*Journal of Fluids and Structures* (1998) **12**, 491–507
Article No. fl970144

# FEEDBACK CONTROL OF COMBUSTION OSCILLATIONS

S. Ziada* and H. Graf
*Sulzer Innotec Limited, CH-8401 Winterthur, Switzerland*

(Received 11 November 1996 and in revised from 1 December 1997)

Combustion oscillations are excited by a feedback mechanism which results from the coupling between the flame instability and an acoustic resonance mode of the combustion chamber. The most important event of this mechanism is believed to be the upstream feedback from the acoustic resonance to the initial region of the flame, where new disturbances are generated. A control system has been implemented into a commercial household burner to counteract this upstream feedback. This is achieved by means of pulsating either the fresh air for combustion or the fuel. The actuators are activated by the signal of a sensor measuring the pulsation inside either the combustion chamber or the fresh air supply pipe. Combustion oscillations of this burner have been eliminated altogether, also when two modes were simultaneously excited, without destabilizing other acoustic modes. This results in 35 dB reduction in the amplitude of the pressure pulsation. Furthermore, when the combustion process was stable, i.e. no resonances were excited, the use of active control has reduced the broad-band noise by 4 dB. The effect of the input sensor location on the control system performance has also been investigated.


## 1. INTRODUCTION

The occurrence of unstable combustion in utility boilers, furnaces and propulsion systems is often associated with intense pressure pulsations which can cause either structural failure of the system or degradation in its performance (Crocco & Cheng 1956; Price 1969; Putnam 1971; Putnam *et al.* 1985). Combustion oscillations, however, can also produce desirable effects, such as increased rates of chemical heat release and heat transfer or reduced emission of pollutants. This latter feature has been exploited by developing pulsating combustors for domestic boilers (Corliss *et al.* 1984; Keller & Westbrook 1986; Yu *et al.* 1991; Zinn 1992). This paper deals with the former situation in which the unstable combustion is harmful to the system or its performance, and is therefore undesirable.

The excitation mechanism causing combustion instabilities is inherently complex because it involves coupling between different complex phenomena, such as two phase-turbulent flow, unsteady chemical reaction and heat release, hydrodynamic instabilities and acoustic wave propagation (Crocco & Cheng 1956; McManus *et al.* 1991). Due to this complexity, it is difficult to assess the stability of a particular combustor without expensive and time-consuming testing under liable (i.e. reactive) acoustic environments. In spite of such a lengthy development phase, burner manufacturers are often faced with combustion oscillations during the commissioning period of new projects. These problems are generally

*Present address: Department of Mechanical Engineering, McMaster University, Hamilton, Ont. L8S 4L7, Canada.

0889–9746/98/040491 + 17 $30.00

solved by means of *ad hoc* countermeasures, such as changing the pattern of fuel injection, altering the nozzles of fresh air, adding acoustic liners or Helmholtz resonators, or even changing the overall dimensions of the combustion chamber (Heitor *et al.* 1984; Putnam *et al.* 1985; Sivasegaram & Whitelaw 1987; Ziada & Oengoeren 1996). These measures, however, are often not effective in other installations which have slightly different dimensions or higher capacity. In addition, compliance with the present stringent emission regulations leaves little room for such *ad hoc* fixes of combustion oscillation problems. A more efficient method to eliminate combustion oscillations is certainly needed.

The success of using active methods to control hydrodynamic instabilities of shear flows (Crow & Champagne 1971; Ho & Huang 1982; Ffowcs-Williams & Huang 1989; Rockwell 1990; Huang & Weaver 1991; Ziada 1995) has encouraged many researchers to investigate the feasibility of alleviating combustion oscillations by active means. There are several types of combustion control systems that have been investigated in the literature [see the reviews by McManus *et al.* (1991) and Schadow *et al.* (1992)]. They range from simple open-loop controllers, which impose external forcing on the combustion process (Gutmark *et al.* 1989) to sophisticated fast-response adaptive controllers (Billoud *et al.* 1992). Most of the previous work, however, has dealt with simplified, laboratory models of dump and gas turbine combustors and little attention has been given to industrial and household burners. This paper focuses on the implementation of a feedback control system into a commercial oil burner operating in a commercial combustion system.

After introducing the excitation mechanism and the concept of feedback control, the test facility, the burner and the components of the control system are described. The system response is then documented and the frequencies of the resonance modes are calculated by means of a simple acoustic model. Finally, the performance of the control system is addressed.

## 2. EXCITATION MECHANISM AND ACTIVE CONTROL CONCEPT

The excitation mechanism of combustion instabilities, together with the basic idea of the control system used to suppress the resulting oscillations are illustrated schematically in Figure 1. Small disturbances at the initial region near the head of the burner are strongly amplified by means of unsteady chemical reaction and heat release. This results in pressure pulsations which excite an acoustical mode of the combustion chamber. The resonant acoustic mode, in turn, induces new disturbances at the initial region of the combustion

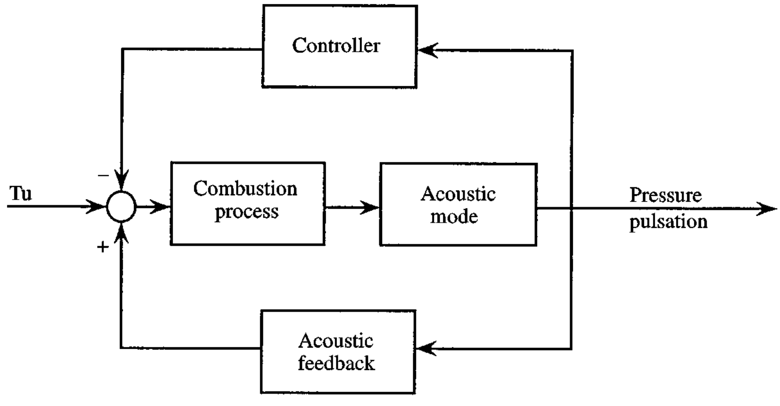


Figure 1. Schematic presentation of the excitation mechanism and the concept of active control.

process. This series of events will be self-enhanced, resulting in large amplitude pulsations, if the "Rayleigh index" (Rayleigh 1945) is positive. This index is given by

$$G = \iint_{X,\ T} Q(x, t)\ P(x, t)\,\mathrm{d}x\,\mathrm{d}t, \tag{1}$$

where $Q(x, t)$ is the unsteady heat release, $P(x, t)$ represents the unsteady pressure, $x$ is the location, and $T$ stands for the time of one oscillation cycle. The above condition means that a favourable phase between $P$ and $Q$ must be maintained over the whole oscillation cycle.

