

INFLUENCE OF COMBUSTION INSTABILITIES ON THE HEATER APPLIANCE WITH ATMOSPHERIC GAS BURNER AND THEIR ELIMINATION BY CROSS FLOW OF AIR

by

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This paper presents results of experimental investigation on elimination of combustion oscillations caused by new low pollution burner which was integrated in a gas heater. The method is known as a passive method, based on introducing the air in the combustion chamber. Since efficiency of the passive method is highly dependent of the way the air is injected, several different methods were investigated and compared. The paper also presents effects on pollution characteristics during its operation in the unstable regime.

Keywords: atmospheric burner, combustion instabilities, CO, NO_x

Introduction

The necessity of decreasing air pollution by domestic gas appliance is not only the matter of legislative or standard; it is rather the matter of engineering conscience and prescience where pollution leads. From that point of view it is normal that appliance producers intend to put new, low pollution burners in existing appliance. The problem arises when new burner working near or over stoichiometric ratio falls (in conjunction with its surrounding) in acoustic instabilities. Manufacturer's intention is always to keep the existing production tools, which means that the problem has to be solved within the same volume and shape. Because of that this paper presents measurements of some heater characteristics when its burner operates in the domain of combustion instabilities, and the way of their elimination.

Combustion instabilities are problem which starts to press engineers from 1940 [1-4], ever since they have begun to work with solid rocket motors and liquid rocket engines. In those machines everything has to have minimal dimensions and to be maximum loaded. For that reason many elements became coupled. There are pressure couplings, velocity coupling, vortex shedding, acoustic interactions, structure interaction, convective waves, *etc.* Even today engineers have a problem to surely design mechanical device in which occur combustion without instabilities. If device is realized and have problems with combustion instabilities, their elimination is some time not very easy, and some time almost impossible [5].

There are two types of combustion instabilities controls: active and passive. Active control requires integration of additional accessories to the burner device (piezo actuators, pressure sensors, valves, *etc.*) which creates negative feedback loop [2]. This method for this type

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of heater appliance (simple and fully independent of electricity) is unacceptable. The second, passive, method is based on implementation of baffles, acoustic liners, Helmholtz resonators, and injection system dynamic. Connection between acoustic and combustion interaction in combustor is studied for a long time and it is still in a focus [6] due its complexity. In a parallel there is lot of work concerned to passive suppression of thermo acoustic oscillations. In [7] was investigated feedback free fluidic oscillator which has a well performance in the particular working regime. In [8, 9] are given engineering tools for solving the problem by introducing the air into the combustion chamber through holes. Tools are commercial although without guaranty that the problem will be slowed. In [10] whole problem is analyzed from acoustic point, derived equation for chamber acoustic impedance which permits combustion instabilities. In [11] is investigated the method of passive control of combustion instabilities for premixed flames anchored on perforated plate. Investigation is connected with mixer chamber acoustic.

Investigation and solution presented in this paper raised from the following motivations. First there is no changing of appliance and burner geometry and dimensions. Next the appliance has to work in whole domain (from 2 to 10 kW) instead in particular working point. For those reasons investigation started with impingement of air mass flow. That means that method is by definition passive, but in reality impingement air mass flow depends of working regime, naturally by the combustion and ambient conditions, without any artificial feedback and accessories.

Existing heater appliance working with old and new burner

Nominal thermal power of domestic heater appliance is 10 kW. Originally it works with atmospheric gas burner in fuel rich regime with mixture of propane-butane as a fuel. In order to decrease pollution emission of NO_x and CO (it is well known that generally doing that by simple methods decreasing of NO_x is bonded with increasing of CO), instead of the old burner in appliance a new one was integrated, which is flexible on fuel type (propane-butane, methane, methane with 20% of CO_2) and operates in fuel lean domain.

Heater appliance cross-section is given on fig. 1. Geometry of integrated new burner is given in fig. 2. Gaseous fuel m_{fuel} is injected by nozzle (fig. 2), which increased its velocity. The nozzle is in the throat of Venturi tube, and because of that the surrounding air is sucked.

