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AN EXPERIMENTAL STUDY OF THERMAL PLUMES OF KITCHEN APPLIANCES

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ABSTRACT
In the kitchen environment, pollutant fumes of the cooking process are released into the ambient air in 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 in the thermal plume. The heat load is 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. Anyhow, it gives reasonable accuracy for practical applications when individually adjusted empirical factor of the virtual origin is applied.

INDEX TERMS
Thermal plume, convection flow, kitchen design, displacement ventilation

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 kitchen environment, pollutant fumes of cooking process are released into ambient air in 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.

It should be noted that with an exhaust system like a hood or 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.

There are many methods available to determine the required exhaust airflow rate. For example face velocity (CP13 2000) where air flow rate is determined by selected capture velocity and the area of the kitchen appliance underneath the hood. This method is not taken into account the actual heat gain of the appliances. Hence in many cases, the estimations always exceed the actual requirements or demands.

The room energy balance approach is used in the previous German VDI (VDI 1984). Based on the sensible load, the requested airflow rate is calculated. More accurate method is based on the heat gain of the appliances (VDI 1999). 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 (VDI 1999) is derived based on momentum and energy conservation equations and assuming Gaussian velocity and temperature distribution in thermal plume cross-sections. The basic formula which is utilized in VDI is originally derived assuming that the complete flow similarity is taking place: 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 aboratory of the Finnish Institute of Occupational Health. Based on the conducted measurements, the accuracy of the generic plume equation is analyzed.

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RESEARCH METHODS

The principal idea of the measurements is 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 kitchen and in more generally in displacement ventilation applications. In this study, temperature and air velocity measurements were conducted to characterize thermal plumes of typical kitchen appliances in a laboratory environment. The extrapolation method has been used to analyze plume characteristics.

The measurements were carried out in a test room built inside a laboratory facility. Construction consists of a steel frame, with floor dimensions of 10 m x 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 ^oC. The supply air flow rate is released, using displacement ventilation principle, at the floor level from six multi-nozzle ductworks, which guarantees 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 meters level. The return air grille was installed at 6 m level. The supply temperature was about 20 ^oC. The used installation principle is different that is used in a laboratory study of kitchen appliances, where volume flow rate of thermal plume is measured together with a hood in a wall-type of installation (Gerstler et al 1999). It should be noted that the island-type installation approach without any local exhaust, which was utilizes in this study, makes it possible to get a generic view of the plume distribution.

Convection plumes from an iron range, a chrome range, a gas range and an induction griddle 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 the surface plate that improves energy efficiency during partial load conditions. Furthermore, the surface is a low-emission material that reduces radiation compared with traditional cast iron surface. Induction griddle has fast and accurate control that maintains the surface temperature constant during different cooking conditions.

During the tests, the actual power of electric appliances was measured with a clip-on-ammeter. The power of the gas range was determined through the consumed quantity of gas. The nominal connection powers, the average gradient of air temperature in the laboratory, the surface temperature of the appliances, the measured convection load and the dimension of the appliances are presented in Table 1.

Table 1. *Description of the kitchen appliances and measurement conditions.*

Appliance	Description
Iron Range, 6 kW Convection load 1750 W ΔT_{grad}= 1.1 ^oC per m $T_{surfacase}$ ~ 400-500 ^oC	Two burners range Surface material: cast iron 500x800x950 (H)
Chrome Range, 6 kW Convection load 1220 W ΔT_{grad}= 0.3 ^oC per m $T_{surfacase}$ ~ 400 ^oC	Two burners range with control of the surface temperature. Surface material: chrome 500x800x950 (H)
Gas Range, 5.4 kW Convection load 3050 W ΔT_{grad}= 0.5 ^oC per m	Two burners range 400x650x460 (H)
Induction Griddle, 6 kW Convection load 180 W ΔT_{grad}= 0.2 ^oC per m $T_{surfacase}$ ~ 200 ^oC	Electronic power control. Surface material: stainless steel 520x440x175 (H)

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The velocity and temperature measurements were performed using a measurement robot, Fig. 1. The convection load is determined based on 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 height of 0.8 m, 1.2 m, 1.6 m and 2.0 m from the appliances. The basic measurement grid of 1.1 m x 1.1 m consists altogether of 121 measurement points (0.1 m interval) in each plane

The air velocity was measured with Kaijo Denki WA- 390 ultrasonic probes, which have an accuracy of 0.02 m/s.

The 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 Fenwale thermistors with an accuracy of 0.1 K.

