

SIMULATED BOILER PRESSURE PULSATION IN COMPARISON WITH EXPERIMENTAL MEASUREMENT

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Abstract: Two approaches, which analyse the interrelationships between the occurrence of pressure oscillations and the geometrical arrangement of flue gas passes in a boiler, are presented. Firstly a complex mathematical description of the process by means of partial differential equations is solved analytically. Then the compartment approach is presented. Using this approach, the complexity of the model can easily be selected to attain the desired accuracy, on the basis of experience. Both approaches have been verified by experimental measurements. The proposed modelling of boiler pressure pulsations can be a useful tool in boiler design. *Copyright © 2002 IFAC*

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1. DEFINITION OF THE PROBLEM

Pressure oscillations occur during operation in furnaces, flue gas passes and air channels of steam and hot water boilers of all types and capacities. The origin of the pressure fluctuations is connected mainly with the primary source – the burning process, which is essentially discontinuous (Fritsch, 1977). The rotating auxiliary machinery (fans, burners etc.) and the eddies generated in the wakes behind obstacles in flue gas passes (Karman's eddies round tube nests) are further sources of fluctuations.

From the acoustic point of view, the flue gas columns in a furnace, both in passes and in chimneys are resonators with a natural frequency defined by the dimensions of the boiler parts and by the temperature and the composition of flue gases. If there is no acoustic feedback among the boiler parts, excessive irregularities in the combustion process (i.e. local pressure and temperature oscillations) tend to fade, and the resulting influence on burning stability is small. However, coincidence between the frequency of the excited fluctuations and the natural frequency of the boiler parts and volumes can lead to major deterioration of the burning process resulting in the following effects:

- fluctuations in the physical parameters of burning (i.e., pressure, temperature) include regular harmonic oscillations among the stochastic components
- there is always a correlation between the oscillations at two different points downstream of the flue gases, or in two different moments at the same point
- the general effect of regular oscillations on the surroundings is not negligible (vibrations, pressure and heat stress of the boiler bearing structure)
- the amplitude of the pressure oscillations inside the furnace is usually more significant than in the case of irregular stable burning

Considerable amplitude of pressure fluctuations can cause problems in boiler operation increasing the noise level and the dynamic stress of the boiler walls and load-bearing structures. At sufficiently high-pressure pulsations, flue gases can penetrate between adjacent boiler rooms, which is not permissible for safety reasons. In this case the combustion process is unstable and may result in a flame outburst accompanied by an explosion. Especially challenging pulsations with high-pressure amplitude arise when the frequency of the pulsation source (fans, burning, Karman's eddies, etc.) matches the natural frequency

of boiler volumes acting as possible resonators. Resonators are defined by the dimensions of the spaces (transverse and longitudinal wave motion), by the lengths and types of the flue gas passageways and air channels, by the configuration of the boiler capacity volumes (furnace, free spaces, precipitators, etc.) and by the heating surfaces in boiler passes. Experiments carried out at the Department of Power Engineering of the Czech Technical University in Prague (Hrdlička, 1977, 1982) determined the frequency spectrum and amplitude of the pressure fluctuations in large-scale furnaces of coal-fired boilers. Two critical ranges of resonant pressure pulsations were located - in a low-frequency range of the spectrum (basic frequency up to about 10 Hz), which mainly corresponds to the source of pulsations from the burning process, and in a high-frequency range (basic frequency up to approx. 200 Hz), which corresponds to the pulsation sources from auxiliary rotating sources, turbulent burners, etc. Frequencies with the maximum amplitude are mostly in a low-frequency range around 1 Hz (pulverized coal combustion).

2. ROLE OF MODELLING IN PULSATION-RESISTANT DESIGNS

In addition to practical experience supported by experimental investigations, very useful and promising support for computer-aided design is offered by mathematical modelling and simulation (Hrdlička and Slavík, 1999; Neuman, *et al.*, 2000). A major task in contemporary boiler design is to minimize the above-mentioned negative impacts associated with pressure oscillations. Recent achievements in computing the natural frequencies of boiler volumes by means of an analogy with acoustic resonators characterized by R-L-C parameters leading to the mathematical determination of critical frequencies have opened new options for the development of simulation models.