During their coaxial flow they are mixing, and on the outlet of the Venturi tube there exits fully fuel lean mixture m_{mix} which occupies burner (1), and flows through flame holes in the combustor (3). After complete combustion, hot flue gases are mixed with secondary air $m_{\text{air}2}$, which decreases its temperature to the level permitted by combustor and heat exchanger structures. Then gas flow through heat exchanger and flue exit (4). Cold flue gas m_{flue} exits the heater. That kind of behavior is present when heater works with old burner and only in the parts of minimum and maximum power with new burner. Instead of smooth working in regimes from 4.3-9.5 kW (depending of fuel type) there are strong combustion instabilities. Existence of the unstable working regime which is bounded by stable parts may be explained as consequence of burner mixing method. Value of air coefficient λ of the naturally aerated atmospheric gas burner depends on its thermal power (fuel mass flow).

When the heater enters unstable working regime it starts to produce loud noise presented on fig. 3. In this case the noise

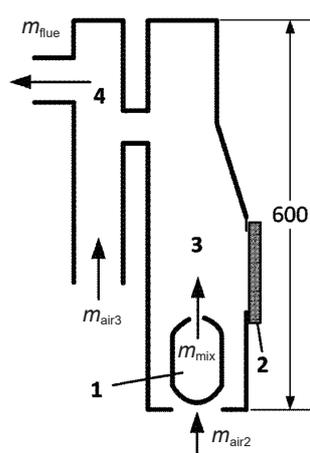


Figure 1. Heater with gas burner (1), window (2), combustion chamber (3), and flue exit (4)

is produced by high frequency combustion thermo-acoustic instabilities which have to be suppressed within mentioned constrains. From engineering point there are no reliable designing tools which may be used and surely solve this kind of problem. Because of that presented solution is experimental, based on theory and experience. First step is to get relevant element characteristics and to make clear insight to their internal connections.

Burner structural and appliance acoustic characteristics

Using the commercial CATIA software, FEM modal analysis of burner has been done. Obtained results are presented in figs. 4 and 5. This analysis was made for real 3-D burner shape, real material, sheet thickness, and type of supports (fig. 2). The forms of the low burner modal frequencies (only bending and torsion) have shapes which may not interact with mixture supply because its internal volume is constant during oscillation. Higher modes of oscillation have symmetrical forms of a deformation. That changes its internal volume and disturbs mixture supply. That interact with acoustic waves occurring in surrounding gas in which burner is submerged (fig. 1). In fig. 4 one may see the coupling between burners 16 oscillation mode and recorded and Furrier analyzed noise. The dominant frequency of highest normalized amplitude is 810 Hz. As the burner flame pattern (fig. 5) changes its shape it changes small vortex

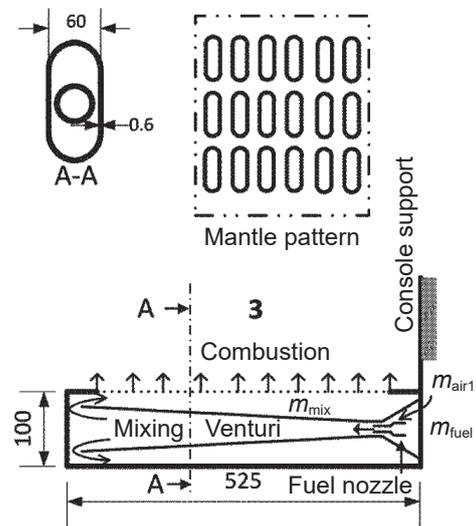


Figure 2. Premixed lean atm. gas burner and its flame openings (mantle pattern)

