Journal of Research and Applications in Mechanical Engineering
ISSN: 2229-2152 (Print); 2697-424x (Online)
(2025) Vol. 13, No. 2, Paper No. JRAME 25-13-031
[DOI: 10.14456/jrame.2025.31]

Research Article

The Air Distribution Simulation inside the Glass Showroom Using Radiant Floor and Convection Cooling Systems

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Received 22 September 2024
Revised 29 October 2024
Accepted 3 November 2024

Abstract:
This research focused on finding velocity and temperature distributions of air inside a glass showroom since showroom demands are growing according to electrical vehicle makers are extended. The showroom was equipped with convection (traditional) and radiant floor cooling systems. The standard k-epsilon model of ANSYS Fluent was utilize to investigate 3 cases of the cooling systems; only traditional, only floor, and both systems. The results showed that the velocity distributions of the first and third cases were related to each other. The distribution in the second case was the smoothest because no forced convection occurred. The average temperatures of 3 cases were 24.21°C, 22.28°C and 20.66°C, respectively. This research could provide important information in using both systems in any showrooms. When chilled water available, the radiant floor cooling system should be used to take sensible heat loads and the traditional system should be used to take fluctuating latent heat loads.

Keywords: *Air Distribution Simulation, Radiant Floor Cooling System, Convection Cooling Systems, Glass Showroom, Heat Transfer*

1. Introduction

Refrigerators have played a significant role in human life since global warming started causing high-temperature gradients. There are several types of refrigerators utilized in human everyday life which consume energy differently. Convection cooling systems, so called traditional cooling systems, are commonly used in residential and commercial spaces and buildings. Radiant floor cooling systems consist of chilled-water tubes embedded inside floors and chilled water takes heat out of the floor structures and off air above the floors. When heat inside refrigerating spaces is taken out continuously, space temperatures are reduced and no heat accumulation inside the spaces and structures occurs, the cooling system consumes less energy. However, the utilization of the radiant floor cooling systems is limited in current high energy demand periods because of the high economic and industrial growth. The electric vehicle (EV) industry is one of many industries whose growth has been noticeable in recent years, vehicle showrooms are constructed vastly, and cooling systems are required and installed in all showrooms. In Thailand. the electric vehicle (EV) market has been growing according to Sustainable Development Goals (SDGs) such as Affordable and Clean Energy, Sustainable Cities and Communities, and Responsible Consumption and Production. The EV showrooms have built in significant amount in the past few years. The showrooms are still utilized to show new technology, product, and ideas continuously. They are the important parts in branding, marketing, and selling, though, in the current digital disruptive society. They are used to create new and impressing customer experience and engagement. Most showrooms need air conditioner to adjust the space living conditions.

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Odyjas and Górka [1] evaluated a floor cooling system performance through numerical simulations, highlighting its reliance on cooling load types. The system shows limited effectiveness with convective loads but improves significantly with direct radiation flux. The research outlines capacity limits under steady thermal conditions and a minimum floor temperature of 20°C, providing design guidelines for various cooling scenarios and optimal performance within acceptable comfort levels.

Wibron et al. [2] numerically explored cooling systems in data centers, focusing on partial aisle containment for hard and raised floor configurations using ANSYS CFX and the Reynolds stress turbulence model. Validation was based on server room velocity measurements and experimental load data. Performance metrics like Rack Cooling Index (RCI), Return Temperature Index (RTI), and Capture Index (CI) showed improved airflow management with raised floors. While RCI achieved 100% at full flow rates for both configurations, the hard floor configuration had lower energy efficiency (RTI ~40%) compared to the raised floor. At reduced flow rates (50%), RTI improved to over 80%, though all setups experienced some hot air recirculation.

Snidvongs [3] presented the "Siam Solar Trough," a 300-kW parabolic trough power plant designed for Thailand's soft-land, medium insolation, and moist climate. The system integrates a high-efficiency steam engine with dual energy sources—solar thermal and biogas backup. Key components include a delta truss structure, reflective materials, a tracking system, dual-phase storage, and a compact steam engine. The plant aims to provide a low-cost, pollution-free, and efficient power solution suitable for remote rural areas, with optimized performance during the monsoon season.

Samsudin et al. [4] evaluated the performance of a solar chimney for enhancing energy efficiency in Malaysian hospitals, focusing on air changes per hour (ACH) in a tropical climate. Using a 2D CFX model and simulating four different times of day, results show that the solar chimney's effectiveness varies with solar irradiance, with peak ACH at 2 pm and lowest at 8 am. Simulations, validated against previous studies, utilized steady-state and laminar flow models to assess air turbulence. Ongoing experimental work aims to further investigate these findings.

