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# THERMAL ENVIRONMENT ASSESSMENT OF GAS STOVE SURROUNDINGS

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***Abstract***

*Residential kitchens with gas stoves are unique spaces inside dwellings.Occupants directly exposed to the combustion products and thermal effects of the stove. Applying a general model with the knowledge of describing parameters, designer could taken thermal aspects also into consideration in order to provide a save, healthy and comfortable environment in residential kitchens and homes. The aim of this study to develop this model and a new ventilation method.*
*Results of field studies shows disadvantageous effect of kitchen exhaust unit during the operation of oven. However turbulence intensity was decreased from 123% to 25% , indoor temperature was increased by 3,7K in the occupied zone due to the forced air movement. PMV index remained within the recommended intervals (between -1,0 and +1.9), therefore it could be applied to evaulate thermal environment aroun the residential gas stove. Unpleasent effect of draught is not expected.*

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***Key words***

- Gas stove,
- Thermal comfort,
- PMV,
- CFD.

## 1 INTRODUCTION

According the survey of the Hungarian Central Statistical Office, there are approximately 3, 8 million occupied dwellings and 53% of these households are equipped with gas stove. This means that indoor environment in more than 2 million kitchens is directly influenced by the combustion products and the heat gain from an open fronted gas-fired equipment.

Based on data provided by CECED (Hungary Domestic Equipment Producing and Distributing Interest Grouping and Representing Society) approximately 13% of the Hungarian kitchens are equipped with a range hood or an exhaust device. Therefore, in most of the cases thermal environment issues and air quality problems will be solved with just by opening a window, irrespectively of energy or other aspects (Sorgato and Melo, 2016).

Under free-running conditions heat emitted in to the space trough radiation and convection. The convective part can be reduced by ventilation. Special attention should be paid to the ratio of radiation, as it cannot be removed from the space even by using the most efficient hood other exhaust device (Livchak et al., 2006). Thermal radiation between human body and warm surfaces could increase the level of dissatisfaction significantly. Consequently, numerical description of the heat load is a fundamental issue, as well as determination of the key parameters. It is neccesseary to examine the applicability of the comfort models and the validity of other microclimate parameters as well (Parsons, 2014).

The main goal of this study is to provide a general model of residential gas stove, and to define the desciptive parameters, in order to facilitate improved ventilation systems in residential kitchens (V.Leitner and Kajtár, 2016). This can be achieved through a simplified source model of the stove at a reasonable level of accuracy. It is crucial that source model should be based on results from free-running (ready to cook) measurements. Therefore, subjective parameters such as cooking habits or food culture can be excluded.

## 2 METHODS AND MATERIALS

Methodology is based on the complex analysis of thermal environment and indoor air quality. Investigation process could be divided two major part: objective evaluation – including laboratory and field measurements - and numerical simulation using computational fluid dynamic (CFD) method. Objective evaluation includes laboratory and field studies, additional calculations and preliminary conclusions. In the course of the numerical simulations different scenarios have been developed by CFD program FLOVENT.

Working conditions were distinguished according the used burner. Three situations were evaluated separately: effects of the largest and the smallest stovetop burner and the oven. The scenarios were characterized by the amount of combusted gas. The burning process was the only significant heat source in the space.

Type of ventilation was described by the amount of exhausted air. Basically, there was no mechanical ventilation ($V_{ex}$=0 m$^3$/h). However, in that case natural air exchange were occurred through the building envelope, rate of this was measured by using tracer gas technique. Mechanical ventilation was provided by an exhaust hood. Amount of airflow was measured previously.

The main indoor environmental monitor point was placed accordingly the expected maximal exposure (Simone and Olesen, 2013) at the height of 1,5m, and at a distance 20 cm from the front of the stove. The measured thermal parameters and the type of applied instruments and sensors are listed in Tab. 1.

### 2.1 Numerical evaluation of the thermal environment

Dry air temperature, humidity and air speed data provide fundamental information about thermal environment. However additional parameters also should be determined.

