexes are listed in Table 4, indicating that the model has an acceptable accuracy [28].

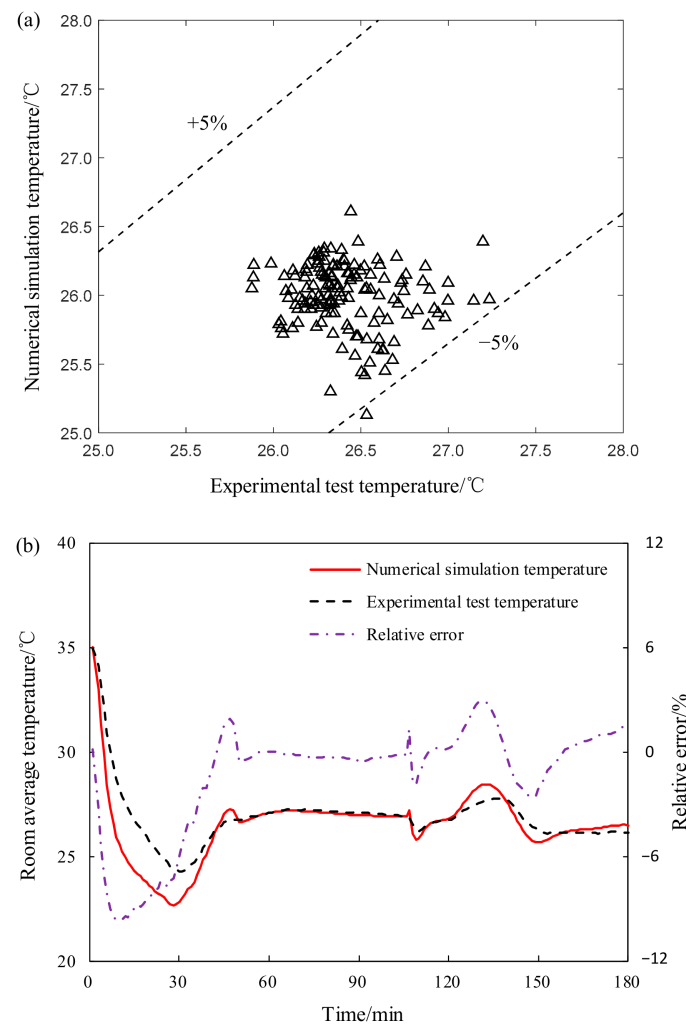


Figure 7. Comparison of the simulation and experimental results at room temperature: (a) steady state; (b) transient process.

Table 4. Error evaluation indexes of the simulation results.

Type of Error	Steady	Transient
RMSE	1.44	3.20
MAPE (%)	3.19	11.48
CVRMSE (%)	0.89	−2.32

4. Collaborative Simulation Results and Discussions

4.1. Prediction and Evaluation of the Thermal Environment Using Collaborative Simulation

During the collaborative simulation, the structural parameters, listed in Table 1, were adopted to the air conditioning system model, and the boundary conditions listed in Table 3 were used in the room thermal environment model. The target room temperature was set at 27 °C according to the standard requirements [1]. The system controller used the difference

between the return air and set temperatures as the input and the compressor operating frequency as the output; thus, the room temperature could be adjusted. Table 5 describes the designed control strategy. Meanwhile, when sharing the data files, the process blocking time was set to 30 s.

Table 5. System controller strategy.

Specific Control Logic	
A	When the room temperature is 2 °C higher than the set temperature, the operating frequency gradually increases to the target high frequency of 85 Hz from the strike frequency of 15 Hz, then the compressor temporarily operates at this frequency.
B	When the temperature difference e is in the range of 0–2 °C, the operating frequency decreases step by step. We assume $f = f_{t-1} - 3e$, where f_{t-1} is the operating frequency at the last interaction moment.
C	When the room temperature is lower than the set temperature, the operating frequency continues to decrease, and it is assumed that $f = f_{t-1} + 9e$ until the operating frequency reaches a minimum of 15 Hz.