The process of initial disturbance growth into unsteady heat release as well as the interaction of this heat release with the acoustic pulsation are regarded here as "black boxes" and are not directly targeted by the control system. The active control concept is based on the hypothesis that *the upstream feedback of acoustic pulsations is crucial* to the generation of new disturbances at the initial region and maintaining a favourable phase between $P(x, t)$ and $Q(x, t)$, such that the oscillation is amplified. Thus, the objective of the control system is reduced to counteracting the effect of acoustic feedback within the initial region; see Figure 1. This can be achieved by measuring the system response, with the aid of a suitable input sensor, adjusting its phase and amplitude, and then feeding it to a secondary source (an actuator) which is *focused on the initial region only*. The concept of actively cancelling the effect of upstream feedback has already been applied very successfully to suppress the global oscillations of impinging shear flows (Ziada 1995). The present study represents an additional application which demonstrates the effectiveness of this control method in suppressing self-excited global oscillations.

Two different actuators are used in this study. The first is a servo valve installed in the fuel (oil) supply pipe to modulate the rate of fuel injection. The second actuator is a loudspeaker which acts on the inlet duct of the fresh air to oscillate the combustion air at the head of the burner. This arrangement of actuators has the following advantages:

(i) the actuators are not subjected to the harsh combustion environment;
(ii) the actuators do not act directly on the system resonator (i.e. the combustion chamber) but rather on a small portion of the whole field, and therefore the power level needed for control is greatly reduced; additional power saving is achieved by focusing the effect of the actuators on the initial region only, where the disturbances exciting the combustion process are still small;
(iii) the actuators act directly on the burner and, therefore, all acoustic modes which may be excited by the burner can also be influenced by the control system; this would not be the case where a secondary burner to be used because its effect would be dependent on its location with respect to the excited modes; in fact, secondary burners may well excite other acoustic modes which are naturally stable (Sivasegaram & Whitelaw 1992).

## 3. EXPERIMENTAL TECHNIQUE

### 3.1. Test Facility

The study was carried out on a standard, low-emission burner which is widely used in household heating systems. It has a power of 25 kW and burns heating oil with a heat value of 11.9 kWh/kg. The experiments were carried out on a standard test boiler Type CEN 7261. The geometry and main dimensions of this boiler are shown in Figure 2. It has two observation pipes, one at each side, and a throttling device. The flue gases recirculation pipes were closed at the burner side.

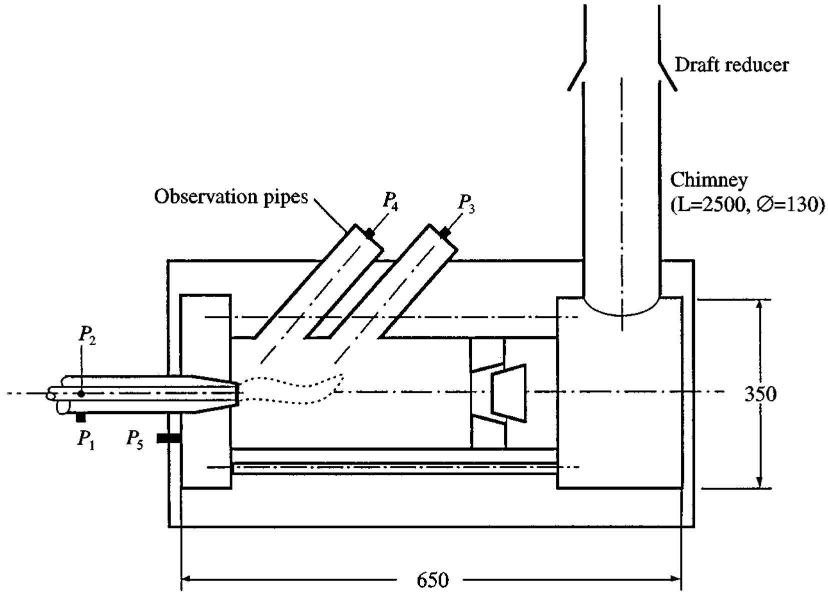


Figure 2. Geometry of the test boiler, showing the main dimensions and the locations of measurement of pressures $P_1$ to $P_5$. Dimensions are in mm.

### 3.2. Control System

Either the combustion oil or the fresh air could be pulsated *separately*. The former was induced by means of a servo valve Type Moog. It was mounted between the oil pump and the burner head to pulsate the oil pressure at the atomization nozzle. A commercial 40 W loudspeaker was connected to the fresh air pipe downstream of the inlet air throttle and the blower to produce pressure pulsations inside the burner pipe which delivers the fresh air to the burner head. Every effort was made to mount the servo valve and loudspeaker as close as possible to the burner head, but still outside the combustion chamber.

The signal activating the actuators was conditioned by either a narrow band controller or a wide band digital one. The former consisted of a general purpose filter, a Sulzer phase shifter and a power amplifier. Its gain and phase were manually adjusted until the best attenuation was achieved. In the second case, a Digisonix 57 adaptive controller featuring a recursive infinite impulse response digital filter was used. It employs a least mean-square algorithm to adjust the filter coefficients. The signal of the pressure pulsation inside the combustion chamber was used as the input signal for both controllers.

### 3.3. Instrumentation

The pressure fluctuations were measured at five locations; see Figure 2. A microphone, Type Sennheiser, was used to measure the pulsation in the fresh air pipe near the burner head ($P_1$). The pressure at all other locations was measured by means of Kistler pressure transducers. These locations are: inside the oil pipe downstream of the servo valve ($P_2$), at the end of the observation pipes ($P_3$, $P_4$), and on the inside wall of the boiler door ($P_5$). All signals, including the inputs to the actuators, were recorded on an eight-channel digital recorder, Type Heim. The data analysis was carried out at a later time by means of a four-channel frequency analyser.