Effects of radiation can be describe by calculating the mean radiant temperature. Mean radiant temperature can be determined based on black globe temperature and indoor air temperature measurement (Parsons, 2014). However, calculation methods depend on the mode of heat transfer which can be either free (natural) or forced. Mean air speed of 0,1m/s can be defined as upper limit for unforced convection (Bánhidi and Kajtár, 2017). Calculation for natural convection it is given by Equation (1).

$$t_{mrd} = \left[ \left(t_g + 273\right)^4 + \frac{0{,}25 \cdot 10^8}{\varepsilon} \left( \frac{\left|t_g - t_{in}\right|}{d} \right)^{0{,}25} \left|t_g - t_{in}\right| \right]^{0{,}25} - 273 \quad (1)$$

For forced convection $t_{mrd}$ can be expressed by the followings:

$$t_{mrd} = \left[ \left(t_g + 273\right)^4 + \frac{1{,}1 \cdot 10^8 \cdot v^{0{,}6}}{\varepsilon \cdot d^{0{,}4}} \left|t_g - t_{in}\right| \right]^{0{,}25} - 273 \quad (3)$$

In Equations (1) and (2) there is ε for emissivity of the black globe, and d is the globe diameter (see Tab.1).

In residential areas thermal comfort can be generally expressed with the predicted mean vote (PMV) index which is recommended to use in range of -2 to +2. However it is neccesseary to analyze that PMV index is suitable for application in residential kitchens with gas stoves or not (EN ISO 7730:2006). Mathematical determination of PMV is requires additional calculation of partial water pressure and the convective heat transfer coefficient. Partial water pressure can be determined on the basis of measured dry and wet air temperature according the fundamental psychrometric relations. The convective heat transfer coefficient also depends on type of convection:

$$\alpha_c = 2{,}38\left(t_{clo} - t_{in}\right)^{0{,}25} \text{ if } v \le 0{,}1\text{m/s} \quad \text{or} \qquad \alpha_c = 12{,}1 \cdot v^{0{,}5} \text{ if } v > 0{,}1\text{m/s} \quad (3)$$

where $t_{clo}$ is the surface temperature of clothing, which is given as a complex function of personal data and measured thermal parameters, in close connection with PMV theory. Mathematical expression can only be solved by an iterative process (Awbi, 1998).

## 3 RESULTS

Data presented in this paper were recorded in Göd, Hungary. On field measurements had been conducted in a residential kitchen (*$V_{room}$=29,5m$^3$*), under real-life circumstances. Investigated stove was a six-year-old, regulary maintained equipment, which could be considered as a popular model in Hungary. During the assessments, the gas stove was the only heat and pollutant source in the kitchen and internal moisture load was equal to zero. Mechanical ventilation was produced by just the exhaust hood unit, and there was no there was no additional fresh air intake. Convection and air movement around the stone was not disturbed by any other object or effect. Assessments were carried out under windless climatic conditions therefore natural air exchange was generated by temperature difference.

This study is focusing thermal environmental results.

### 3.1 Basic input data

Dimensions of the stove were 600x600x850 mm, stovetop openings were placed for oven ventilation in the rear wall. Oven was equipped with non-insulated glass. The largest stovetop burner was placed near to the wall (Fig. 1).

### 3.2 Preliminary measurements

Fundamental parameters of types of ventilation and characteristic data of the three working conditions were determined by gas con-

**Tab. 1** *Thermal environmental monitoring*

| Thermal parameter | | Instrument, sensor |
|---|---|---|
| Climatic conditions | outdoor air temperature | TESTO data loggers |
| | outdoor realativ humidity | |
| Indoor thermal environment | dry air temperature | TESTO 454 / WBGT sensor (DIN 33403, ISO 7243); d=0,15m; ε=0,95 |
| | wet-bulb temperature | |
| | black globe temperature | |
| | air speed | TESTO 454 / Thermal Comfort Probe (EN 13779) |