Figure 8a demonstrates the dynamic variations in the operating frequency, air supply temperature, and average room temperature. The air conditioning system and room thermal environment become stable after approximately 30 min. Subsequently, a cross section of $X = 0$ m was selected. As shown in Figure 8b, after 1 min, the operating frequency of the air conditioner increases gradually during the start-up, the air supply temperature is still high, the room is in the initial stage of cooling, and the air is not appropriately mixed. As illustrated in Figure 8c, after 10 min, the operating frequency gradually decreases; however, the air supply temperature reaches its lowest value. When the air hits the top surface of the wall, it spreads rapidly along the wall, and the air temperature near the top surface is approximately 22 °C. As the rest of the room is not uniformly cooled, the temperature is within the range of 25–27 °C. Thus, there is an evident temperature gradient in the room, and the temperature distribution is uneven. When the collaborative simulation time reaches 50 min, the ultimate temperature gradient is smaller and the temperature distribution is more uniform, as shown in Figure 8d.

Additionally, the thermal environment indices were analyzed and evaluated.

(1) Temperature deviation represents the difference between the stable temperature of the air-conditioned room and set temperature. The temperature deviation in the simulation was 0.17 °C.

(2) Temperature uniformity was utilized to evaluate the differences between the concurrent air temperatures at different measuring points. The instantaneous temperature uniformity at a certain moment is represented by the standard deviation of all the measuring points at this time. In the simulation, the temperature uniformity of the entire indoor thermal environment was constant at 0.307 according to the instantaneous temperature uniformity over the last 20 min.

(3) The vertical air temperature difference was used to evaluate the thermal discomfort of the human body. Temperatures of the measuring points at the head (assumed height = 1.6 m) and ankle (assumed height = 0.1 m) in the same vertical line direction were obtained, and the average value of differences was calculated over a period of 20 min. As the temperature distribution difference at the air supply outlet could not be considered in the simulation, the vertical air temperature difference was -0.365 °C, that is, less than 0 °C.

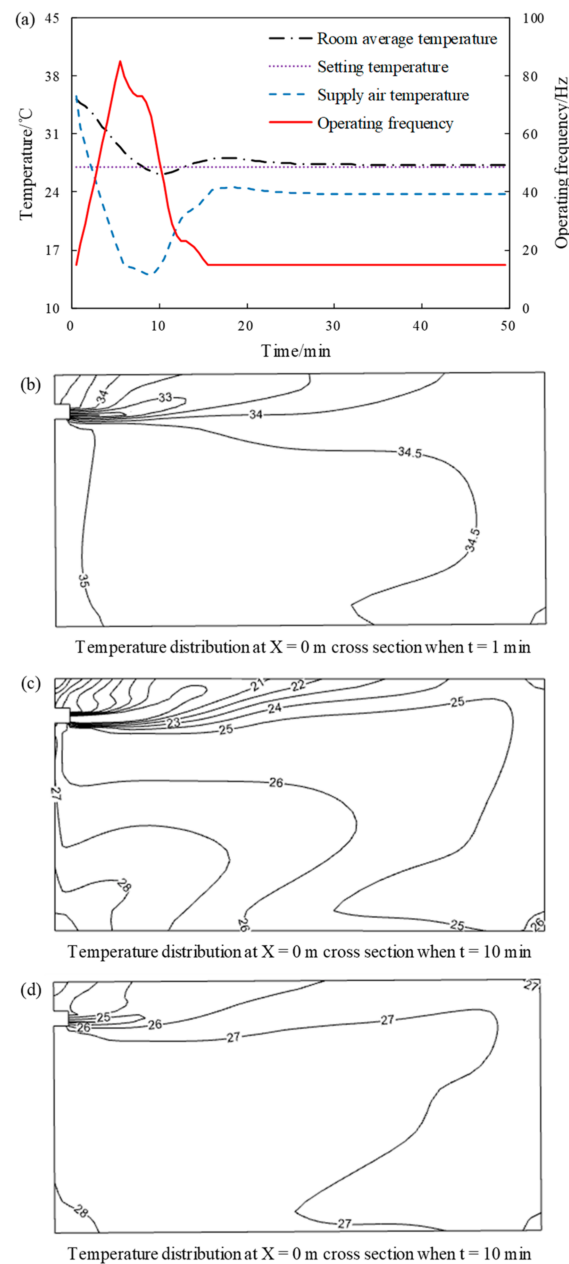


Figure 8. Change process of the room local temperature: (a) Variations of the main dynamic parameters with time; (b–d) Temperature distribution of profile $X = 0$ m at different times.

