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Evaluation of Interaction Between Load of Refrigerated Display Cases and Air Conditioners in a Grocery Store Using CFD

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Abstract

In this paper, the cooling load of refrigerated display cases and the load of air conditioners in a grocery store were simulated by using computational fluid dynamics (CFD) models to understand the relationship between the loads of refrigerated display cases and air conditioners in summer and winter. In the summer, the results show that a higher preset temperature of the air conditioner led to an increase in the cooling load of the display cases and a decrease in the cooling load of the air conditioners. For this reason, the total electricity consumption was minimized when the preset temperature was set to 25°C, and the temperature was neither the highest nor the lowest explored. In the winter, the total electricity consumption of both display cases and air conditioners increased with increasing preset temperature. Furthermore, by replacing the frozen multi-deck display cases with frozen reach-in cases in the summer, the electricity consumption of the display cases decreased to half, while that of the air conditioners somewhat increased; in the winter, the electricity of the display cases and air conditioners both decreased.

Keywords: Grocery store; Air conditioner; Refrigerated display case; Energy consumption; CFD

1. Introduction

Supermarkets and grocery stores play an important and energy-intensive role in the commercial sector. The average electrical energy intensity of these stores varies from 1500 kWh/m^2 to 850 kWh/m^2 per year as the sales area increases from 280 m^2 to 1400 m^2 [1]. In Japan, grocery stores occupy 8% of the energy consumption in the commercial sector [2].

Refrigerated and frozen display cases (DCs) are widely used in grocery stores. According to the research data presented by Narumi, 84% of the energy consumption of grocery stores was related to chillers [3], while another study in the UK reported that the power consumption ratio of refrigeration and HVAC (heating, ventilation, and air conditioning) equipment was 60% and 26%, respectively [4].

In addition, HVAC equipment and refrigerators can influence each other in energy consumption, but most studies investigate air conditioners and DCs separately rather than regarding them as a combined working system. Hence, it is essential to explore the interaction between the load of refrigerators and air conditioners in grocery stores. To achieve this goal, the energy consumption of stores should be measured. However, the energy consumption measurement in stores is not easily performed because of the high costs and difficulties in controlling conditions.

CFD models can simulate the leakage of DCs and airflow of air conditioners; thus, the abovementioned obstacle can be overcome. Based on a series of experimental results, we built models of DCs in this study that can imitate their load and low-temperature air leakage under different temperatures or humidity environments.

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Nomenclature

$Q_{MD\text{-}R}$	Cooling load of an open refrigerated multi-deck display case [kW]
$Q_{SMD\text{-}R}$	Cooling load of an open refrigerated semi-multi-deck display case [kW]
$Q_{MD\text{-}F}$	Cooling load of a frozen multi-deck display case [kW]
$Q_{RI\text{-}F}$	Cooling load of a frozen glass door reach-in display case [kW]
$Q_{TUB\text{-}F}$	Cooling load of a frozen tub display case [kW]
H_{in}	Influx absolute humidity of a measurement point in front of a display case [kg/kg]
T_{in}	Influx temperature of a measurement point in front of a display case [°C]

Abbreviations

DC	Display case
HVAC	Heating, ventilation, and air conditioning
A&C	Air conditioner
MD-R	Open refrigerated multi-deck display case
SMD-R	Open refrigerated semi-multi-deck display case
MD-F	Frozen multi-deck display case
RI-F	Frozen glass door reach-in display case
TUB-F	Frozen tub display case
DAG	Discharge air grille of a display case
RAG	Return air grille of a display case

A small grocery store situated in Kobe, Japan, was chosen to be simulated to evaluate the interaction between the load of the DCs and air conditioners in the store. The DC CFD models were arranged in the CFD model of the store.

2. Methodology

The main contents of this study can be divided into three parts: (1) setting up an experiment to measure the performance of 5 kinds of DCs at different temperatures and humidities, (2) building the CFD models of the DCs from the data obtained in experiments, and (3) evaluating the interaction between the load of the DCs and that of the air conditioners in a grocery store at different air conditioner preset temperatures and applying energy-saving strategies. These details are provided in the following sections.

2.1. Performance of Display Case Measurements

The experiment measuring the performance of DCs in a laboratory with constant temperature and humidity was divided into 2 types: refrigerated and frozen experiments. The refrigerated DCs measured in the experiment consisted of one open refrigerated multi-deck DC (MD-R) and one open refrigerated semi-multi-deck DC (SMD-R); the frozen DCs consisted of a frozen multi-deck DC (MD-F), a frozen glass door reach-in DC (RI-F) and a frozen tub DC (TUB-F). Table 1 lists specifications of the 5 kinds of DCs. The 2 experimental plans are separately shown in Fig. 1(a) and Fig. 1(b). Plastic cardboard of approximately 1.7 m in height were fixed at the air exhaust to simulate a real store environment. A series of experiments were conducted to evaluate the effect of indoor air temperature and relative humidity on the cooling load of the 5 kinds of DCs, and the environment of the outdoor chamber remained at steady-state conditions (32°C and 40% relative humidity). The experimental conditions are summarized in Table 2. To build CFD models and to check the reliability, temperature and humidity and flow rate of the inlet and outlet vents of the DCs, the room temperature distribution and the cooling load of the DCs were measured.