3. MODELS OF PULSATION

Two ways of modelling have been proposed and applied in order to analyse the interrelationships between the occurrence of pressure oscillations and the geometrical arrangement of boiler flue gas passes. The two models use very similar concepts of simplified modelling of physical phenomena derived from quite different views of their causes. They also differ in the goals of the solutions. The background of the first approach is acoustic resonance and its description using analogy with electric R-L-C circuits, whereas the second approach derives a similar R-L-C model from a simplified description of the flow process of flue gases in a boiler. There is also a difference in the final goals. The first method analytically computes the values of natural frequencies, while in the second method equations for simulation models are derived and the simulation model is used for an experimental evaluation of the possible occurrence of a resonance effect.

3.1 Analytical approach

The set of volumes and heat exchange surfaces inside the boiler (furnace, tube and sheet heat exchangers, transition passes, connecting ducts and other parts, including dust precipitators and chimney) has the form of a closed acoustic circuit with a retarding component and feedback. Such an acoustic circuit behaves like an electrical Helmholtz resonator with definite natural frequencies. It is possible to determine these natural frequencies of boiler parts by a dynamic description of the flow of air and flue gas columns through the boiler structure.

The analogy between an R-L-C electric circuit and the model of a boiler created from analogical elements of acoustic resistance, capacitance and inductance is described by the following differential equation

$$L \frac{d^2 \dot{V}(t)}{dt^2} + R \frac{d \dot{V}(t)}{dt} + \frac{1}{C} \dot{V}(t) = -\frac{d}{dt} P(t) \quad (1)$$

where \dot{V} [$\text{m}^3 \cdot \text{s}^{-1}$] is transported gas volume, P [Pa] is pressure, L , C , R are acoustic inductance, capacitance and resistance of the boiler. The natural frequency of this system can be calculated by the relations

$$f_o = \frac{1}{2\pi} \sqrt{\frac{1}{C \cdot L}} \quad [\text{Hz}] \quad (2)$$

$$C = \frac{V}{w^2 \cdot \rho} \quad (3) \quad \text{and} \quad L = \frac{\rho \cdot l}{S} \quad (4),$$

where V [m^3] is volume, w [$\text{m} \cdot \text{s}^{-1}$] is speed of sound, ρ [$\text{kg} \cdot \text{m}^{-3}$] is average density of flue gases, l [m] is length of the duct section and S [m^2] is mean clear area of this.

Generally, the system complexity is given by the number of considered capacitances. In accordance with this, systems with one, two, three or more capacitances can be created. Mathematical solution of the one-capacitance model describing only the combustion chamber of a boiler is relatively simple. The natural frequency of the furnace is

$$f_{o1} = \frac{w_1}{2\pi} \sqrt{\frac{S_1}{l_1 \cdot V_1}} \quad [\text{Hz}] \quad (5)$$

where V_1 [m^3] is combustion chamber volume and w_1 [$\text{m} \cdot \text{s}^{-1}$] is speed of sound in flue gases at mean logarithmic temperature in the furnace.

Models with several such R-L-C substitutions for following parts of a boiler were created in a subsequent step of the described approach, and relations for calculating the natural frequencies were derived. As a consequence, the accuracy of the detected natural frequencies increased, but the complexity of the mathematical relations for calculating them became greatly complicated. For practical use of this approach, the number of considered boiler parts is limited to three. In this case

the boiler is described by a differential equation system

$$L_1 \frac{d^2}{dt^2} \dot{V}_1(t) + R_1 \frac{d}{dt} \dot{V}_1(t) + \frac{1}{C_1} \dot{V}_1(t) + \frac{d}{dt} P_1(t) = 0 \quad (6)$$

$$L_2 \frac{d^2}{dt^2} \dot{V}_2(t) + R_2 \frac{d}{dt} \dot{V}_2(t) + \frac{1}{C_2} \dot{V}_2(t) - \frac{d}{dt} P_2(t) + \frac{d}{dt} P_1(t) = 0 \quad (7)$$

$$L_3 \frac{d^2}{dt^2} \dot{V}_3(t) + R_3 \frac{d}{dt} \dot{V}_3(t) + \frac{1}{C_3} \dot{V}_3(t) = 0 \quad (8),$$

$$\text{where } \frac{d}{dt} P_1 + \frac{1}{C_2} (V_2 - V_1) = 0 \quad (9)$$

$$\text{and } \frac{d}{dt} P_2 + \frac{1}{C_3} (V_3 - V_2) = 0 \quad (10)$$

From this equation system the final solution can be derived in the following symbolic form

$$X^6 - a.X^4 + b.X^2 - c = 0 \quad (11)$$

where a , b and c are cumulated constants. Consequently three values of natural frequencies as physically real roots can be calculated.

