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Thermal plumes of kitchen appliances: Idle mode

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Abstract

In the kitchen environment, pollutant fumes of the cooking process are released into the ambient air by the convection plumes. The practical problem is to compute the requested extract air flow rate to maintain good indoor air quality in an energy efficient manner. In the most accurate design method, the design of a kitchen ventilation system is based on the flow rate of the thermal plume. In this method, the amount of heat carried in a convective plume over a cooking appliance at a certain height is calculated. The heat load is then assumed to be a point heat source and the velocity and temperature profiles are approximated to be Gaussian distributed. In commercial kitchens, the location of the extraction point is at a height of 0.9–1.4 m above the heat source where the convection flow is not yet fully developed. This paper demonstrates that the generic plume equation, derived in the region of complete flow similarity, is not accurate in this intermediate zone. However, it gives a reasonable accuracy for practical applications when an individually adjusted empirical factor of the virtual origin is applied. The power intensity of the heat gain has a much more significant effect on the plume characteristic than the previous studies indicate. The plumes are narrower and the spreading angle is smaller with higher heat gains.

Keywords: Heat load based design; Thermal plume; Kitchen ventilation; Displacement ventilation

1. Introduction

Increased interest in indoor air quality and energy consumption has increased the need to evaluate the convection flow of thermal plumes generated by heat sources. In the kitchen environment, pollutant fumes from the cooking process are conveyed into the ambient air by convection plumes. The practical problem is to compute an extract air flow rate capable of maintaining good indoor air quality in an energy efficient manner.

It should be noted that with an exhaust system like a hood or a ventilated ceiling, it is only possible to capture the convection part of the load. The thermal radiation always ends up into the room space. The main purpose in design practice has been the adjustment of the airflow rate, which is sufficient to extract the convective heat and contaminants from the occupied zone. Undersized air flow rates could lead to indoor air problems and an oversized ventilation system increases unnecessary energy consumption and the life-cycle costs of the ventilation system.

There are many methods available to determine the required exhaust airflow rate. For example, face velocity [1] where air flow rate is determined by the selected capture velocity and the area of the kitchen appliance underneath the hood. This method does not take into account the actual heat gain of the appliances. Hence in many cases, the estimations exceed the actual requirements or demands.

Some codes (e.g., [2]) use either a prescriptive or an engineered procedure for hood design. This engineered procedure is a performance-based approach that allows to utilize a suitable technology to reach the set targets. The solutions should be reviewed in the field or proven with appropriate calculations.

The room energy balance approach is used in the previous German VDI ([3]). The requested airflow rate is calculated based on the sensible load. A more accurate method is based on the heat gain of the appliances ([4]). In this method, the consideration is made for the convective heat output, the area of the appliance and the distance between hood and appliance. The formula of the convection air flow rate calculation ([4]) is derived based on momentum and energy conservation equations and assuming Gaussian velocity and temperature distributions in thermal plume cross-sections. The basic

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Nomenclature

Ar	Archimedes' number
c_p	specific heat at constant pressure (kJ/kg K)
D_h	hydraulic diameter (m)
E	kinetic energy (W)
g	gravitational acceleration (m/s^2)
H	enthalpy flux (W)
L	length of a heat source (m)
m	velocity distribution factor
M	momentum flux (kg/s^2)
p	temperature distribution factor
q_v	air flow rate (m^3/s)
r	radial distance (m)
R_T	profile width where 1/e of the maximum temperature is obtained (m)
R_v	profile width where 1/e of the maximum velocity is obtained (m)
T	temperature (°C)
v	velocity (m/s)
W	width of a heat source (m)
z	vertical distance (m)

Greek symbols

α	entrainment factor
β	thermal expansion coefficient (1/K)
Φ_{conv}	convective heat gain (W)
δ	spreading angle of the plume profile
λ	ratio between temperature and velocity profile widths
ρ	density of air in the plume (kg/m^3)

formula utilized in VDI is originally derived assuming that the complete flow similarity is taking place i.e.: the plume spreads linearly and the ratio between velocity and temperature excess profiles is constant.

In this paper, the actual plumes of typical kitchen appliances are presented using measurements in the laboratory of the Finnish Institute of Occupational Health. Based on the conducted measurements, the actual plume characteristics of typical kitchen appliances are analyzed in order to improve the accuracy of the air flow rate determination and thus to improve the energy efficiency of the total ventilation system.

2. Review of the theory of thermal plumes

Basic assumptions about a plume's axial velocity, excess temperature and plume width from a point heat source have been presented by Schmidt [5] and Morton et al. [6]. Schmidt makes basic assumptions about the flow from a point heat source. The axial velocity, excess temperature and the width are presented as a function of the vertical distance in Eqs. (1)–(3).

$$v_c(z) = f(z^{-1/3}) \tag{1}$$

$$\Delta T(z) = f(z^{-5/3}) \tag{2}$$

$$r(z) = f(z) \tag{3}$$

Rouse et al. [7] introduced expressions for the volume flux, momentum flux and kinetic energy flux by using the similarity hypothesis. The following expression for the air flow rate q_v, momentum flux M and kinetic energy E are obtained:

$$q_v = 2\pi \int_0^\infty vr\,dr \approx z^{5/3} \tag{4}$$

$$M = 2\pi \int_0^\infty \rho v^2 r\,dr \approx z^{4/3} \tag{5}$$

$$E = 2\pi \int_0^\infty \frac{1}{2}\rho v^3 r\,dr \approx z \tag{6}$$

The method to calculate the volume flow rate in plumes based on Gaussian form is used, e.g., by Popiolek [8]. In that method, the velocity distribution were approximated by Gaussian distribution calculated from maximum v_c, width R_v, and distance r:

The mean velocity distribution can then be calculated from:

$$v(r) = v_c(z)\,e^{-r^2/R_v^2} \tag{7}$$

The temperature difference is calculated using the following equation, respectively

$$\Delta T(r) = \Delta T_0\,e^{-r^2/R_T^2} \tag{8}$$

where ΔS_0 is the maximum excess temperature, and R_T is the width of the temperature profile.