(4) PMV is an internationally recognized comprehensive evaluation index that considers many factors related to human thermal comfort. In particular, PMV considers the seven-level thermal sensation evaluation standard, set using a large sample of people, as the thermal comfort index. PMV evaluation and human thermal load calculation models are expressed as follows:

$$PMV = (0.303e^{-0.036M} + 0.028)TL \quad (12)$$

$$TL = (M - W) - 3.05[5.733 - 0.007(M - W) - P_a] - 0.42 \times \\ - 58.15) - 1.73 \times 10^{-3}M(5.867 - P_a) - 1.4 \times 10^{-3} \times M(34 - t_a) \\ - 3.96 \times 10^{-8}f_{cl}[(t_{cl} + 273)^4 - (t_r + 273)^4] - f_{cl}h_c(t_{cl} - t_a) \quad (13)$$

where M is the metabolic rate of the human body, W is the external work, P_a is the partial pressure of water vapor, t_a is the air temperature, f_{cl} is the clothing area coefficient, which is related to the thermal resistance of clothing, t_{cl} is the surface temperature of clothing, t_r is the mean radiant temperature, and h_c is the convective heat transfer coefficient, which is evaluated from the relative air velocity v_{ar} , a function of the local air velocity v_a and the metabolic rate M [29] as noted in Table 6. The six basic parameter values involved in the calculation of PMV are presented in Table 6. The metabolic rate of the human body was selected under the active state of sitting position. Moreover, the thermal resistance of clothing was selected when wearing underpants, short sleeve shirts, light pants, thin shorts, and shoes. Other parameters could be deduced or iterated using the above basic parameters, and the PMV was 0.84. It should be noted that although including human occupants by organizing them in a series test may give different results, this study aims to simulate the behaviors of the air conditioners when they are in a thermal environment/comfort test. In this test, the PMV is one of the indices with the theoretical correlation provided in the standard. Therefore, in the current study, the experiments with human occupants are not involved.

Table 6. Basic parameters calculated using PMV.

Parameter	Value	Parameter	Value
Air temperature (°C)	27.2	Mean radiant temperature (°C)	30.0
Relative humidity (%)	39.40	Human metabolic rate ($W \cdot m^{-2}$)	70
Relative air velocity ($m \cdot s^{-1}$) *	0.26	Clothing thermal resistance ($m^2 \cdot K \cdot W^{-1}$)	0.080

* $v_{ar} = v_a + 0.0052(M - 58.2)$.

(5) DR represents the percentage of people who are dissatisfied owing to the air flow dissipating human body heat. Its magnitude depends on the indoor temperature, air velocity, turbulence intensity, physical activity level, and clothing of the person. The local DR of measuring point i during collection time can be calculated as follows:

$$DR_i = (34 - t_a)(v_a - 0.05)^{0.62}(0.37v_a T_u + 3.14), \quad (14)$$

where T_u is the local turbulence intensity, which is 40%. The DR value calculated using the collaborative simulation was 6.67%.

(6) The air cooling rate is defined as a rate at which the thermal environment of a room becomes steady state. In the simulation, when the average room temperature dropped from 35.5 to 27.2 °C, the stabilization time was approximately 31 min, and the cooling rate of the air-conditioned room was $0.269 \text{ } ^\circ\text{C} \cdot \text{min}^{-1}$.

Therefore, the proposed collaborative simulation method can represent the interactions among the system structure parameters, control parameters, and thermal environment to a certain extent. Although manufacturers have to mass produce HVAC systems integrated in multiple zones, the background of this study is to simulate the behaviors of the air conditioners when it is in a thermal environment/comfort test. Such a test is the one that each type of the products need to pass before putting into the market. Thus, the simulation is based on the same laboratory and the standard conditions that are used in the thermal environment/comfort test.

4.2. Effects of Structural and Control Parameters on the Thermal Environment

In exploring the thermal comfort in an air-conditioned room, environmental factors, such as outdoor working conditions, room size, and air conditioner position need to satisfy or maintain the requirements in the standards. Therefore, we focused on analyzing the effect of the evaporator structure and air supply angle on the prediction and evaluation of the room thermal environment. Table 7 lists the benchmark values of structural and control parameters used in the simulation.

