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Development of a ventilation control system for a commercial kitchen

Eri NAGASHIMA [#1], Takashi KURABUCHI [#2], Yoshihiro TORIUMI [#3], Shiwan LEE[#4], Tomoro HORI[#5], Megumi AIBARA[#6], and Atsushi OKUDA[#7]

[1,2,5] *Tokyo University of Science, Tokyo, Japan*

[1]marinebluepink@gmail.com

[2]kura@rs.kagu.rus.ac.jp

[5]supernova0619@gmail.com

[3] *Polytechnic University, Tokyo, Japan*

[3]toriumig@gmail.com

[4] *Shinshu University, Nagano, Japan*

[4]shany@shinshu-u.ac.jp

[6,7] *Tokyo Gas Co.Ltd., Tokyo, Japan*

[6]m-aibara@tokyo-gas.co.jp

[7]a-okuda@tokyo-gas.co.jp

Abstract

Ventilation control system has recently been attracting attention as a method for saving energy in commercial kitchens, but the quantitative effects in detail has not been understood. The purpose of this study is to verify the energy-saving effects, ventilation effectiveness, and thermal comfort by introducing the ventilation control system in a commercial kitchen. First, we carried out measurements to verify the effects of ventilation control system in a commercial kitchen, which is susceptible to the influence of outside air temperature in the summer. The data indicated that the fan consumed less energy, resulting in a net savings effect while not significantly affecting thermal comfort. Secondly, we reproduced the measurements through the development of a computational fluid dynamics (CFD) simulation to analyze ventilation effectiveness and contribution to the thermal environment. The results of the CFD simulation regarding the ventilation effectiveness for a room indicated that indoor air quality was maintained when a ventilation control system was introduced. During the introduction of the ventilation control system, the energy-saving effect became higher, and the influence on thermal comfort was not significant.

Keywords – commercial kitchen, ventilation, thermal comfort, energy-saving, computational fluid dynamics

1. Introduction

In a commercial kitchen, because of the large amounts of heat and water vapor generated by heated cooking appliances, a large amount of ventilation and air conditioning are required for thermal comfort. However, energy consumption has become a concern by the fans and air-conditioning system. To solve this problem, the use of ventilation control system has drawn attention as a method for saving energy in commercial kitchens. The ventilation control systems can be expected to reduce the energy use by the fans and air conditioning system, because the ventilation rate is constant regardless of the usage conditions of the cooking appliance. The ventilation control system detects the operation of a cooking appliance based on an increase in temperature within the exhaust hood, and then controls the ventilation rate by varying the exhaust fan output in response to the operating conditions. Funado et al. (2012) and Aibara M. et al. (2012) carried out verification effect of the ventilation control system in commercial kitchens. It was confirmed that combine energy saving performance and conservation of thermal comfort by measurement in winter season. In this study, the validity of the ventilation control system is examined based on indoor air quality and the energy saving effects in summer season.

2. Methods

In this study, we carried out measurements and CFD simulation. First, we carried out measurements to introduce a ventilation control system in the commercial kitchen, which is susceptible to the influence of outside air temperature during the summer. Secondly, in order to verify the effect of the outside airflow generated from the double hood, we reproduced the measurements by CFD simulation in relation to ventilation effectiveness and contribution to thermal environment.

2.1 Measurement

2.1.1 System Overview

The layout of the area where the ventilation system was installed, the controlled hoods, design air flow rates, and cooking appliances are shown in Fig. 1. The exhaust system consisted of a hood exhaust system and is located above the cooking appliance. The air supply consisted of two systems. The first system consisted of six vertical horizon shutter (VHS) units that provided a constant amount of outside air adjusted to 22º C, as shown in Fig. 1. The second system consisted of a double-hood air supply providing air from the edge of the hoods in the main kitchen. This air was unconditioned, outside air using a variable air volume system (VAV) in

conjunction with total emissions. Moreover, gas heat pump type air conditioner (GHP) was installed in the main kitchen and cleaning room maintaining the air conditioning temperature at 25° C.