When approximating the results with the Gaussian distribution, it is possible to calculate the buoyant jet properties [9].

The volume flow rate (q_v) can be calculated with the Eq. (9):

$$q_v = \pi v_c R_v^2 \tag{9}$$

The vertical momentum flux, M:

$$M = \frac{\pi}{2}\rho v_c^2 R_v^2 \tag{10}$$

where ρ is the local plume density.

The kinetic energy flux, E:

$$E = \frac{\pi}{3}\rho v_c^3 R_v^2 \tag{11}$$

The enthalpy flux, H:

$$H = \pi\rho c_p v_c\,\Delta T_0 \frac{R_v^2 R_T^2}{R_v^2 + R_T^2} \tag{12}$$

where c_p is the specific heat at constant pressure.

Further, the local Archimedes' number Ar defined on a local velocity radius scale as well as the width ratio between the temperature excess and velocity profile (λ) can be calculated as:

$$Ar = \frac{\beta g\,\Delta T_0 R_v}{v_c^2} \tag{13}$$

$$\lambda = \frac{R_T}{R_v} \tag{14}$$

where β is the thermal expansion coefficient, and g is the acceleration due to gravity.

Popiolek [8] has analyzed the plume from a point heat source assuming Gaussian shaped profiles, and presents a local approximation by using a model of a plume above a point heat source. Expressions for the local Archimedes' number and the ratio factor, by means of the temperature excess and velocity distribution factors gives:

$$Ar = \frac{2p}{3m^{2/3}} \tag{15}$$

$$\lambda = \left(\frac{m}{p}\right)^{1/2} \tag{16}$$

The parameter m is called the velocity distribution factor and it describes the width of the velocity profile. It also characterizes the angle of spread. The parameter p is called the temperature distribution factor, having the same function for temperature.

If Eq. (14) is introduced, the temperature excess and velocity distribution factors, p and m can be calculated explicitly as:

$$p = \frac{4}{9} Ar^{-2} \lambda^{-6} \tag{17}$$

$$m = \frac{4}{9} Ar^{-2} \lambda^{-4} \tag{18}$$

The entrainment factor α describes how effectively the convection flow induces room air. The entrainment factor can be calculated according to Eq. (19).

$$\alpha = \frac{5}{6} m^{1/2} \tag{19}$$

In this study, two methods are used for the calculation of the plume parameters: (1) the fitting of the Gaussian distribution, and (2) an integration method where the flow rate is calculated as a sum of the measured velocities multiplied by the respective areas.

Anyhow, there is a practical problem as pointed out by Mundt [10] in using the measured velocities for the air flow rate determination, the ignoring of the tails of the Gaussian curves can cause a significant error to the results. That error of the air flow rate is strongly dependent on the velocity ratio of the velocity in reference point (v_0) and the accuracy of the velocity probe (v_x) [10]. For example, if v_x is 0.05 m/s and v_0 is 0.25 m/s, the calculated flow rate is then 80% of the real flow rate, Eq. (20).

$$q_v = \pi v_c R_v^2 \left(1 - \frac{v_x}{v_0}\right) \tag{20}$$

The accuracy is much better if v_x is 0.02 m/s and v_0 is 0.9 m/s. Then this average velocity approach gives 97% of the theoretically Gaussian distributed flow.

The theory of plumes is widely exploited, e.g., in displacement ventilation design practice. For the practical application, analytical equations to calculate velocities, temperatures and air flow rate in a thermal flow above point and line sources are presented, e.g., Mundt [10] and Baturin [11].

In practice, heat sources are seldom a point source. The most common approach to account for the real source dimensions is to use a virtual origin from which the air flow rate are calculated, Eq. (21).

$$q_v = 0.05(z + z_0)^{5/3} \Phi_{conv}^{1/3} \tag{21}$$

In the VDI kitchen standard [4], which utilizes thermal plume approach, the virtual origin (z_0) is set to be at 1.7 D_h below the surface of the appliance. D_h is the hydraulic diameter of the heat source

$$D_h = \frac{2LW}{L + W} \tag{22}$$

where is the L is the length of the heat source, and W is the width of the heat source.

It should be noted that the plume follows the Gaussian distribution only after a certain distance. The flow in a buoyant round jet can be divided in three distinct regions where only the last pure plume region follows the Gaussian distribution. In Kofoed's study [9], the third region starts from the height of 1.8 m above the heat source. Determination of boundaries between the three regions of a buoyant jet is not easy because there is no abrupt change from one region to the other. Therefore, the plume models handle these regions as one group and the model of the plume above the heat source is described by flow of the pure plume region.

Since zero stratification very seldom occurs in real ventilated spaces, the question about the influence of the vertical temperature gradient arises. The effect of the temperature gradient has been studied by several researchers (Fitzner [12], Kofoed and Nielsen [13] and Mundt [10]). In these studies, they have reported that the vertical volume flux from a heat source decreases when the gradient increases. Anyhow based on Mundt [10] study, the effect of the temperature gradient on the convection flow is very small up to the height of 1 m above the heat source. At the level of 2 m, the effect of the temperature gradient was about 8% when the heat source and the temperature gradient were 80 W and 3 K/m.

It should be noted that in all the published studies the used heat sources are relatively small like a person, a desk lamp, a computer or a heated cylinder. In the commercial kitchen applications, the heat gains are much higher (3–60 kW) which means that the effect of the temperature gradient is not so significant than, e.g., in the office environment. Nonetheless, the generic model for the effect of the temperature gradient on the convection flow is missing.