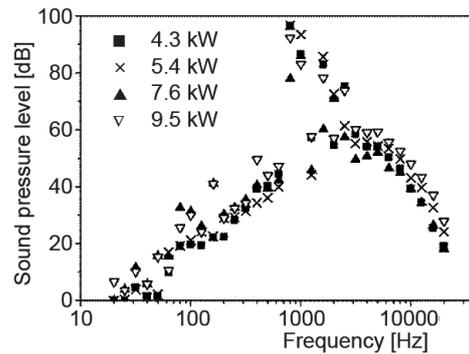


Figure 3. Appliance generation of noise

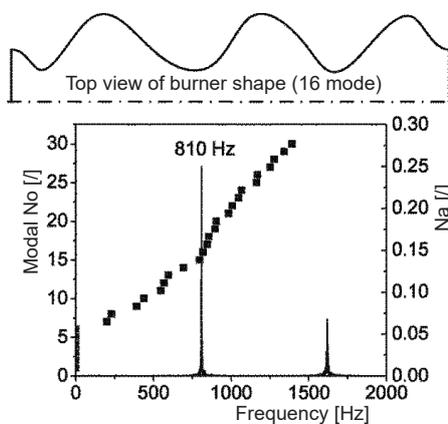


Figure 4. Burner modal and main combustion instability frequencies

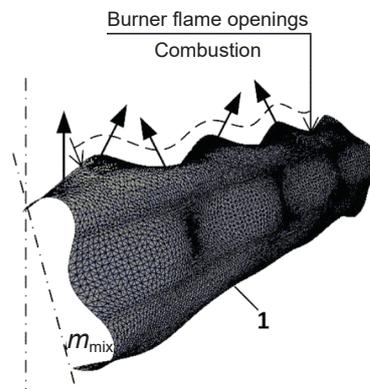


Figure 5. 16 modal burner oscillation 810 Hz

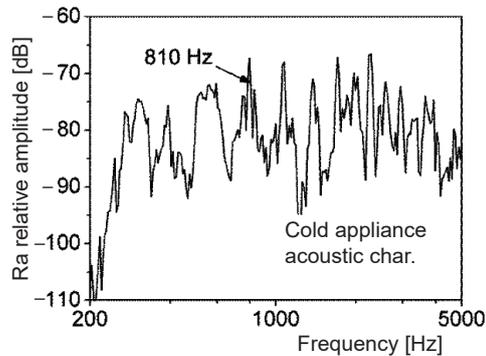


Figure 6. Appliance acoustic characteristics

spectrum of frequencies for which appliance has no capabilities for their attenuation. This conclusion is approved with generated noise presented in figs. 3 and 4 and dominant tone of 810 Hz.

Strategy of heuristic experimental research

Even heuristic research is based on the trial and error approach it cannot be provided without its directing to the methods which are reasonable and within existent constrains. In

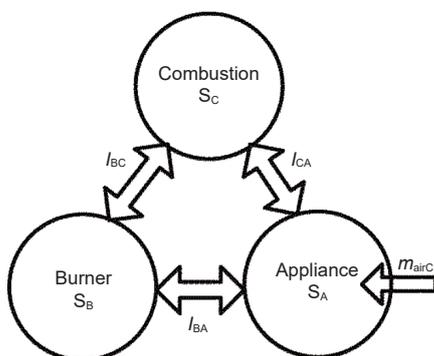


Figure 7. Interrelations between elements

fig. 7 is presented the relation between burner, combustion, and appliance. Every element state (S_A , S_B , and S_C) has influence on each other thought vectors (I_{BC} , I_{AC} , and I_{BA}).

First, experimentally proved solution is gotten by suppressing the burner oscillations in its zone of largest deformation for its 16 mode. This was done by adding mechanical supports. This solution is not acceptable because there is necessity of major burner structural changes which are disallowed by its producer. Nevertheless, this solution confirms that S_A contains stabile acoustic wave structure state which is created and supported by S_B and S_C . This state cannot be destroyed by parallel air flow, because that flow already exists as m_{air2} (fig. 1) during combustion instabilities existence. That means that there is only one experimental research path based on impinge of air cross flow m_{airC} . This method, with its variant is presented in fig. 8.

From fig. 8 one can see that there are two possible ways for introduction air cross flow m_{airC} into the combustion chamber 3. First is through horizontal slit (case 1). Second way is through multiple vertical slits (case 2). Both methods uses appliance window opening 2. Both experiments can be realized with open and closed secondary opening (m_{air2}), and for defined fuel and power range (tab. 1).

Targeted measurements were done during experimental research of the appliance leading from combustion instabilities into the stabile working regimes. Since it is very difficult, sometimes even impossible, to do experiments (with ordinary equipment) in which pressure of few Pascal's may have great importance on appliance behavior; measurements of flue flow rates m_{flue} and m_{flueC} are in some way indirect (fig. 8).

zone between neighboring flames. It disarranges flame to flame heat and mass (radicals) transfer and it works in radiate mode.

Measurements of appliance acoustic characteristics are presented in fig. 6. The characteristics are presented as relative amplitude; relative to the measured SPL (sound pressure level) of speaker in the same disposition but without obstacles (appliance). The measurement is for cold state (room temperature). In reality the appliance is filed with hot gas which temperature and velocity fields are very complex. Because of that this characteristics has informative value which expresses that there is a very broad continual

Conduction of experiments and measure of suppression efficiency

Before every experimental research all necessary equipment is prepared in accordance to fig. 8. Heater power is adjusted by valve (5). Gas analyzer measures the composition of flue gases and its temperature T_{flue} . On those values is calculated mass flow rate of flue gas m_{flue} .