Lopez et al. [5] evaluated the underfloor air distribution (UFAD) system in comparison to traditional overhead (OH) systems, focusing on various UFAD ventilation layouts. It finds that swirl-type diffusers provide more uniform temperature and reduced air recirculation compared to rectangular grille-type diffusers. Side-mounted return vents enhance cooling efficiency by isolating recirculating air. The study concludes with a design guide matrix detailing how different supply and return vent configurations impact air conditioning performance.

Nawale et al. [6] explored improving thermal systems for heat recovery in various settings by inducing turbulence through twisted tape inserts in pipes within concrete floors. Experimental and numerical analyses, using ANSYS-FLUENT, examined 13 tape designs, revealing that swirling flow enhances thermal efficiency. The most effective configuration increased water temperature by 1.62°C over 1.8 meters, with simulation results aligning with experimental data.

Akbarpoor et al. [7] examined an integrated cooling system combining an earth-to-air heat exchanger (EAHE) and a domed roof, evaluated through simulations with ANSYS FLUENT and MATLAB. Experimental validation confirmed the system's ability to meet thermal comfort standards with a cooling capacity of about 1000 W, reducing electrical consumption by up to 0.360 kWh compared to traditional cooling methods. Environmental benefits include reductions of 361.89 kg and 216.06 kg in CO_2 emissions when replacing split inverter air conditioners and evaporative coolers, respectively. The system also effectively provides cooling for a two-story building with a demand of 300 W per floor.

Jafari and Kalantar [8] addressed the challenge of excessive fossil fuel use by focusing on energy consumption in building ventilation systems. They explored an innovative method combining windcatchers, solar chimneys, and a water spray system (WSS) to improve evaporative cooling in a multi-story building in a hot, dry climate. Their CFD simulations using ANSYS FLUENT revealed that incorporating WSS could lower temperatures by 6–12°C and increase humidity by 80%, enhancing system efficiency and maintaining thermal comfort even without wind.

Nuitchanthuek et al. [9] In Thailand's hot and humid climate, greenhouses face challenges that impact farming. This study compares the effectiveness of evaporative and radiant floor cooling systems in reducing greenhouse temperatures during the day and night. Daytime results show the evaporative system alone reduces temperatures by

1-2°C, the radiant floor system by 3-4°C, and the combined systems by 4-5°C. At night, the evaporative system lowers temperatures by 2-3°C, the floor system by 4-6°C, and the combined systems by 5-7°C. The radiant floor system alone closely matches the combined systems' nighttime cooling efficiency.

Pawlak et al. [10] developed a computational model to analyze dynamic heat transfer in a room with radiant floor cooling, validated against reference models and full-scale experiments with high accuracy (0.23°C for air temperature and 0.16°C for floor temperature). Simulations explored solar, long-wave, and convective heat gains, examining their impact on cooling floor performance and control systems. The research highlights the importance of considering dynamic heat flow and developing design methods that predict operational characteristics without excessive complexity.

Khovalyg et al. [11] explored the use of modular prefabricated load-bearing elements, specifically glass fiber-polymer composite profiles, for building construction. The study focuses on a new slab design called P-TACS (Prefabricated Thermally Activated Fiber-Polymer Composite Slab), which integrates water channels for active heating and cooling. The P-TACS slab shows superior structural performance, allowing for a 10-meter span with uniform surface temperature. Thermal and hydraulic analyses using ANSYS Fluent indicate that P-TACS outperforms standard radiant systems (ESS Type A and RCP) in both heating and cooling efficiency, with a three-fold faster response time and lower operational energy use. Fire safety measures are also addressed in the design.

Kamil and Abd [12] presented a unique radiant floor cooling system using phase change materials (PCMs) and thermal energy simulation (TES) with spiral and counter configurations. Simulations via ANSYS FLUENT 2022R2 revealed limited impact from water flow rate, with the counter setup achieving superior thermal balance. The PCM in the counter model dissolved more slowly (144 min) than in the spiral model (130 min), enhancing energy efficiency during peak loads. Temperature ranges for melting and discharge were 27–29°C and 22–26°C, respectively.

Janthasri et al. [13] evaluated the impact of double-pipe heat exchangers (DPHExs) on energy consumption (EC) and thermal performance in agricultural postharvest refrigeration. The research compares two refrigerating systems using HFC-32 fixed-speed (FSC) and variable-speed compressors (VSC) across four experimental setups. Installation of DPHExs significantly reduced EC—by up to 52.33% for FSC and 50.63% for VSC at 18°C, and by 17.19% for FSC and 20.00% for VSC at 22°C. The VSC system with DPHExs showed improved energy efficiency and more stable conditions compared to FSC systems, offering valuable insights for optimizing refrigeration systems in fluctuating climates.