3. Research method

It is still quite common practice to estimate exhaust airflow rates based on rough methods. The characteristic feature of these methods is that the actual heat gain of the kitchen appliance is neglected. These kinds of rough estimation methods lead not optimal solutions: the size of the whole system will be oversized and energy consumption will increase. The principal idea of the measurements was to analyse the convection flows of the actual kitchen appliances. In practical applications, the measured data can be utilized in improving the accuracy of the existing design practice of the ventilation air flow rates in commercial kitchens and also more general displacement ventilation applications. In this study, temperature and air velocity measurements were conducted to characterize the thermal plumes of typical kitchen appliances in a laboratory environment.

The measurement of thermal plumes is a demanding task. The buoyant jet flow is very sensitive to outside disturbances such as room draft and enclosing walls, and the flow itself may influence the conditions of the room. Further, the flow is unstable and the velocities and temperature excesses are small presenting demands for measurement instruments and their calibration.

In this study, the extrapolation method has been used to analyze plume characteristics. The measurements were carried out in a test room built inside a laboratory facility. Its construction consists of a steel frame, with floor dimensions of 10 m × 4 m and a height of 6 m. The room space is thermally insulated with 50 mm thick polystyrene elements from the surroundings. During measurements, the temperature difference between the average wall temperature of the test room and ambient air temperature at 1.1 m level was less than 2.0 °C. The vertical temperature gradient in the room space varied between 0.2 and 1.1 °C/m depending on the tested appliance.

The supply air flow rate was released, using the displacement ventilation principle, at the floor level from six multi-nozzle ducts, which guaranteed an undisturbed convection flow of a kitchen appliance. The total supply air flow rate of 600 l/s was adjusted to cover the induced air flow rate of the convection flow above the 3 m level. The return air grille was installed at the height of 6 m. The supply temperature was about 20 °C.

The installation principle is different from that is used in laboratory studies of kitchen appliances, where the volume flow rate of thermal plume is measured together with a hood in a wall-type of installation [14]. It should be noted that the island-type installation approach without any local exhaust, which was utilized in this study, makes it possible to get a generic view of the plume distribution.

Convection plumes from an iron range of 6 kW (500 mm × 800 mm × 950 mm (*H*)), a chrome range of 6 kW (500 mm × 800 mm × 950 mm (*H*)), a two burners gas range of 5.5 kW (burners of 2 and 3.5 kW, 400 mm × 650 mm × 460 mm (*H*)) and an induction griddle of 6 kW (520 mm × 440 mm × 175 mm (*H*)) were studied under idling conditions. The studied appliances represent the state-of-the art technology of kitchen appliances. The iron range is a typical range used in commercial kitchens. The chrome range has a temperature sensor in its surface plate that improves energy efficiency during partial load conditions. Furthermore, the surface is made of a low-emission material that reduces radiation compared with traditional cast iron surface. The induction griddle has a fast and accurate control that maintains the set surface temperature constant during different cooking conditions.

During the tests, the actual power of the electric appliances was measured with a clip-on-ammeter. The power of the gas range was determined through the consumed quantity of gas by weighting the gas bottle continuously during the test. It should be noted that the power of kitchen appliances can also be determined from the energy balance in the test room as described by ASTM [15]. However, the direct power measurement was selected in this study because of the simplicity and accuracy of the measurement procedure.

The velocity and temperature measurements were performed using a measurement robot. The convection load was determined from the temperature and velocity measurements in a horizontal plane. The probes were attached to a computer-controlled traversing system moving them from point to point and scanning the determined four measurement planes at the heights of 0.8, 1.2, 1.6 and 2.0 m above the appliances. The basic measurement grid of 1.1 m × 1.1 m consists altogether of 121 measurement points (0.1 m interval) in each plane.

In the literature, hot wire, vane and hot sphere anemometers, smoke, video recording and digital image processing procedure as well as laser-Doppler-anemometers, have been used to measure the velocity distribution in convection flows (e.g., Kofoed [9], Mundt [10] and Welling [16],). In this study, the air velocity was measured with Kaijo Denki WA-390 ultrasonic probes having an accuracy of 0.02 m/s. These sensors measure air velocity vector components with three pairs of ultrasonic transducers by registering the flight time of an ultrasonic pulse. The air temperatures were measured with Fenwal thermistors with an accuracy of 0.1 K. Measurements of air velocity and temperatures were time averages over 60 s and the values were recorded with a data logger.

4. Results

Velocity and temperature profiles from an electric range, a chrome range, a gas range and an induction griddle were measured under idling conditions. Based on the conducted measurements, the actual plume characteristics were analyzed to improve the accuracy of the air flow rate determination.

4.1. Velocity and temperature profiles of the iron range

During idle mode, the maximum temperature on the cooking plate is about 510 °C when the actual power of the appliance was 5.17 kW (the nominal capacity of 6 kW). The measured convection power was 1.79 kW (34.6%). Thus, radiation was 3.38 kW (65.4%). On the corners of cooking plates, temperatures varied between 374 and 438 °C. The measured face surface temperature range was between 31 and 73 °C.

Velocity and temperature profiles of an iron range were measured at four different heights (0.8, 1.2, 1.6 and 2.0 m) above the surface of the appliance. The measurement results of velocity distributions at two levels are shown in Fig. 1. In all conducted measurements, the average temperature gradient was 1.1 °C/m between the heights of 1 and 3 m.

The velocity profiles and distributions depict that the plume is quite compact and the maximum velocity does not significantly decrease as a function of the distance. The maximum velocity at different levels maintains between 0.9 and 1.1 m/s.

The maximum air temperature is 52 °C at 0.8 m level. At level of 1.2 m, the maximum temperature is decreased to 38.7 and 34.0 °C at 1.6 m level. Still at the 2.0 m level, the maximum temperature is 32 °C. During the measurements, the surrounding air temperature was between 22.4 and 22.6 °C measured at 2.0 m level from the floor. Temperature profiles measured above the iron range are shown in Fig. 2.