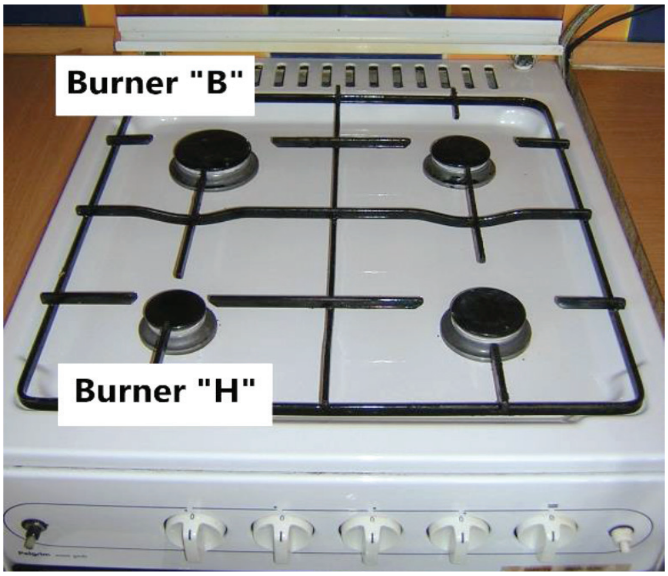


**Fig. 1** *Gas stove – Type 2*

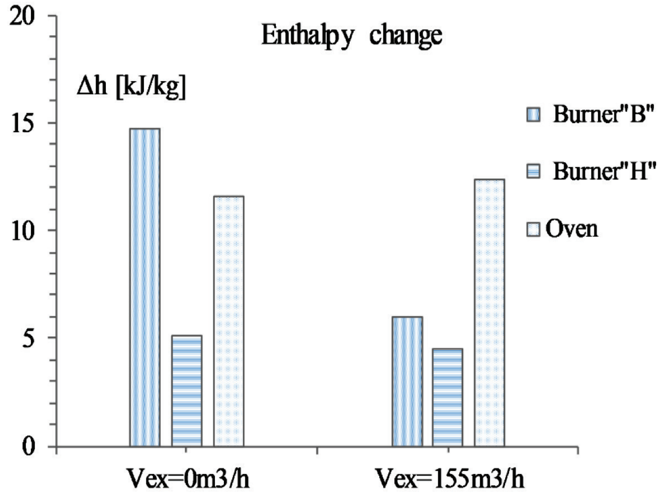


**Fig. 2** *Enthalpy change (τ=20min)*

sumption *NGC* (Tab. 2). Type of the gas was *2H*, net calorific value 34190kJ/$m^3$. Based on gas consumption data the most convective heat load is expected during the operation of oven.

Air exchange rate refers to high insulated, air thigh openings. Based on literature data air change rate is approximately is equal to the recommended minimum (Bánhidi and Kajtár, 2017). Measured exhausted airflow can be considered as a characteristic value (Abanto and Reggio, 2005), higher ventilation volume can lead to noise problem, which can be a subject of further investigations.

### 3.3 Thermal parameters

#### *3.3.1 Climatic conditions*

Outdoor air temperature and air humidity were measured continuously by a sensor sheltered from wind, rain and sunshine (Tab. 3)

Considering the relative short period of time, indoor environment was not influenced significantly by the changes in climatic conditions due to the thermal resistance and heat capacity of the building. Furthermore, air change rate of 0,8 1/h (Tab. 2) indicated limited possibility of air mixing.

#### *3.3.2 Indoor thermal environment*

Thermal comfort monitoring point was placed in front of the stove at a distance of 20cm and at a of 1,5m. This monitor point was determined according to the expected maximal exposure. It is not a question that indoor air temperature was increased proportionally to the convectional heat gain. Naturally, if the air temperature is rising and there is no additional moisture load, relative humidity will decrease. Combined impact of these can be expressed in terms of the enthalpy changes which are corresponding to the heat load (Fig. 2)

Using the hood, enthalpy change was decreased by 59% in case of burner "B" and by 12% in case of burner "H". Presumably, the enthalpy difference must be related to heat removed from the space. In case of the oven further investigation required (Fig. 3).

During the operation time of the oven harmful effect of forced convection was revealed. Without mechanical ventilation hot combustion products are released into the kitchen through the stovetop openings in the rear wall and (Fig. 1), moreover buoyancy-driven convection will evolve in front of the oven. By using the exhaust device forced convection was developed and additional heated airflow was delivered from the front zone of the oven into the occupied zone.