Janthasri et al. [14] explored traditional vapor compression and new radiant floor cooling systems in both non-heat-loaded and heat-loaded storage rooms. The radiant floor systems, used alone or combined with compression systems, were found to reduce energy consumption and minimize temperature and humidity fluctuations. When tested at 18°C and 22°C, the radiant floor systems improved stability and efficiency. The combined systems are particularly recommended for sensitive products like ready-to-eat and fresh fruits, offering valuable insights into operational costs and storage effectiveness.

This research focused on the theoretical investigation of air inside a clear glass showroom equipped with radiant floor and/or convection cooling systems. The investigation was aimed at finding velocity and temperature distributions of the air and presenting guidelines for using both cooling systems in the glass showroom in high cooling demand period of Thailand. The information from this research can be used in managing showroom floorplan, furniture and residential areas, as well as comfort zone management. Cooling system operators and owners can take this research information into account of selecting the cooling systems, setting system operational schedules, managing power consumption of both cooling systems by operating individual or coupled systems, and increasing cooling-system performance by operating only required systems to take heat load.

2. Methodology

2.1 Cooling Systems

Convection cooling systems refers to the vapor compression cooling system which consists of an evaporator, compressor, condenser, and refrigerant flow control. The convection systems are widely used in refrigerating spaces and showrooms. The radiant floor cooling system consists of chillers, chilled water, and tube systems. Since the chilled water flowing inside the tubes embedded in floors takes heat from refrigerating spaces out to the water chiller, temperatures of the refrigerating spaces are reduced accordingly. The floor cooling system creates low noise because air moves by natural convection. The convection system takes heat from the refrigerating spaces out faster than the latter system because air moves and is circulated by forces of convection. However, the air temperatures inside the space created by the floor cooling systems fluctuated less than those provided by the convection systems [13,15]. Normally, the floor cooling system is coupled with air dehumidifiers and/or outdoor air units (OAUs) which clean and take fresh air to circulate inside the refrigerating spaces, therefore, the air inside rooms installed with the floor cooling is fresh and clean. In the floor cooling system, radiation is the main heat transfer mechanism to remove heat while the convection system utilizes the convection heat transfer as the main mechanism [16,17]. During the floor cooling operation, low noise and quiet atmosphere are perceived because of no vibration no moving component required. Since the OAUs and humidifiers are operated with the floor cooling system to control indoor humidity, mold and fungus would be controlled and limited, the system helps promoting good health and well-being of residents. When the floor and convection cooling systems were operated at the same time, the latter systems would consume energy lower; at least 14% less than the individual-convection-system consumption [14]. The floor cooling system can be turned into floor heating systems by simply changing temperatures of water inside the embedded tubes. The limitations of the floor cooling systems are acknowledged as more controllers involved in managing indoor-air conditions and higher first investment compared with the convection system investment if no water chiller is available.

2.2 Theoretical investigation

Among many simulation programs, Ansys Fluent is used to simulate fluid behaviors as domains in engineering fields. Users can choose a variety of models available in Ansys Fluent such as K-epsilon, transition SST, Reynolds Stress, Inviscid, K-omega, Scale-Adaptive Simulation, Spalart-Allmaras, Transition k-kl-omega, Detached Eddy Simulation and Large Eddy Simulation, Laminar.

2.3 The k-epsilon $(k-\varepsilon)$ turbulent models

The model is one famous model in Computational Fluid Dynamics (CFD) using Partial Differential Equation (PDE). The first transport variable is the turbulent dynamic energy (k) and the second variable is the distribution rate of the turbulent dynamic energy. The k-epsilon model may be classified as Standard, Renormalization Group (RNG), Realizable k-epsilon turbulent models.

2.3.1 The Standard k-epsilon model

This model solves both transport equations separately, the users can specify turbulent velocities and lengths freely. This model can be used to investigate flow and heat transfer at the same time. The Standard k-epsilon model is usually used in modeling uncomplicated, turbulent, and internal flow of water, oil, and air, as well as air flow inside and over buildings. In Ansys Fluent, the following equations are solved continuously; transport equations for the Standard k-epsilon model;

$$\frac{\partial}{\partial t}(\rho k)+\frac{\partial}{\partial x_i}(\rho k u_i)=\frac{\partial}{\partial x_j}\left[\left(\mu+\frac{\mu_t}{\sigma_k}\right)\frac{\partial k}{\partial x_j}\right]+G_k+G_b-\rho\varepsilon-Y_M+S_k \tag{1}$$

and

$$\frac{\partial}{\partial t}(\rho \varepsilon)+\frac{\partial}{\partial x_i}(\rho \varepsilon u_i)=\frac{\partial}{\partial x_j}\left[\left(\mu+\frac{\mu_t}{\sigma_\varepsilon}\right)\frac{\partial \varepsilon}{\partial x_j}\right]+C_{1\varepsilon}\frac{\varepsilon}{k}\left(G_k+C_{3\varepsilon}G_b\right)-C_{2\varepsilon}\rho\frac{\varepsilon^2}{k}+S_\varepsilon \tag{2}$$

the turbulent viscosity modeling;

$$\mu_t=\rho C_\mu \frac{k^2}{\varepsilon} \tag{3}$$

where C_μ , $C_{1\varepsilon}$, $C_{2\varepsilon}$ and $C_{3\varepsilon}$ are constant while σ_k and σ_ε are Prandtl number for k and ε turbulent flow, respectively. S_k and S_ε are source term, constant adjusted by users. In the current work, the constant values are described in Table 1 and the Ansys Fluent default values.