Difference of overall mass flow rate Δm_{flueC} is determined in two steps done at the same power. In the first step the silted window (2) is closed, and consequently cross flow does not exist. The mass flow rate m_{flue} is calculated. If, at that power the combustion instabilities exist, then the segments of window are moved apart increasing the A_{airC} and m_{airC} until the noise (combustion oscillations) are fully suppressed. Then m_{flueC} is measured (in the same way as m_{flue}). The difference is:

$$\Delta m_{flueC} = m_{flueC} - m_{flue} \tag{1}$$

Calculation of flue mass flow rates of m_{flue} and m_{flueC} are on the bases of measured flue concentration of O_2 and fuel mass flow rate. Both values depend of heater combustion efficiency, which depends on: thermal power set up, fuel type, and presence of combustion instabilities. Since those efficiencies are not known, and because they are close to 1, all calculations are provided for same combustion efficiency which is 1. This is the best reasonable assumption, affirmed with maximal value of flue temperature difference of 7% when heater operates with and without combustion instabilities.

The mass flow rates m_{air1} (fig. 8 air entering Venturi tube 1₁), m_{air2} (air in secondary opening) and m_{airC} (cross flow air) are not measured because all of them enter the zone of combustion chamber 3 with completely unknown internal redistribution. Because of that, Δm_{flueC} is set as a representative of mass flow rate which has to be reorganized in the combustion chamber for suppression of the acoustic oscillations.

To make the experimental research comparable it must be defined some kind of rating. In this case it is the efficiency. Efficiency depends of wideness of the stable working regime, and the amount of Δm_{flueC} . If the whole working range (2-10 kW) is not stable (including oscillation triggering with 100 dB sound pulse next to the opening 2), the method is not effective. If the whole working range is stable, higher Δm_{flueC} defines lower efficiency. Instead Δm_{flueC} for efficiency representation can be used its relative percentage value $RP\Delta m_{flueC}$.

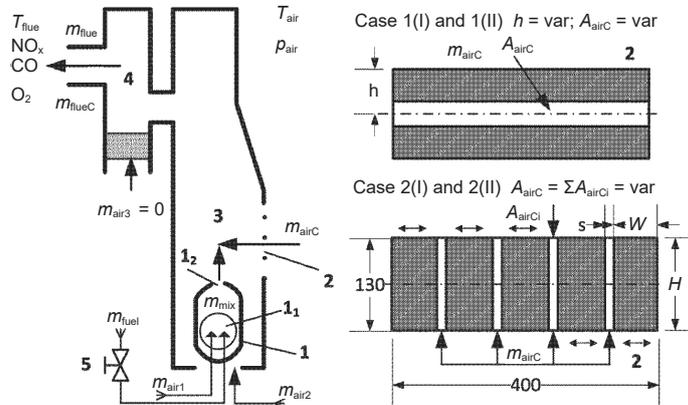


Figure 8. Equipment for planned research

Table 1. Schedule of experimental research with propane-butane as fuel

No.	Slits type/number	A_{air2} [open/closed]	A_{airC} [m ²]	Power range [kW]	h
1(I)	horizontal/1	closed	var	2-10	variable
1(II)	horizontal/1	open	var	2-10	variable
2(I)	vertical/variable	closed	var	2-10	—
2(II)	vertical/variable	open	var	2-10	—

Experimental research case 2(I)

In this section is presented the most efficient method in details. It includes appliance characteristics when it operates with oscillations and with suppressed oscillations.

Experimental research case 2(I) was provided with closed secondary opening. Segment high $H = 130$ mm and width $W = 50$ mm. In the first stage, all 7 segments (2) presented in fig. 9, occupies left side of the window opening. Whole area A_{airC} is on the right side. The burner power is on the minimum, and the burner is ignited. The power is raising slowly and on the 4.3 kW the appliance start to produce laude noise (fig. 3). That sound is captured and presented on fig. 9. On the beginning sound amplitude growth has exponential character, and after that the amplitude growth decreases, until reach the constant value. From fig. 9 that time is about 5 s, but it depends of the start type (cold or worm), and of the fuel type.