## 4. FEATURES OF SYSTEM OSCILLATIONS

### 4.1. Acoustic Characteristics of the System

At first, the system response to acoustic excitation of the fresh air was studied. The loudspeaker was activated by a frequency sweep (50 to 250 Hz) and the transfer functions between the fluctuation pressures ($P_1$, $P_3$, $P_4$, $P_5$) and the input signal ($S$) to the speaker were measured. This was made with the fresh air flowing, but first without and then with combustion.

Typical results are shown in Figure 3. The frequency response ($P_1/S$) is relatively flat, indicating that the fresh air channel has little effect on the system response. Without combustion, the pressure inside the combustion chamber shows system resonances at the frequencies 75, 109 and 188 Hz: see ($P_5/S$) in Figure 3(a). When the burner is switched on,

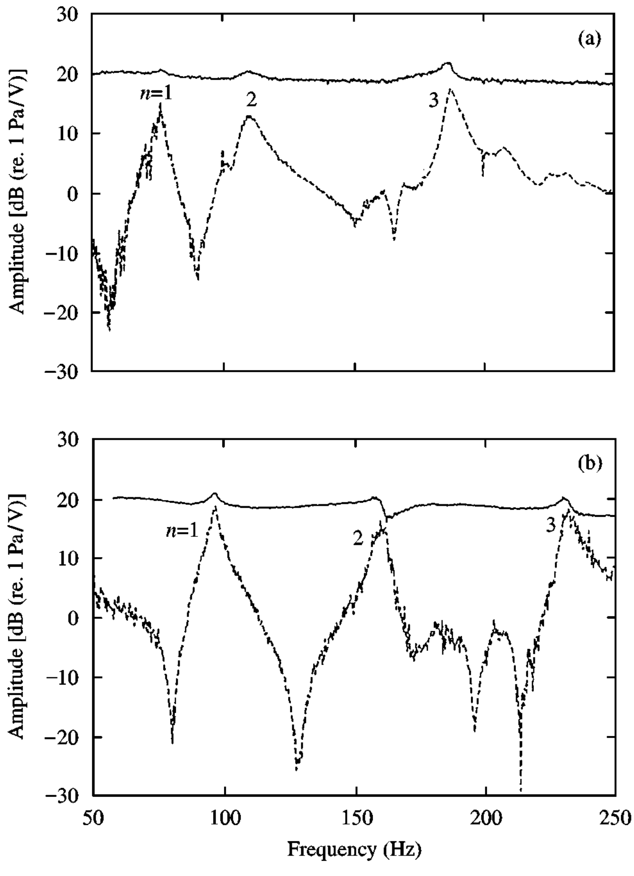


Figure 3. Frequency response of the system to acoustic excitation of the fresh air pipe by means of the loudspeaker, (a) without combustion and (b) with combustion:—-, $P_1/S$; - - - -, $P_5/S$; $S$ is the loudspeaker signal, and $n$ is the mode number.

Figure 3(b), the resonance frequencies become higher, to 95, 160 and 230 Hz, due to the increase in the temperature and the sound speed. Measurements at all other locations showed similar results. Therefore, self-excited combustion oscillations are expected to occur at one or more of the latter set of frequencies.

The above resonance frequencies could not be estimated with sufficient certainty by means of simple calculation which considers the combustion chamber or the chimney separately. To eliminate any doubt concerning the nature of these resonance modes, an acoustic model combining the combustion chamber, the chimney and the burner (i.e. the fresh air pipe) was formulated. In this model, the system components consisted of pipe sections with different lengths and cross-sectional areas (and also different sound speeds in the case with combustion). Further details of the acoustical model are given in the Appendix.

The frequency response of the model ($P_5/P_1$) at room temperature is given in Figure 4(a), and that corresponding to hot conditions in Figure 4(b). Comparison with Figure 3 indicates that the model reproduces the measured responses rather well, despite the uncertainty of the assumed temperature distribution. Thus, the system resonance frequencies can be

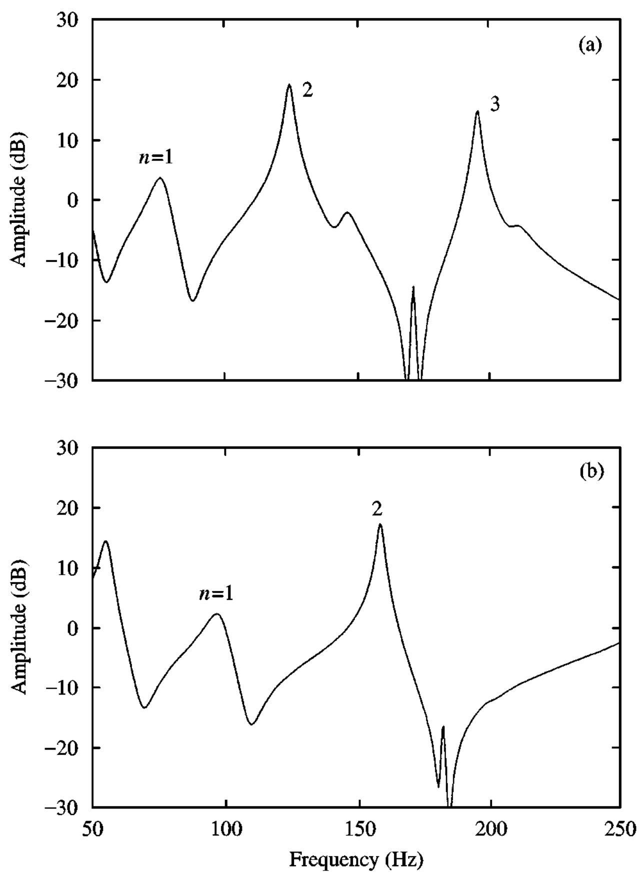


Figure 4. Frequency response ($P_5/P_1$) of the acoustic model consisting of the burner, the combustion chamber and the chimney: (a) at cold conditions; (b) at hot conditions; $n$ is the mode number.

predicted by means of an appropriate model which combines the relevant system components.

### 4.2. Self-Excited Oscillations

It was not possible to produce self-excited oscillations at normal operating conditions of the burner. However, when the air/fuel ratio was reduced from that of normal operation, and the through-draft reducer of the chimney was fully opened, to make the chimney acoustically more reactive, self-excited oscillations occurred. The fact that these oscillations occurred outside the normal operation range of the burner does not detract from the value of this study, because the concept of the control system is not dependent on the burner test conditions.