Table 1 Specifications of the display cases

Display case	MD-R	SMD-R	MD-F	RI-F	TUB-F
Effective unobstructed capacity (L)	1940	1477	1507	1077	900
Outside dimension (mm)	Length 2440, Width 1100 Height 1890	Length 2440, Width 1100 Height 1700	Length 2440, Width 1100 Height 1880	Length 1430, Width 760 Height 1905	Length 1810, Width 1520 Height 810
Required refrigerating capacity (kW)	2.22	3.04	3.75	0.6/0.5	1.20/1.51
Evaporating temperature (°C)	-10	-10	-40	-40	-40

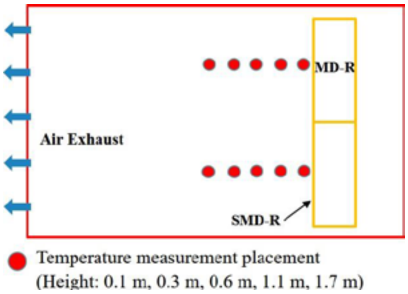


Fig. 1(a) Refrigerated experiment plan

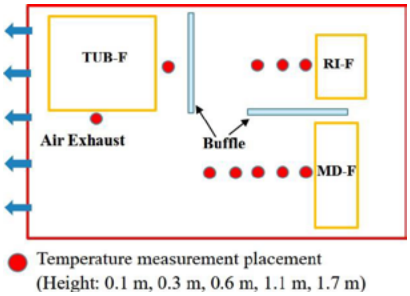


Fig. 1(b) Frozen experiment plan

Table 2 Experimental conditions

Item	Conditions
Outdoor environment	Temperature: 32℃ Relative humidity: 40%
Temperature	DC preset temperature: MD-R, SMD-R: 1℃; MD-F, RI-F: -28℃; TUB-F: -25℃ Indoor: 19℃, 22℃, 25℃, and 28℃
Indoor humidity	30%, 40%, 60%, and 70%
Cases	16 cases
Measurements	(1) Temperature, flow velocity and humidity of DCs RAG and DAG (2) Indoor temperature distribution (Measuring placements and heights are shown in Fig. 1 (a), (b)) (3) Cooling load of DCs

Table 3 Conditions of the CFD analysis of DCs

Items	Conditions
Employed code	STREAM Ver.14 (Software Cradle Co., Ltd)
Dimension	9.36 m (X)*6.5 m (Y)*4.0 m (Z)
Object	Air (Incompressible)
Analysis method	Steady-state analysis (Interval: 0.1 s)
Turbulent model	RANS (high-Reynolds)
Vapor	Diffusivity: 0.0000256 m2/s, Power law: 0.67
Number of mesh	From 2.1 to 6.6 million (Depending on DCs)
Boundary surface	No-slip surface
Boundary heat transfer	Thermo insulation
Air influx of the air conditioner	Speed: 0.065 m/s Turbulent: k=0.0001 m^2/s^3, ε=0.0001 m^2/s^3 Temperature: 19℃, 22℃, 25℃, and 28℃ Humidity: 30%, 40%, 60%, and 70%
Air exhaust	Natural outflow and inflow with aperture ratio while under 1.7 m (5% for refrigerated DCs and 10% for frozen DCs)
Radiation	View factor radiative heat transfer

2.2. Display case CFD Model

2.2.1. Detailed CFD Model

The laboratory model is shown in Fig. 2, which was built on the basis of the real laboratory. Detailed DC models, using the MD-R as an example shown in Fig. 3, were built based on the data obtained from the experiments. The commercial code STREAM Ver. 14 (November 2018. Software Cradle Co., Ltd. Oosaka, Japan) was employed for calculations in this study, and a high-Reynolds number-type κ-ε turbulence model was used to solve a steady-state simulation because the inaccurate imitation of viscous sublayers close to solid walls can be ignored in this study. A termination criterion of 10^{-4} is used for all variants in this study. The view factor method was used to calculate radiation. Additional details of the conditions are summarized in Table 3.

As the boundary conditions, the heat was insulated at all boundaries between the defined and undefined areas. The speed of flow from the air conditioner outlet surfaces was maintained at a constant speed. The air

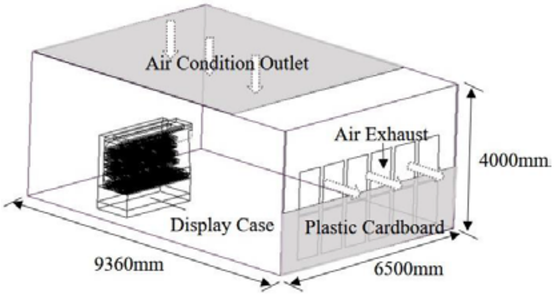


Fig. 2. Laboratory CFD model

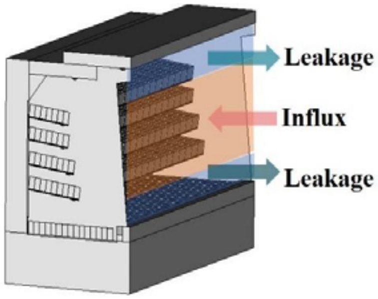


Fig. 3. Detailed model of MD-R

conditioner outlet temperature and relative humidity conditions were in accordance with the experiment (Table 3). The air conditioner exhaust surface was fixed as natural air outflowing and inflowing (the aperture ratio was fixed while under 1.7 m to simulate the plastic cardboard). Because plastic cardboard is not spliced tightly, the aperture may not remain constant through experiments. The aperture ratio values of the refrigerated and frozen experiments are discussed separately and determined as shown in Table 3. The DAG and RAG settings are shown in Fig. 4, using the MD-R as an example. The flow rate, temperature, and humidity of RAG and DAG were based on data measured from the experiment.