#### *3.3.3 Air velocity and turbulence intensity*

In accordance with the technical specifications of the microclimate analyser, air speed data were recorded in every minute during eight measurement periods. Turbulence intensity was calculated by

**Tab. 2** *Working conditions and ventilation parameters*

| Working conditions | | | |
|---|---|---|---|
| **Stove parameter** | **Large burner ("B")** | **Small burner ("H")** | **Oven** |
| **NGC [$m^3$h-1]** | 0.11 | 0.05 | 0.14 |
| **P [kW]** | 1.0 | 0.5 | 1.3 |
| **Ventilation parameters** | | | |
| **n [h-1]** | 0.8 | | |
| **$V_{ex}$ [$m^3$h-1]** | 155 | | |

**Tab. 3** *Outdoor climatic parameters*

| **Outdoor parameters** | | **$V_{ex}$=0$m^3$/h** | | | **$V_{ex}$=155$m^3$/h** | | |
|---|---|---|---|---|---|---|---|
| **"B"** | | **"H"** | **Oven** | **"B"** | **"H"** | **Oven** | |
| $t_{out}$[°C] | min | 17.0 | 11.4 | 15.9 | 17.0 | 9.4 | 15.9 |
| | max | 18.3 | 14.8 | 16.5 | 19.0 | 10.0 | 17.9 |
| | mean | 17.6 | 13.2 | 16.2 | 18.3 | 9.6 | 16.7 |
| | std.dev. | 0.63 | 1.28 | 0.25 | 0.53 | 0.19 | 0.73 |
| $\varphi_{out}$ [%] | min | 32.4 | 36.6 | 34.3 | 30.7 | 56.0 | 33.7 |
| | max | 34.7 | 47.4 | 35.6 | 34.4 | 59.3 | 36.0 |
| | mean | 33.5 | 41.6 | 34.9 | 32.5 | 57.7 | 35.0 |
| | std.dev. | 0.90 | 3.92 | 0.56 | 1.08 | 0.97 | 0.84 |

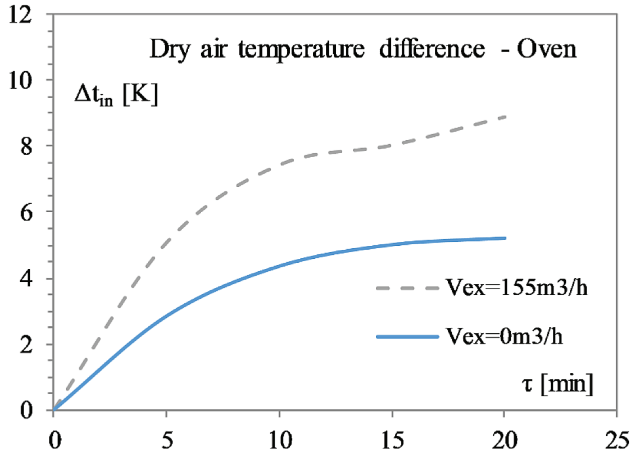


**Fig. 3** *Indoor air temperature difference due to oven (τ=20min)*

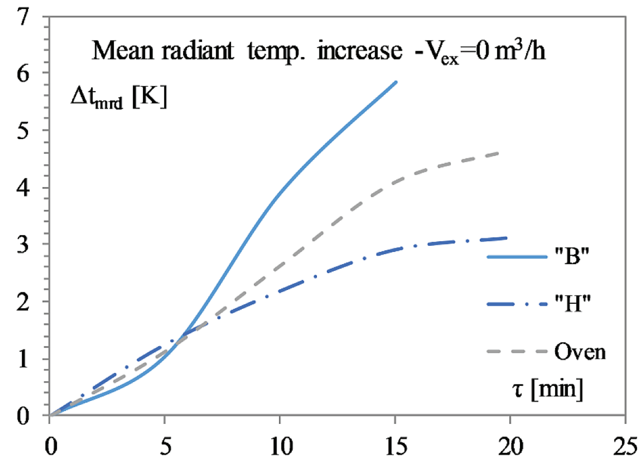


**Fig. 4.** *Mean radiant temperature difference ($V_{ex}$=0m³/h)*

summarising 180 individual data. To characterize the airflow pattern, average values of air speed data and the turbulence intensity could be applied (Tab. 4).