Table 7. Benchmark parameter values in the simulation.

Parameter	Tube Spacing (mm)	Tube Outer Diameter (mm)	Fin Spacing (mm)	Air Supply Angle (°)
Benchmark	19.05	5.00	1.40	90

4.2.1. Effect of the Evaporator Structure

Tube outer diameter, tube spacing, and fin spacing are important structural parameters of a finned tube heat exchanger, which affect its heat transfer capacity (the pressure drop characteristics were not considered). Here, a finned-tube evaporator was used to obtain ten groups of structural parameters by varying the reference parameters by 40%, as presented in Table 8. Because the structural parameters do not directly affect the room thermal environment, dynamic variations of the relevant parameters in the simulation are not displayed.

Table 8. Structural parameters of different evaporators.

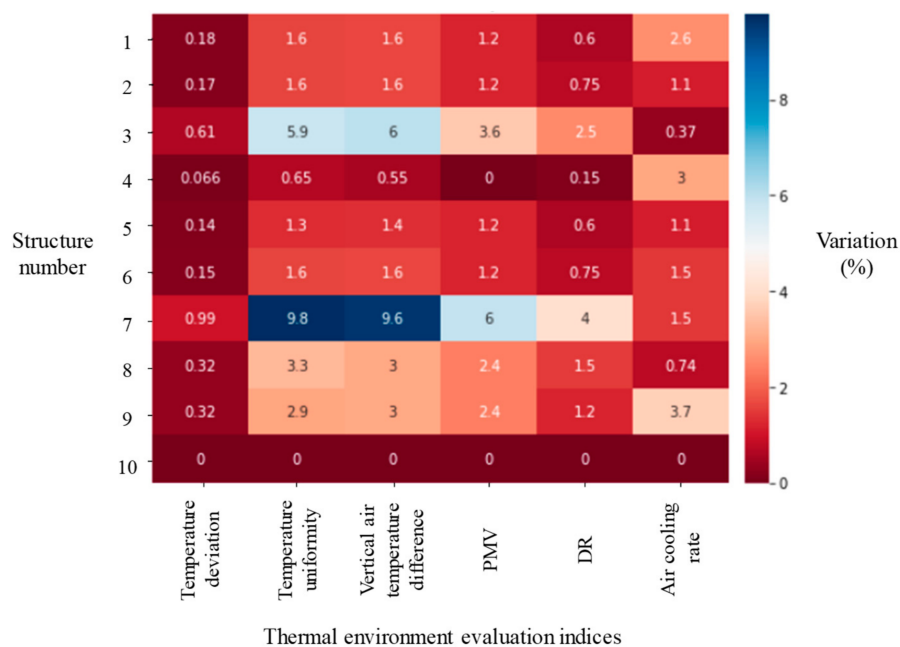
Structure Number	Tube Spacing (mm)	Tube Outer Diameter (mm)	Fin Spacing (mm)
1	18.10	5.00	1.40
2	20.00	5.00	1.40
3	22.86	5.00	1.40
4	19.05	4.75	1.40
5	19.05	6.00	1.40
6	19.05	7.00	1.40
7	19.05	5.00	1.12
8	19.05	5.00	1.33
9	19.05	5.00	1.47
+10 *	19.05	5.00	1.40

* Evaporator reference structure.

As shown in Table 9 and Figure 9, within the specified range, larger tube spacing, larger tube outer diameter and smaller fin spacing (structures 3, 6, and 7) tend to increase the amount of heat transfer, resulting in a lower room temperature when the room thermal environment reaches a steady state, whereas the temperature uniformity and vertical air temperature difference increase, PMV decreases, and DR increases. Considering the results for structure 3, variations of the stable temperature, temperature uniformity, vertical air temperature difference, PMV, DR, and air cooling rate are -0.61 , 5.86 , 6.03 , -3.57 , 2.55 , and 0.37% , respectively. Moreover, the evaporator structure has a significant effect on the temperature uniformity, vertical air temperature difference, PMV, and DR. When the benchmark parameters vary identically, the relevant evaluation indices are sensitive to the evaporator structural parameters, i.e., fin spacing, tube spacing, and tube outer diameter, in that order. Considering the PMV values of structures 7, 3, and 5 with a 20% change from the benchmarks, the variations are -5.95 , -3.57 , and -1.19% , respectively.