It was found that the air in front of a DC model was always divided into two air leakage parts and an influx part, as shown in Fig. 3.

The cooling load, which was calculated from enthalpy differences between the DAG and RAG of a DC, and vertical temperature distribution in front of the DC were used to inspect the reliability of the detailed DC CFD model. The MD-R is taken as an example, whose results are presented in Figs. 5 and 7. The positive reliability of the detailed DC CFD models was confirmed.

2.2.2. Simplified CFD Model

If the scale of a domain expands, the computational load increases when simulating the energy consumption of the store by using CFD with detailed DC models. Hence, a simplified DC CFD model was used to decrease the computational load.

Without building models in detail, such as shelves and goods, only leakage and influx surfaces are set, as shown in Fig. 6. The airflow volume, temperature, humidity of leakage and influx surface were fixed from the association between them and the average temperature of the influx surface. The relations were obtained from the results of the detailed models. Fig. 8 displays the volume of influx surfaces as an example. Formula (1)-Formula (5) show the relationships between the cooling load of the DCs and the average temperature and humidity of the influx surface, which can be used to obtain the energy consumption of the stores.

(a) Discharge air grille (DAG) (b) Return air grille (RAG)

Fig. 4. DAG & RAG settings of the MD-R model

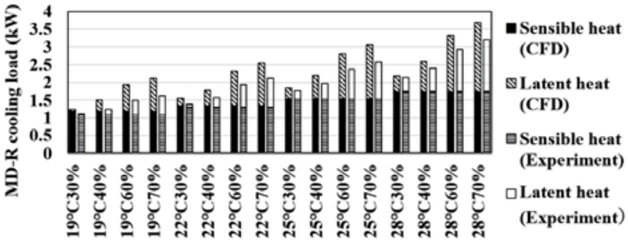


Fig. 5. MD-R cooling load reliability

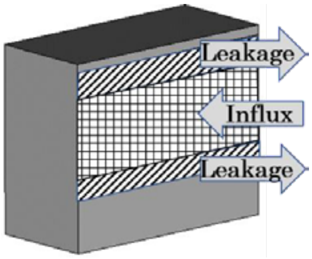


Fig. 6. Simplified model of the MD-R

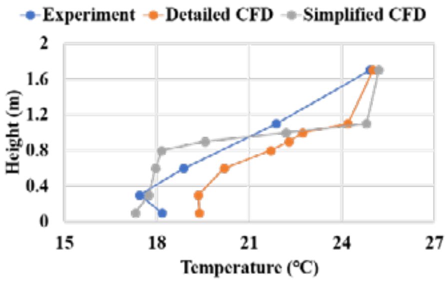


Fig. 7. Vertical temperature distribution
(2 m away from the MD-R)

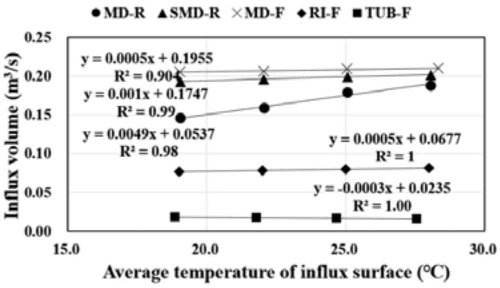


Fig. 8. Relationship between the influx volume and a measurement point temperature

The vertical temperature distribution in front of the DC were used to inspect the reliability of the simplified DC CFD model. The MD-R is taken as an example, whose results are presented in Figs. 7. The positive reliability of the simplified DC CFD models was confirmed.

$$Q_{MD-R} = 143.75 \times H_{in} + 0.083 \times T_{in} - 1.02 \quad (1)$$

$$Q_{SMD-R} = 182.17 \times H_{in} + 0.070 \times T_{in} - 0.67 \quad (2)$$

$$Q_{MD-F} = 124.62 \times H_{in} + 0.099 \times T_{in} + 7.43 \quad (3)$$

$$Q_{RI-F} = 0.070 \times H_{in} + 0.00215 \times T_{in} + 2.5 \quad (4)$$

$$Q_{TUB-F} = 4.286 \times H_{in} + 0.0022 \times T_{in} + 2.54 \tag{5}$$

2.3. Evaluation of the Energy Consumption of a Grocery Store

2.3.1. Description of the Grocery Store

The grocery store simulated in this study is a typical small grocery store that is approximately 261 m^2 in size and is located in Kobe. The plan of the store is provided in Fig. 9. There are 9 open refrigerated multi-deck DCs (MD-R), an open refrigerated semi multi-deck DC (SMD-R), 2 frozen multi-deck DCs (MD-F) and 2 hybrid-type frozen DCs, which were regarded as 2 frozen tub DCs (TUB-F) in this study. For the HVAC equipment, there are 4 four-way cassette indoor air conditioner units, 3 ducted exhaust ventilation fans used in the summer and a ducted in-line ventilation fan with four vents used in the winter.

