CONTROL OF AIR FLOWS IN COMMERCIAL KITCHENS

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ABSTRACT

Kitchen exhaust air flow rate should be defined in such a way that it prevents the dispersion of effluents from cooking process into the kitchen and achieve satisfactory thermal conditions by capturing the excess heat that is generated during cooking and frying. Control of air flow rates is one way to influence indoor air quality, thermal conditions and energy economy. Two different air flow control methods were tested during this research. The first stage, was to test manual control where by the users boosted air flow. The second stage, the automatic control method where by air flow rates were controlled by wall mounted temperature sensor, was tested. Finally, the measurement results were compared to the simulation results. The conclusion of these control tests was that it is possible to improve thermal conditions and to save heating and fan energy simultaneously. The heating energy savings were 25 to 45 %, and fan energy savings 50 to 80 % depending on control system. The best control method was VAV system. With simulations it was possible to predict thermal conditions and energy consumption, quite accurately. The simulations are the best methods for comparing temperature levels and energy consumption reached with the different ventilation, and control systems.

KEYWORDS

Kitchen ventilation, Thermal comfort, Energy, Control technology

INTRODUCTION

Kitchen exhaust air flow rate should be defined in such a way that it prevent the dispersion of effluents from cooking process into the kitchen and achieve satisfactory thermal conditions by capturing the excess heat that is generated during cooking and frying. There are many alternating design methods to define this capture and containment air flow rate. These methods are normally based on kitchen floor area, hood area, hood type, heat gains, or number of portions to be prepared daily. Common aspect for all methods is that exhaust air flow rate is usually dimensioned to handle worst-case cooking requirements, so air flow rates are often overstated.

Normally, if the exhaust air flow rate is inadequate to achieve acceptable indoor air quality and thermal conditions, energy economy is good, and when the exhaust air flow rate is sufficient to achieve acceptable indoor air quality and thermal conditions, energy economy is forgotten. The indoor air consideration should have priority, but the energy economy should not be forgotten. This complex problem of designing and operating commercial kitchen exhaust and HVAC systems has led to a situation where there have been a lot of discussions about reducing exhaust air flow rates (ASHRAE 1995).

When both heating energy and fan energy costs are taken into account the average national costs for kitchen ventilation air is \$ 2.0/yr/dm^3/s in Finland. That is four times greater than in USA on average (Claar 1996). Air flow rates, operating hours, supply air temperature and outdoor air temperature have an effect on heating energy costs for kitchen ventilation air.

Control of air flow rates is one way to influence indoor air quality, thermal conditions and energy economy. Thermal conditions and energy consumption of ventilation systems were evaluated in 12 Finnish commercial kitchens by measurements and calculations (Heinonen 1997). According to these measurements, designed air flow rates were needed only occasionally. To guarantee acceptable thermal conditions in all situations the air flow rates should be controlled when cooking equipment or part of them are in idle position. By controlling the air flow rates, it is possible to improve thermal conditions in kitchens and to save heating energy simultaneously.

The control of air flows could be carried out manually or automatically, and it could be continuous or stepwise. This research focused on the comparison of the different air flow control methods, and what kind of effects these control methods have on thermal conditions and energy economy. Both measurements and simulations were included to this research.

METHODS

Measurements

Two different air flow control methods were tested during kitchen ventilation project carried out at the Helsinki University of Technology, and at Olof Granlund Oy, in Finland. The air flow control systems were installed in an existing school kitchen (figure 1) in order to get information how the control systems functioned in real conditions. The type of selected kitchen was heating kitchen. These tests were carried out during autumn 1997 and spring 1998.