Table 9. Effects of the evaporator structure on the thermal environment.

Structure Number	Temperature Deviation (°C)	Temperature Uniformity (-)	Vertical Air Temperature Difference (°C)	PMV (-)	DR (-)	Air Cooling Rate (°C·min ⁻¹)
1	0.220	0.302	−0.359	0.85	6.63%	0.276
2	0.125	0.312	−0.371	0.83	6.72%	0.266
3	0.004	0.325	−0.387	0.81	6.84%	0.270
4	0.188	0.305	−0.363	0.84	6.66%	0.277
5	0.133	0.311	−0.370	0.83	6.71%	0.266
6	0.128	0.312	−0.371	0.83	6.72%	0.265
7	−0.099	0.337	−0.400	0.79	6.94%	0.273
8	0.083	0.317	−0.376	0.82	6.77%	0.267
9	0.256	0.298	−0.354	0.86	6.59%	0.279
10	0.170	0.307	−0.365	0.84	6.67%	0.269

**Figure 9.** Color map of the evaporator structure effects on the thermal environment.

4.2.2. Effect of the Air Supply Angle

As illustrated in Figure 10, the air supply angle of the air conditioner indoor unit was set to 105°, 90°, and 75°. As shown in Figure 11a, when the air supply angle is 105°, the low-temperature air from the air supply outlet is closer to the return air inlet, which directly results in a low return air temperature as the input for the control model. Consequently, after the air conditioner starts, the operating frequency does not reach the set high-frequency and subsequently declines. Therefore, the room cooling process is slow, and a high stable temperature is achieved. Figure 11b shows that when the air supply angle is 75°, the temperature overshoot process causes the time to reach a steady state temperature to be approximately identical to that of the air supply angle of 105°; however, the stable temperature is lower and there is a certain temperature fluctuation.

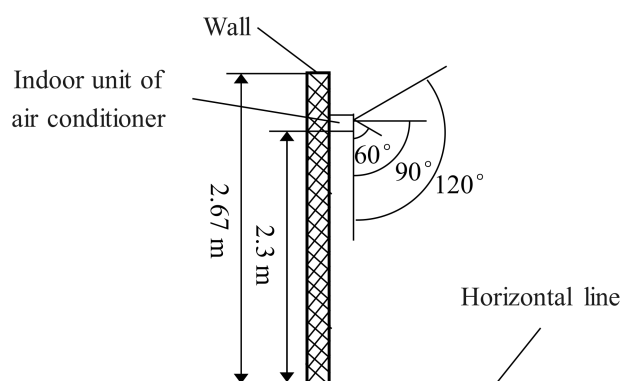


Figure 10. Air supply angle of indoor unit.