The oscillation occurred at two frequencies (97 and 152 Hz) which are close to the resonance frequencies described above (95 and 160 Hz). With small adjustments of the air–fuel ratio, it was possible to produce oscillations at either the first mode, the second mode, or both modes simultaneously, Figures 6(a), 5(a) and 6(b), respectively. Thus, it was possible to study the performance of the control system at different modes of combustion oscillation.

## 5. FEEDBACK CONTROL WITH NARROW-BAND CONTROLLER

This control method was applied when the system was oscillating at the second mode. The loudspeaker exciting the fresh air upstream of the burner head was activated by the controller, using the signal of the pressure transducer measuring $P_5$ which was mounted in the combustion chamber and closest to the burner as the input signal. The phase and gain of the controller were manually adjusted until the spectral peak of the second mode was minimized. With these settings, the phase and gain of the speaker excitation are optimally adjusted only at the second-mode frequency.

The spectra with and without control are illustrated in Figure 5(a). The amplitude of the second mode at 152 Hz is seen to be reduced by an amount of 18 dB, and its harmonic at 304 Hz is eliminated from the spectrum. The elimination of this harmonic, despite the fact that the controller settings are not optimal at this frequency, indicates that this harmonic results from nonlinear amplitude distortion of its fundamental component, i.e. the second mode. The secondary peak at the first-mode frequency, 97 Hz, is not reduced because the phase of the speaker excitation is not optimally adjusted at this frequency. In fact, when the gain of the control circuit was increased beyond this optimal value, the combustion became unstable and large amplitude oscillation occurred at frequencies which are not necessary related to the resonance frequencies. In this case, the controller provides a favourable feedback, via the speaker, to the initial region of the combustion process such that the oscillation is amplified. With the optimal settings, the electric power needed to keep the system under control was 0.16 W only.

## 6. FEEDBACK CONTROL WITH ADAPTIVE DIGITAL CONTROLLER

### 6.1. Control by Forcing the Fresh Air

Since wide-band, adaptive controllers (referred to hereafter as digital controllers) have built-in digital filters with adjustable coefficients, the phase and amplitude of the actuator

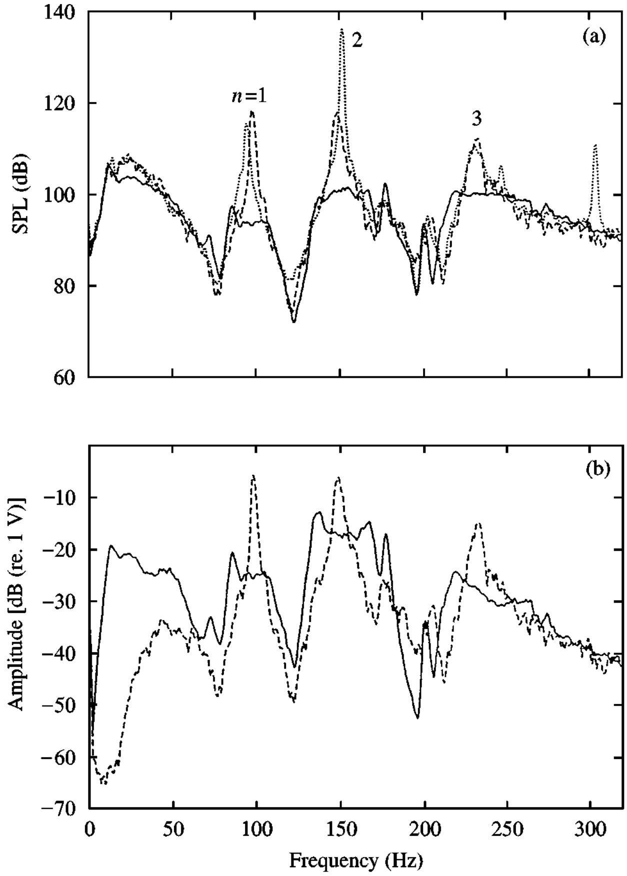


Figure 5(a, b). Frequency spectra of (a) pressure pulsation $P_5$ and (b) activating signal of the loud speaker; $\cdots\ \cdots$, without control; ----, with narrow-band controller; —, with adaptive digital controller. $P_5$ is used to activate the speaker; $n$ is the mode number.

excitation can be correctly adjusted over a wide band of frequency. In this case, active suppression of a particular mode does not necessarily destabilize the system at other modes. The amount of noise reduction, therefore, can be increased substantially in comparison to the case of narrow-band control. Moreover, if the system oscillates at two modes simultaneously, digital controllers can attenuate both oscillation modes at the same time.

Figure 5 compares the performance of the digital controller with that of the narrow-band controller. The digital controller reduces the amplitude of the second mode at 152 Hz by 35 dB, that of the first mode at 97 Hz by 20 dB, and that of the broad-band peak near 230 Hz by 10 dB. This is equivalent to a reduction of 16 dB in the *total* SPL (sound pressure level), compared with only 10 dB in the case of narrow band control. The superior performance of the digital controller is achieved with an input power to the loudspeaker of 0.1 W, which is less than that needed by the narrow-band controller.

Figure 5(b) shows frequency spectra of the signals activating the speaker for the narrow-band and the digital control cases, and Figure 5(c,d) depicts the transfer functions (amplitude and phase) of both controllers. The transfer function of the narrow-band controller changes gradually with frequency and the speaker signal is very similar to the pressure