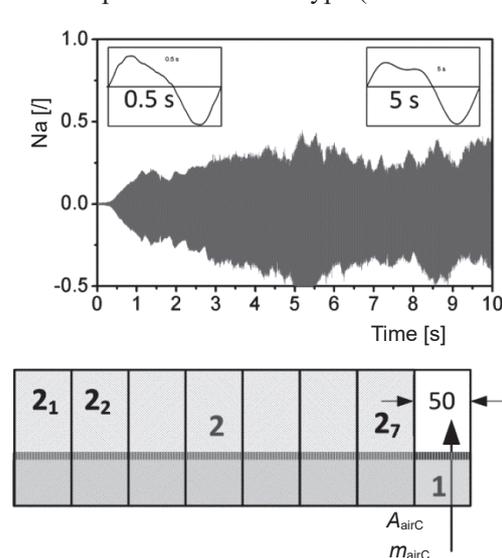


Figure 9. Development of combustion instabilities

which gives opportunity for creating the stabile acoustic oscillation geometry inside appliance. Whole amount of air cross flow are passing through right opening (fig. 9), without ability to suppress the oscillations.

Those oscillations (combustion instabilities) affects appliance characteristic. Some of them are measured and presented on figs. 10-13. For simplicity they are presented as relative percent of changed value. Figure 10 presents influence on NO_x production which is slightly decreased. Even the mechanism is not well understood some researches are working on that trying to decrease ordinary NO_x pollution using external acoustic oscillations [13].

Thermo acoustic oscillations increase CO emission (fig. 11) for more than 2000%, while in same time has a minor influence on flue temperature (fig. 12). Almost all mentioned elements of appliance behavior during oscillatory operating mode have to be used as is. In fig. 13 are presented two cold appliance acoustic characteristics; with entirely closed window $A_{\text{airC}} = 0$, and with $A_{\text{airC}} > 0$ as in fig. 9. It appears that the characteristics are practical same (as expected), and aloud to interpret suppression of combustion oscillations as a real consequence of m_{airC} .

After measured flue emissions of CO, NO_x , and temperature the window segments 2_1 to 2_7 are moved apart right side (fig. 14). Between them are formed 8 slits $s = 6$ mm. The air

Sound wave form, after 0.5 s is presented on fig. 9. This shape [12] is characteristic for rising thermo acoustic oscillations in which the heat is supplied into the high pressure part of wave. Added heat deforms sinusoidal wave form, increase the duration of positive pressure and the whole system starts to be nonlinear. Similar is within 5 s. The wave amplitude is almost constant, and the part of the heat added to the wave is converted into the mechanical acoustic power. In this case 500 burner flame ports are sources of acoustic power which operates synchronously. Simultaneously the burner structure (1 on fig. 4, 5, and 8) is in state of oscillation (16. modal mode) which changes mixture flow. Instead of smooth mixture flow from flame openings the flow is with oscillations synchronized with overall acoustic oscillations (810 Hz). Burner works in radiate regime. The flame is anchored for its surface