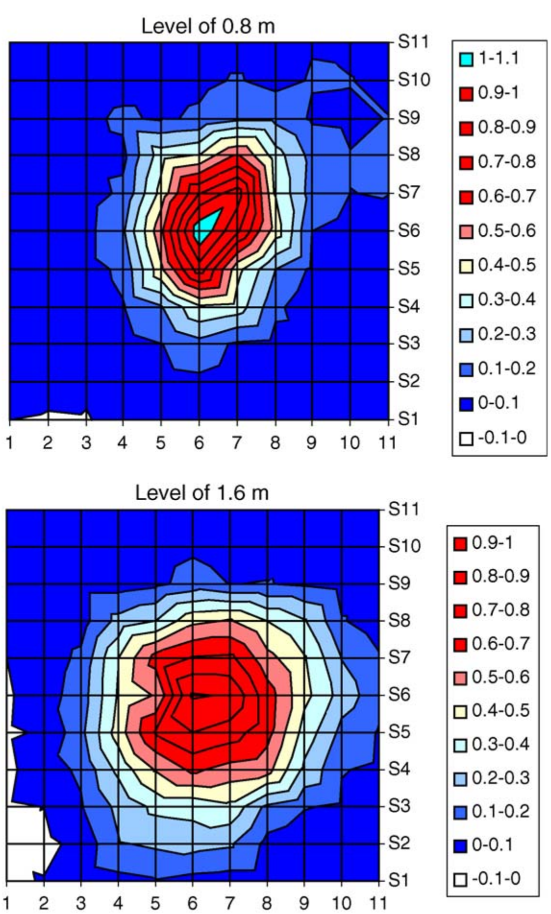

Fig. 1. Velocity distribution of the iron range (500 mm (*W*) × 800 mm (*L*)) at two different levels. The length of a range is accordant with *Y*-axis. The centre point is at (6–S6). The measurement grid is 10 cm × 10 cm.

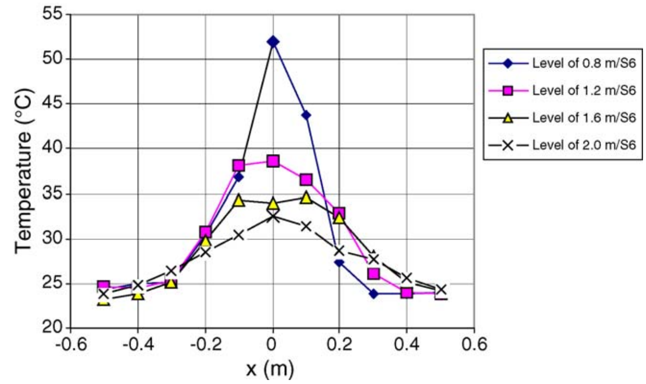

Fig. 2. Maximum temperature profiles above the iron range.

4.2. Velocity and temperature profiles of the chrome range

The control system of the chrome range reduces the electric power in the idle mode by maintaining the set surface temperature. The actual requested power of the appliance was 2.50 kW out of 6 kW connection power. The measured convection power was 1.10 kW (44.3%). Thus, radiation was 1.39 kW (55.7%). In the idle mode, the surface temperatures of the cooking area was between 307 and 439 °C. The system maintains the surface temperature in certain reference point within a 35 °C temperature range (400–435 °C). The measured face surface temperatures were between 28 and 52 °C.

It should be noted that there is a small vertical collar on the rear side of the cooking area. Also behind that collar, the ventilation air of the internal natural ventilated cooling air is released. Both of the previous factors influence on the spreading of the thermal plume.

Velocity and temperature profiles of a chrome range have been measured. The measurement results of velocity distributions are presented at two levels in Fig. 3. In all conducted measurements, the average temperature gradient was 0.35 °C/m between the heights of 1 and 3 m.

The vertical collar of the rear side and the internal ventilation air affected the convection plume. Thus, the maximum velocity is not at the middle of the range at 0.8 m level. At higher levels, the section of the maximum velocity moves 30 cm from line S7–S10. The maximum velocity at different levels was between 0.7 and 0.8 m/s.

The temperature profiles are presented in Fig. 4. The maximum temperature is 34.9 °C at 0.8 m level. At level of 1.2 m, the maximum temperature is decreased to 32.1 and 28.8 °C at 1.6 m level. At the 2.0 m level, the maximum temperature is 26.7 °C. During the measurements, the surrounding air temperature was between 20.4 and 20.6 °C measured at 2.0 m level from the floor.

4.3. Velocity and temperature profiles of the gas range

The thermal plume of a gas range burners was measured without any cooking process. The nominal capacity of the range is 5.5 kW. In the range, there are two different sized burners which capacities are 2 and 3.5 kW. The actual measured power

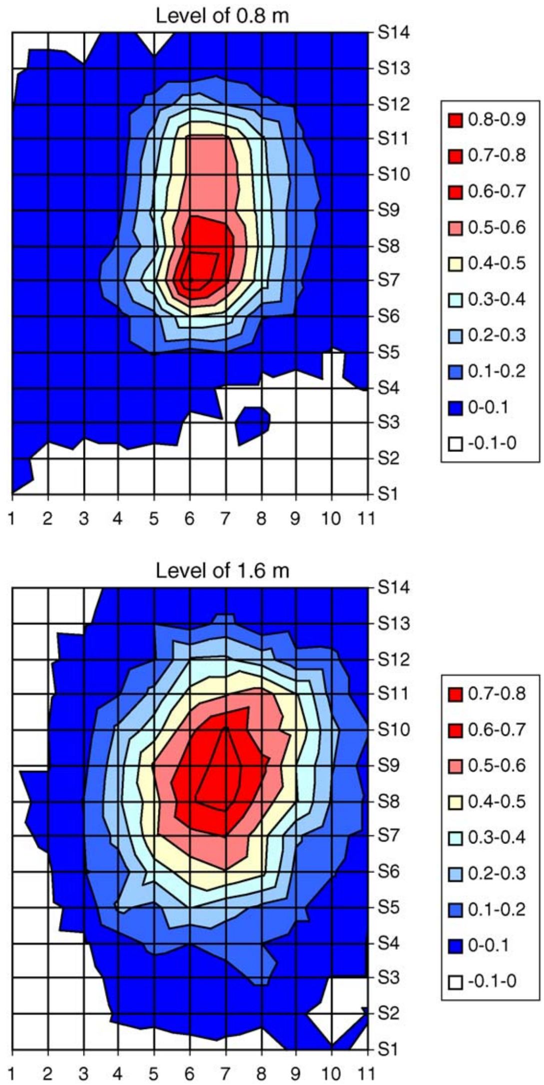

Fig. 3. Velocity distribution of the chrome range (500 mm (W) × 800 mm (L)) at two different levels. The length of a range is accordant with Y-axis. The centre point of a range is at point (6–S9). The measurement grid is 10 cm × 10 cm.