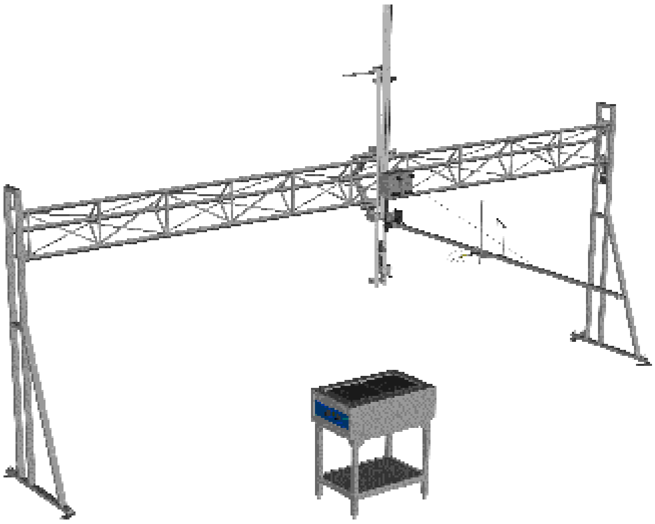


Figure 1. *The principle of the traversing system.*

RESULTS

In this study, both Gaussian distribution and numerical integration methods were used. In the numerical integration method, the flow rate is calculated as the sum of the measured velocities multiplied by the 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 exactly valid. Direct use of the measured velocities gives an easy platform for air flow calculation. But there is a practical problem as pointed out by Mundt (Mundt 1996) in using directly the velocities for air flow rate determination: there could be a significant error to ignore the tails of the Gaussian curves. The affect of the tails is reduced with a better accuracy of the probe. Also, the effect of the tails is not so significant when the maximum air velocity of plume is high.

The measured convection flows were compared with the generic plume equation (VDI 1999), Eq. 1 and 2. In VDI, the virtual origin is set to be at 1.7 D_h below the surface of the appliance. It should be noted that normally in displacement applications the virtual origin is set at 1.8 –2.1 D_h, which gives much higher air flow rate than the VDI equation.

$$q_{v,p} = k \cdot (z + 1.7D_h)^{5/3} \cdot \Phi_{conv}{}^{1/3} \qquad (1)$$

where

$q_{v,p}$ is the airflow in convective plume, [m^3/s]

z is the height above the cooking surface, [m]

D_h is the hydraulic diameter of the appliance, [m]

Φ_{conv} is the cooking appliance convective heat output, [W]

k is an empirical coefficient, k is 5 for a generic hood

$$D_h = \frac{2L \cdot W}{L + W} \qquad (2)$$

L,W are the length and width of the cooking surface, [m]

In addition, the effect of the product specific virtual origin on the accuracy of the air flow rate was studied, Eq.3. The empirical factor of each appliance was adjusted to get reasonable correlation with the measurements.

$$q_{v,p} = 5 \cdot (z + a \cdot D_h)^{5/3} \cdot \Phi_{conv}{}^{1/3} \qquad (3)$$

where
a is the product specific factor of the virtual origin

The measured and estimated air flow rates of a low capacity griddle are presented in Fig. 2. The results of the more energy intensive ranges are presented in Fig. 3.