Table 1: the Ansys Fluent default values used in Equations (1) to (3).

Constant	Values
C_μ	0.09
σ_k	1.00
σ_ε	1.30
$C_{1\varepsilon}$	1.44
$C_{2\varepsilon}$	1.92
$C_{3\varepsilon}$	1.00

Since all showroom walls were made of glass, energy equation for constant wall heat flux boundary in the Standard k-epsilon model can be used to calculate heat transfer of turbulent indoor air. Kinetic energy of the air effects heat transfer and air temperatures from the following equation;

$$\frac{\partial}{\partial t}(\rho E)+\frac{\partial}{\partial x_j}\left(u_j(\rho E+p)\right)=\frac{\partial}{\partial x_j}\left(k_{eff}\frac{\partial T}{\partial x_j}\right)+\Phi+S_h \tag{4}$$

The Standard k-epsilon model should be used to analyse internal, uniform, irrotational, and inviscid flow. This model is simple and fast, as well as less processing power and time consumption.

2.3.2 The Renormalisation Group (RNG) k-epsilon model

This model was developed from the Standard k-epsilon model to extend the limitations of the Standard k-epsilon model such as flow with high shear stress, vortex, separated, and sudden velocity changing. The model provides more accurate results in the mentioned flow while more processing power and time consumption are required. The model may not be suitable for some near-wall flow.

2.3.3 The Realizable k-epsilon model

This model has been utilized as the turbulence model relating to Computational Fluid Dynamics (CFD) for complicated domains and various flow fields. The model was developed to solve the limitations of the Standard k-epsilon model in flow with high velocity, separation, and vortex. The model may also not be suitable for some near-wall flow.

The Standard k-epsilon model was chosen to simulate the turbulent flow in this research because of its simplicity and stability. The model could be used to simulate simple turbulent flow such as indoor air flow and air flow inside closed spaces. The empirical constants for this model are available and simple to set in the simulations. The results obtained from this model were acceptable.

2.4 Simulation cases

There were three investigated cases analyzed in the current research which are listed below;

- Air inside the showroom is cooled by only the convection cooling systems.
- Air inside the showroom is cooled by only the radiant floor cooling system.
- Air inside the showroom is cooled by both cooling systems.

In the simulations, the boundary conditions of all clear glass walls were set as constant wall heat flux at 500 W/m^2 as informed by the literature [3]. The theoretical investigations delivered velocity and temperature distributions of air provided by each case. Then, the distribution and energy consumption information from the literature [14,15] were analyzed to find suggestions and guidelines for users to utilize both cooling systems.

3. Results and Discussion

3.1 Model Setup

3.1.1 Model Validation

Firstly, the standard k-epsilon model of the greenhouse was validated with the experimental greenhouse in the literature [9], the same geometry was considered and the radiant floor cooling system was installed and operated in the greenhouse. When only the radiant floor cooling system was utilized, the temperatures in the experimental results obtained from Points 1, 2, 3, 4, 5 and 6 (Figure 1); 35.1°C, 35.1°C, 32.9°C, 33.9°C, 33.2°C and 31.2°C, respectively, were compared with the temperature results of the validated simulation. With the 10% of errors accepted, the temperature errors were found as 5.69 %, 5.69 %, 0.60 %, 2.36 %, 0.30 % and 5.75 %, respectively. Since the errors were acceptable, the standard k-epsilon model was applied in the further simulation.

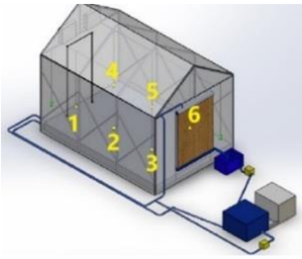


Fig. 1. Temperature measuring points in the literature [9]