during the test was 4.71 kW. The measured convection power was 3.17 kW (67.2%). Thus, radiation was 1.54 kW (32.8%). The power was determined from the consumed constant gas flow (0.370 kg/h) and the caloric value of the liquid gas (12.8 kW h/kg). In the idle mode, the measured face surface temperatures were between 24 and 42 °C.

The measured velocity distributions are presented in Fig. 5. In all conducted measurements, the average temperature gradient was 0.55 °C/m between the heights of 1 and 3 m. The maximum velocity at different levels stayed between 0.9 and 1.1 m/s.

The maximum temperature was 54 °C at 0.8 m level. At level of 1.2 m, the maximum temperature was 44.7 and 41.6 °C at 1.6 m level. At the 2.0 m level, the maximum temperature was 38.0 °C.

It should be noted that the thermal plume of two flames of the range was quite similar than, e.g., with the iron range. The temperatures above the gas range were only slightly higher than above the iron range.

During the measurements, the surrounding air temperature was between 22.5 and 23.7 °C measured at 2.0 m level from the floor. Temperature profiles measured above a gas range are shown in Fig. 6.

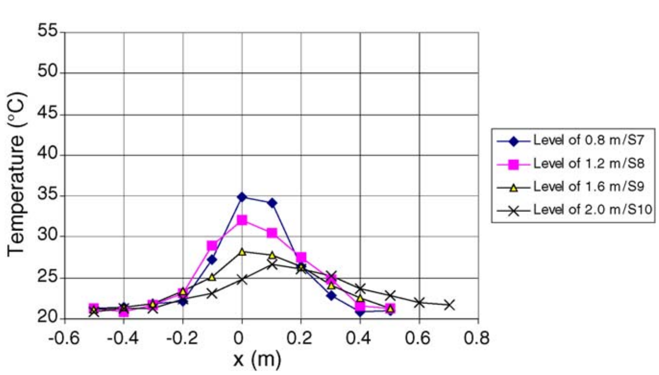

Fig. 4. Maximum temperature profiles above the chrome range.

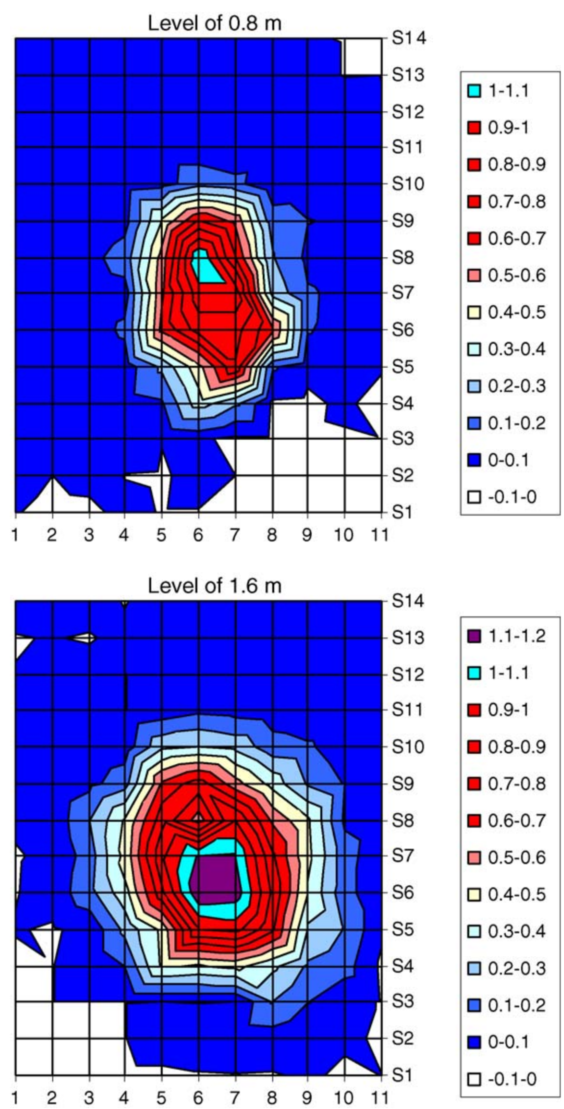

Fig. 5. Velocity distribution of the gas range (450 mm (W) × 650 mm (L)) at two different levels. The length of the range is accordant with Y-axis. The centre point is 6–S6 The measurement grid is 10 cm × 10 cm.

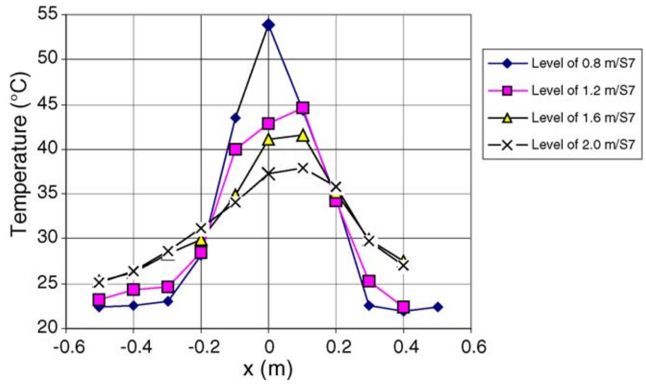

Fig. 6. Maximum temperature profiles above the gas range.