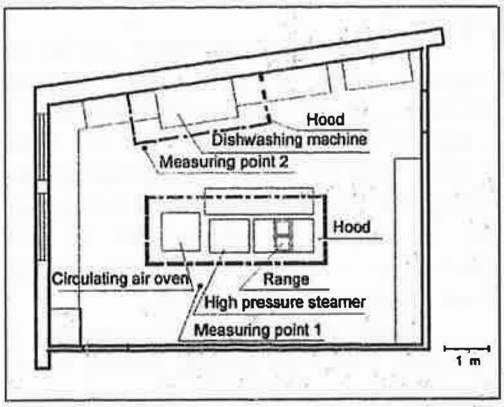

Figure 1. Layout of the kitchen (54 m^2) where the measurements were carried out.

In the first stage the control method where by the users boosted the exhaust and supply air flow rates manually, was tested. Normally, the air flow rates were at minimum level (half of maximum). The boosting was activated with the push button which gave automatically one hour boosting for air flow rates, and after that the air flow rates automatically returned to the minimum level.

In the second stage the automatic control method where by the air flow rates were controlled by wall mounted temperature sensor installed near the dishwashing area was tested. The control of air flow rates was based on PID-controller where the aim was to keep the indoor air temperature at a set value (23 °C). During automatic control test, the minimum air flow rate was also half of the maximum.

Reference measurements were done during January 1998, when supply and exhaust air flow rates were constant, and it was based on maximum air flow rate level. So it is possible to compare thermal conditions in situations in which the control system was in operation versus maximum air flow rate situation.

During the control tests, temperatures were measured at the dishwashing area and at the cooking area. The temperature sensors located near working place at the height of 1,6 meter (figure 1). Also supply and exhaust air temperatures and pressure level of supply air duct were measured. The logging interval was 3 minutes, and the logged value was the average of values measured every 10th second.

The air flow rates were controlled by frequency changers. The minimum frequency for both frequency changers was selected in such a way that the air flow rates were half of the maximum when there was not boosting.

Finally, the measurement results were compared to the simulation results to confirm that with simulation method it is possible to predict thermal conditions and energy savings in commercial kitchens, when air flow rates are controlled.

Simulations

Thermal condition and energy simulations were carried out with integrated building simulation tool RIUSKA developed by Olof Granlund Oy. RIUSKA covers thermal simulation needs of whole building life cycle. The main components of the simulation system are a simulation database, user interfaces, a result module, a building geometry modeller and a calculation engine. The building geometry modeller generates 3-D surface model of the building. The calculation engine is DOE 2.1

Table 1 shows the input power, sensible heat emission and power utilization rate of kitchen equipment which were needed in thermal condition and energy simulations. The power utilization rate describes the average input power when equipment was in use.

Table 1. Technical details needed in thermal condition and energy simulations.

Equipment	Input power (kW)	Sensible heat emission (kW/kW)	Power utilization rate
High pressure steamer	36,0	0,046	0,70
Air circulation oven	27,8	0,105	0,70
Dishwashing machine	32,5	0,427	0,80
Range	6,0	0,418	0,60
Refrigerators	1,7	2,000	0,50

Operation times of kitchen equipment, table 2, were defined according to observations. The hoods were equipped with capture jets which made the hoods more efficiency than traditional hoods. That is why in thermal condition simulations the efficiency of the exhaust hoods was assumed to be 80 %, i.e. the instantaneous heat gain to kitchen was 20 % of total sensible heat gain. The instantaneous heat gains (figure 2) were defined according to table 1 and 2, and equation 1.

Table 2.Typical operation times of kitchen equipment.

Equipment	Operation times
High pressure steamer	09:00-10:00, 11:00-12:00
Air circulation oven	09:00-10:00, 11:00-12:00
Dishwashing machine	09:00-09:30 11:00-12:00 13:00-14:00
Range	-
Refrigerators	00:00-24:00

$$P_g = \Sigma k_g \cdot k_{ur} \cdot P_e \cdot (1-\eta)$$

Where P_g heat gain to kitchen
k_g sensible heat emission, table 1
k_{ur} power utilization rate, table 1
P_e input power, table 1
η efficiency of exhaust hood

Heat gain from lighting was also taken into account in simulations. Assumed heat gain was 20 W/m^2.