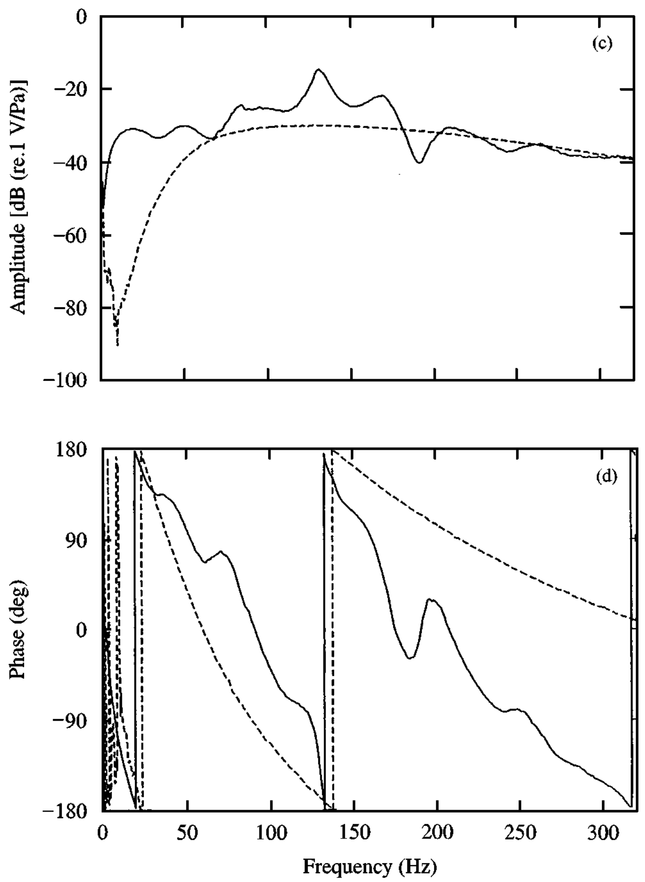


Figure 5(c, d). Controller transfer function (c) amplitude and (d) phase:- - -, narrow-band controller; —, adaptive digital controller. $P_5$ is used to activate the speaker.

signal. In the digital controller case, the transfer function has a wavy shape, reflecting the fact that correct setting requires rather complex adjustments in frequency. It is interesting to note that the setting of the narrow-band controller at 152 Hz differs by 6 dB in magnitude and 46° in phase from the optimum setting as determined by the digital controller.

### 6.2. Controller Performance at Different Modes of Instability

As described in the preceding section, the performance of the digital controller is found to be excellent when applied to combustion oscillations at the second acoustic mode. In this section, the controller performance is studied under three additional test conditions.

First, when the first mode was excited, Figure 6(a), the controller reduced its amplitude by 35 dB, and the broad-band peaks near 150 and 250 Hz were effectively eliminated from the spectrum. The *total* SPL was therefore reduced by 9 dB. Second, when the first and the second modes were simultaneously excited, Figure 6(b), their amplitudes were reduced by 19 and 30 dB, respectively, and all other peaks at the sub- and super-harmonics were eliminated. The equivalent reduction in the *total* SPL was 14 dB. Finally, at normal

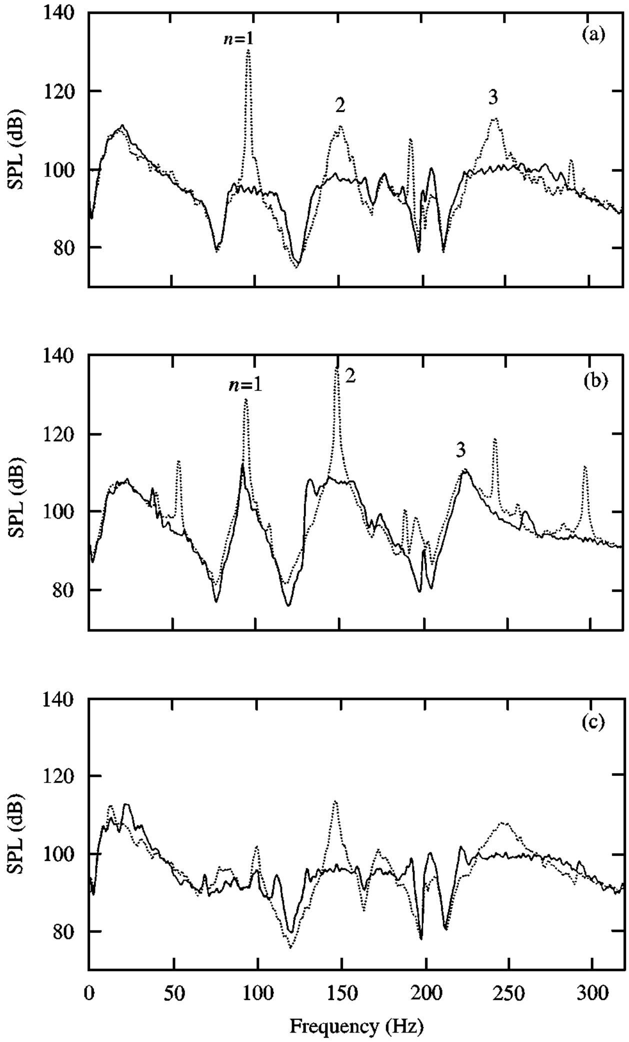


Figure 6. Pressure spectra inside the combustion chamber with and without adaptive digital control: (a) control of first mode oscillation; (b) first and second mode oscillations; (c) broad band oscillations. $P_5$ is used to activate the speaker; $n$ is the mode number.

operating conditions, the pressure spectrum resembled broad-band peaks and neither mode was excited, Figure 6(c). In this case, the controller eliminated these broad peaks, resulting in a "smooth" spectrum and a reduction of 4 dB in the total SPL inside the combustion chamber. Although the present study has not been focused on broad-band noise, the latter case underlines the possibility of using active control techniques to reduce broad-band noise of combustion systems.

### 6.3. Effect of Position of Input Transducer

In order to check the influence of the location of the transducer which activates the controller, the signals of different transducers were used consecutively as input to the controller. This test series was conducted when the second mode was excited and the results are shown in Figure 7.

The best performance resulted when the transducer measuring $P_5$, which is the closest to the flame, was used as the input sensor, a reduction of 35 dB was achieved. When the transducer at the end of the observation pipe was used, the attenuation was only 1 dB lower and the speaker power was 2 dB higher. The attenuation factor was substantially lower (a reduction of 18 dB only) when the transducer in the fresh air pipe was used for control. Moreover, the system stability was not satisfactory, because large-amplitude pulsation occurred sporadically and the controller calibration had to be repeated often.

These results confirm the supposition that the input sensor should measure the acoustic pressure as close as possible to the flame in order to reduce the time lag between the flame oscillation and the effect of the active feedback on the flame. The poor performance observed when the input sensor was located in the air supply is rather disappointing, because at this location the sensor is better protected from the hazardous combustion environment.