Considering the limitation ($v_{max}$=0,1m/s), average values indicate the determinant impact of buoyancy forces. In case of oven, data also shows the strong influence of the forced air movement, in accordance the air temperature measurements (Fig. 3). Using the hood, average air speed was measured 0.175 m/s at the monitor point, which is 150% higher than if no mechanical ventilation is used. However, due to the higher air temperature and the decreased turbulence intensity (from 123% to 25%), undesirable draught is not expected.

### *3.3.4 Mean radiant temperature*

To evaluate the radiant thermal parameters, black globe temperature was measured at the monitor point. With respect to the measured air velocity and the dry air temperature, mean radiant temperature can be calculated. Depending on the characteristic of the convection, Equation (1) or Equation (2) should be used. Mean radiant temperature is a fundamental parameter to evaluate the thermal environment in terms of radiant heat load (Fig. 4 and Fig. 5).

### *3.3.5 Predicted mean vote (PMV)*

In order to determine PMV, personal parameters had to be established: $M/A_{Du}=1{,}6met$, $\eta=0$, $I_{clo}=0{,}5clo$, $f_{clo}=1{,}1$. In addition to measured data (Fig. 2, Tab. 4) mean radiant temperature is also need to be calculated (Fig. 4-5). In addition further parameters (partial water pressure, convective heat transfer coefficient and the surface temperature of clothing) also required (Gunnarsen, 2003), which can be evaluated based on the measurement results (Tab. 5).

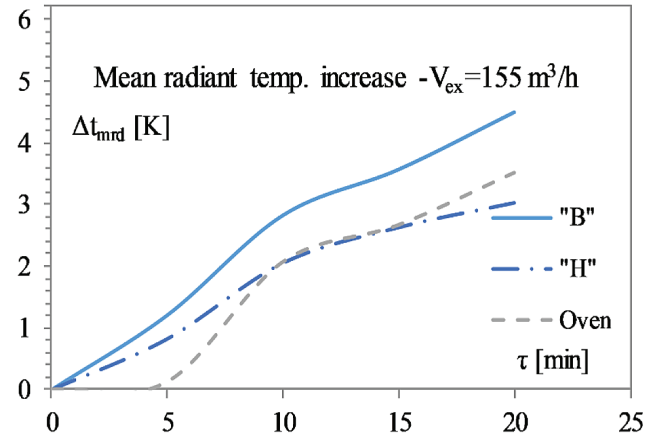


**Fig. 5** *Mean radiant temperature difference ($V_{ex}$=155m³/h)*

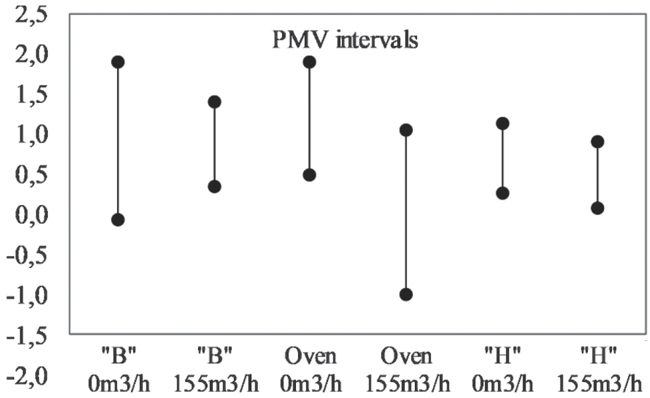


**Fig. 6** *Minimal and maximal values of PMV (20min)*

**Tab. 4** *Air velocity and turbulence intensity*

| Air speed monitoring "B" | | $V_{ex}$ = 0m³/h | | | $V_{ex}$ = 155m³/h | | |
|---|---|---|---|---|---|---|---|
| | | "H" | Oven | "B" | "H" | Oven | |
| v [ms-1] | min | 0.009 | 0.069 | 0.019 | 0.010 | 0.023 | 0.145 |
| | max | 0.018 | 0.075 | 0.155 | 0.022 | 0.039 | 0.215 |
| | mean | 0.012 | 0.071 | 0.070 | 0.016 | 0.031 | 0.175 |
| | std.dev. | 0.005 | 0.003 | 0.074 | 0.006 | 0.008 | 0.036 |
| Tu [-] | mean | 113 | 47 | 123 | 45 | 54 | 25 |