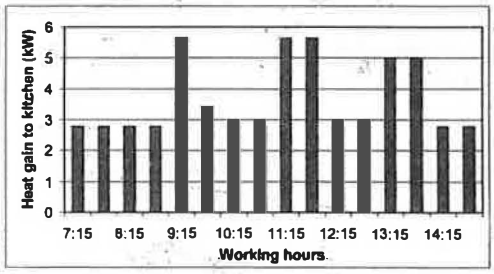

Figure 2. Instantaneous heat gains to kitchen during typical working day (including lighting).

The weather was an average standard year measured at the Helsinki-Vantaa airport during 1979. In simulations the kitchen ventilation system operated during working days 07:00-15:00.

RESULTS

Manual boosting (MB)

In figure 3, the constancy of outdoor air temperatures during 3 measuring weeks at autumn 1997, is shown. Both manual boosting (MB) and reference measurements (CAV) were carried out during heating season (outdoor temperatures were below the supply air set value, 18 °C) which made the results comparable.

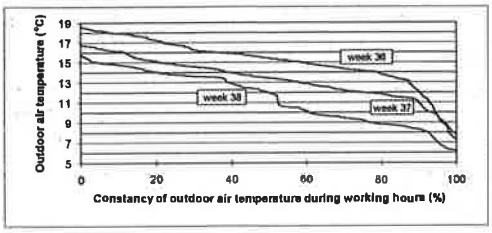

Figure 3. Constancy of outdoor air temperatures during weeks when the manual boosting system was tested.

Table 3. Indoor air temperatures at cooking area and at dishwashing area during reference week and three control weeks.

	Reference week (°C)	Control weeks (°C)
Cooking area		
Average	20,8	22,7
t_r-t_{sup}	3,0	3,1
t_{exh}-t_{sup}	6,1	6,3
Dishwashing area		
Average	21,6	23,8
t_r-t_{sup}	3,7	4,2
t_{exh}-t_{sup}	3,3	4,9

t_r-t_{sup} = average indoor air temperature – supply air temperature

t_{exh}- t_{sup}= average exhaust air temperature – supply air temperature

It can be seen from table 3 that the average indoor temperature level rose about 2,0 °C during manual boosting both at the cooking area and at the dishwashing area.

When average temperature differences between indoor air and supply air (t_r-t_{sup}) are compared, it can be noticed that there was only 0,1 °C rise at the cooking area, and 0,5 °C rise at the dishwashing area. This means that the average supply air temperature rose over 1 °C (18 °C→19,5 °C) during manual boosting. This was due to lower air flow velocity in supply air duct.

When average temperature differences between exhaust air and supply air (t_{exh}- t_{sup}) are compared to average temperature differences between indoor air and supply air (t_r-t_{sup}) it can be seen that these differences were almost same at the dishwashing area during reference week and during control weeks. It means that the exhaust hood was not so effective (spilling) even at maximum air flow rate level.

When the workers were asked opinions about thermal conditions during constant air flow method (CAV) and manual boosting method (MB) they considered that MB was better because during MB method the minimum indoor air temperature was at higher level. So there did not appear to be a draught anymore during no-load conditions.

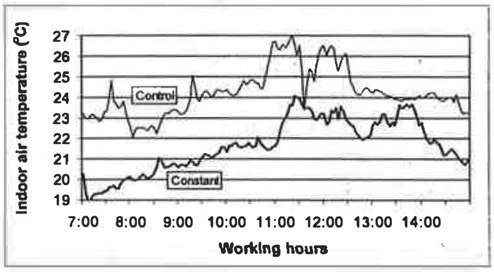

Figure 4. The indoor air temperatures during one working day at the dishwashing area.