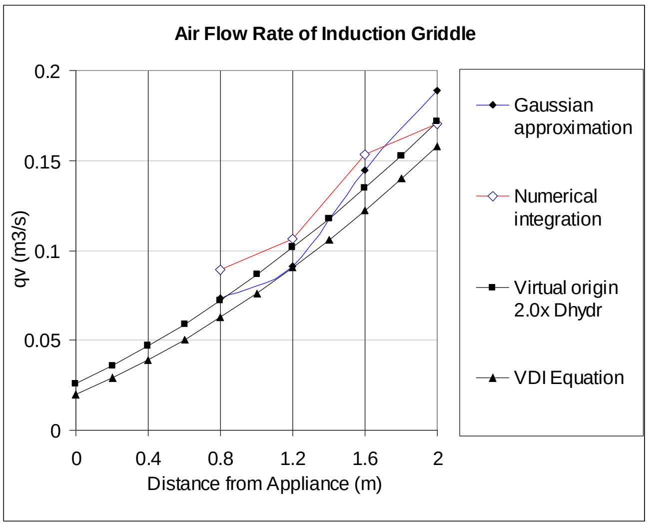
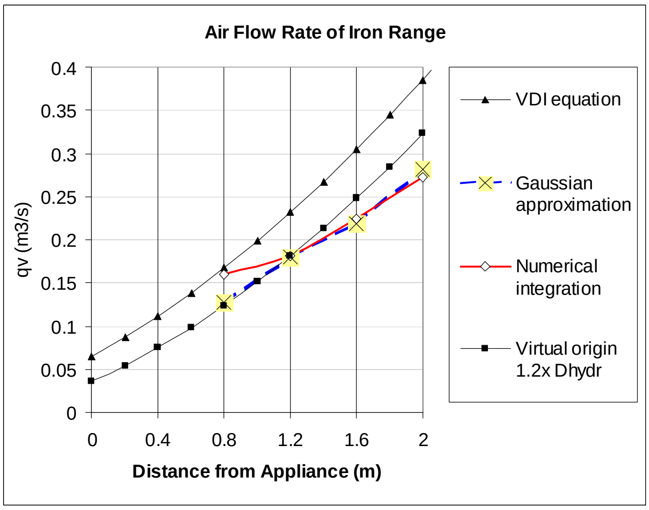
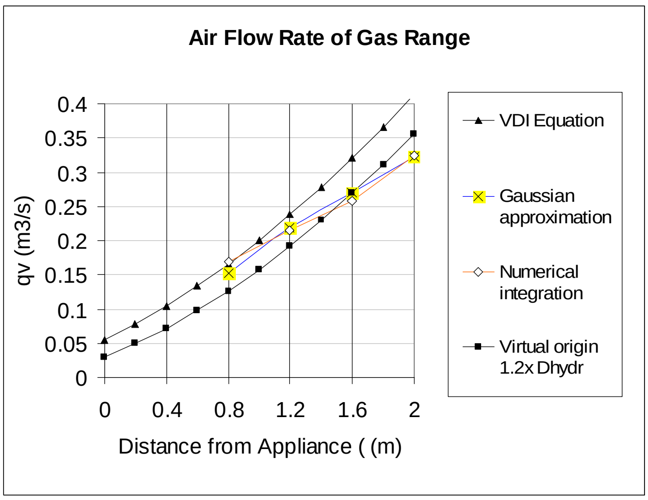


Figure 2. *Measured and calculated air flow rate of an induction griddle.*





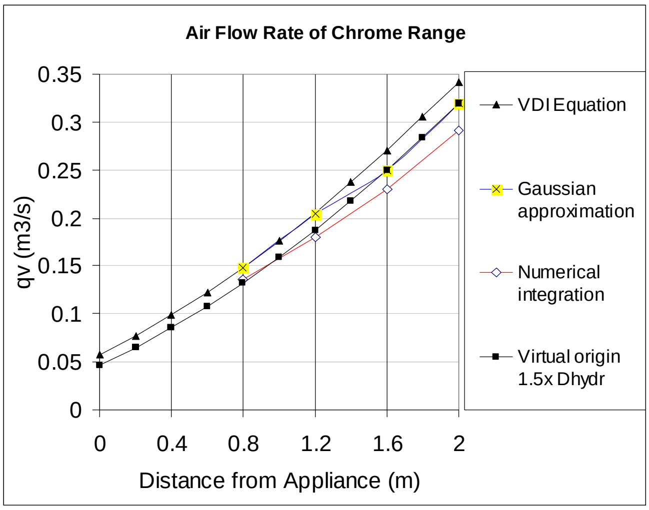


***Figure 3.** Measured and calculated air flow rate of ranges.*

With the energy intensive loads, both Gaussian approximation and the direct use of measured velocities give almost the same air flow rates. However, there is a significant difference with low heat gains like an induction griddle.

With high heat gains, 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. It should be noted that the most important distance from appliance is between 0.9 – 1.4 m. 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 is not adequate.

DISCUSSION

From the results of the induction griddle where the heat gain is low, the Gaussian approximation gives lower air flow rates than the direct numerical integration of the velocities. This indicates that the affect of the missing velocity tails is not significant when the accuracy of the probe is sufficient. This means that the numerical integration in tandem with the Gaussian approximation gives an accurate platform for air flow rate determination.

Previous studies depict the problem with accuracy in the intermediate zone (Popiolek 1981 and Kofoed 1991). Kofoed's measurement show that the zone of complete similarity starts at 2 m level from the heat gain. The conducted measurement supports Kofoed's findings and shows that the power of distance in the generic plume equation is not exactly valid in the intermediate zone. Better accuracy could be reach by developing a novel plume equation for the intermediate zone.

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.

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 – 1.1 ^{o}C per m. Since the excess temperature of the plume was in order of 10 - 35 ^{o}C in this study, the effect of the gradient in the room space was assumed to be negligible.

CONCLUSION

The generic plume equation derived in the region of complete flow similarity is not exactly valid in the intermediate zone. However, it gives reasonable accuracy for practical applications when an individually adjusted empirical factor of the virtual origin is applied.

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