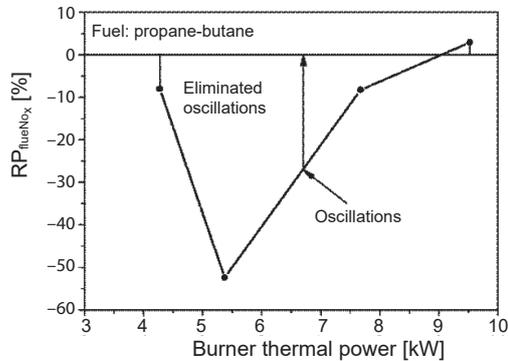


Figure 10. Influence on NOx emission

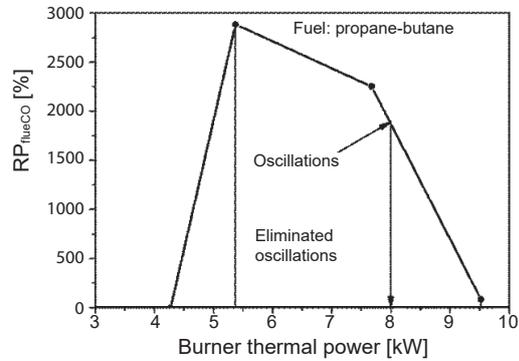


Figure 11. Influence on CO emission

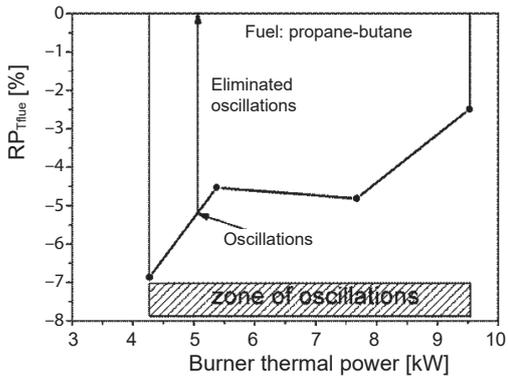


Figure 12. Influence on flue gas temperature

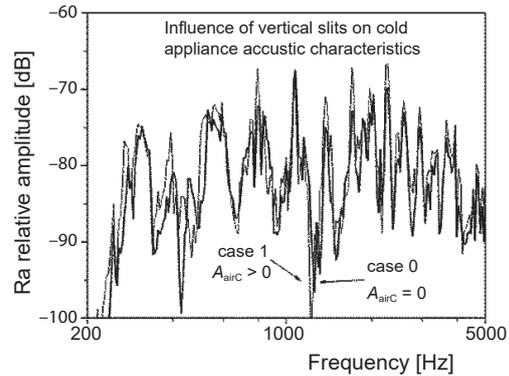


Figure 13. Appliance acoustic characteristics

mass cross flow m_{airC} is divided and redistributed in combustion chamber. The combustion instabilities are suppression. The normalized noise amplitude falls from $Na = 1$ to $Na = 0.01$. The CO emission is decreased for more than 2000%, and the burner works normal.

The influence of air mass cross flow can be better understood with aid of CFD simulation presented in fig. 15. Because the air enters into the combustion chamber through the narrow slit the air flow forms intensive vortices. They are the part of recirculation zone, which cannot be planar since the presence of combustion chamber gas mainstream. In the slits zone starts formation of helicoidally vortices. They penetrate to the central zone of combustion chamber. In this way inside the chamber exists intensive chaotic convective transport. The convection includes transport of mass, energy, species, but in this case most important is acoustic waves [14]. For that reason inside the appliance can no more be sustained stable wave acoustic structure which interacts with burner and combustion. From the sound spectrogram (fig. 16) one can straightforwardly distinguish zone with thermo acoustic instabilities (a) from the zone where they are suppressed by this method (b). Region (b) is practically white noise, which is random as vortices produced by slits.

Cumulative results of the experimental research

In this research were investigated four different methods. Results are presented in fig. 17 and tab. 2. The simplest solution of using the holes [8, 9] for air mass cross flow intro-

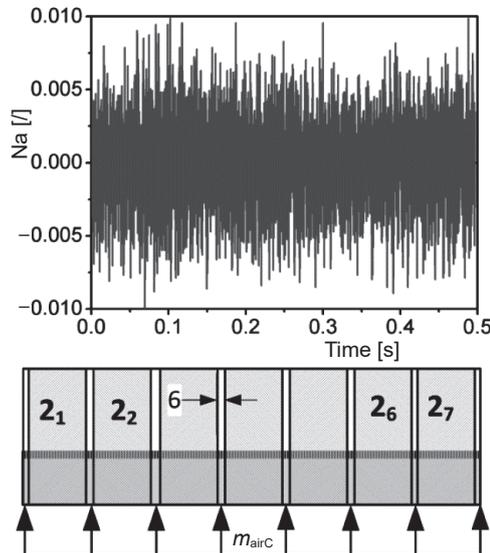


Figure 14. Appliance noise with eliminated combustion oscillations

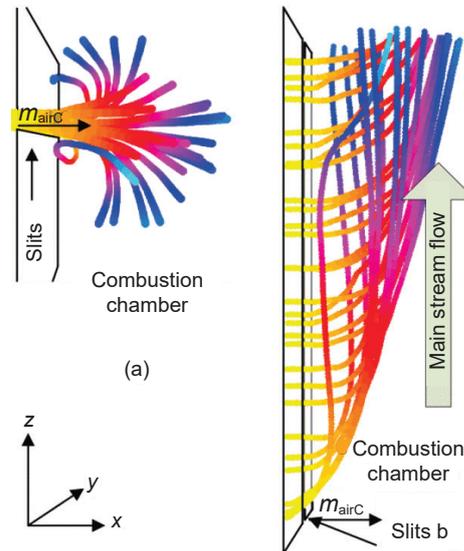


Figure 15. CFD simulation of flow through slits into the combustion chamber

duction was tried, and after few unsuccessful attempts discarded. Because of the burner shape (long cylinder), practically there are infinite possibilities of their position, diameter, and number.