Figure 4 shows the indoor air temperatures at the dishwashing area during one working day when MB was in use (control curve), and when maximum air flow rates, CAV, was in use (constant curve). It can be seen from figure 4 that in both cases the temperature level rose during working day, and the shapes of these curves were also very similar. It is also possible to notice MB at 12:00 o'clock. There was not boosting at 7:30 although the temperature level dropped at this point. The figure also clearly pointed out that during manual boosting the average indoor room air temperature was 2 °C higher than during maximum air flow rates.

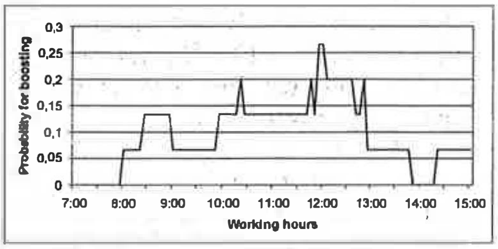

Figure 5. Probability of boosting air flow rates during weeks when control system was tested.

Figure 5 shows the recorded probability of boosting during weeks when MB was tested. The probability of boosting air flow rates was defined by diving the sum of instantaneous boosting hours by total operating days (15 d). For example, at 12:00 o'clock the probability for boosting air flow rates was 26,7 %, which means that air flow rates were boosted during 4 days out of 15 at 12:00 o'clock. The average probability for boosting the air flow rates was 9,5 % which means that kitchen ventilation operated at the maximum air flow rate level only 12 hours during these three weeks.

During MB, the heating energy saving was 45 % and the fan energy saving was 79 % compared to the maximum air flow rate.

Automatic boosting based on clock (AB)

A step further in air flow control method is boosting air flow rates automatically. If it analyzed figure 5, it can be predicted the need for boosting. In this case, it could be recommended automatic boosting between 8:30 and 9:00, and between 10:00 and 13:00. The boosting need between 8:30 and 9:00 was based on dishwashing, as well as between 12:00 and 13:00. The cooking period was normally between 10:00 and 12:00.

If the boosting of air flow rates will be arranged automatically with schedule mentioned above, the heating energy saving will be 30 %, and the fan energy saving will be 50 %, compared to the maximum air flow rate situation.

Automatic control (VAV)

Table 4 shows the indoor air temperatures at cooking area and at dishwashing area during reference week and one control week when automatic control system was tested.

Table 4. Indoor air temperatures at cooking area and at dishwashing area during reference week and one control week.

	Reference week (°C)	Control week (°C)
Cooking area		
Average	20,8	21,7
t_r-t_{sup}	3,0	3,4
t_{exh}-t_{sup}	6,1	6,3
Dishwashing area		
Average	21,6	22,1
t_r-t_{sup}	3,7	3,8
t_{exh}-t_{sup}	3,3	3,6

It can be seen from table 4 that during the automatic air flow control method the average indoor air temperature levels rose less than 1,0 °C both at the cooking area and at the dishwashing area. When average temperature differences between indoor air and supply air are compared, it can be noticed that they were almost equal at the reference week and at the control week.

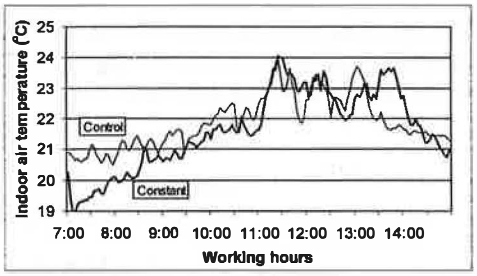

Figure 6. The indoor air temperatures during one working day at the dishwashing area.

It can be seen in figure 6 that the indoor air temperature was more stable with VAV method (thin curve) during morning when there were no heat gains from the dishwashing machine. The rise of the temperature level with the constant air flow rates, CAV, were due to heat gains from cooking equipment. In both cases the indoor air temperature rose above the set value which means that the maximum air flow rate level was insufficient, or the supply air temperature was too high to maintain the set value. However, indoor air temperature level did not reach the set value when there was no-load condition which means that minimum air flow rate level was too high.