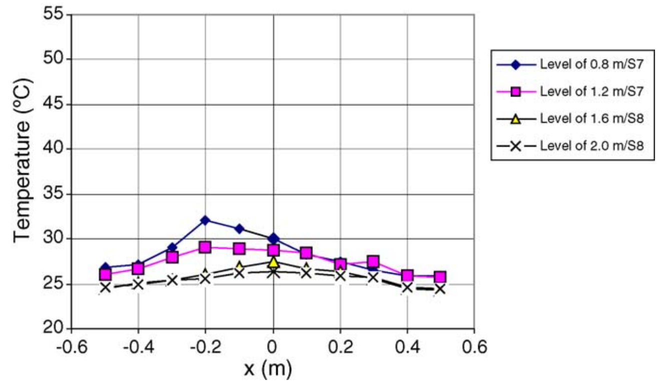

Fig. 8. Maximum temperature profiles above the induction griddle.

4.4. Velocity and temperature profiles of the induction griddle

The control system of the induction griddle reduces significantly the electric power in the idle mode by maintaining the set surface temperature. In the induction griddle, the actual required power of the appliance was 0.87 out of 6 kW connection power. The measured convection power was 180 W (20.6%). Thus, radiation was 692 W (79.4%). In the idle mode, the surface temperatures of the cooking area were about 215 °C (200–227 °C). The measured face surface temperatures were between 33 and 88 °C.

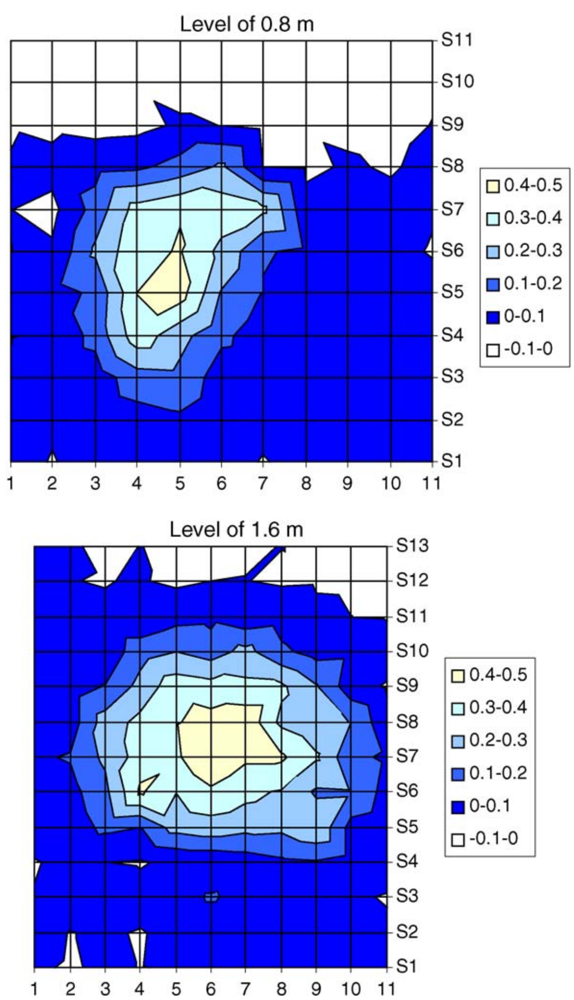

Fig. 7. Velocity distribution of the induction griddle (500 mm (*W*) × 800 mm (*L*)) at two different levels. The length of the range is accordant with *Y*-axis. The centre point is 6–S6 The measurement grid is 10 cm × 10 cm.

Velocity distributions measured above the induction griddle are shown in Fig. 7. In all conducted measurements, the average temperature gradient was lower than 0.25 °C/m between the heights of 1and 3 m. The maximum velocity at different levels stayed between 0.4 and 0.5 m/s.

The temperature distribution is presented in Fig. 8. During the measurements, the surrounding air temperature was between 24.5 and 26.0 °C measured at 2.0 m level from the floor. The maximum temperature was 32.1 °C at 0.8 m level. Higher up the maximal plume temperatures were 29.0 °C at 1.2 m level, 27.4 °C at 1.6 m level and 26.3 °C at 2.0 m.

4.5. Plume characteristics and convection air flow rate

Based on the conducted velocity and temperature measurements, the plume characteristics of the kitchen appliances were

Table 1
Plume characteristic of the iron range

Plume parameter	Distance from the appliance			
	0.8 m	1.2 m	1.6 m	2.0 m
v_c (m/s)	1.07	1.03	0.90	0.92
R_v (m)	0.19	0.23	0.28	0.31
δ_v (°)	7.2	6.7	6.8	6.5
R_T (m)	0.18	0.25	0.30	0.33
δ_T (°)	6.8	7.3	7.2	6.9
ΔT_a (K)	29.4	16.1	12.2	10.0
ρ_a (kg/m^3)	1.09	1.13	1.15	1.16
H (W)	1941	1668	1653	1748
q_v (m^3/s)	0.127	0.170	0.219	0.282
M (kg/s^2)	0.074	0.098	0.114	0.150
E (W)	0.053	0.067	0.069	0.092
Ar	0.155	0.115	0.138	0.123
λ	0.95	1.08	1.07	1.07
p	25	22	16	20
m	23	25	18	23
α	0.18	0.17	0.20	0.17

Table 2
Plume characteristic of the gas range

Plume parameter	Distance from the appliance			
	0.8 m	1.2 m	1.6 m	2.0 m
v_c (m/s)	1.18	1.11	1.20	1.11
R_v (m)	0.20	0.25	0.27	0.30
δ_v (°)	8.3	8.0	7.0	6.7
R_T (m)	0.20	0.24	0.27	0.31
δ_T (°)	8.1	7.6	7.0	6.8
ΔT_a (K)	36.0	27.6	18.7	14.1
ρ_a (kg/m^3)	1.07	1.09	1.12	1.14
H (W)	2838	3167	2874	2639
q_v (m^3/s)	0.152	0.219	0.270	0.323
M (kg/s^2)	0.095	0.133	0.182	0.204
E (W)	0.075	0.099	0.146	0.152
Ar	0.162	0.175	0.111	0.113
λ	0.97	0.96	1.01	1.02
p	20	19	33	31
m	19	17	34	32
α	0.19	0.20	0.14	0.15

computed according to Eqs. (10)–(22). The results are presented in Tables 1–4.