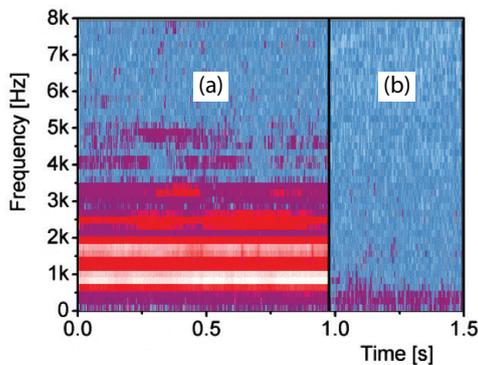


Figure 16. Sound spectrogram

Method 1(I) is based on injection of m_{airC} through horizontal opening, which length is the same as the length of the window, fig. 8. The middle line of opening was of different heights and different A_{airC} values. Openings for secondary air A_{air2} are closed. Efficiency is very low (tab. 2) because of extremely high Δm_{flueC} . This method is discarded.

Method 1(II) is the same as 1(I), except for the openings for secondary air A_{air2} are fully open. Efficiency is very low and the method has been discarded.

Method 2(I) applies injection of m_{airC} through vertical slits, presented in figs. 8, 9, and 14. There are 8 slits, and the procedure has already been explained in the section *Experimental research case 2(I)*. In this case the openings for secondary air A_{air2} are closed. The efficiency is the highest (fig. 17 and tab. 2).

Method 2(II) differs from 2(I) in the openings for secondary air which are open A_{air2} . Irrespective to the fact that acoustic instabilities in this method start on higher power (5.6 kW) it has low efficiency (fig. 17 and tab. 2).

Positive results from research are clearly presented in tab. 2. From there it can be seen that suppression of thermo acoustic oscillations (in this particular case) by homogenous horizontal sheet of cross flow air is almost ineffective compared with segmented and vertically injected.

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Experiment set-up

Scheme of experimental set-up for research of appliance (heater) combustion instabilities suppression by air mas cross flow is presented in fig. 18. The fuel (propane-butane) is in the pressurized cylinder (6), partially in liquid and partially in gaseous phase. Measurement of fuel pressure p_{fuel} (2-5 bar) and temperature T_{fuel} (293-303 K) are necessary for determination of gaseous phase composition and the lower fuel heating value accordingly. Heater power is adjusted by valve (5) and measuring of flow rate by beck pressure compensated Rotameter (7). Rotameter is Dwyer Instruments VA-K. The appliance air valve is closed ($m_{air3} = 0$) during all provided experiments. Gas analyzer (TESTO 350-XL with CO low ad NO low cells) measures the composition of flue gases and its temperature T_{flue} . The known flue and fuel composition determines mass flow rate of flue gas m_{flue} [15]. Accuracy ε of measured values is calculated as highest probability random propagation error [16]. The values are: $\varepsilon_{CO} = \pm 5\%$, $\varepsilon_{NO} = \pm 5\%$, $\varepsilon_{O_2} = \pm 5\%$, $\varepsilon_T = \pm 5\%$, $\varepsilon_P = \pm 3.5\%$, and $\varepsilon_{\Delta m_{flueC}} = \pm 8.8\%$.

Sound and acoustic measurements are presented in fig. 19(a). Source of sound is the loudspeaker (8), driven by wave generator (11), which is PC with sound card and the Spectral-ab software. Wave generator generates pink noise in the continuous range from 0.2 to 10 kHz.

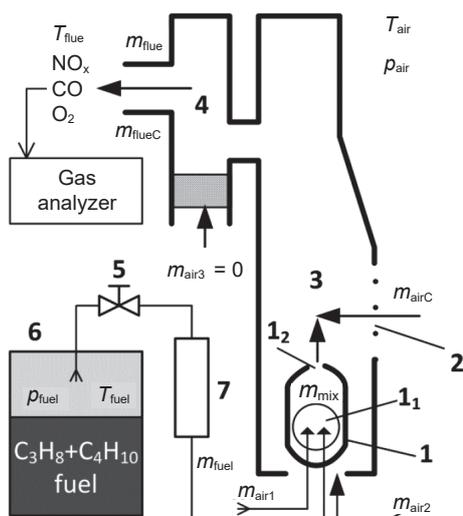


Figure 18. Experimental set-up

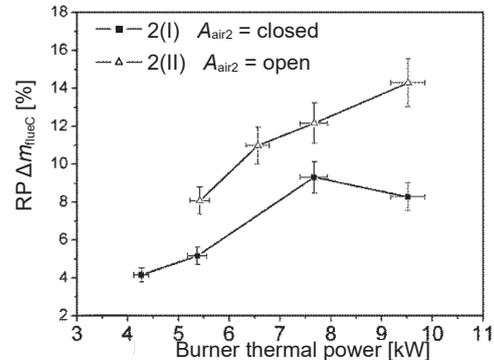


Figure 17. Efficiency of methods 2(I) and 2(II)