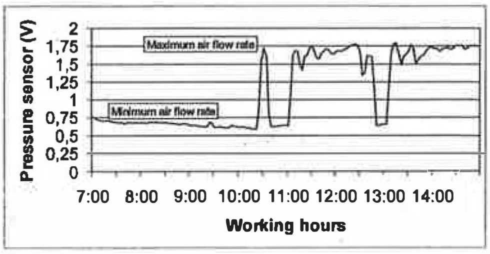

Figure 7. Air flow rate levels with VAV system during one working day.

The VAV system functioned with minimum and maximum basis as can be seen from figure 7. It is possible to evaluate the energy savings by calculating the area between constant line (y = 1,75 V) and the pressure curve.

During the automatic control, the heating energy saving was 28 % and the fan energy saving was 49 %, compared to the maximum air flow rate situation.

Summary of measurements

Table 5. Average indoor air temperatures with different air flow control method

	Cooking area (°C)	Dishwashing area (°C)	Average (°C)
CAV	20,8	21,6	21,2
VAV	21,7	22,1	21,9
MB	22,7	23,8	23,3

CAV constant air volume, reference case

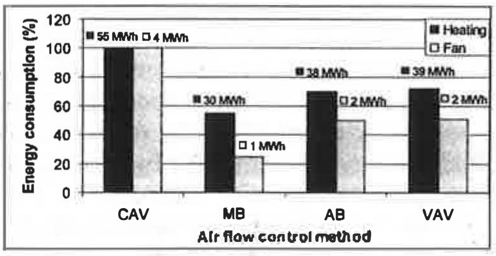

Figure 8. Measured heating energy consumption (%), and electric energy consumption (%). Energy consumption have been proportioned to reference case (CAV).

Simulations

Figure 9 shows the simulated indoor air temperatures during one working day with different control systems. Instantaneous heat gains to room were based on operation times of kitchen equipment, table 2. The simulated ventilation systems were constant air volume system with air flow rate 800 dm^3/s (CAV/800), variable air volume system (VAV), and constant air volume system with air flow rate 1600 dm^3/s (CAV/1600). The constant air volume system with air flow rate 800 dm^3/s described the situation with MB when air flow rates were not boosted during working day. The constant air volume system with air flow rate 1600 dm^3/s described the situation with maximum air flow rates.

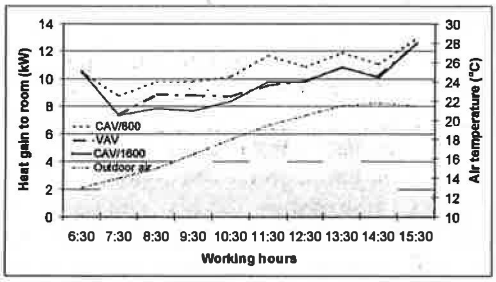

Figure 9. Simulated indoor air temperatures during one working day with different control systems.

When figures 4, 6 and 9 are compared it can be noticed that the temperature profiles and levels were very similar between simulations and measurements. In simulations the indoor air temperatures fluctuated less than in measurements which was due to one hour time step. The average indoor air temperature rose about 2,0 °C with CAV/800 system compared to CAV 1600 system. With VAV system the average indoor air temperature rose about 0,5 °C. When these temperature rises are compared to the measurements (table 5) it can be noticed that values are almost equal. However, in simulations the average temperature levels were about 1,5 °C higher than in measurements which were mainly due to weather used in simulations. Outdoor air temperature level rose during working day up to 22 °C.

Figure 10 shows the constancy of indoor air temperatures with different ventilation system during whole working year, except summer.

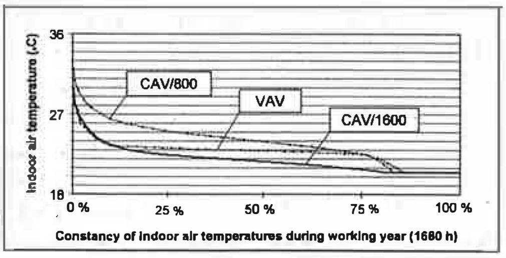

Figure 10. The constancy of indoor air temperatures during whole working year.