2.3.2. Grocery Store CFD Model

Fig. 10 displays an exploded view of the CFD store model. The model consisted of refrigerated and frozen DCs, unrefrigerated DCs, HVAC equipment, architecture components, and people. The architecture components were external walls, an internal wall, the ceiling, the roof, the floor and windows, whose detailed settings are provided in Table 4.

A high-Reynolds number-type κ-ε turbulence model was used because the inaccurate imitation of viscous sublayers close to solid walls can be ignored in the store. Additional details of the basic conditions are summarized in Table 5. Radiation analysis was omitted due to the limitations of the CFD analysis code employed in this analysis, where radiation analysis and solar radiation cannot be activated simultaneously. Both surfaces of the DCs and HVAC devices were fixed as boundary conditions; therefore, only the walls and floors were influenced by radiation. Therefore, the error induced by neglecting radiation slightly influenced the airflow and temperature distribution results in this study.

The simulation time was set to Aug. 8th and Dec. 21st, whose solar radiation data were input from a database called METPV-11 [5]. In this paper, we only simulated starting at 12:00 by transient analysis whose interval time was 0.1 s per calculation cycle. The number of guests generating heat from bodies and leading to influx

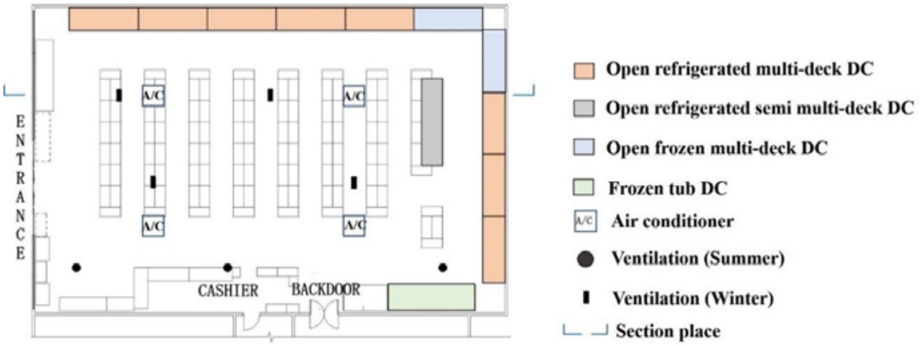
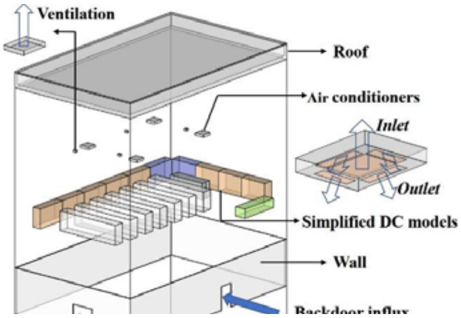


Fig. 9. Plan of the grocery store

Table 4 Architecture components conditions

Component	Material	Thickness (mm)	Thermal conductivity (W/(m K))
External wall	Tile	10	1.3
	Glass wool	100	0.034
	Gypsum	10	0.14
Internal wall	Tile	10	1.3
	Air layer	30	Temperature-dependent
	Gypsum	10	0.14



Ceiling and roof	Stainless	5	16
	Air layer	975	Temperature-dependent
	Glass wool	10	0.034
	Gypsum	20	0.14
Floor	Tile	20	1.3
	Concrete	200	1.2
	Glass wool	100	0.034
Window	Glass	50	0.75

Fig. 10. Exploded view of the CFD store model

Table 5. Conditions of the CFD analysis of the store

Items	Conditions
Employed commercial code	STREAM Ver.14 (Software Cradle Co. Ltd)
Dimension of domain	19.82 m (X)*13.14 m (Y)*4.15 m (Z)
Object	Air (Incompressible)
Analysis Method	Transient Analysis (Interval: 0.1 s)
Turbulent model	RANS (high-Reynolds), k= 0.0001 m^2/s^3, ε=0.0001 m^2/s^3
Boundary surface velocity	No-slip surface
Boundary heat transfer	Thermo insulation
Inflow boundary	(1) Outlet surfaces of air conditioners (2) Leakage surfaces of simplified DC models (3) Outside influx surfaces of entrance and backdoor
Outflow boundary	(1) inlet surfaces of air conditioners (2) air exhausted surfaces of ventilation fans
Vapor	Diffusivity: 0.0000256 m^2/s Power law: 0.67
Meshes	1,699,629 meshes
Human	Body temperature: 36.5°C Number: Change by the hour and 63 people in total in the case of 12 o'clock
Solar radiation	City: Kobe Time: 12:00, Aug. 8th and Dec. 21st Database: METPV-11

from outside was determined to be 631 people per day based on the *Annual Statistical Report on Supermarkets* [6]. The total amount was distributed per hour according to the guest distribution of the store, which was obtained from Google Maps data [7]. The average time of stay of each guest was fixed as 10 minutes, which matched the size of the store. In addition, a staffer was set in the model. As a result, there were 63 people in total being set in the store model at the start of simulation at 12:00. The surface temperature of the guests and the staff was maintained constant at 36.5°C (Table 5).