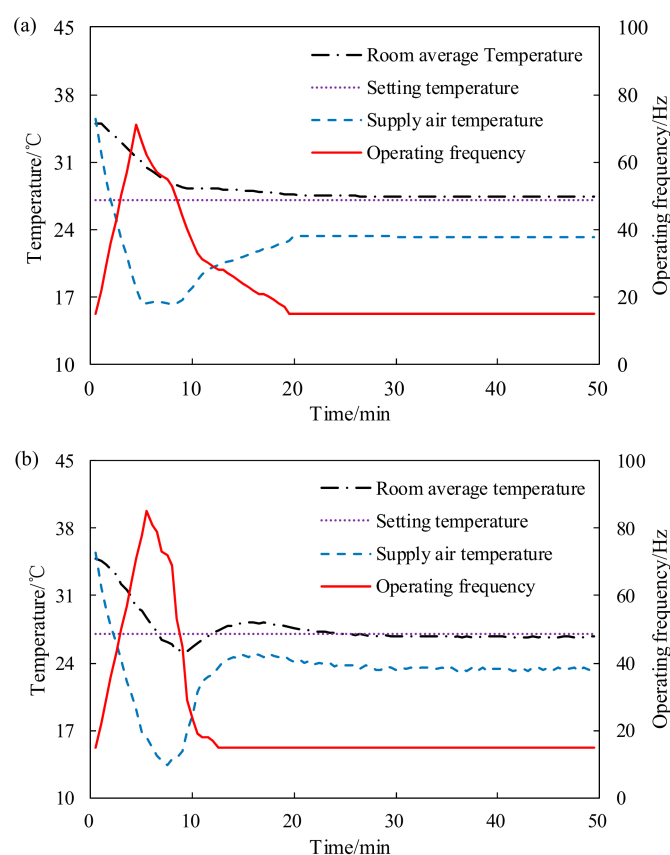


Figure 11. Dynamic variation of room temperature when varying air supply angle: (a) 105°; (b) 75°.

Table 10 presents the effects of the air supply angle on the thermal environment. When the air supply angle is 90°, the steady state temperature, temperature uniformity, vertical air temperature difference, PMV, DR, and air cooling rate change by 0.33, 5.54, 40.00, 13.10, −21.44, and −2.60% compared to those of the air supply angle of 105°. Moreover, when the air supply angle is 75°, variations of each evaluation index are −2.40, 22.48, 117.79, −17.86, 18.59, and 7.43%, respectively. Thus, the effect of air supply angle on the vertical air temperature difference is evident; however, the effect on the PMV, DR, temperature uniformity, and other indices does not indicate consistent differences. Different indices may lead to different optimal air supply angles. Therefore, variations of air supply angle has a complex influence on the room thermal environment.

Table 10. Effects of the air supply angle on the thermal environment.

Air Supply Angle	Temperature Deviation (°C)	Temperature Uniformity (-)	Vertical Air Temperature Difference (°C)	PMV (-)	DR (-)	Air Cooling Rate (°C·min ⁻¹)
75°	−0.482	0.376	−0.784	0.69	7.91%	0.289
90°	0.170	0.307	−0.365	0.84	6.67%	0.269
105°	0.259	0.324	−0.511	0.95	5.24%	0.262

Consequently, the structural and control parameters of air conditioning systems differently affect the thermal environment state and evaluation indices of the air-conditioned room. The obtained results help test the thermal comfort of air conditioners, reasonably improve the detection efficiency of the laboratory, and provide a theoretical reference for designing parameters and optimizing air conditioners.

It should be noted that the current model may not be able to analyze the non-uniform flow caused by the indoor heat exchanger and the air supply outlet. However, it is able to analyze the effect caused by the structure variations of main components and control methods in the vapor compression cycle. Structural parameters and control strategies have impacts on the heat transfer capacity, response time, and efficiency, all of which further affect the supply air status and the dynamic variations of thermal environment state. The differences in the supply air condition and cooling rate may lead to the variations on the other evaluation indices of the thermal environment.

5. Conclusions

To predict dynamic variations of a thermal environment in an air-conditioned room, a collaborative simulation method was proposed to achieve efficient coupling and collaborative prediction between the air conditioning system model and room thermal environment model. The prediction error of the proposed model was less than 10%. Due to the lack of actual control strategy, the idealization of the room model, and the simplification of the air outlet structure, the collaborative simulation model has some deviation with the experimental results. However, the collaborative simulation method can represent the interactions among the structural parameters, control parameters, and room thermal environment. The effects of the evaporator structure and air supply angle on the thermal environment state and evaluation indices were thoroughly analyzed using the collaborative simulation method.

The results demonstrated that both the structural and control parameters differently affected the thermal environment state and evaluation indices of the air-conditioned room within a certain parameter range. The evaporator structure exhibited a more significant effect on the temperature uniformity and vertical air temperature difference, followed by PMV and DR. The evaluation indices were sensitive to the evaporator structural parameters, i.e., fin spacing, tube spacing, and tube outer diameter, in this order. The effect of the air supply angle on the vertical air temperature difference was evident; however, its influence on the PMV, DR, and temperature uniformity did not indicate consistent differences.

Author Contributions: Conceptualization, L.H., S.Z. and S.C.; methodology, G.X., X.W. and S.C.; validation, X.W.; formal analysis, G.X. and X.W.; investigation, S.Z.; resources, S.Z.; writing—original draft preparation, G.X.; writing—review and editing, S.C. and J.X.; supervision, J.X.; project administration, L.H. and S.C.; funding acquisition, L.H. All authors have read and agreed to the published version of the manuscript.