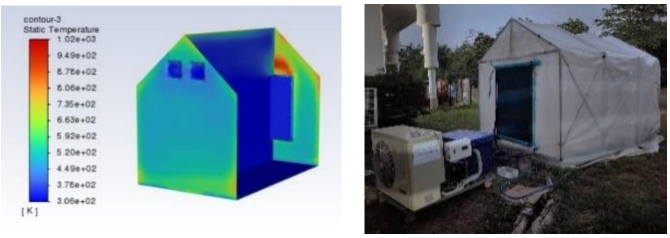


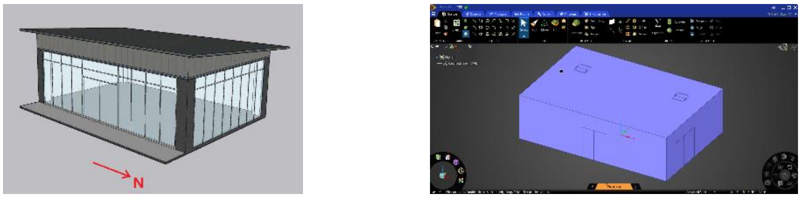
Fig. 1. The validated model and experimental setup [9]

3.1.2 Model Geometry

The air domain inside the standalone glass showroom (Fig. 1a) was shown in Fig. 1b having dimensions as detailed in Table 1. The air domain and 0.84 m x 0.84 m convection cooling systems were created in Ansys Fluent while the cooling area of the radiant floor cooling system was set as one boundary condition as shown in Table 2. As in the normal operation, the floor cooled by the chilled water system was controlled to have a constant wall temperature. The floor-surface temperatures are usually set and controlled to be above dewpoints to avoid vapor condensation on floors causing wet floors and accidents caused by any condensed water. Thermocouples or temperature sensors are used to measure the floor-surface temperatures, if the surface temperatures are higher than the set temperature, manifolds will allow chilled water with a higher flow rate and/or lower temperatures to flow into the embedded tube systems. On the other hand, if the surface temperature is reduced and close to the set temperatures, manifolds will allow lower flow rate water to pass into the tube systems. All boundary conditions were detailed in Table 3. Temperatures and relative humidity values of air in the indoor comfort zone are in ranges of 22.0 - 27.0°C and 20-60 %RH, respectively. The highest dew point of air in the comfort zone is about 15.0°C. To avoid any condensation, especially from water vapor in the air, floor cooling surfaces will be controlled above 15.0°C. A typical surface temperature of 20.0°C was selected for this showroom. If water condensed on the floor surfaces, it can cause serious accidents for residents who slip on water drop. The constant wall heat flux of 500 W/m^2 [3] from all glass walls of the showroom was calculated in the source term, S_h, in Equation (4).

Table 1: Model Geometry.

Model	dimensions
The air domain inside the glass showroom	12 m x 8 m x 3 m
Each convection cooling system (total of two systems)	0.84 m x 0.84 m
The cooling area of the radiant floor cooling system	12 m x 8 m



(a) (b)

Fig. 2. The 3-dimensional glass showroom model (a) and the Ansys Fluent air domain inside the showroom (b).

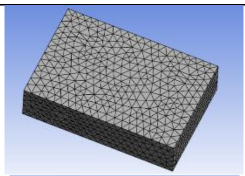
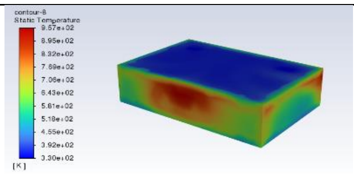
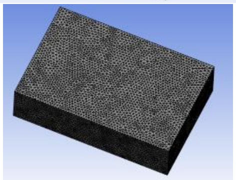
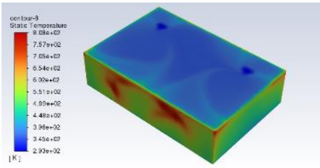
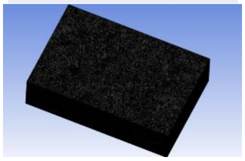
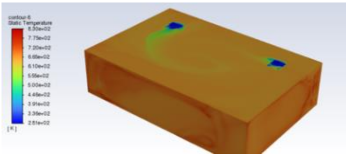
Table 2: Boundary Conditions.

Properties	Value/Unit
Air density	1.225 kg/m^3
Specific heat	1006.43 J/kg K
Thermal conductivity	0.0242 W/m K
Viscosity	1.7894 x 10^{-5} kg/m s
Average velocity inlet	1.337 m/s
Constant wall heat flux (all glass walls)	500 W/m^2 [3]
The cooling surface temperature of the radiant floor fooling system	20°C
Supply air temperature from the convection cooling system	20°C

3.1.3 Grid Independent

The model was tested for grid independent since the fine grids required more calculation periods as shown in Table 4. The default grid size generated by Ansys Fluent for the current model was 0.6 m, this grid size required 4.05 minutes for 1000 iterations calculation. Then, the grid size was changed to 0.2 m, this size required 37.23 minutes for the same calculation but provided less fluctuated results. Additionally, the 0.08-m grid size was applied, similar fluctuated results were created while this size took longer period for the same calculation. Therefore, the 0.2-m grid size and triangle grid shapes were chosen in the current work as shown in Figure 3.