As the boundary conditions, the heat was insulated in all boundaries between the defined and undefined areas. Inflow and outflow boundaries are shown in Fig. 10. Additional details of the boundary conditions are summarized in Table 5. The inflow boundaries consisted of the outlet surfaces of the air conditioners, the leakage surfaces of the abovementioned simplified DC models, and the outside influx surfaces of the entrance and backdoor. The outflow boundaries consisted of the inlet surfaces of the air conditioners and air exhaust surfaces of the ventilation fans. Additional details are supplied in the following sections.

Simplified refrigerated and frozen DC CFD models were arranged in accordance with the real store Although there is a range of set temperatures in the real store, the set refrigeration temperature was unified at 1°C, and the freezer temperature was at -28°C in the simulation because an accurate reproduction is not the purpose of this study. Unrefrigerated cases were placed in the center of the store. Moreover, to simulate the gap between goods placed in unrefrigerated cases, the refrigerated cases were built by surfaces with an opening ratio of 60%.

For air conditioner models, the outlet temperature and humidity were calculated from those of the inlet flow, which can be determined in a solver during the simulation. The power of the air conditioners set up in the store model was controlled by the inlet temperature. The power changed linearly over a range of ± 0.5°C around the set temperature from 0 kW to the maximum temperature and remained at constant values outside the temperature range. For example, when the preset temperature was 25°C in the summer, the power remained at 0 kW before 24.5°C, then the power increased as the temperature rose and was maintained at a maximum after 25.5°C. For the purpose of making the computations easy to converge, the power control function was set in a simplified way despite being different from the real function. Other detailed settings of the HVAC system, such as the ventilatory volume, are shown in Table 6.

Table 6 HVAC system and influx conditions

Item	Season	Conditions
Air conditioner	Summer	Mode: Cooling Volume: 0.0875*4=0.35 m^3/s, Maximum power: 14 kW
	Winter	Mode: Heating Volume: 0.15*4=0.6 m^3/s, Maximum power: 15 kW
Ventilation fan	Summer	The number of vents: 3 Total volume: 2115 m^3/h
	Winter	The number of vents: 4 Total volume: 610 m^3/h

Regarding the outside influx volume from the entrance or backdoor, 3 assumptions were considered in this study: (1) the automatic door at the entrance keeps opening for a duration of 10 s once someone passes, (2) the influx only originates from the entrance when the door keeps opening; in the other situation, the influx originates from the backdoor where the temperature remains at 22°C in the winter and 25°C in the summer, (3) without simulating the door opening and closing, the influx originates from both the backdoor and the entrance all the time in the CFD model to make the computation converge easily. Hence, the influx volume compensating for the outflow ventilation volume was separated from the backdoor and the entrance by the ratio of the time of the door opening and not opening.

The calculations were conducted for six cases each season, as summarized in Table 7. They were divided into two types: various preset temperature experiments and energy-saving strategies experiments. CASE3 was the standard of these cases. Four preset temperature experiments (from CASE1 to CASE4) were adopted to evaluate the effectiveness of the air conditioner temperature changes in the energy consumption of the store. One energy-saving strategy was halving the ventilation volume in the summer and tripling it in the winter (CASE3-V), and the other strategy was replacing the frozen multi-deck DCs with frozen glass door reach-in DCs (CASE3-R). In each case, an initial temperature indoor was set to be the preset temperature of air conditioners to make the computations faster to converge.