Table 2. Characteristics of implemented methods (fig. 8)

No.	Slits type/number	m_{air2} [kg s ⁻¹]	m_{airC} [kg s ⁻¹]	Stable power range [kW]	Efficiency
1(I)	horizontal/1	0	> 0	2-10	very low
1(II)	horizontal/1	max	> 0	2-10	very low
2(I)	vertical/8	0	> 0	2-10	high
2(II)	vertical/8	max	> 0	2-10	low

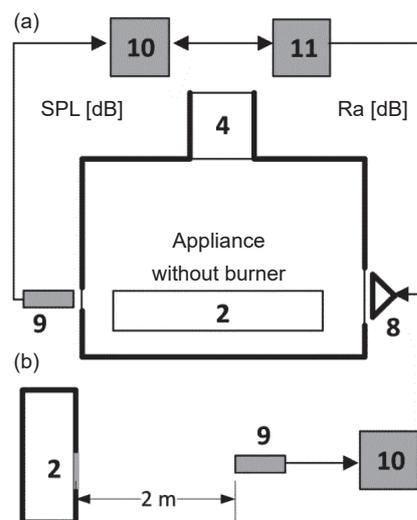


Figure 19. Sound measurements

Sound is captured by MiniSPL microphone (9) and digitally analyzed by the data acquisition system Minilyzer ML1 (10). In fig. 19(b) are presented measurements of sound generated by heater appliance [17].

Conclusions

From presented experimental research, and from measured characteristics of this appliance in which technical solution 2(I) is implemented follows next conclusions.

Introduction of air mass cross flow into the combustion chamber with lengthy pre-mixed gas burner suppress complex combustion thermo acoustic oscillations and drive the whole system into the normal working stage.

This research exemplifies that most effective method for suppression of thermo acoustic combustion oscillation (for this type of burner) is by divided air cross flow. Segmented air ($W/H \sim 0.4$) has to enter the combustion chamber through narrow slits ($s/W \sim 0.12$). Spaces between slits create recirculation zones with production of helicoidally vortices which are parallel to mainstream. The needed air mass flow rate for that purpose is of the order of 7% of overall combustor air mass flow rate.

Implemented method cover whole burner working regime. It is self-adaptable, because heater naturally ingests needed air mass flow rate for cross flow.

There were experiments, not presented in this paper, with other gaseous fuels (propane-butane, methane, methane with maximum 20% CO₂). They showed that this heater, with implemented combustion instability suppression method, is still flexible in using different types of fuel, and that is low CO and NO_x emission (for both lower than 50 mg/kWh).

This method permits burner producers to design long and lightweight burners. They are chipper because they needs less material and smaller machines for plastic deformation and punching.

This method increases the level of design autonomy for both producers (burner and heater).

Another conclusion is that appliance with gas burner, which suffers from combustion instabilities, extremely increases emission of CO, and for that reason, although the oscillations are relatively small, or occurred from time to time, or from regime to regime, the problem should be tackled.

The applied method may be helpful to other researchers which try to facilitate other types of lean burners [18, 19] with the view of decreasing NO pollution and combustion instabilities.

Nomenclature

A	– surface, [m ²]
m	– mass flow rate [kgs ⁻¹]
P	– power [W]
p	– pressure [Pa]
T	– temperature [K]
RP X	– relative percentage of X defined for same power. X_c is measured with air cross flow injection, and X without it, [$= 100(X_c - X)/X$], [%]
Na	– normalized amplitude by maximal measured noise signal amplitude for 4.3 kW combustion oscillations, [-]

Greek symbols

λ	– air-fuel ratio, [$m_{air}/m_{airStoichiometric}$], [-]
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Subscripts

C	– during air cross flow injection
mix	– air/fuel mixture
1	– Venturi inlet
2	– secondary air enter
3	– air vent
4	– flue exit

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