Figure 10 clearly shows the benefit of VAV system. The VAV system was able to maintain the indoor air temperature at the set value (23 °C) almost whole working year. With VAV and CAV/1600 the indoor air temperature was above 24 °C only during 120 hours per year (7 % of total working hours). CAV/800 presents the situation when air flow rates were not boosted, so this constancy was the upper limit for manual boosting, and constancy of CAV/1600 was the lower limit.

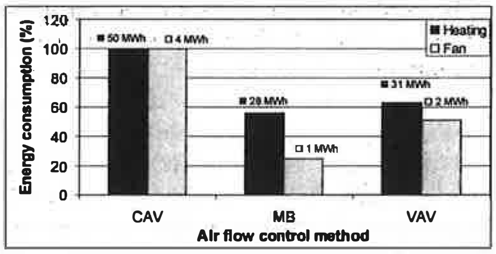

Figure 11. Simulated heating energy consumption (%), and electric energy consumption (%). Energy consumption have been proportioned to reference case (CAV).

The simulated heating energy saving was 45 % with manual boosting, and 38 % with VAV system compared to maximum air flow rate situation (CAV). The simulated fan energy saving was 76 % with manual boosting, and 49 % with VAV. When simulated energy savings and consumption are compared to measurements it can be noticed that the relative energy consumption values were almost equal. Simulated and measured heating energy consumption values differed only with the VAV system. The difference might be due to measurements in which MB and VAV values were defined based on CAV value, and measurement period was only one work week.

DISCUSSION

Although, the control tests were carried out at the kitchen where food production process mostly based on heating, these results are also valid in other process types, like traditional cooking. This statement is based on research (Heinonen 1997) in which thermal conditions were measured in 12 Finnish commercial kitchens with different kitchen and process types. According to this research, in all types of kitchens have situations when cooking equipment are in no-load position.

These control tests and simulations clearly pointed out that it is possible reduce maximum air flow rate level, occasionally. The effects on thermal conditions were minimal. It was even possible to make thermal conditions better when VAV system was in use. With VAV system the minimum temperature level rose, which reduces the draught problems during cold seasons.

The conclusion of these air flow control tests was that it is possible to improve thermal conditions (especially with VAV system) and to save heating and fan energy simultaneously. The heating energy savings were 25 to 45 %, and fan energy savings 50 to 80 % depending on control system.

The best control method was automatic control system (VAV) because it can maintain the acceptable thermal conditions in almost all conditions. The situation would be even better if VAV system is equipped with manual boosting. Automatic boosting based on the clock (AB) would be also a good solution, if it is also equipped with manual boosting. With these two control systems there is always the maximum air flow rates available.

If the control of air flow rates is based only on manual boosting (MB) there might appear some dissatisfaction caused by increased temperature level, which is due to forgotten boosting. MB was the most effective control method when the heating and fan energy savings were compared.

All these air flow control methods are designed to operate only during heating season. So, during hot season, end of spring to autumn, these systems are assumed to operate at maximum air flow rate level.

The conclusion of simulations was that it is possible to predict thermal conditions and energy consumption, quite accurately. There might appear to be small differences in temperature fluctuations and the average temperature levels compared to measurements. However, the simulations are the best methods for comparing temperature levels and energy consumption reached with the different ventilation, and control systems.

ACKNOWLEDGEMENTS

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REFERENCES

ASHRAE applications handbook (1995) *Kitchen ventilation*, Chapter 28, 1-20.

Claar, C. N. (1996) *IFMA's commercial kitchen research - an update*. FM Journal, 9, 30-32.

Heinonen, Jarkko (1997) *Indoor climate in commercial kitchens and possibilities to improve it (Master's thesis)*. Helsinki University of Technology, Espoo