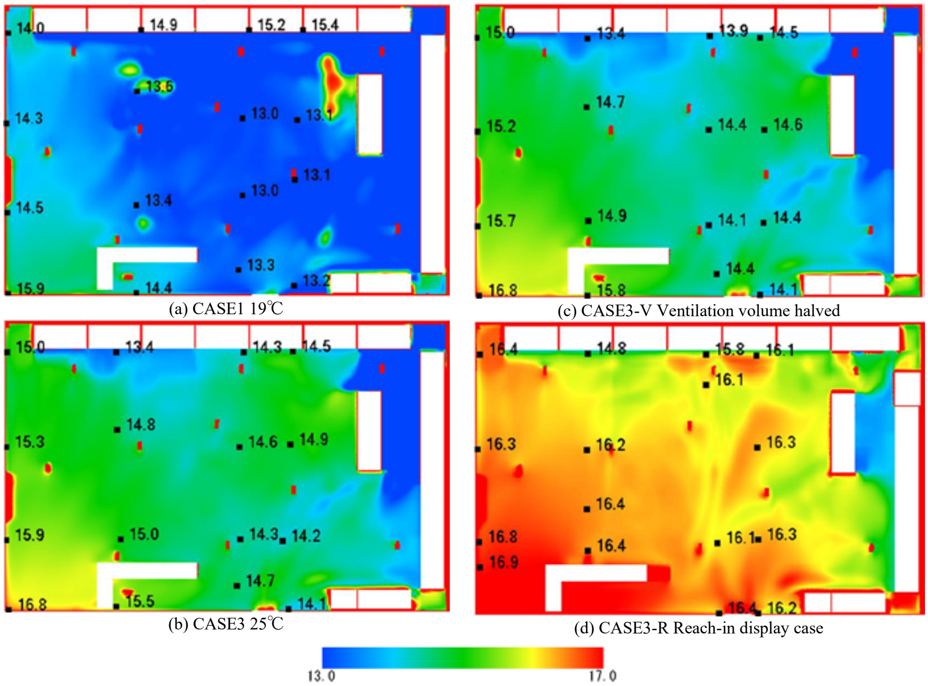
Table 7. Cases list

Experiment type	Case name	Display cases	Ventilation Volume(m^3/h)	Preset temperature
Temperature experiment	CASE1	MD-R, SMD-R, MD-F, TUB-F	Summer: 2115, Winter: 610	19°C
	CASE2	MD-R, SMD-R, MD-F, TUB-F	Summer: 2115, Winter: 610	22°C
	CASE3	MD-R, SMD-R, MD-F, TUB-F	Summer: 2115, Winter: 610	25°C
	CASE4	MD-R, SMD-R, MD-F, TUB-F	Summer: 2115, Winter: 610	28°C
Energy-saving strategies	CASE3-V	MD-R, SMD-R, MD-F, TUB-F	Summer: 1058, Winter: 1830	25°C
	CASE3-R	MD-R, SMD-R, RI-F, TUB-F	Summer: 2115, Winter: 610	25°C

2.3.3. Energy and Electricity Calculation Method

The energy (load) consumption calculation was divided into 2 parts: DC and air conditioner. The formulas summarized above were used to calculate the cooling load of the DCs. For air conditioners, the cooling load or the heating load was calculated from enthalpy differences between the outlet air and inlet air.

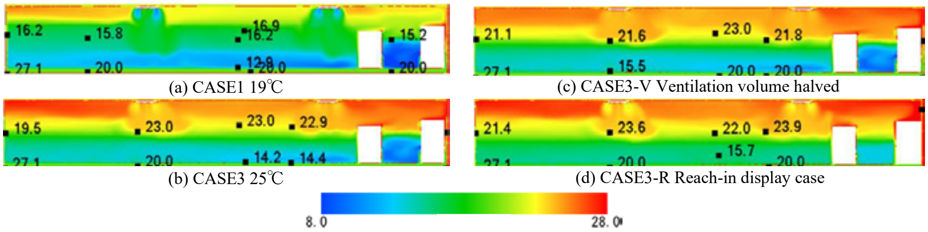
The electricity can be calculated from the load by using the coefficient of performance (COP). In the summer, the COP of the air conditioners was 3.59, the refrigerated DC COP was 1.90, and the frozen DC COP was 0.75. In the winter, the COP of air conditioners was 4.06, the refrigerated DC COP was 1.86, and the frozen DC COP was 0.79.



(a) CASE1 19℃ (c) CASE3-V Ventilation volume halved

(b) CASE3 25℃ (d) CASE3-R Reach-in display case

Fig. 11. Temperature plan distribution at the height of 0.7 m



(a) CASE1 19℃ (c) CASE3-V Ventilation volume halved

(b) CASE3 25℃ (d) CASE3-R Reach-in display case

Fig. 12. Section plan temperature distribution

3. Results

3.1. Summer

3.1.1. Stability

The transient temperatures of the air conditioner outlet air and an indoor measurement point at a height of 1.6 m were used to check the stability of the simulations. The outlet air temperatures remained stable after 1000 cycles, while the indoor air temperatures remained stable after 6000 cycles. Therefore, the results of 10000 cycles were stable enough to be adopted.

3.1.2. Temperature

Fig. 11 compares the temperature plan distribution at a height of 0.7 m for different cases, and the temperature distribution of the section plan shown in Fig. 9 is provided in Fig. 12.

As a result, the higher the preset temperature was, the higher the temperature was in the upper part of the store, where high-temperature air accumulated. On the other hand, low-temperature air accumulated in the lower part of the store and hardly changed as the preset temperature increased. Therefore, the vertical temperature distribution was obviously separated into a low-temperature part and a high-temperature part. The

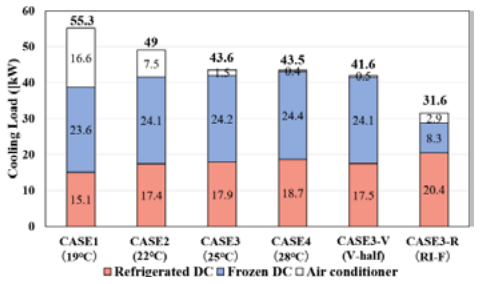


Fig. 13. Cooling load of the air conditioner & the DCs in the summer

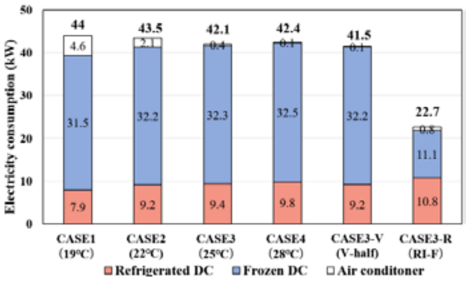


Fig. 14. Electricity consumption in the summer

maximum difference can reach 10℃. For the temperature plan distribution, the temperature decreased from the entrance to the innermost part of the store in every case. The average temperature of the 0.7 m plan was 13.6℃ when the preset temperature was 19℃, and it increased to 15.2℃ as the preset temperature increased to 28℃, which illustrates that the average plan temperature increased as the preset temperature increased.