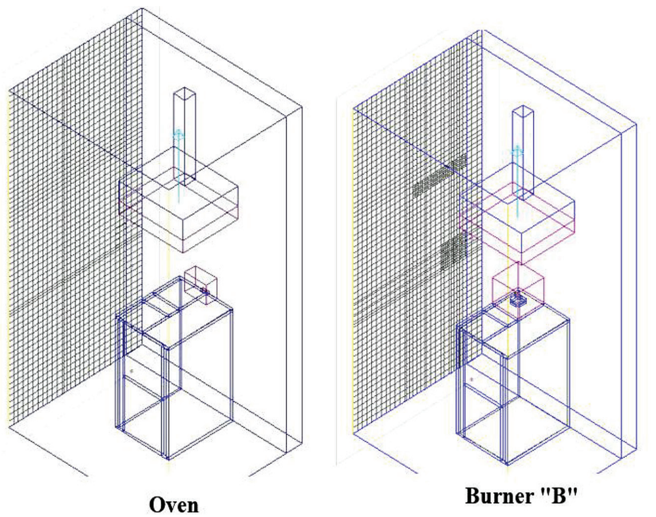


**Fig. 7** *Geometrical CFD models and grid systems*

Final results shown that calculated PMV index was inside the recommended range of -2 and +2 during the assessments (Fig. 6), therefore it can be applied to describe thermal environment around the residential gas stove.

## 4 CFD MODELLING

Three-dimensional CFD models of a gas stove (nominal power 5kW) were developed to simulate running conditions regarding burner "B", "H" and the oven. For the different scenarios similar initial and boundary conditions and the same turbulence model (k-ε) were used (Lim and Lee, 2008). Validation process has been carried out by evaluating the air temperature results in a horizontal plane at the cooking line. Gas stove primarily release heat into the space due the burning process, moreover additional radiation occurs due to the increasing surface temperature of the casing. To provide an adequate CFD model this effect cannot be neglected. During this part of the study stove casing was divided into ten geometrical parts. Surface temperature was measured as a function of time for each part. Thermal characteristic of the stove casing is presented on the simulation by approximating functions. In conclusion, CFD models were based on two general assumptions:

- change of surface temperature is simulated as a function of time, initial value is equal to the indoor air temperature (20°C);
- initial exhaust airflow of 0 $m^3/h$, and a constant value of 155 $m^3/h$ under running conditions.

Modelling grid is a main issue as regards accuracy of CFD models (Awbi, 1998). It is highly recommended to avoid any over dimensioning or using too poor grid system. In order to provide an adequate solution at the decreased level of computer capacity, different grid regions need to be established. These regions can be characterised by the maximal size of the cells. Basically it should be formed a fined grid in every part of space where gradients are expected to be high. Therefore, the base grid (50mm) were used in occupied zone, however it was subdivided into a secondary grid and localized grids. Secondary grid (20mm) was generally used in the surroundings and a fined gird (10mm) was generated in every location of particular interest (Fig. 7).

Considering results of assessments and calculation processes two basic CFD model was developed. Operation of oven and the large gas stove needed to simulate separately due to the different characteristic of heat load and pollutant emission. Scenarios are referred to free-running conditions, and validation was provided via air temperature results and residual analysis. CFD modelling revealed the airflow patterns and temperature distributions around the stove under different running conditions (Fig. 8 and Fig. 9). Based on validated models of the stove it is possible to evaluate complex scenarios in further studies. Vertical plane of visualisation is designated at the monitor point in accordance with the filed studies.

By the operation of the exhaust device CFD visualisations showed a more concentrated air flow, and limited mixing across the occupied zone. Results are in accordance with experiences of the field studies (Fig. 3 and Tab. 4).