The exhaust rate of the exhaust hoods were patterned as shown in Table 1. As a rule, it was decided that the designed air supply rates would be secured when each kitchen appliance was operating. It was also ensured that 30KQ (KQ denotes theoretical quantity of combustion gas) or more of air was exhausted when a kitchen appliance was not operating. With a focus on the steam convection oven and rice cooker, which have short operating times and a large air flow reduction effect, the total ventilation and exhaust rates of exhaust hoods were patterned into 10 steps. The type of control mode used was dependent on the operational status of a specific cooking appliance. It was also ensured that 30KQ or more of air was exhausted when a cooking appliance was not operating.

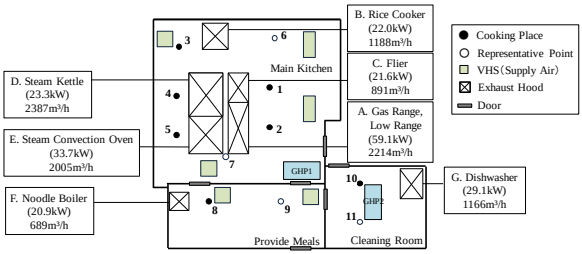


Fig. 1 Test room plan

Table 1. Control mode

Mode	Exhaust Flow Rate [m³/h]							Supply flow rate [m³/h]
	Hood:A	Hood:B	Hood:C	Hood:D	Hood:E	Hood:F	Hood:G	
1	1712	1085	918	1914	1670	1385	881	8398
2	1682	1042	887	1815	809	1346	883	8022
3	1582	979	840	639	1545	1263	922	7756
4	1596	549	843	1730	1364	1209	916	7899
5	1623	979	851	642	778	1152	945	7584
6	1663	561	875	1780	796	1178	858	7838
7	1595	546	845	651	1560	1105	918	7689
8	1683	560	853	676	805	1188	866	7681
9	1325	498	793	589	859	1232	898	6997
10	1298	602	658	629	897	726	697	6095

■･･･Working, □･･･Non-Working

2.1.2 Outline of the measurement

Environmental measurements during system operation (control) and non-operation (non-control) were performed for 19 days from July 29th–Aug. 8th, 2014, and from Aug. 18th–Aug. 29th, 2014. The measurement

items are shown in Table 2. Each measurement item was measured at intervals of 1 m at Points 1–11 as shown in Fig. 1. The CO_2 concentrations were measured every 5 m using a gas selector at Points 1–5. There were no days when the conditions (e.g., operational status of the cooking appliances, outside air temperature, etc.) corresponded completely between either control days or non-control days. Therefore, the results on Aug. 5th and 6th were compared and discussed due to the similarity in kitchen and outside environmental conditions. The time range between 9:00–14:00 was used to compare results as this was the time when cooking functions were primarily performed. The gas consumption rates for the two days are shown in Table 3.

Table 2. Measurement items

Measurement item	Point	Height [mm], notes
Air temperature (Air humidity)	1–5,8,10	Floor Line +100,600,1100,1600,2100
	6,7,9,11	Floor Line +1100,2100
	Outside air	Fan inlet, 1 point
	Supply air	VHS outlet, 6 points
	Return air	GHP outlet, 2 points
	Transport air	Cafeteria, 2 points
Glove temperature	2,4	Floor Line +600,1100,1600
	1,3,5–11	Floor Line +1100
Air velocity	2,4	Floor Line +600,1100,1600
	1,3,5–11	Floor Line +1100
Concentration of CO_2	1–5	Floor Line +1100

Table 3. Gas consumption [m^3]

	Integrated flow rate [m^3]							
	Gas range	Rice cooker	Flier	Steam kettle	Steam convection oven	Noodle boiler	Dishwasher	Total
Non-Control day(5/8)	1.98	0.39	0.75	0.00	1.22	5.40	1.98	11.72
Control day(6/8)	3.24	0.38	0.85	0.37	1.67	4.62	2.12	13.25

2.2 CFD simulation

2.2.1 Outline of the model

The outline of the model is shown in Fig. 2. The main kitchen was the object room used for analysis. Unconditioned outside air was provided through a 25 mm opening at the edge of each exhaust hood. Transfer air flow from an adjacent room (i.e., cafeteria) was blown into the main kitchen through three doors as shown in Fig. 2.