Table 3: Grid Independent.

Elements Size (m)	Calculation period (min)	Grid Elements	Mesh	Simulation
0.6	4.05	24937		
0.2	37.23	251937		
0.08	333.21	1930169		

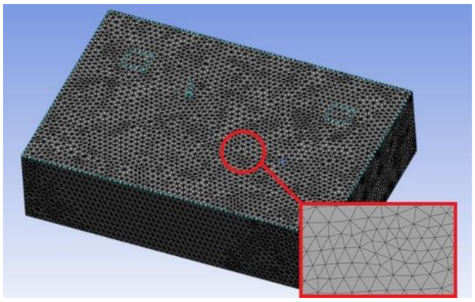


Fig. 3. The 0.2-m triangle grids in the investigations.

3.2 Velocity and temperature distributions

3.2.1 The first case results; only convection cooling systems operated

The air velocity distribution (Figure 4) displayed the spots inside the room having quite different air movement as different color spots. This character was caused by the forced convection of air from the convection cooling systems as the main system creating the air movement. The air velocities near the north and south walls were higher than those near the east and west walls. The average air velocity inside the showroom was 0.107 m/s. The same character occurred in the temperature distribution as shown in Figure 5. The cold spots were along air-drop paths from the cooling system. Since there was heat generated by solar radiation as heat flux on the glass walls, the highest temperatures were on the glass walls. The air temperatures near the floor, in the middle of the showroom, and near the room ceiling were 25.81°C, 24.17°C and 22.64°C, respectively.

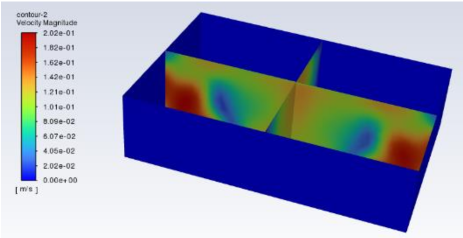


Fig. 4. The velocity distribution provided by only convection cooling systems.

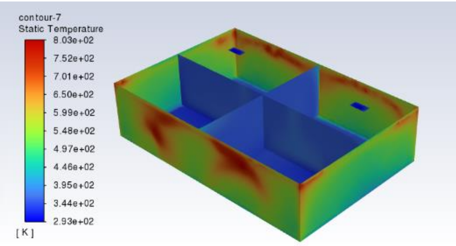


Fig. 5. The temperature distribution provided by only convection cooling systems.

3.2.2 The second case results; only radiant floor cooling system operated

The velocity and temperature distributions (Figs 6. and 7.) were smoother than those provided by the convection system since, in this case, the natural convection of air played an important and main role, the air velocities were lower than those of the first case. The average air velocity inside the showroom was very low; $9.50 x 10^{-13}$m/s because no forced convection created any air movement in this case. On the other hand, the natural convection play the important role in this case. As one may notice, the smooth air velocity and temperature distribution could create more comfortable living conditions than the fluctuated conditions. The average temperatures near the floor and in the middle of the room were at 20°C while the temperature near the ceiling was at 26.85°C.

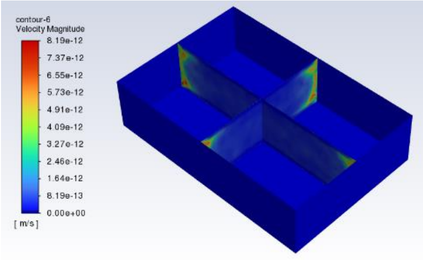


Fig. 6. The velocity distribution provided by only radiant floor cooling system.

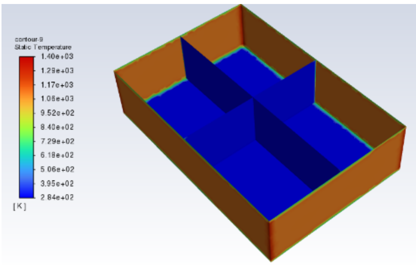


Fig. 7. The temperature distribution provided by only radiant floor cooling system.

3.2.3 The third case results; both cooling systems operated

When both systems were utilized, the average air velocity (Figure 8) was the same as in the first case result at 0.107 m/s because there was no forced convection created by the floor cooling system. While the air movement of the third case was similar to that of the first case, the temperature distribution (Figure 9) was different from other two cases. The average floor surface temperature was at minimum temperature 19.91 °C and the average air temperature near the ceiling was at 22.17 °C.

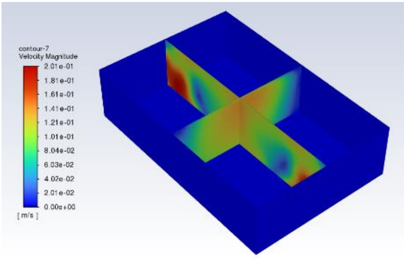


Fig. 8. The velocity distribution provided by both cooling systems.

