Investigation of Air Supply Conditions in the Room of a B_{11} Type Gas Appliance

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Abstract: In Hungary, the prevalently used "B_{11}" type gas appliances equipped with atmospheric burner and they have a draught hood beyond the outlet of the appliance. For the appropriate adjustment of the gas boiler to the conditions of the building, computer-generated calculation results have to be obtained which can help the positioning of the appliance and the air supply outlets in the room in which the appliance is installed. Chimneys of natural draught are very sensitive to the changes in the amount of combustion air due to inside or outside ambient phenomena or forced effects.

For the purposes of modelling, a numerical simulation (CFD method) can be used. The aim of numeric modelling is to investigate velocity and temperature conditions around the flue gas outlet and in the room and, subsequently, to define design approaches and the requirements for different conditions. The conclusion of our investigations is that the air supply of the room can be accomplished with correctly sized air outlets and inlet elements.

Keywords: air supply, chimney, gas appliances, CFD method, numerical modelling, design requirements

1 Introduction

In Hungary, the heating of about 80% of family houses and blocks of flats is provided by gas boiler operated central heating systems. 9-10 million gas appliances are estimated to operate in Hungary, most of which are "B" type appliances (see CEN TR 1749 [6]). These appliances have an open combustion chamber; combustion air comes from the room in which the equipment operates, while flue gases leave through a chimney. The prevalently used B_{11} type gas appliances are equipped with atmospheric burner and they have a draught hood beyond the outlet of the appliance. They are connected to a chimney of natural draught.

In the last decade, quite a few carbon-monoxide intoxications occurred in Hungary due to the inadequate operation of gas appliances. These cases emphasized the importance of the correct air supply of appliances and the safe removal of the incipient flue gases. For the appropriate adjustment of the gas boiler to the conditions of the building and its heating system, computer-generated calculation results have to be obtained which can help position the appliance and planning the heating system.

For the examination of the operation of the gas appliance and the heated space, a mathematical model is needed to compute the air supply conditions of the room and the removal of flue gases, and to examine the non-steady-state condition of the room's air supply – appliance – chimney.

In Hungary, regulations have not been updated to follow the innovations of gas appliance designs and they do not include the recent drastic decrease of air-change rates due to air-tight windows and doors. This is the reason why we paid special attention to the modelling of B_{11} type gas appliances. With a theoretically established background, the placement, design and operation of the appliance all become easier.

The mathematical modelling and its results for the B_{11} type gas appliances are included in the 2005 WSEAS Conference Proceedings ([1]) and in ([2]).

The simultaneous or variance-based examination of several factors cannot be carried out analytically because of the large number of equations and their complexity. For the modelling of changes caused by the variations in the inside or outside ambient conditions, numerical investigation can be used. For the numerical investigation, we use the computational fluid dynamics (CFD) method.

The aim of numeric modelling is to examine the velocity and temperature conditions in the room, in the chimney and around the flue gas outlet. The results of the calculations can help in defining designing approaches and the requirements for different conditions.

Fundamentals of the modelling were introduced in [2], [3], while the CFD modelling and some results for the B_{11} type gas appliances were demonstrated in 2006, in a WSEAS Conference Proceedings ([5]).

2 Problem Formulation

2.1 The developed physical model

The developed physical model of B_{11} type gas appliance and its room is presented in Fig.1.

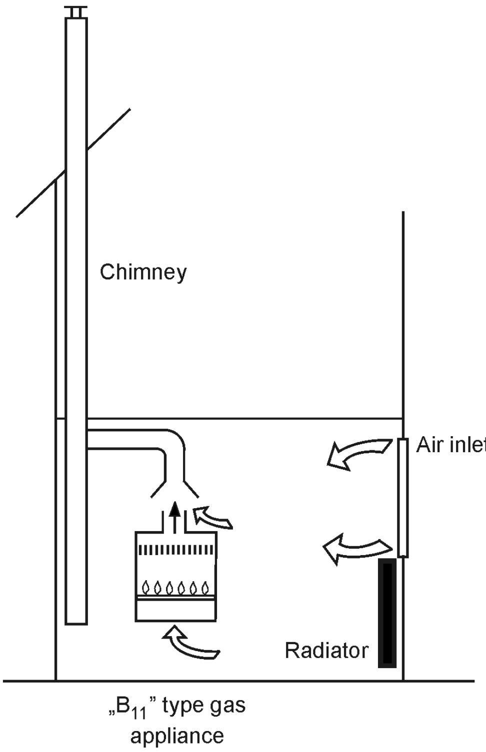

Fig.1. The physical model of a room and a B_{11} type gas appliance

The windows and doors of the room are air-tight structures made of wood or plastic, sealed with several layers of rubber sealing. Outside air can barely or cannot enter at all in the room through natural (gravitational) means. The air necessary for combustion is provided via air inlets that in the model were inserted in the window frame in different ways, under and above the window. Under the window a radiator is situated, controlled by a thermostatic radiator valve which adjusts the heat loss so that the desired room temperature is achieved.