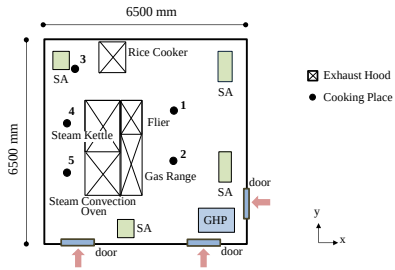


Fig. 2 Plan of model

2.2.2 Outline of the boundary condition

The boundary conditions used for the CFD simulation are shown in Tables 4 and 5. The three cases of the air flow rate for non-control days and control modes 7 and 9 were analyzed as shown in Table 4. Equation 1 was used to calculate the heat provided by the cooking appliances. Half of the heat added to the kitchen space was considered as convection heat while the remaining half was considered as radiation heat. The convection heat added due to the use of gas appliances was calculated using Equation 2. The input of radiation heat considered uniform throughout the room.

Table 4. Boundary conditions for air flow rate

Control mode	Air flow rate [m^3/h]					Temperature [° C]			
	Outside air	Exhaust air	Supply air	GHP	Transfer air flow	Outside air	Supply air	GHP	Transfer air flow
Non-Control	2984	7053	3129	1550	940	32.0	22.0	17.0	23.0
Mode 7	1781	5197			287				
Mode 9	816	4064			119				

Table 5. Boundary conditions for the appliances

Cooking appliance	Declared power [kW]	Thermal efficiency [-]	Load factor [-]	Mode 7				Mode 9			
				Status of use	Amount of heat [kW]	Convection heat [kW]	Radiation heat [kW]	Status of use	Amount of heat [kW]	Convection heat [kW]	Radiation heat [kW]
Gas range	47.4	0.43	0.40	○	8.9	4.5	9.3	×	0.0	0.0	1.9
Low range	11.7	0.24	0.40	○	3.1	1.5		×	0.0	0.0	
Rice cooker	22.0	0.44	0.40	×	0.0	0.0		×	0.0	0.0	
Flier	21.6	0.64	0.68	○	3.8	1.9		○	3.82	1.9	
Steam kettle	23.3	0.46	0.50	×	0.0	0.0		×	0.0	0.0	
Steam convection oven	33.7	0.63	0.30	○	2.7	1.4		×	0.0	0.0	

$$Q = P \times (0.9 - \varepsilon) \times \varphi \tag{1}$$

$$Q_{ex\_in} = C_P \times \rho \times S \times u(T_{g\_in} - T_o) \tag{2}$$

where Q is the amount of heat [kW], P is the declared power [kW], ε is the thermal efficiency, φ is the load factor, Q_{ex_in} is the amount of heat from combustion gas [kW], S is the area of the combustion gas outlet [m^2], ρ is the density [kg/m^3], C_P is the constant pressure specific heat [J/(g·K)], u is the velocity of combustion gas [m/s], T_{g_in} is the exhaust gas temperature [K], and T_o is the base temperature [K].

3. Results

3.1 Measurement

3.1.1 Energy saving

The mode selection status and gas flow rate on a control day is shown in Fig. 3. It was found that the exhaust rates changed depending on operational status of the cooking appliances while the system operated as assumed. A comparison of the fan energy consumption between control and non-control days is shown in Fig. 4. It was found that the energy consumed by the fans was reduced by 56 % creating a significant energy savings effect. A comparison of the air conditioning input energy between control and non-control days is shown in Fig. 5. It was recognized that a control day had a reduction effect of 10% in GHP1. By contrast, a difference in GHP2 was not observed. GHP2 was installed in a cleaning room that did not have a double hood setup, thereby preventing introduction of unconditioned outside air as with other spaces. Consequently, it was considered there was no difference in the outdoor air load as a result of ventilation control.