Funding: This research was funded by the State Key Laboratory of Air-conditioning Equipment and System Energy Conservation (Grant No. ACSKL2019KT11).

Institutional Review Board Statement: The study was conducted according to the guidelines of the Declaration of Helsinki and approved by the Institutional Review Board (or Ethics Committee) of Huazhong University of Science and Technology (430074, 27 August 2021).

Informed Consent Statement: Not applicable.

Data Availability Statement: The data that support the findings of this study are available from the corresponding author upon reasonable request.

Conflicts of Interest: The authors declare no conflict of interest.

References

1. ISO. ISO Standard 7730. *Ergonomics of the Thermal Environment—Analytical Determination and Interpretation of Thermal Comfort Using Calculation of the PMV and PPD Indices and Local Thermal Comfort Criteria*; International Organization for Standardization: Geneva, Switzerland, 2005.
2. ISO. ISO Standard 17772-1. *Energy Performance of Buildings—Indoor Environmental Quality—Part 1: Indoor Environmental Input Parameters for the Design and Assessment of Energy Performance of Buildings*; International Organization for Standardization: Geneva, Switzerland, 2017.
3. Carlucci, S.; Bai, L.; de Dear, R.; Yang, L. Review of adaptive thermal comfort models in built environmental regulatory documents. *Build. Environ.* **2018**, *137*, 73–89. [\[CrossRef\]](#)
4. Mishra, A.K.; Loomans, M.G.L.C.; Hensen, J.L.M. Thermal comfort of heterogeneous and dynamic indoor conditions—An overview. *Build. Environ.* **2016**, *109*, 82–100. [\[CrossRef\]](#)
5. Horikiri, K.; Yao, Y.; Yao, J. Numerical optimisation of thermal comfort improvement for indoor environment with occupants and furniture. *Energy Build.* **2015**, *88*, 303–315. [\[CrossRef\]](#)
6. Chang, H.; Cheng, L. Simulation calculation of air conditioning system under complex vehicle conditions. *E3S Web Conf.* **2021**, *233*, 04024. [\[CrossRef\]](#)
7. Lee, Y.; Park, S.; Kang, S. Performance analysis of a solid desiccant cooling system for a residential air conditioning system. *Appl. Eng.* **2021**, *182*, 116091. [\[CrossRef\]](#)
8. Danca, P.; Bode, F.; Nastase, I.; Meslem, A. CFD simulation of a cabin thermal environment with and without human body—Thermal comfort evaluation. *E3S Web Conf.* **2018**, *32*, 01018. [\[CrossRef\]](#)
9. Zhang, F.; Ryu, Y. Simulation study on indoor air distribution and indoor humidity distribution of three ventilation patterns using computational fluid dynamics. *Sustainability* **2021**, *13*, 3630. [\[CrossRef\]](#)
10. Wang, Z.; Li, J.; Li, T.; Li, S. Research on Simulink/Fluent collaborative simulation zooming of marine gas turbine. *Appl. Comput. Intell. Soft Comput.* **2017**, *2017*, 1–8. [\[CrossRef\]](#)
11. Han, T.; Tang, S.Q.; Huang, C.Q.; Dong, K.S. Fluent/Simulink collaborative simulation for missile separation from cavity. *Adv. Mater. Res.* **2012**, *591–593*, 1902–1906. [\[CrossRef\]](#)
12. Liu, B.; Chen, H.; Zhang, T.; Pei, P. A vehicular proton exchange membrane fuel cell system co-simulation modeling method based on the stack internal distribution parameters monitoring. *Energy Convers. Manag.* **2019**, *197*, 111898. [\[CrossRef\]](#)
13. Nunes, T.K.; Vargas, J.V.C.; Ordonez, J.C.; Shah, D.; Martinho, L.C.S. Modeling, simulation and optimization of a vapor compression refrigeration system dynamic and steady state response. *Appl. Energy* **2015**, *158*, 540–555. [\[CrossRef\]](#)
14. Bonilla, J.; Yebra, L.J.; Dormido, S.; Cellier, F.E. Object-oriented modeling of switching moving boundary models for two-phase flow evaporators. *IFAC Proc. Vol.* **2012**, *45*, 1069–1074. [\[CrossRef\]](#)
15. Kim, D.; Ziviani, D.; Braun, J.E.; Groll, E.A. A moving boundary modeling approach for heat exchangers with binary mixtures. *Energy Procedia* **2017**, *129*, 466–473. [\[CrossRef\]](#)
16. Bonilla, J.; Dormido, S.; Cellier, F.E. Switching moving boundary models for two-phase flow evaporators and condensers. *Commun. Nonlinear Sci.* **2015**, *20*, 743–768. [\[CrossRef\]](#)
17. Zhai, Z.J.; Chen, Q.Y. Performance of coupled building energy and CFD simulations. *Energy Build.* **2005**, *37*, 333–344. [\[CrossRef\]](#)
18. Singh, S.; Abbassi, H. 1D/3D transient HVAC thermal modeling of an off-highway machinery cabin using CFD-ANN hybrid method. *Appl. Eng.* **2018**, *135*, 406–417. [\[CrossRef\]](#)
19. Gowreesunker, B.L.; Tassou, S.A.; Kolokotroni, M. Coupled TRNSYS-CFD simulations evaluating the performance of PCM plate heat exchangers in an airport terminal building displacement conditioning system. *Build. Environ.* **2013**, *65*, 132–145. [\[CrossRef\]](#)
20. Cucca, G.; Ianakiev, A. Assessment and optimisation of energy consumption in building communities using an innovative co-simulation tool. *J. Build. Eng.* **2020**, *32*, 101681. [\[CrossRef\]](#)
21. Yang, C.; Chen, Y.; Li, N.; Sun, Y.; Wu, R. In EnergyPlus and Python co-simulation model to support machine learning-based control of ground-source heat pump system. In Proceedings of the 11th International Symposium on Heating, Ventilation and Air Conditioning, ISHVAC, Harbin, China, 12–15 July 2019; pp. 759–766.
22. Seo, B.; Yoon, Y.B.; Yu, B.H.; Cho, S.; Lee, K.H. Comparative analysis of cooling energy performance between water-cooled VRF and conventional AHU systems in a commercial building. *Appl. Therm. Eng.* **2020**, *170*, 114992. [\[CrossRef\]](#)
23. Lymperopoulos, G.; Ioannou, P. Distributed adaptive HVAC control for multi-zone buildings. In Proceedings of the 2019 IEEE 58th Conference on Decision and Control (CDC), Nice, France, 11–13 December 2019; pp. 8142–8147.
24. Bleil De Souza, C. Contrasting paradigms of design thinking: The building thermal simulation tool user vs. the building designer. *Autom. Constr.* **2012**, *22*, 112–122. [\[CrossRef\]](#)