2.2 The geometric model

The first step of CFD modelling is to create the geometry of the model presented in Fig.2.

For the modelling of the B_{11} type gas appliance a conventionally sized room is used. The volume of the room is 15 m^3, and its size is: 2 m (width), 2,5 m (length), 3 m (height).

The *U*-value of the external wall is 0.45 $W/m^2 \cdot K$, while the window has a *U*-value of 1.4 $W/m^2 \cdot K$.

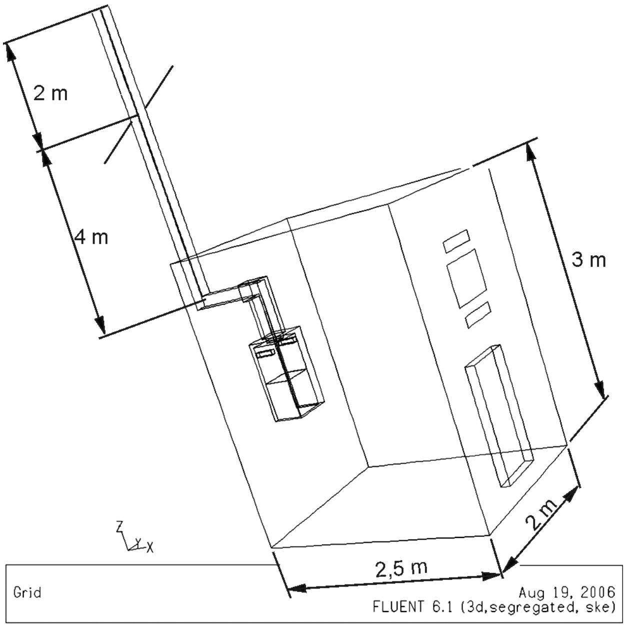

Fig.2. The geometric model of the room equipped with a B_{11} type gas appliance

The connecting flue pipe consists of: a 0,5 m long vertical section, a bend, and a 1 m long horizontal section. The pipe is made of aluminium and has a maximum absolute roughness of 1 mm.

The chimney is situated partly in the heated space and partly outside. The length of the section in the heated space is 4 m, while the section outdoors is 2 m long.

Nominal heat output of the gas appliances in the investigation are: 12 kW, 24 kW, 28 kW and 36 kW.

The outdoor air temperature is -15 °C, which is the best condition regarding the chimney but is the worst from the room's comfort point of view.

2.3 The differential equations for the numeric model

The numeric model, based on the geometric model, was developed by adding the principal initial and boundary conditions.

The air movements of closed areas are described by the differential equations of continuity and *Navier-Stokes*. The thermo balance of the areas is expressed by the equation of energy; its distribution of concentration is described by the differential equation of material balance.

Assuming an incompressible agent the listed equations are formed as follows:

Continuity:

$$\text{div}\,(\rho \cdot u_i) = 0 \,, \quad (1)$$

where ρ is the air density and u_i are the air velocity components in *x, y, z* direction.

Equation of movement:

$$\frac{\partial}{\partial x_i}(\rho \cdot u_i \cdot u_j) = \frac{\partial}{\partial x_i}\left((\mu + \mu_t)\left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i}\right)\right) - \\ - \frac{\partial}{\partial x_j}\left(p + \frac{2}{3} \cdot \rho \cdot k \cdot \delta_{ij}\right) + g_i(\rho_x - \rho) \,, \quad (2)$$

where μ is the viscosity, and μ_t is the turbulent viscosity, *p* is the pressure, *k* is the kinetic energy, and δ_{ij} is the *Kronecker* symbol.

Equation of energy:

$$\frac{\partial}{\partial x_i}(\rho \cdot u_i \cdot h) = \frac{\partial}{\partial x_i}\left\{\left(\frac{\mu}{\sigma} + \frac{\mu_t}{\sigma_t}\right)\frac{\partial h}{\partial x_i}\right\} + Q \,, \quad (3)$$

where *h* is the enthalpy, *Q* is the quantity of heat per volume, σ_t is a factor, depends on *Prandtl*- and *Schmidt*-numbers.