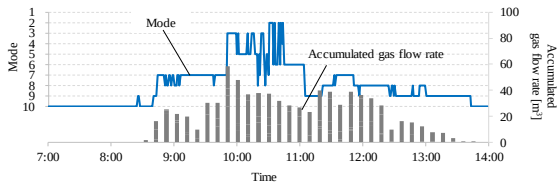


Fig. 3 Mode selection status and accumulated gas flow rate

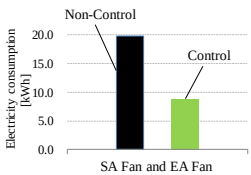


Fig. 4 Fan energy consumption

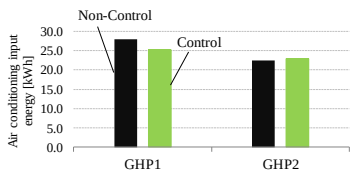


Fig. 5 Air-conditioning input energy

3.1.2 Thermal environment and comfort

The average of the air temperature and mean radiant temperature (MRT) between 9:00–14:00 taken at five points on a control and a non-control day is shown in Fig. 6. The measured data was based on the assumption the kitchen was fully operational. Although the air temperature was high at points 4 and 5 on the non-control day, a significant influence was not seen between those two days. Taking into account the difference in gas consumption between the two days, it was noted the steam kettle was not operated on the non-control day. Additionally, it was noted a significant temperature difference between the control and non-control days did not occur when comparing the gas consumption rates of the steam convection oven for those days.

The predicted mean vote (PMV) was calculated to evaluate thermal comfort. The assumed conditions were the amount of metabolism 2.0 met and 0.4 clo as well as the amount of clothes worn by the people performing the cooking duties. Actual measurements were used for air temperature, glove temperature, air velocity, and relative humidity. The time average of PMV at Points 1–11 are shown in Fig. 7. Although the difference between the PMV for Point 6 was 0.7 and for Point 11 was 1.2, there were only slight differences between the two days. The CO_2 concentration was also measured for those two days with no difference noted. Additionally, there was no influence regarding the escape of exhaust gas by the ventilation control system.

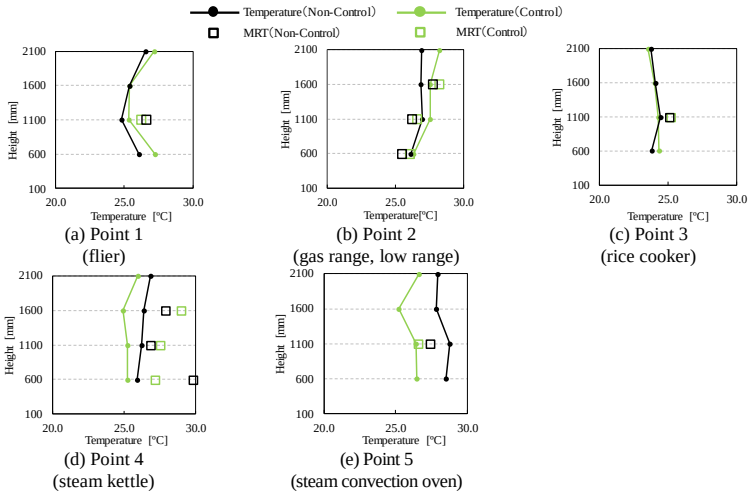


Fig. 6 Average of temperature and MRT for Points 1–5

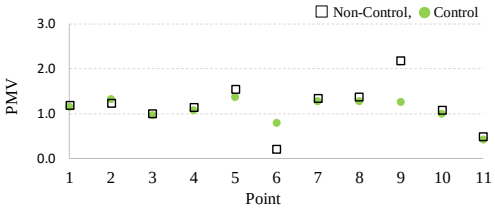
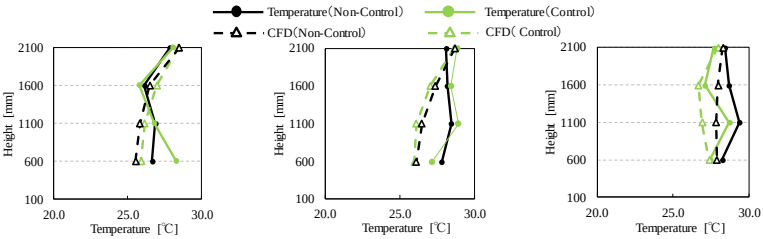


Fig. 7 Time average of PMV for Points 1–11

3.2 CFD simulation

3.2.1 Distribution of air temperature

Fig. 8 shows the air temperature distribution for Mode 7 on a control and non-control day. No major difference was evident for the control situation as represented by Points 1 and 2. In contrast, the air temperature was high at Point 5 on the non-control day. The tendency for a temperature rise near the steam convection oven was confirmed thereby matching the measured values.