25. Moreno Santamaria, B.; Del Ama Gonzalo, F.; Pinette, D.; Gonzalez-Lezcano, R.; Lauret Aguirregabiria, B.; Hernandez Ramos, J.A. Application and Validation of a Dynamic Energy Simulation Tool: A Case Study with Water Flow Glazing Envelope. *Energies* **2020**, *13*, 3203. [[CrossRef](#)]
26. Lam, K.P.; Wong, N.H.; Mahdavi, A.; Chan, K.K.; Kang, Z.; Gupta, S. SEMPER-II: An internet-based multi-domain building performance simulation environment for early design support. *Automat. Constr.* **2004**, *13*, 651–663. [[CrossRef](#)]
27. Jaegler, A.; Burlat, P. Carbon friendly supply chains: A simulation study of different scenarios. *Prod. Plan Control* **2012**, *23*, 269–278. [[CrossRef](#)]
28. Haberl, J.; Claridge, D.; Culp, C. ASHRAE guideline 14-2002 for measurement of energy and demand savings: How to determine what was really saved by the retrofit. In Proceedings of the 5th International Conference for Enhanced Building Operations, Pittsburgh, PA, USA, 11–13 October 2005.
29. D'Ambrosio Alfano, F.R.; Olesen, B.W.; Palella, B.I.; Pepe, D.; Riccio, G. Fifty Years of PMV Model: Reliability, Implementation and Design of Software for Its Calculation. *Atmosphere* **2020**, *11*, 49. [[CrossRef](#)]