The model can be improved by dividing the boiler into more parts, but an analytical solution is impracticable. This can be done much more easily in the compartment approach described in the following part.

3.2 Compartment modelling

The generally valid equations based on energy and mass balance

$$\int_{x_1(t)}^{x_2(t)} \left(S\rho \frac{\partial u}{\partial t} + Su \frac{\partial \rho}{\partial t} + M \left(w \frac{\partial w}{\partial x} + \frac{\partial w}{\partial t} \right) \right) dx = \int_{x_1(t)}^{x_2(t)} q dx + M_1 i_1 - M_2 i_2$$

and force balance

$$\int_{x_1(t)}^{x_2(t)} \rho \left(w \frac{\partial w}{\partial x} + \frac{\partial w}{\partial t} \right) dx = \int_{x_1(t)}^{x_2(t)} - \frac{\partial P}{\partial x} dx - \zeta \frac{M w_1}{2S}$$

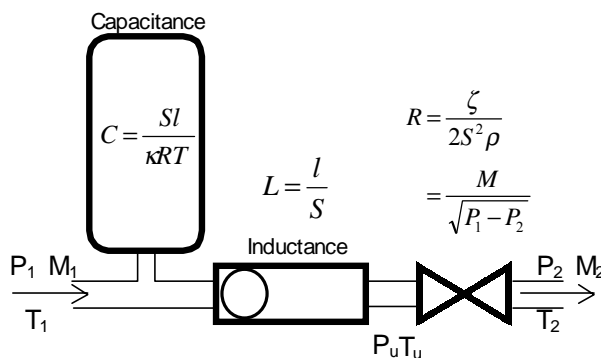


Fig. 1 Ordering of imaginary elements in the compartment model of a segment

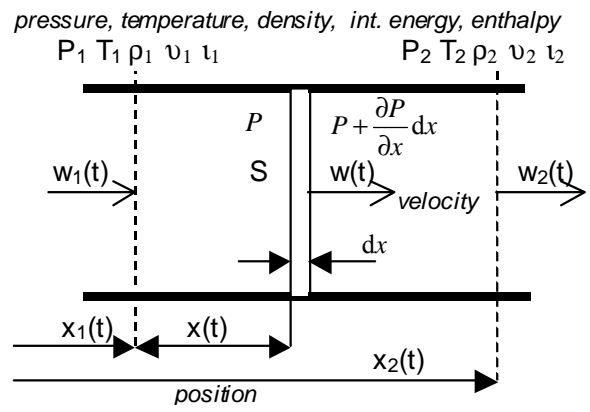


Fig. 2 Quantities in a distributed parameter model of a segment of the pipe

provide the basic mathematical background for a simplified description of three basic phenomena:

- energy dissipation
- compressibility of the medium
- inertia of the flowing medium

In the approximate arrangement of mutually separated imaginary elements representing these phenomena, as shown in Fig. 1, the following state equations can be derived from the previous partial differential equations, if the position dependence within a pipe segment is neglected

$$\frac{d}{dt} P_1 = \frac{\kappa R}{V} (M_1 T_1 - M_2 T_u) + \frac{R}{c_v V} Q$$

$$\frac{d}{dt} T_u = \frac{\kappa}{m} (M_1 T_1 - M_2 T_u) + \frac{T_u}{m} (M_1 - M_2) + \frac{1}{c_v m} Q$$

$$\frac{d}{dt} M_2 = \frac{S}{l} (P_1 - P_u) \quad (12)$$

where the concentration of the parameters within a pipe segment of length l , cross section S and volume $V=S.l$ is performed under the assumption that temperature changes are excluded from the compartment of inertia effect, where only the flow of an incompressible medium without friction and heat exchange is assumed (no thermodynamic changes).