### 6.4. Control by Forcing Combustion Fuel

Modulations of the fuel flow rate was also effective in suppressing the excitation mechanism, as can be seen in Figure 8. The signal of the transducer measuring $P_5$ was used to activate the servo valve. The pulsation amplitude at 152 Hz is seen to be reduced by 34 dB (only 1 dB lower than that achieved by the loudspeaker), and its harmonic is also eliminated. The broad-band peak near the first mode however is not much influenced by the servo valve. The reduction in the total SPL (14 dB) is only 2 dB lower than that achieved by the loudspeaker (Figure 5).

The signals activating the servo valve and the loudspeaker, to control the system under similar conditions, are compared in Figure 8(b). The valve requires a much higher input level than the speaker, about 20 dB higher. The total input power to the valve, however, is only 0.14 W, which is somewhat higher than that needed to activate the speaker (0.1 W).

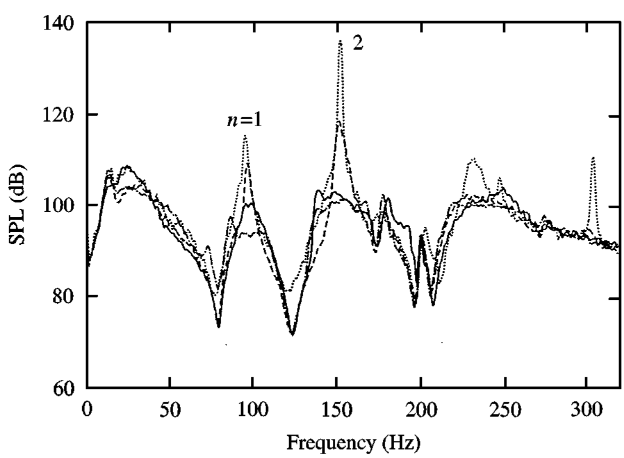


Figure 7. Effect of location of input transducer on the controller performance: ⋯ ⋯, without control; - - - -, control with $P_1$ as input; —-, control with $P_3$ as input; –·–·–·, control with $P_5$ as input. $n$ is the mode number. Reduction in total SPL:- - - -, 12 dB; —-, 15 dB; –·–·–·, 16 dB.

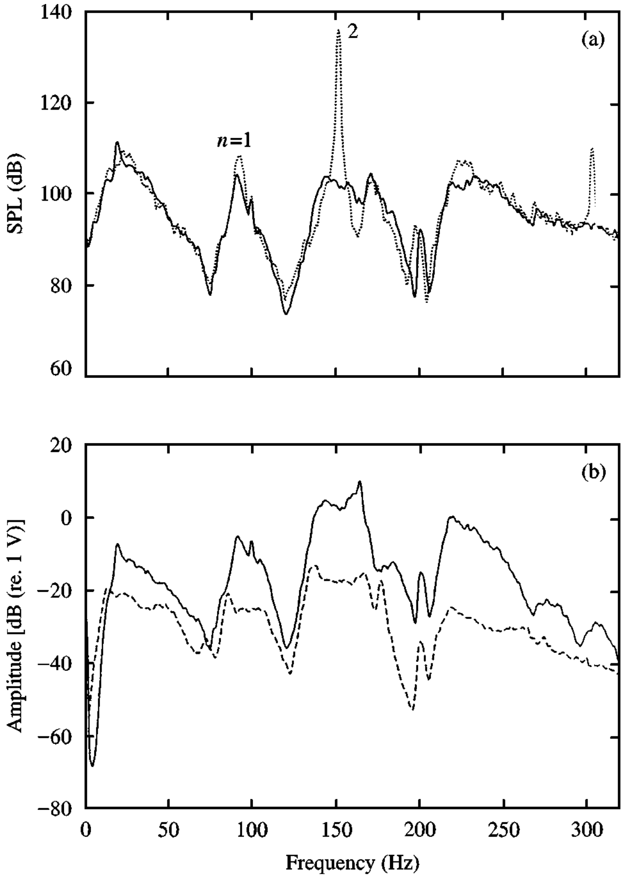


Figure 8. Adaptive control by pulsating the combustion fuel. (a) Spectra of pressure pulsation: ⋯ ⋯, without control; —-, with control ($P_5$ activated the servo valve). (b) Comparison between the control signal activating:—-, the servo valve; - - - -, the speaker. $n$ is the mode number.

## 7. TRANSIENT RESPONSE AT THE INITIAL PHASE OF CONTROL

Typical time traces of the signal activating the loudspeaker and the signal of the pressure pulsation in the combustion chamber ($P_5$) are shown in Figures 9 and 10. As can be seen in Figure 9, when the controller is switched on at time near $t \approx 0.1$ s, the pressure pulsation is damped out within about 0.05 s. During this initial phase, the loudspeaker signal also decreases rapidly to less than $\pm$ 0.1 V. This is so because the loudspeaker is activated by the signal of the pressure pulsation. Thus, once the feedback excitation mechanism is suppressed (by cancelling the effect of the upstream feedback), the system oscillation will vanish, and therefore a relatively small amount of power will be needed to keep the system stable, i.e. to keep the upstream feedback weaker than the threshold required to initiate the oscillation. This feature is distinct from the situation of active noise cancellation, in which the secondary sources do not influence the noise source, but continuously generate sound levels as high as those of the source, and therefore a continuous much higher power level would be required than in the present case.

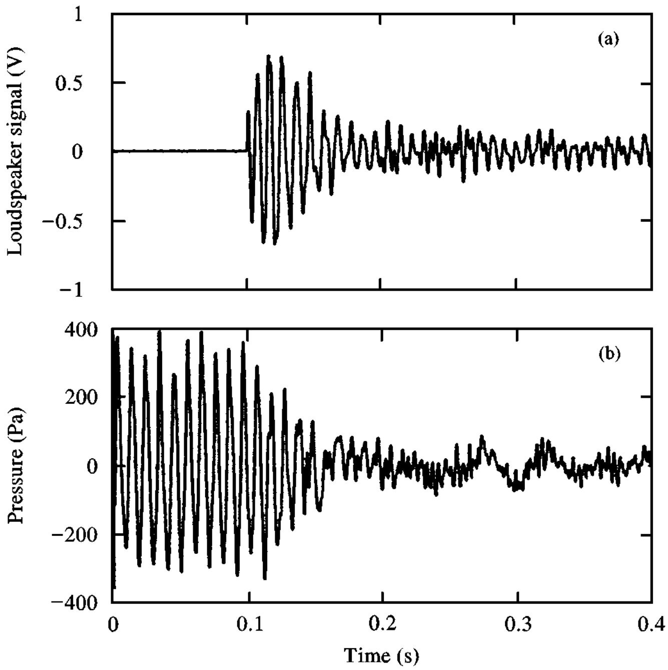


Figure 9. Time traces showing the transient response of the system when the controller is switched “on”: (a) signal activating the loudspeaker; (b) pressure pulsation in the combustion chamber. Adaptive control with $P_5$ activating the loudspeaker.