As we are talking about turbulent airconduction, also the proportion of the kinetic energy and the dissipation (*k-ε*) of the airflow has to be determined. Resulting from a system of equations, this is the mathematical model of closed spaces.

The *k-ε* transport equation is created from *Navier-Stokes*-equation on the condition that the turbulence effect dominates over the whole flow field.

The turbulent viscosity after the dimension analysis is:

$$\nu_t = \frac{\mu_t}{\rho} = C_\mu \frac{k^2}{\varepsilon} \,. \quad (4)$$

With the *k-ε turbulence model* it becomes possible to manage turbulent effects as a transport equation. It is an important advantage that numerical methods can handle transport equations and thus, besides the known transport (diffusion) processes, turbulence can be modelled as well.

The differential equations used for the calculations were demonstrated in the Proceedings of WSEAS Conference in Crete, 2006 [5].

3 Problem Solution

3.1 Calculation of supply-air volume flow rate

Before starting the numerical modelling, general designing tools were used to calculate the required supply airflow rate, and the chimney was sized according to the model described in [5].

According to the regulations, in the case of B_{11} type appliances sufficient air has to be supplied for the combustion process and to compensate for the air leaving through the draught hood from the room.

For the calculations we used a gas type from the 2H gas family, which has the following components:

Methane (CH_4)	0,9700	m^3/m^3
Ethane (C_2H_6)	0,0110	m^3/m^3
Propane (C_3H_8)	0,0030	m^3/m^3
Butane (C_4H_{10})	0,0010	m^3/m^3
Carbon dioxide (CO_2)	0,0050	m^3/m^3
Nitrogen (N_2)	0,0100	m^3/m^3

The calculation results are presented in Table 1.

Table 1. Results of supply airflow rate calculations based on chimney dimensioning

Nominal heat output of the appliance in kW		12	24	28	36
Efficiency of the gas appliance in %		90	90	90	90
Volume flow rate of the gas consumption in m^3/h		1,41	2,83	3,3	4,24
Mass flow of the flue gas in kg/h		24,1	48,2	56,2	72,3
Mass flow of the combustion air in kg/h		21,69	43,38	50,61	65,07
Mass flow leaving through the draught hood in kg/h, when the outside air temperature in °C is	–15	56,21	73,44	69,54	96,12
	+15	30,82	41,27	38,52	54,76
	+32	21,56	27,8	25,2	36,67
Combustion air in m^3/h + volume flow rate through the draught hood, m^3/h (the density of air is in 1,19 kg/m^3), when the outside temperature is	–15	65,46	98,17	100,9	135,4
	+15	44,13	71,13	74,90	100,7
	+32	36,34	59,82	63,71	85,50

Table 2. Results of the modelling for winter

Nominal heat output of the appliance, kW	12	24	28	36
Mass flow of the flue gas, kg/h	22.3	49.3	75.6	108
Temperature of the flue gas leaving the appliance, °C	101	121	122	118
Temperature of the flue gas in the beginning of the chimney's indoor section, °C	88	104	112	108
Temperature of the flue gas in the beginning of the chim-ney's outdoor section, °C	70	75	89	82
Temperature of the flue gas at the chimney outlet, °C	60	62	84	71
Supply air flow rate, m^3/h	23	45	53	73.3
Pressure difference by the air-inlet, Pa	3.9	4.4	4.5	4.5
Average air temperature of the room, °C	24.1	23.6	23.5	23.6
Average air velocity in the room, m/s	<0.1	<0.1	<0.1	<0.1

3.2 The results of CFD modelling

The calculation results of the CFD simulation (carried out with the programme FLUENT) for winter are presented in Table 2.

Fig.3 and 4 show the temperature distribution and the temperature flow lines in the room in the case of gas equipment with a nominal heat flow rate of 24 kW.

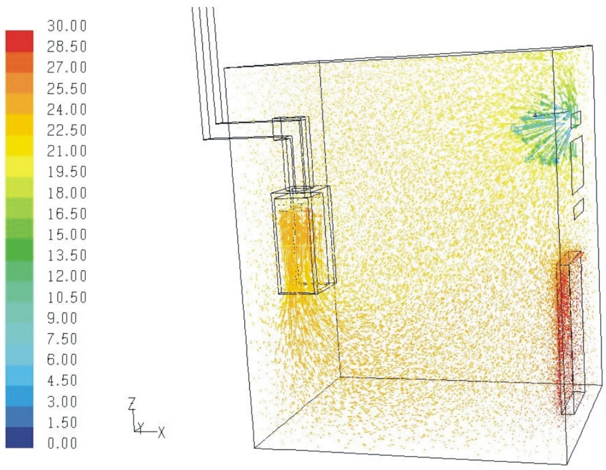

Fig.3. Temperature distribution in the room

Fig.3 indicates that the air coming through the air inlet is heats up quickly and the temperature in the occupied zone is between 20 and 22 °C. The supply air flow rate is about 45 m^3/h (see Table 2), which means that the air change rate is 3 1/h in the room.