Pressure – Flow rate model

Assuming that in the first stage of exploration, the complicated thermal processes in a boiler can be neglected and the effects of temperature changes are accounted for by a constant factor for fluid density, we can derive a model based on two quantities only – pressure and flow rate:

$$\frac{d}{dt} P = \frac{1}{C} (M_1 - M_2)$$

$$\frac{d}{dt} M_2 = \frac{1}{L} (P_1 - P_u) \quad (13)$$

$$M_2^2 = \frac{1}{R} (P_1 - P_u)$$

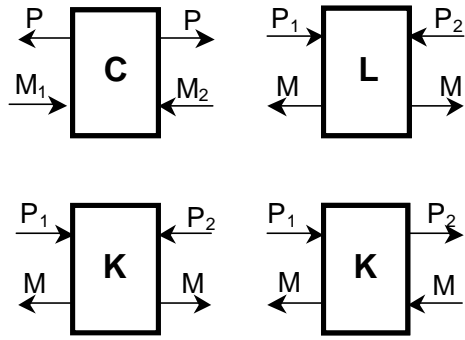


Fig. 3 Four-port arrangement of blocks for modelling compressibility, inertia and pressure losses

where the following parameters have been introduced by analogy with electrical elements:

$$\text{resistance} \quad R = \frac{\zeta}{2S^2\rho},$$

$$\text{capacitance} \quad C = \frac{Sl}{\kappa RT},$$

$$\text{and inductance} \quad L = \frac{l}{S}.$$

Library blocks

Using tools for creating and masking a subsystem provided by Simulink, a library of blocks representing each of the properties described by (13) was created. They are graphically arranged in the form of blocks with four ports (Fig. 3 and Fig. 4). The advantage of this arrangement is that it preserves the basic attribute of block schemes – the separation of inputs from outputs – and it is a simple way of providing a well-arranged interconnection scheme. The left-hand side of the four-port block can be interpreted as an inlet side, and the right-hand side as an outlet side (from the viewpoint of energy or mass flow).

This means that index 1 is always a variable characterizing the inlet flow, while index 2 is a variable in the outlet flow. Variables without index 1 or 2 have the same value on the inlet and outlet sides. Fig. 4 depicts a Simulink block as an example of the appearance of four-port blocks in this programme.

4. SIMULATION RESULTS

Both models divide the whole boiler flue passageway into segments that approximately correspond to a typical layout (combustion chamber, reverse chamber, convention pass, etc.). Simulation and verification of the models was performed for the tested boiler as described below, using these dimensions and operational data (temperatures, flows etc.). A scheme of the boiler with its main dimensions is shown in Fig. 5.

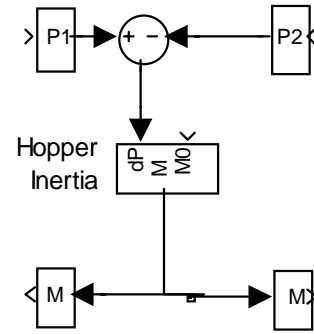


Fig. 4 Example of a four port block from the user defined Simulink library

4.1 Analytical approach

Applying the analytical model is relatively simple. The tested boiler is divided into three parts – combustion chamber plus louvre separator, reverse chamber plus convective pass with tube bundles, and hopper plus flue gas channel – representing three R-L-C substitutions in the model. The volumes, longitudes and free areas of these parts were calculated from the drawing documentation of the boiler. The temperature of the flue gases, which is important for calculating the gas density and sound speed in each part, was taken from the operational data of the boiler. Cumulative constants a , b , c were calculated and inserted into Eq. (11), which was solved numerically. Three real values of the natural frequencies of the boiler were obtained

$$f_1 = 1.03 \text{ Hz} \quad f_2 = 17.8 \text{ Hz} \quad f_3 = 52.7 \text{ Hz}.$$

4.2 Compartment modelling

From several ways of carrying out the simulation experiments, we selected the response to flue-gas mass flow perturbation at the combustion chamber inlet. This flow rate perturbation was performed in such a way that prior to the input of harmonic

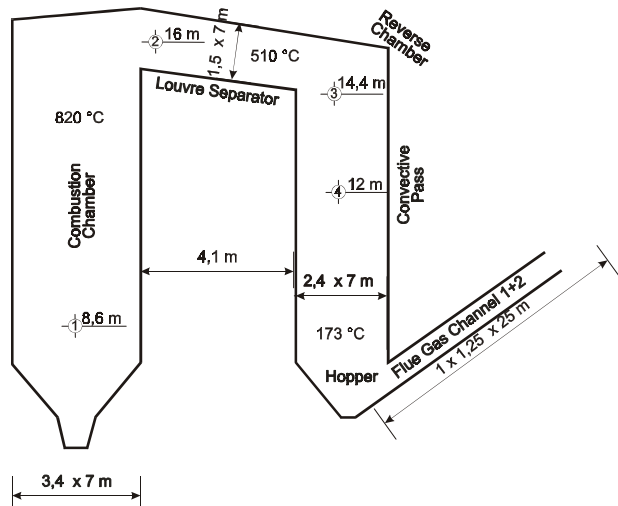


Fig. 5 Scheme of the tested boiler depicting the placement of the pressure probe in the experiment