Figure 10 illustrates the system response when the controller is switched off. The pressure pulsation grows in place with time, indicating that the system oscillation is caused by a global instability. Another feature which can be seen in Figure 10 is that when the controller is switched off, the pulsation growth rate is substantially slower than the attenuation rate when the controller is switched on. The time signal of Figure 10(b) can be used to analyse the negative damping which induces the combustion instability.

## 8. SUMMARY

An active control system has been implemented into a household burner to suppress combustion instabilities. Systematic tests have been carried out on an actual boiler to better understand the performance of the control system when implemented with different actuators, activated by different input sensors, and operating under different instability modes.

Excellent control performance was observed by pulsating either the fresh air or the fuel. In both cases, an input power less than 0.2 W was needed to reduce the pressure pulsation by an amount of $\sim$35 dB. It was possible to totally suppress the combustion instability, whether it occurred at the first acoustic mode, at the second one, or at both modes simultaneously. The best performance was achieved when the sensor located nearest to the flame was used to activate the actuator.

The active control concept developed in this study is based on the hypothesis that the upstream feedback of acoustic pulsations is crucial to the amplification of oscillation. The

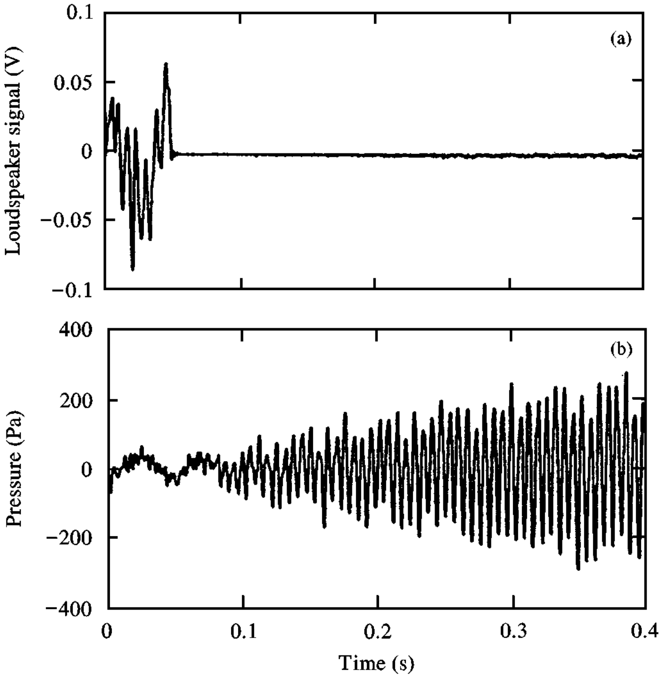


Figure 10. Time traces showing the transient response of the system when the controller is switched "off": (a) signal activating the loudspeaker; (b) pressure pulsation in the combustion chamber. Adaptive control with $P_5$ activating the loudspeaker.

success in suppressing the oscillations by actively cancelling the feedback effect confirms this basic hypothesis and the control concept.

The findings that excellent performance can be achieved by means of either the servo valve or the speaker, and that comparable input power is needed in both cases are of practical importance. This is because the choice of the actuator type is actually dictated by the frequency under consideration. Combustion instabilities in large boilers occur at low frequencies (50 Hz or lower). In these cases, loudspeakers are very inefficient and servo valves are more promising. Conversely, loudspeakers are more suitable to control high-frequency oscillations which occur in small boilers.

## ACKNOWLEDGEMENT

This work is jointly supported by the National Energy Research Fund of Switzerland (NEFF) and Sulzer Innotec Limited. The authors are grateful to M. Hauswirth and L. Lehmann, of Oertli Waermetechnik AG in Schwerzenbach, for their help and permission to carry out these tests in their development laboratory.

## REFERENCES

Billoud, G., Galland, M. A., Huynh Huu, C. & Candel, S. 1992 Adaptive active control of combustion instabilities, *Combustion Science and Technology* **18,** 257–283.

Corliss, J. M., Putnam, A. A., Murphy, M. J. & Locklin, D. W. 1984 $NO_x$ emissions from several pulse combustors. 105th ASME Winter Annual Meeting, Paper 84-JPGC-APC-2.

Crocco, L. & Cheng, S. I. 1956 Theory of combustion instability in liquid propellant rocket motors. *AGARDograph* **8,** London: Butterworths.

Crow, S. C. & Champagne, F. H. 1971 Orderly structure in jet turbulence. *Journal of Fluid Mechanics* **48,** 547–591.

Ffowcs Williams, J. E. & Huang, X. Y. 1989 Active stabilisation of compressor surge. *Journal of Fluid Mechanics* **204,** 245–262.

Gutmark, E., Parr, T. P., H anson-Parr, D. M. & Schadow, K. C. 1989 Stabilisation of combustion by controlling the turbulent shear flow structure. *Proceedings 7th Symposium on Turbulent Shear Flows*, Stanford University, 21–23 August, Paper No. 23(1), pp. 1–6.

Heitor, M. V., Taylor, A. M. K. P. & Whitelaw, J. H. 1984 Influence of confinement on combustion instabilities of premixed flames stabilised on axisymmetric baffles. *Combustion and Flame* **57,** 109–121.

Ho, C. M. & Huang, L. S. 1982 Subharmonics and vortex merging in mixing layers. *Journal of Fluid Mechanics* **119,** 443–473.

Huang, X. Y. & Weaver, D. S. 1991 On active control of shear layer oscillations across a cavity in the presence of pipeline acoustic resonance. *Journal of Fluids and Structures* **5,** 207–219.

Keller, J. O. & Westbrook, C. K. 1986 Response of a pulse combustor to changes in fuel composition. *Proceedings 21st International Symposium on Combustion*, pp. 547–555, The Combustion Institute, Pittsburgh, PA, U.S.A.