(a) Point 1 (flier)

(b) Point 2 (gas range, low range)

(c) Point 5 (steam convection oven)

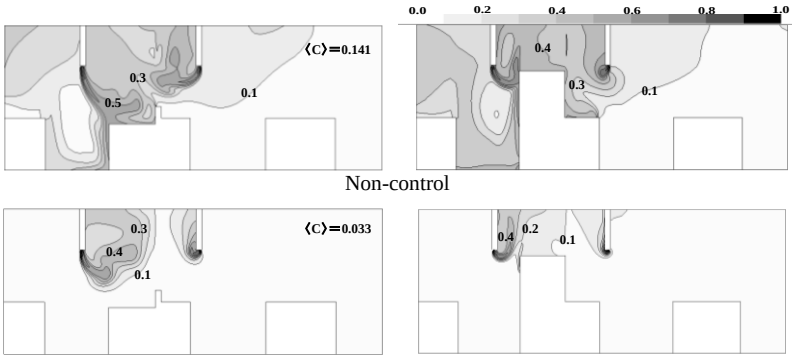
Fig. 8 Comparison of measured values and CFD about the air temperature (Mode 7)

3.2.2 Concentration distribution and contaminant removal efficiency

Fig. 9 shows the concentration distribution of the staining material used for the case where unconditioned outside air was supplied to the outlet of the double hood system for a control and a non-control day. It can be seen the unconditioned outside air from the double hood is masked by the air entering the exhaust hood and does not enter kitchen space. It was also confirmed the unconditioned outside air remained near the steam convection oven on the non-control day. Therefore, it is believed that the

unconditioned outside air had no influence on the temperature rise near the kitchen equipment.

The contaminant removal efficiency (CRE) of the room was determined by Equation 3. The CRE of each analyzed case is shown in Table 6. The CRE was significantly greater than 1 in case of complete air mixing. This confirmed that the CRE tends to be lower as the input of raw outside air into the room increased.



(a) Steam kettle and flier (b) Steam convection oven and gas range

Fig. 9 Concentration distribution of raw outside air (Mode 9)

Table 6. Contaminant removal efficiency (CRE)

	Non-control	Mode 7	Mode 9
CRE	2.33	2.78	6.01

$$CRE = \frac{C_e}{\langle C \rangle} \tag{3}$$

where Ce is the exhaust concentration, and <C> is the average concentration within the room.

4. Discussion

The measured results indicated a major energy saving effect was obtained by introducing a ventilation control system. When the air temperature at the time of actual cooking was measured, there was no significant difference between the control day and non-control day for the majority of the measurement points. The PMV was calculated as a factor of the thermal environment indicating the results were equivalent for air temperature and MRT.

The results of CFD simulation indicated the ventilation efficiency was high on a control day and indicated it was possible to maintain indoor air quality (IAQ) while introducing the ventilation system. The people cooking just beneath a double hood where outside air is provided may be affected. However, despite the higher temperature of the outside air at the double hood, it was determined the effect of the outside air was minimal and did not spread directly into the kitchen space. We concluded the thermal comfort of individuals who were preparing the food improved by introducing a ventilation control system and by reducing the air flow rate.

5. Conclusions

1) From the measured results, the energy savings effect increased with the introduction of a ventilation system. However, the influence on thermal comfort was not significant.
2) The results of CFD simulation indicated that ventilation efficiency was high while controlling the level of ventilation.
3) Indoor air quality was maintained during the introduction of the ventilation control system.
4) It was concluded that the ventilation control system is effective.

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