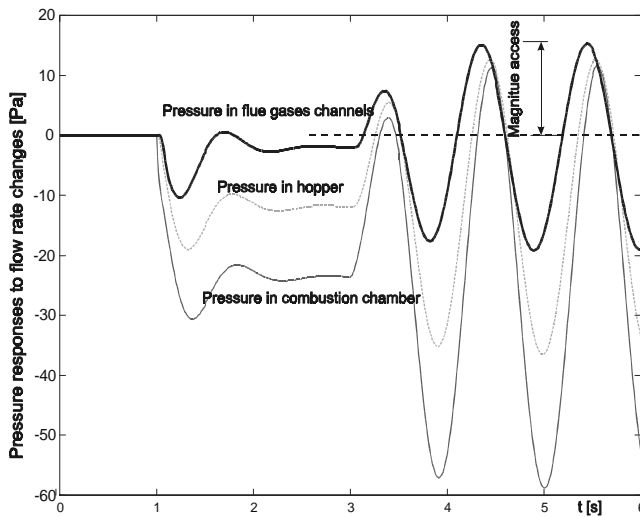


Fig. 6 Responses (differences from steady state values) to flow rate changes in the combustion chamber

oscillations with the magnitude $\pm 0,2$ kg/s starting at time 3 seconds, a step change of $-0,2$ kg/s was introduced. This enabled a very simple evaluation of whether the magnitude of the forced oscillation is greater or smaller than the magnitude of the exciting perturbation. Fig. 6 shows the responses at several locations along the flue path of the gases for the exciting frequency close to the resonance frequency. It should be noted that while the size of the pressure changes goes down as the pressure losses increase along the path, the gain for the harmonic responses grows and achieves its maximum at the end, although here the magnitude of the variations is absolutely the smallest.

This increase on gain is shown in the plot (Fig. 7), which depicts the ratio of the pressure oscillation magnitude in selected locations to the steady pressure changes at the same location. The frequency of the peaks in this dependence varies a little according to the location of the pressure changes, but it is in very good agreement with the experimentally measured values of about 1 Hz.

5. EXPERIMENTAL RESULTS

The measurements used for verification of the models were carried out in December 2000 on the K 13 fluidized bed coal-fired boiler operated by the Sokolov Chemical Works, working with significant pressure pulsations (Hrdlička and Dlouhý, 2001). K 13 is a drum steam boiler with nominal output of 50 tonnes of steam per hour with a temperature of 445 °C and a pressure of 3,73 MPa. The temperature of the feed water is 105 °C. The boiler has two flue gas passes. An evaporator with natural circulation covers the walls of the combustion chamber and the vertical walls of the horizontal pass. A fluidized-bed reactor with an adiabatic lining is situated in the lower part of the furnace. The flue gases are led away from the combustion chamber through the louvre separator located in the horizontal pass of the

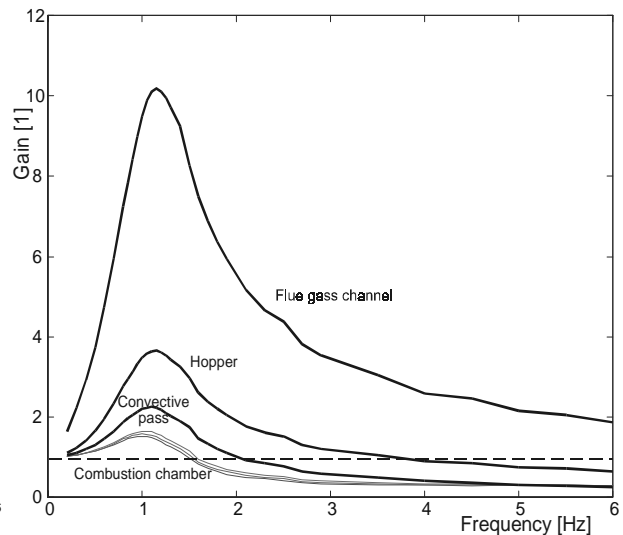


Fig. 7 Gain of frequency responses evaluated at several locations along the flue gas path

boiler. The ash particles separated from the gases are returned back to the fluidized bed. The convective tube bundles of the superheater and economizer occupy the subsequent vertical pass. The flue gases are blown from the economizer through the ash hopper and two parallel channels out from the boiler. The positions of the measuring points are shown in Fig. 5. The load of the boiler in the course of measurement was 29 and 42 tonnes of steam per hour. In order to detect the pressure fluctuations inside the K 13 boiler we used two tensiometric pressure probes with an overlaid membrane and an amplifier giving the required 5 V signal at overpressure 8 kPa. The natural frequency of the probes was over 500 Hz.