## 5 CONCLUSIONS

In Hungarian households more than 2 million gas stoves are used, mostly without any controlled ventilation or makeup air. In the kitchens like that, occupants are directly exposed to the effects of the burning process. Therefore, it is essential to describe the stove as a source of heat and air pollutants. However, in design practice, there is a need to establish models and methods in order to provide adequate solutions for kitchen ventilation with acceptable level of accuracy and reliability. It is a fundamental issue, that air quality and thermal aspects should be evaluated by combined analysis.

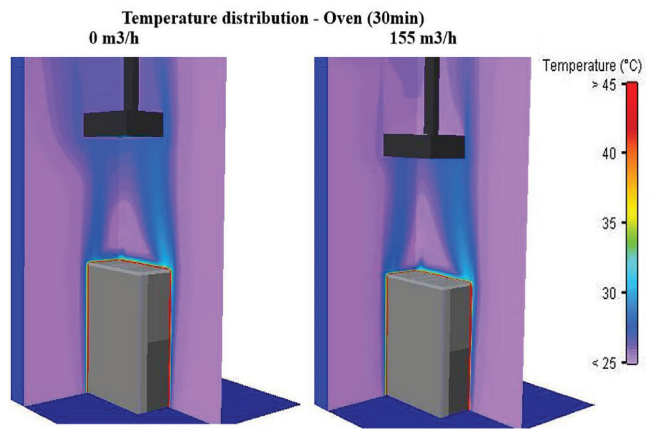


**Fig. 8** *Temperature distribution around the stove during the operation of oven*

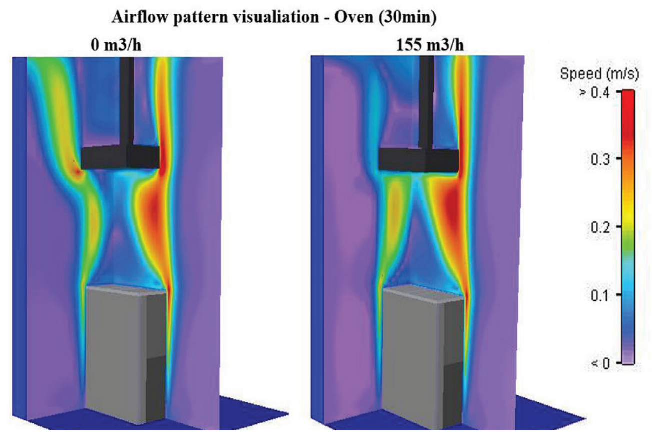


**Fig. 9** *Airflow pattern visualisation during the operation of oven*

During the operation of the stove heat load leads to increased indoor air temperature in the occupied zone, moreover surface temperature of the appliance is obviously rising. Therefore, convective and radiant heat gain released into the occupied zone. Shortcomings of the heat gains can be generally overcome by using a kitchen exhaust device. Operation of the oven can be considered as an exception in terms of thermal environment. Indoor air temperature rises at the monitoring point increased due to the air movements generated by the hood. During a 20 minutes period dry air temperature change of 5,6K was measured.

Development of forced convection has clearly demonstrated the airflow pattern analysis. During the operation of the oven average air velocity of 0,070 m/s raised to 0,175 m/s, turbulence intensity reduced from 123% to 25% due to the hood.

In case of stovetop burners air movements and thermal conditions are basically determined by buoyancy forces and flame radiation. By the operation of the exhaust device CFD visualisations showed a more concentrated air flow, and therefore, there is a limited mixing across the occupied zone.

Combined effect of the thermal parameters was described by the *PMV* index. During the investigations calculated values remain within the recommended range of -2 to +2. Therefore, research has shown that *PMV* index is suitable for predict the thermal comfort around the residential gas stoves. Harmful effect of draught is not expected. Suitability analysis of the other discomfort indexes and additional microclimate parameters can be recommended for further research.

In this study general issues and methods for evaluation have been determined, and a database has been established in terms of residential kitchen ventilation. The developed CFD models of the gas stove provide fundamental basic for complex simulation processes. Results will allow taking into consideration both of energy and indoor environmental aspects in the design practice.

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