In Table 5, measured values of entrainment, velocity and temperature distribution factors are shown with corresponding values presented in the literature by Schmidt [5], Morton et al. [6], Rouse et al. [7], Kofoed [9], George et al. [17], Nakagome and Hirata [18], Popiolek and Mierzwinski [19] and Shepielev [20].

The convective load is one of the main inputs for the calculation of the air flow rate. The breakdown of the heat gain and the actual value of the convective power are computed from the plume velocity and temperature measurements are presented in Table 6.

The measured convection flows were compared with the generic calculation method of VDI [4] where the virtual origin is set to be at 1.7 D_h below the surface of the appliance. In addition, the effect of the product specific virtual origin on the accuracy of the air flow rate was studied. The empirical factor of each appliance was adjusted to give reasonable correlation with the measurements. The measured and estimated air flow rates of the appliances are presented in Fig. 9.

Table 3
Plume characteristic of the chrome range

Plume parameter	Distance from the appliance			
	0.8 m	1.2 m	1.6 m	2.0 m
v_c (m/s)	0.82	0.75	0.77	0.74
R_v (m)	0.24	0.29	0.32	0.37
δ_v (°)	7.9	7.9	7.3	7.2
R_T (m)	0.23	0.27	0.31	0.35
δ_T (°)	7.6	7.2	7.0	6.8
ΔT_a (K)	14.2	11.6	7.7	6.2
ρ_a (kg/m^3)	1.15	1.16	1.17	1.18
H (W)	1168	1248	1101	1106
q_v (m^3/s)	0.149	0.205	0.250	0.320
M (kg/s^2)	0.070	0.089	0.112	0.140
E (W)	0.038	0.044	0.057	0.070
Ar	0.167	0.203	0.143	0.139
λ	0.96	0.91	0.97	0.95
p	20	19	27	31
m	19	16	25	28
α	0.19	0.21	0.17	0.16

Table 4
Plume characteristic of the induction griddle

Plume parameter	Distance from the appliance			
	0.8 m	1.2 m	1.6 m	2.0 m
v_c (m/s)	0.46	0.43	0.49	0.48
R_v (m)	0.23	0.26	0.31	0.35
δ_v (°)	7.3	6.9	6.8	6.8
R_T (m)	0.22	0.23	0.22	0.24
δ_T (°)	7.1	6.0	4.9	4.6
ΔT_a (K)	6.12	3.72	2.91	1.87
ρ_a (kg/m^3)	1.16	1.17	1.18	1.18
H (W)	251	173	168	129
q_v (m^3/s)	0.073	0.091	0.145	0.189
M (kg/s^2)	0.019	0.023	0.042	0.053
E (W)	0.006	0.007	0.014	0.017
Ar	0.224	0.176	0.125	0.098
λ	0.97	0.88	0.72	0.67
p	11	31	212	523
m	10	24	108	233
α	0.26	0.17	0.08	0.05

Table 5
Entrainment, velocity and temperature distribution factors

Author	Plume parameters			
	α	m	p	s
Popiolek [8]	0.036	42.5	32.8	1.14
Schmidt [5]	0.124	45	45	1.00
George et al. [17]	0.078	55	65	0.92
Nakagome and Hirata [18]	0.103	65	70	0.96
Shepielev [20]	0.045	74	59	1.12
Morton et al. [6]	0.093	80	N/A	N/A
Rouse et al. [7]	0.036	96	71	1.16
Kofoed [9]	0.080	110	115	0.98
Present work (at 0.8 m)	0.18–0.26	10–23	11–25	0.95–0.97

Table 6
The breakdown of the thermal power of the kitchen appliances

Appliance	Total power (W)	Convective power (W)	Radiation and rest (W)
Iron range: 500 mm × 800 mm × 950 mm (H)	5170	1788 (34.6%)	3382 (65.4%)
Gas range: 400 mm × 650 mm × 460 mm (H)	4710	3165 (67.2%)	1545 (32.8%)
Chrome range: 500 mm × 800 mm × 950 mm (H)	2496	1106 (44.3%)	1390 (55.7%)
Induction griddle: 520 mm × 440 mm × 175 mm (H)	872	180 (20.6%)	692 (79.4%)