Six measuring points were selected for detection of the static pressure inside the K 13 boiler and for frequency analysis:

- 1 – combustion chamber – level 8,6 m
- 2 – horizontal pass – level 16 m
- 3 – free admission space to the convective pass – level 14,4 m
- 4 – convective pass under superheater – level 12 m
- 5 – outlet channel of left primary air fan
- 6 – outlet channel of right primary air fan

The signal from the pressure probes was fed into the ADVANTEST two-channel low-frequency analyzer working in real time. The analyzer was used for recording and plotting the measured signals, for evaluating the pressure fluctuation amplitude, and for frequency analysis of the signal standard deviation in the range from 50 mHz to 20 Hz, in order to determine the dominant frequencies of the pressure fluctuations.

Fig. 8 shows an example of a typical measured frequency spectrum. For all measured cases, the most significant frequency peak was found in the range from 0.95 to 1.15 Hz.

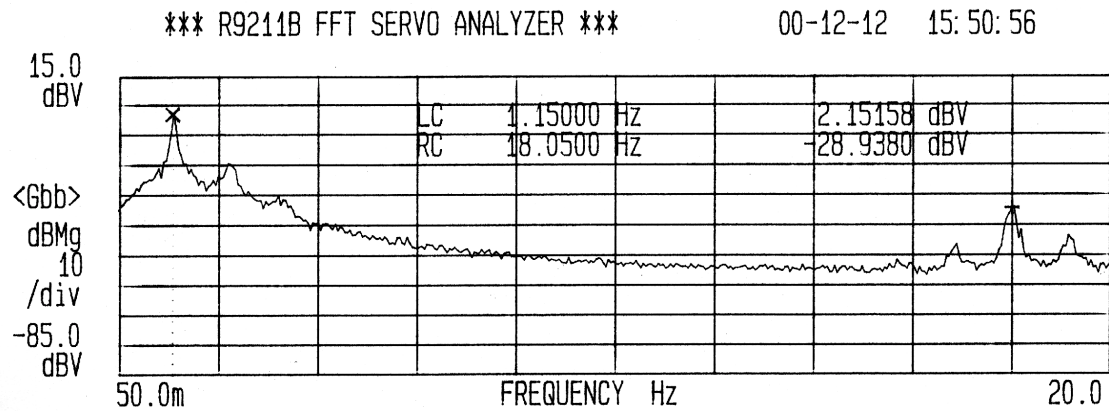


Fig. 8 Standard deviation spectrum of a pressure probe signal at measuring point 1 (furnace)

The dominant frequency was accompanied by the 1st and 2nd harmonics, which were clearly evident in the air channel spectra. The amplitude of the harmonics in the furnace and flue gas passes was considerably lower due to absorption of the higher-frequency fluctuations in large volumes. The magnitude of the dominant frequency peak in the range around 1 Hz can be rated as the natural frequency of the boiler passes amplified by the resonance with pressure fluctuations of the burning process. Another natural frequency peak, but much lower, was detected in the range from 16.45 to 18.05 Hz.

The measured frequency peaks match acceptably with those calculated by both models. The magnitude of the dominant frequency peak in the range around 1 Hz can be rated as the natural frequency of the boiler passes amplified by resonance with the pressure fluctuations of the burning process.

7. CONCLUSIONS

An evaluation of simulated pressure oscillations by two models has shown that the results are very close to the values obtained from the experiments on a real boiler. Successful creation and verification of the models opens up the possibility of estimating the dynamics of flue gas transport inside boilers during the course of design and construction in order to avoid the occurrence of damaging resonance effects after coming into operation. Neither of the methods differed essentially in the estimated values of the undesirable frequencies, i.e., they can be used equally, according to the circumstances. Analytical computation seems to be more suitable for a rapid evaluation of problems in existing boilers, while the simulation approach is advantageous for the design of new boilers.

8. ACKNOWLEDGEMENT

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