When the ventilation volume was halved, the high-temperature part of the store became cooler because of the lower high-temperature influx air from outside; in contrast, the low-temperature part of the store hardly changed. The low-temperature part of the store became warmer when the frozen multi-deck DCs were replaced with frozen reach-in DCs.

3.1.3. Load and Electricity Consumption

The energy and electricity consumption in the summer are shown in Fig. 13 and Fig. 14, respectively. The result shows that with an increase in the preset temperature, the cooling load of the DCs increased while the space cooling load of the air conditioner decreased. The total cooling load of the DCs and the air conditioner increased with the increasing preset temperature of the air conditioners. Although the highest preset temperature of air conditioners and the highest total cooling load of DCs and the air conditioner were in the 28°C case, the total electricity consumption was minimized when the preset temperature was set to 25°C. Furthermore, the frozen multi-deck DCs strongly affected the total energy consumption of the store. By replacing the frozen multi-deck DCs with frozen reach-in DCs, the indoor air temperature increased. Therefore, the electricity consumption of DCs was halved, while that of air conditioners somewhat increased. In the aggregate, the total electricity consumption of the DCs and air conditioners obviously decreased while replacing the frozen multi-deck DCs with frozen reach-in DCs.

3.2. Winter

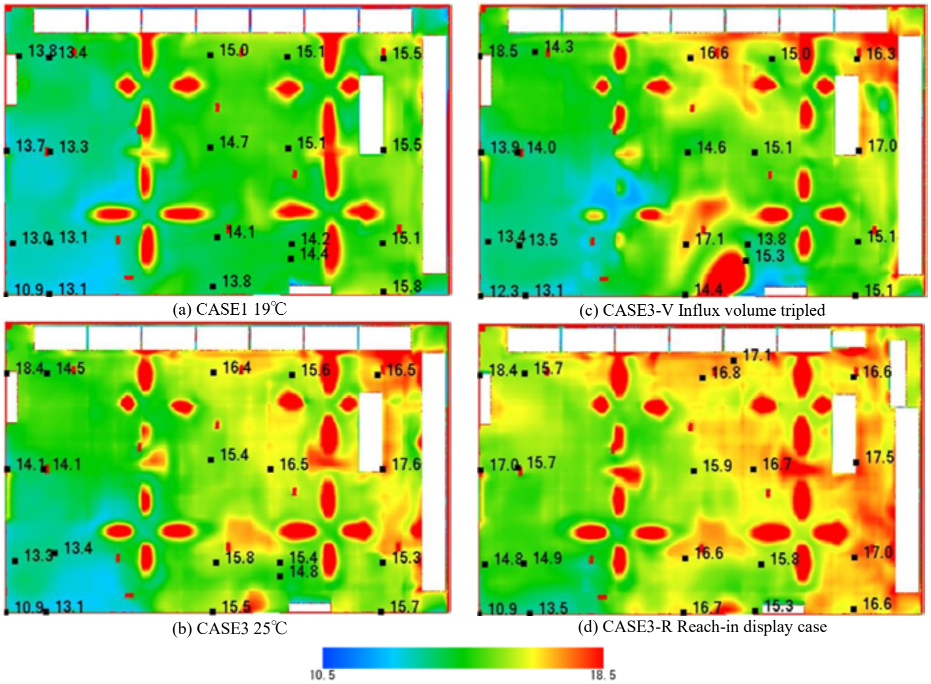
3.2.1. Stability

In the winter, the outlet air temperature remained stable after approximately 1000 cycles. Therefore, the results of 3000 cycles can be adopted.

3.2.2. Temperature

Fig. 15 compares the temperature plan distribution at a height of 0.7 m for different cases, and the temperature distribution of the section plan is provided in Fig. 16.

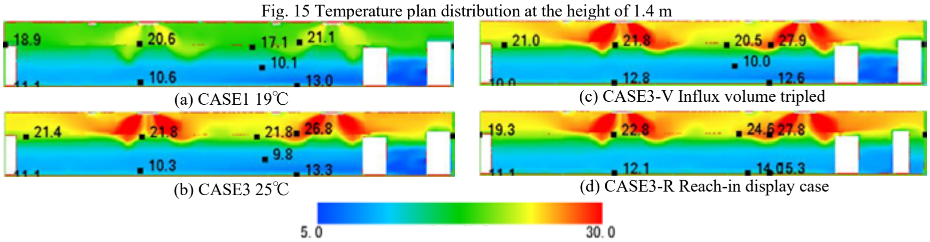
The results were similar to those for summer; the higher the preset temperature was, the higher the temperature was in the upper part of the store, where high-temperature air was accumulated. Simultaneously, low-temperature air accumulated in the lower part of the store and increased as the preset temperature increased, which is different from the summer. Also, with winter having a general opposite temperature compared to the summer temperature, the temperature increased from the entrance to the innermost part of the store. The indoor temperature decreased slightly when the ventilation volume was tripled. The low-temperature part of the store became warmer when the frozen multi-deck DCs were replaced with frozen reach-in DCs.