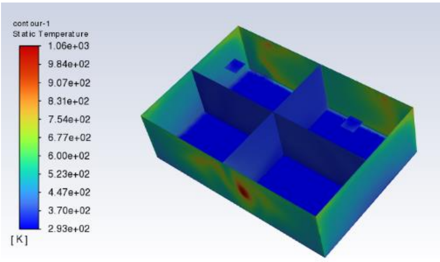
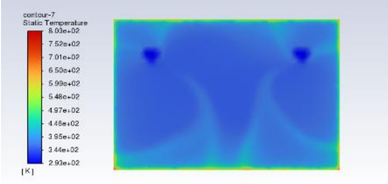
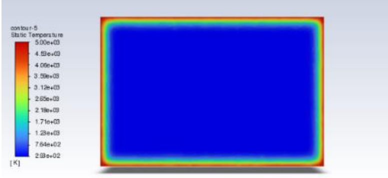
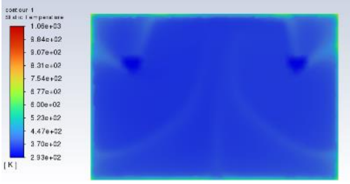


Fig. 9. The temperature distribution provided by both cooling systems.

Results from Table 5 displayed the lowest room temperature occurred in the third case; both cooling systems operated, and in the comfort zone [15]. The air movement in the first case may cause some disturbances on occupants because of fluctuating air temperatures and velocities. Although the convection cooling systems could reduce temperatures near the showroom ceiling, there were more fluctuating temperatures inside the showroom. When both cooling systems were operated inside the showroom, there were less fluctuations, the temperatures were lower than those provided by only the floor cooling system operated. From the literatures [13,14], one could choose the convection cooling system with the variable-speed compressor to reduce power consumption when both systems were utilized.

Table 4: Temperature distributions from different planes of all three cases.

	Ceilings	
Only traditional systems	Only radiant floor cooling system	Both systems
		
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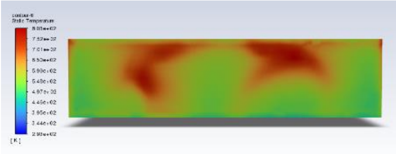	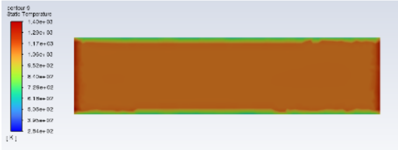	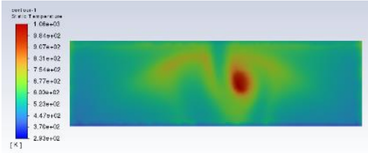
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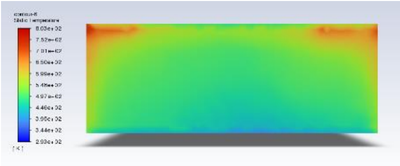	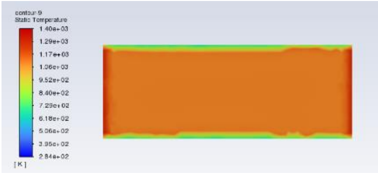	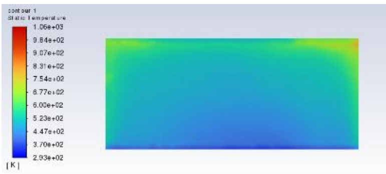
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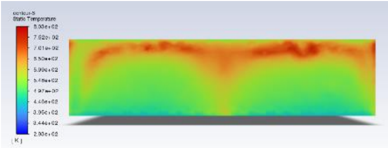	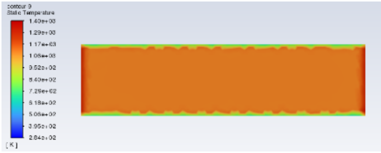	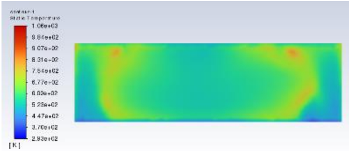
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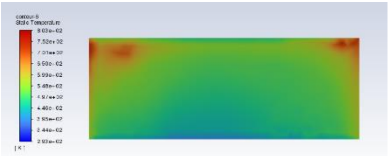	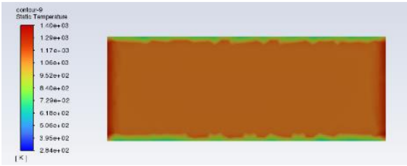	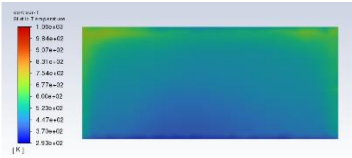
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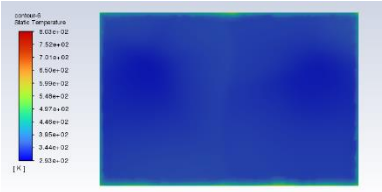
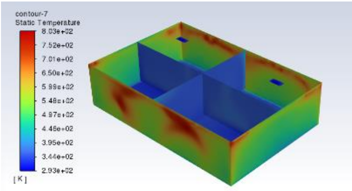
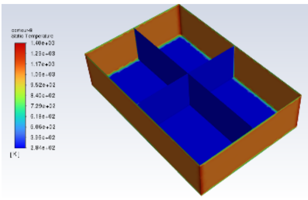
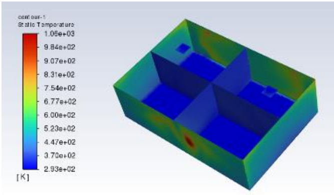
		