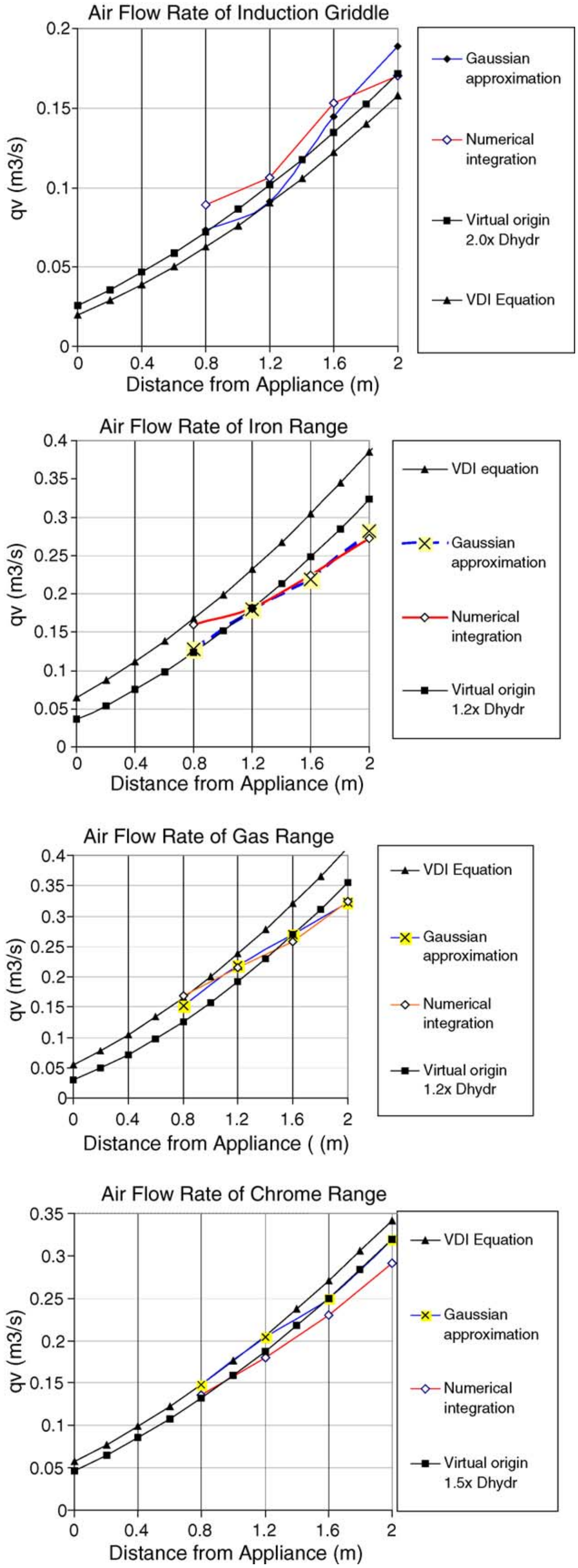

Fig. 9. Measured and calculated induced air flow rates of the plumes.

5. Discussion

Previous studies have pointed out the effect of the temperature gradient on convection flow. In the current measurements, the temperature gradients of the ranges were between 0.3 and 1.1 °C/m. Since the excess temperature of the plume was in order of 10–35 °C in this study, the effect of the gradient in the room space was assumed to be negligible. That assumption is supported Mundt's [10] study where the effect of the temperature gradient on the convection flow was about 8% up to the height of 2 m above the heat source when the heat source and the temperature gradient were 80 W and 3 K/m.

Kofoed's measurements [9] show that the zone of complete similarity starts at 2 m level from the heat gain. The conducted measurement supports Kofoed's findings. The velocity of the plume is not decreasing as a function of the distance from the appliance as the plume theory expects Eq. (1): the velocity of the plume is more or less constant within the measured range of the distance.

In the normal applications of kitchen ventilation and more generally of the displacement ventilation, the consideration point is always below the level of 2 m from the heat source. At this level, the plume has not reached the zone of the complete similarity where the flow is fully developed. The flow is still in the intermediate zone where the plume spreads non-linearly.

In the previous studies, the velocity and distribution factors are much larger than in the present study. In addition, the entrainment factors of the previous studies are much smaller than in this study. It should be noted that the heat gain of the kitchen appliances is much higher than the measured heat output in the earlier studies. The measurements indicate that the heat gain has a significant effect on the spreading angle and the entrainment factor. The plumes with high heat gains are narrower and the convection flow induces more room air than the previous studies have pointed out.

In this study, both Gaussian distribution and numerical integration methods were used for the establishment of the air flow rate. In the numerical integration method, the flow rate is calculated by summing up of the measured velocity samples multiplied by their respective areas. There are pros and cons with both of these methods. The traditionally used Gaussian approximation is accurate if the velocity follows the Gaussian curve. However, close to the appliance the Gaussian approximation is not valid. Direct use of the measured velocities gives an easy platform for air flow calculation but the accuracy of the probes should be sufficient. In this study, both Gaussian approximation and the direct integration of measured velocities gave almost the same air flow rates.

Because the generic plume equation derived in the region of complete flow similarity is not exactly valid in the intermediate zone, the generic power 5/3 of the distance in the plume equation can not fully describe the induced air flow rate even when the virtual origin is adjusted. With the adjusted virtual origin, it is possible to reach a reasonable accuracy for practical applications. However, the principle of the constant virtual origin as utilized in VDI [4] is not adequate. The appliance specific factor of the virtual origin should be exploited.

In this study, the adjusted virtual origin gives a reasonable accuracy. The constant virtual origin approach of VDI standard overestimates the high temperature appliances and underestimates the low temperature appliances. The location of the product specific virtual origin is strongly dependent on the surface temperature of the appliance. Still, some details of appliances have also a significant effect on the location of the virtual origin: the surface temperature of a chrome range is relatively high but the location of the virtual origin is at low level.

6. Conclusions

The main purpose in design practice has been the adjustment of the airflow rate, which is sufficient to extract the convective heat and contaminants from the occupied zone. Undersized air flow rates could lead to indoor air problems and an oversized ventilation system increases unnecessary energy consumption and the life-cycle costs of the ventilation system. In the most accurate design method, the design of a kitchen ventilation system is based on the flow rate of the thermal plume.

The generic plume equation derived in the region of complete flow similarity is not exactly valid in the intermediate zone where the air velocity stays nearly constant from the surface of the appliance to the level of 2 m. The power intensity of the heat gain has a much more significant effect on the plume characteristic than the earlier studies indicate. Especially, the plumes are narrower and the spreading angle is smaller with relatively high heat gains.

However, the generic plume equation gives reasonable accuracy for practical applications when an individually adjusted empirical factor of the virtual origin is applied. In this study, the adjusted virtual origin gives a reasonable accuracy. The constant virtual origin overestimates the high temperature appliances and underestimates the low temperature appliances. The location of the product specific virtual origin is strongly dependent on the surface temperature of the appliance. Still, some details of appliances could have also a significant effect on the location of the virtual origin: the surface temperature of a chrome range is relatively high but the location of the virtual origin is at low level.

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