(a) CASE1 19℃ (c) CASE3-V Influx volume tripled

(b) CASE3 25℃ (d) CASE3-R Reach-in display case

Fig. 15 Temperature plan distribution at the height of 1.4 m



(a) CASE1 19℃ (c) CASE3-V Influx volume tripled

(b) CASE3 25℃ (d) CASE3-R Reach-in display case

Fig. 16 Temperature section distribution

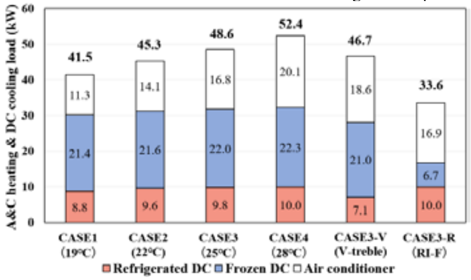


Fig. 17. Air conditioner heating & DC cooling load in the winter

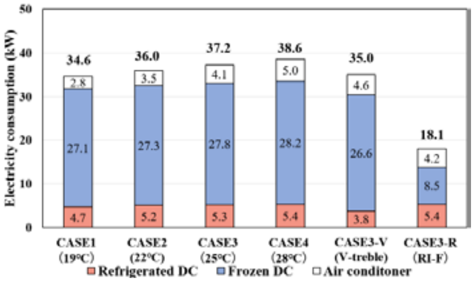


Fig. 18. Electricity consumption in the winter

3.2.3. Energy and electricity consumption

The energy and electricity consumption in the winter are shown in Fig. 17 and Fig. 18. The cooling load of the DCs and the heating load of the air conditioner both increased with increasing preset temperature. Additionally, the total electricity consumption of the DCs and air conditioners increased. Similar to summer, frozen multi-deck DCs also strongly affected the total energy consumption of the store in the winter. The total electricity consumption of the DCs and air conditioners decreased to half by replacing frozen multi-deck DCs with frozen reach-in DCs. In addition, the total load and electricity consumption both decreased when the ventilation was tripled.

4. Discussion

We have shown that the CFD method can assist in simulating the energy consumption of DCs and air conditioners in a store and evaluating the interaction between them without assuming that the temperature is uniform anywhere indoors. By applying simplified DC CFD models, which can be adopted to calculate the cooling load of DCs in different conditions, to study changes in the energy consumption of a grocery store under different conditions, we found that the total energy consumption of the store changed when changing the preset temperature of air conditioners or adopting other energy-saving strategies. In the summer, the result shows that a higher preset temperature of the air conditioner leads to increases in the cooling load of the display cases and a decrease in air conditioners cooling load. In the winter, both the total electricity consumption and the total load increased with increases in the preset temperature. Furthermore, by replacing the frozen multi-deck DCs with frozen reach-in DCs in the summer, the electricity consumption of DCs decreased to half, while that of air conditioners somewhat increased; in the winter, the electricity and the load of the DCs and air conditioners both decreased.

The main cause of the load interaction between the DCs and air conditioners may be explained by the fact that the indoor temperature distribution changed under various conditions. It is surprising that the total electricity consumption in the summer was minimized when the preset temperature was set to 25°C, and the temperature was neither the highest nor the lowest explored. In contrast, the total cooling load of air conditioners and DCs became largest when air conditioners were set to 28°C. This is because the COP of air conditioners is larger than that of the DCs. Tripling the ventilation in the winter, which can lead to a more low-temperature air influx from outside, reduces the energy consumption of the store. An explanation is that the influx air restrained the cooling load of the DCs and comparatively induced a slight increase in the heating load of air conditioners because of the delamination of air. We believe that there might be an appropriate ventilation volume that can lead to the lowest energy consumption of the store, which is worth investigating in the future.

The method we adopted to build the DC CFD models was partly successful because it was able to simulate the cooling load under different temperatures and humidity environments. However, the limitation of the models we built is that they can only be applied under a particular condition, where the preset refrigerant evaporation temperature is -10°C and -40°C for refrigerated DCs and frozen DCs, respectively.

We also found that a vertical air temperature distribution was obviously separated into a low-temperature part and a high-temperature part. Research by Endo et al. shows a similar result [8]. A possible explanation for this is that the density of high-temperature air is lower than that of low-temperature air, which makes the lower air easily accumulate in the upper part of a room. In addition, to make the store model easy to converge computationally, we ignored the air mixing caused by guests moving or doors opening and closing, which might be able to strengthen the delamination in this study. The influence of air mixing should be evaluated in the future.

5. Conclusion

This study set out to develop a CFD store model to imitate the cooling load of DCs and air conditioners in a store and to evaluate the interaction between them. We changed the preset temperature of air conditioners and applied some energy-saving strategies to understand the interaction between the load of air conditioners.

The findings clearly indicated that changing the air conditioner preset temperature or altering the equipment arrangement in stores is in favor of reducing the energy consumption of stores.

- The DC CFD models in this study can be used to calculate the cooling load of DCs under different conditions and to imitate the transient energy consumption of grocery stores.
- If the temperature was increased by the air conditioners, both the load of the DCs and air conditioners increased in the winter; however, in the summer, the load of the DCs increased, while that of air conditioners decreased. The lowest electricity consumption occurred when the preset temperature was 25°C in the summer. An appropriate temperature should be set to save energy rather than being set as high as possible in the summer.
- Replacing open DCs by reach-in door DCs can obviously reduce the load of the store. Reducing the ventilation volume in the summer or increasing it in the winter can also help save energy in the store.

- In the future, factors such as people moving and doors opening should be considered to improve the precision of store CFD models.

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