Longhorne, P. J., Dowling, A. P. & Hooper, N. 1990 Practical active control system for combustion oscillations. *Journal of Propulsion Power* **6,** 324–330.

Mcmanus, K. R., Poinsot, T. & Candel, S. M. 1991 A review of active control of combustion instabilities. *Proceedings of International Symposium on Pulsating Combustion*, Paper GI, pp. 1–34, Monterey, CA, U.S.A.

Price, E. W. 1969 Recent advances in solid propellant combustion instability. *Proceedings 12th International Symposium on Combustion*. pp. 101–113, The Combustion Institute, Pittsburgh.

Putnam, A. A. 1971 *Combustion Driven Oscillations in Industry* (ed. I. M. Beer), New York: Elsevier.

Putnam, A. A., Murphy, M. J. & Locklin, D. W. 1985 Burner Technology Bulletin: Control of Combustion noise for small gas burners. Report GR1-85/210, *Gas Research Institute*, Chicago, Il., U.S.A.

Lord Rayleigh 1945 *The Theory of Sound*, Vol. 2. New York: Dover.

Rockwell, D. 1990 Active control of globally unstable separated flows. *Proceedings of ASME Symposium on Nonsteady Flows* (eds J. A. Miller & D. P. Telionis) pp. 379–394, Toronto, 3–9 June 1990.

Schadow, K. C., Hendricks, E. W. & Hansen, R. J. 1992. Recent progress in the implementation of active combustion control. ICAS paper no. 92-2.5.3, pp. 942–952.

Sivasegaram, S. & Whitelaw, J. H. 1987 Suppression of oscillations in disk-stabilised flames, *Journal of Propulsion and Power* **3,** 291–295.

Sivasegaram, S. & Whitelaw, J. H. 1992 Active control of oscillations in combustors with several frequency modes. In *Active Control of Noise and Vibrations* (eds C. J. Radvliffe, K. W. Wang, H. S. Tzou & E. W. Hendricks), ASME Publications DSC-38, pp. 69–74, New York: ASME.

Yu, K., Trouve, A. & Candel, S. M. 1991 Combustion enhancement of a premixed flame by acoustic forcing with emphasis on role of large scale vertical structures. 29th Aerospace Sciences Meeting, Paper AIAA-91-0367.

Ziada, S. 1995 Feedback control of globally unstable flows: Impinging shear flows. *Journal of Fluids and Structures* **9,** 907–923.

Ziada, S., Oengoeren, A. & Vogel, H. H. 1996 Acoustic resonance in the combustion conduits of a steam locomotive. In *Flow Induced Vibration*-1996 (ed. M. J. Pettigrew), ASME Publication PVP-328, pp. 425–437. Also in *Journal of Fluids and Structures* **12,** 409–425.

Zinn, B. T. 1992 Pulsed combustion: recent applications and research issues. *Proceedings 24th International Symposium on Combustion*, pp. 1297–1305, The Combustion Institute.

# APPENDIX: ACOUSTICAL MODEL OF THE BOILER

The burner/boiler system is acoustically modelled by a network of pipe elements. Each element is represented as a two-port transmission line, as shown in Figure A.1. At each node (1–13), the acoustic pressure must be identical for all adjacent elements and the mass flux must be balanced.

The response is computed at each node, using the transfer function of the corresponding element. Calculation is started at the draft reducer and progresses backwards to the air intake, where the excitation is assumed to be located.

At the draft reducer (open end, node 1) the radiation impedance is specified as $0.2c/A$, where $c$ is the speed of sound and $A$ is the cross-sectional area. This parameter provides some attenuation and was selected in order to match the measured frequencies at cold conditions.

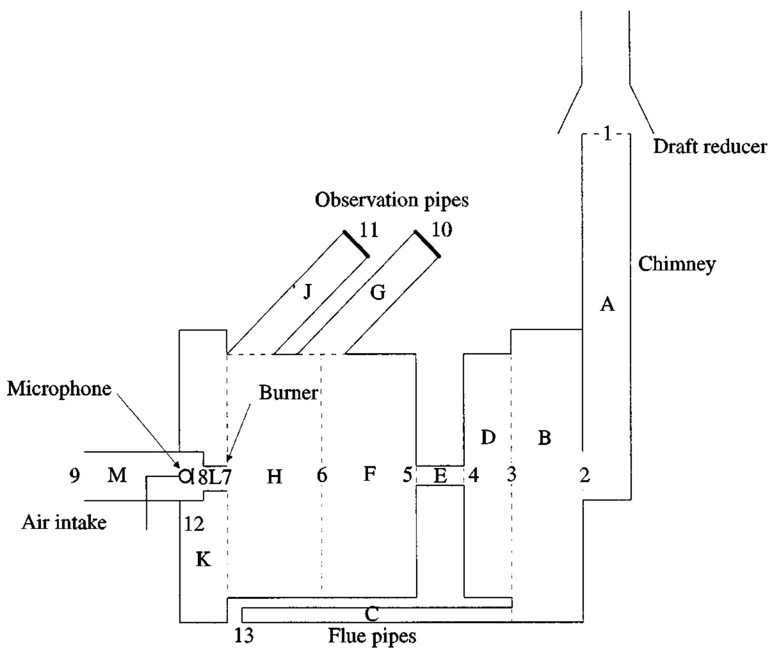


Figure A.1. Schematic of the acoustic model of the boiler; the pipe elements are denoted by letters and the nodes by numbers.

Table A.1
Numerical parameters for the model

| Pipe element | A | B | C | D | E | F | G | H | J | K | L | M |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Temperature (°C) | 180 | 180 | 60 | 200 | 220 | 1000 | 80 | 1000 | 80 | 1000 | 20 | 20 |
| Length (mm) | 2500 | 190 | 460 | 105 | 40 | 180 | 510 | 150 | 510 | 60 | 10 | 500 |
| Cross-section ($mm^2$) | 13 273 | 96 211 | 1590 | 39 761 | 3421 | 39 761 | 3632 | 39 761 | 3632 | 96 211 | 491 | 6940 |
| Adsorption (in % of standard pipe absorption)* | 100 | 100 | 400 | 100 | 800 | 100 | 100 | 100 | 100 | 100 | 800 | 100 |

*The values higher than 100% are due to the smallness of the hydraulic diameter.