Fig.5 (next page) shows the velocity distribution in the room. It can be seen that in the occupied zone velocities are far below 0.1 m/s, despite of the high air change.

Based on the data indicated in Table 2, PMV (Predicted Mean Vote) values were also calculated for the room. Results show that in case of 0.8 clo, which is considered a light clothing ensemble, and for the activity level of 1 met, PMV values fall between -0.3…+0.1. This region stands for a slightly cool and comfortable thermal comfort level.

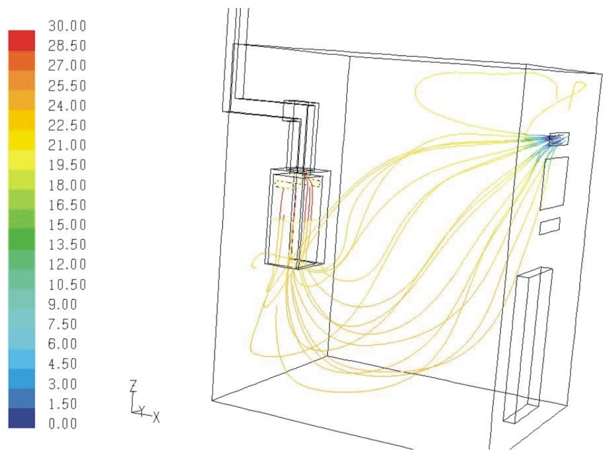

Fig.4. Temperature flow lines in the room

4 Conclusion

Based on the data introduced in Table 2 and on the distributions indicated in the Figures 3 – 5 the following statements can be made:

– the natural draught through the chimney ensures the entry of supply air through the air inlets when the outdoor temperature is –15 °C,

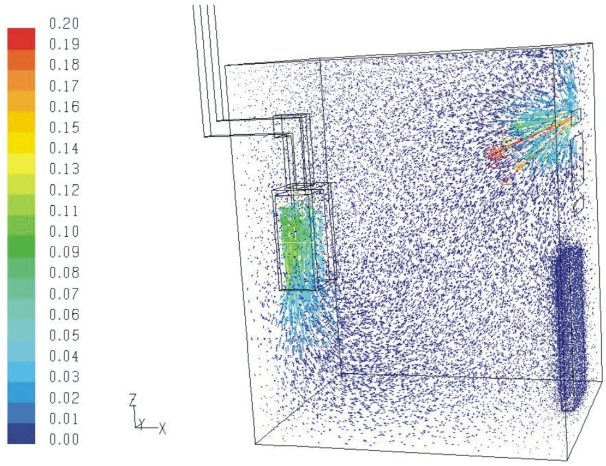

Fig.5. Velocity distribution in the room

- our other investigations show that the entry of supply air is ensured in the case of higher outdoor temperatures,
- the average air temperature in the occupied zone is adequate (23-24 °C) and uniform,
- the relative air velocity in the occupied zone is far below 0.1 m/s,
- the air velocity in the occupied zone has a uniform distribution,
- 30 cm from the air-inlet, the entering fresh air has a temperature that hardly differs from the average air temperature of the room,
- it is reasonable to place a heat-transfer appliance (accurately sized radiator) under the air-inlet so that the cool zone of the air-inlet can be reduced,
- when using the described settings, the comfort conditions are not unacceptable despite the high air change rate and the very cold air entering the space,
- results of our other investigations show that the average temperature in the occupied zone is unaffected whether the air inlet is placed underneath or above the window.

Based on our investigations, we conclude that the air supply of the room can be accomplished with correctly sized air outlets and inlet elements. In order to satisfy comfort requirements, a radiator has to be placed under the air inlet.

Summarising the considerations regarding the air supply of a room with a gas appliance installed in it:

- determination of supply airflow rate from the combustion air of the appliance and – if there is any – from the air leaving through the draught hood,
- ensuring the necessary pressure difference by the natural draught of the chimney so that the supply air can enter the space,
- based on the two above, sizing and selection of the air inlet,
- thought-out placement of the air inlet based on the results of the introduced numeric simulation.

Our future goal is to verify and if needed correct the results from the computer modelling by carrying out measurements.

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