	The middle of the showroom	
		

3.3 The guideline for operating both cooling systems

Results from all three cases could be used to present the guidelines for using both cooling systems in the glass showroom in different conditions. The floor cooling system could be used to provided the the same range of temperatures inside the showroom as provided by only the convection cooling system. If any showrooms have chilled water systems or design to operate cooling systems by using chilled water, the radiant floor cooling system should be considered as the first priority because it could utilize the exist chilled water with no extra power consumption. The convection cooling systems could be installed and used to take care of the extra latent heat loads such as human activities and high humidity in the rainy season. When there are more people come in the showroom or there are events organized in the showroom, the convection cooling systems could be turned on. Otherwise, only the floor cooling system could take off sensible heat loads inside the showroom, especially, if the floor cooling system is operated 24 hours a day. One may consider operating only floor cooling system during night time because no occupant inside the showroom. Both systems will be operated only the daytime and high latent heat loads.

4. Conclusion

The technology, economy, and industry have been growing after the COVID-19 pandemics while many countries strictly focus their developments on Sustainable Development Goals. There are more showcases and showrooms required continuously, all showrooms are equipped with heating, ventilation, and air conditioning (HVAC) systems. In Thailand, the air conditioners are operated to adjust temperature for showroom occupants and users. Then, air velocity and temperature distributions inside the showrooms are two parameters indicating the comfort of the residents. The convection cooling system is well known as the traditional cooling system in Thai HVAC society while the radiant floor cooling system is less popular. This work indicated the air velocity and temperature distributions inside the showrooms equipped with the traditional and radiant floor cooling systems by utilizing the theoretical investigation; the standard k-epsilon model offered by Ansys Fluent program applied. Three cases; only traditional systems, only floor cooling system, and both systems. The results displayed possibilities in using single or both cooling systems as the air conditioners of the showrooms because the temperature distributions of all three cases were in the same range. Noticeably, the floor cooling system provided the smoothest temperature distribution among the three cases. If the velocity and temperature distributions are mainly concerned, the floor cooling system may be the priority system. One could consider using the radiant floor cooling system to take sensible heat loads such as heat gain through the showroom structure and solar heat gain because the floor cooling system consumes less energy when chilled water is available, no energy required to drive vapor compression systems. The traditional system could be considered to take latent heat loads in special situations such as rain and high occupancy days. Both cooling systems could be operated to provide indoor air in comfort zone conditions. Notably, when water chillers are available and energy consumption is the main concern, the floor cooling system should be operated to provide indoor air conditions having suitable temperature and humidity, low noise, and smooth temperature distributions.

Nomenclature

E	Total energy (including kinetic and internal energy)
G_b	The generation of turbulence kinetic energy due to buoyancy
G_k	The generation of turbulence kinetic energy due to the mean velocity gradients
k	Turbulent kinetic energy
k_{eff}	Effective thermal conductivity, $k_{eff} = k + \mu_t c_p / \mathrm{Pr}_t$ where k stands for molecular thermal conductivity
p	Pressure
P_k	Rate of production of turbulent kinetic energy
S_h	Source term
T	Temperatures
t	Time
u_i	Velocity component in the corresponding direction
u_j	Velocity component in the *j* direction

x_j	Coordinates
Y_M	The contribution of the fluctuating dilatation in compressible turbulence to the overall dissipation rate
ε	Rate of dissipation of turbulent kinetic energy
μ_t	Turbulent viscosity
Φ	Turbulent dissipation function
ρ	Fluid density
σ_k	Diffusion constant for k

Acknowledgments

This work was supported by Kasetsart University International College; the Graduate Program Scholarship from the Graduate School, Faculty of Science and Engineering, and Faculty of Engineering, Kasetsart University; and Kasetsart University Research and Development Institute.

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