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Experimental study of periodic heat transfer coefficient in the entrance zone of an exhaust pipe

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Abstract

The application of new standards of pollutant gas emission restrictions has forced the motor-car industry to improve their level of knowledge on heat transfers occurring between intermittent gas flows and exhaust system pipes. In this paper we present a study that we carried out in order to estimate the heat transfer coefficient in exhaust pipes. We developed an experimental device which re-creates engine working conditions. This experiment set up has been designed in order to check all assumptions of the theoretical model. Measurements are carried out with heat flux sensor, gas temperature probes and pressure sensor in the entrance zone of a cylindrical exhaust pipe. First results show that the heat transfer coefficient estimated in the case of an intermittent gas flow is higher than the one measured on a continuous flow with identical inlet conditions: same mass flow and inlet temperature. Measurements show that, for a given flow rate, the intensification of the heat transfer due to the flow intermittency corresponds to the eigen frequency of the exhaust pipe.

Keywords: Heat transfer coefficient; Intermittent flow; Entrance zone; Exhaust pipe; Frequency; Mean mass flow rate

1. Introduction

The application of new standards of pollutant gas emission restrictions has forced the motorcar industry to improve their catalytic exhaust system. In order to reduce the noxious gas emission, the exhaust gas temperature must be maintained into a specific range. Therefore, it becomes necessary to the well-know interaction between gas and inner wall along the whole exhaust system (from combustion chamber to monolith). Such local information can be obtained from numerical simulations computed on fluid mechanic solvers that take into account heat transfers between gas flow and solid wall. Due to the complex geometry of exhaust manifolds and the intermittent feature of the gas flow, the viability of numerical results is not guaranteed. Therefore, the majority of the work carried out on this subject was of experimental nature.

Up to now, the study of the fundamental mechanisms of heat transfers between an intermittent gas flow and a wall, in the case of the established periodic state, did not arouse interest. Thus, all the works listed in the literature are industrial ones. They essentially relate to the study of engine exhaust systems [1–3]. Most of the authors noted that the values of heat transfer coefficients measured on intermittent flows are higher than the ones given, for the same Reynolds number, by the traditional laws of forced convection. Therefore these authors introduced a new coefficient named CAF (Convective Augmentation Factor) that characterizes the intensification of heat transfers due to the intermittency of the flow [4,5]. One can note that, although the studied flows are periodic steady-state ones, all of these authors have only estimated average values of the heat transfer coefficient. Nevertheless it is expected that this coefficient will present the form of a periodic temporal law with an average value and a fluctuating one. Therefore the originality of the present study is mainly due to the instantaneous character of the heat transfer coefficient estimation.

This paper is divided into three parts. The first one deals with the presentation of the method we have selected in or-

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Nomenclature

C_v	specific heat capacity	$\mathrm{J\,kg^{-1}\,K^{-1}}$
D	diameter	m
h	heat transfer coefficient	$\mathrm{W\,m^{-2}\,K^{-1}}$
L	length	m
r	radius	m
T	fluid temperature	K
S	pipe cross section	$\mathrm{m^2}$
t	time	s
u	velocity	$\mathrm{m\,s^{-1}}$
x	axial position	$\mathrm{m^{-1}}$

Greek symbols

λ	thermal conductivity	$\mathrm{W\,m^{-1}\,K^{-1}}$
θ	temperature of the tube	K
ρ	density	$\mathrm{kg\,m^{-3}}$
τ	period	s

Subscripts

i	inner
o	outer
g	gas
c	thermocouple
w	wall

der to estimate the heat transfer coefficient. It is based on a coupled model which takes into account both conduction and convection phenomena. This model makes it possible to calculate for each section of the tube both the temperature in the wall and the mixed-temperature of the gas. In the second part, we present the new experimental device we designed for the needs of the study. In this section, we are going to pay here a particular attention on the thermal metrology we used. The last part of this paper deals with the estimation of the heat transfer coefficient in itself. We then present an example of results obtained on thermal transfers occurring between an intermittent gas flow and exhaust tube walls.

2. The theoretical model: Measurement principle of heat transfer coefficient

We develop a rather realistic model that deals with the characteristic of the entry region of our experimental device for that one controls the inlet conditions such as temperature and pressure levels. The aim of this model is to apprehend the influence of thermal and dynamical parameters of the experimental device on the heat transfer coefficient at the interface wall-flow in the two different cases: the case of reference of a continuous gas flow presenting some rate flow and the one of an established intermittent flow presenting a same mean rate flow. The estimated value of heat transfer coefficient in the case of intermittent flow is systematically compared to the convective heat transfer coefficient value measured on the continuous flow of reference (same mean mass rate flow). The comparison is more appropriate considering Nusselt number values. Before developing the principle of measurement, we have some opening remarks to make on the concept of heat transfer coefficient in the case of intermittent flow.

2.1. Heat transfer at the interface wall–intermittent gas flow

In the case of periodic intermittent flows, one cannot find in the literature a definition devoted to the heat transfer coefficient at the wall that is the equivalent of the convection coefficient in the case of continuous flows. In fact, there are no fundamental studies on this kind of flows, neither in the field of the fluid mechanics nor on that of the heat transfer [6,7]. In the absence of such a definition founded on theoretically established physical bases or by the experimental observation, the authors retain the model of conductance which allows the writing of a boundary condition of third kind to characterize the transfer to the wall. The few authors interested by the intermittent flows describe the heat transfer at the wall by either a constant heat transfer coefficient based on the mean mass flow rate or a heat transfer coefficient depending of the crank angle. Indeed, the thermal phenomena involved are highly dependent of the valve position [3].

A heat transfer coefficient is defined in the case of the thermal steady state. But it is commonly used for transient and periodic regimes since the time-constant of boundary layer is much smaller than the characteristic time of the studied system. If it is not the case we can consider a time dependent thermal conductance. It is the case in the present study. We do not need to know what can be the mechanical and thermal events during the share of the period when the valve is closed. These aspects are difficult to control and in our priority, we are interested initially by the intensity of the heat transfer to the wall compared to the transfer by convection having the same mean mass rate flow. So, we simply define in the entrance zone a heat transfer coefficient at the interface wall—intermittent flow $h(x,t)$ depending on time and axial position of dimension ($\mathrm{W/m^2/K}$) such as:

$$h(x,t) = \frac{\varphi(x,t)}{\theta(r_i,x,t) - T_f(x,t)} \tag{1}$$

where $\varphi(x,t)$ is the heat flux density ($\mathrm{W/m^2}$), $\theta(r_i,t)$ the inner surface temperature (K) and $T_f(x,t)$ the reference temperature that is the mixed-cup temperature of fluid (K) of the considered right section. Note that φ, θ and T_f are the main thermal conditions at the interface and the difference temperature in the denominator is always positive in the exhaust pipe application. In established periodic regime, the heat flux density presents two terms:

$$\varphi(x,t) = \bar{\varphi}(x) + \tilde{\varphi}(x,t) \tag{2}$$

$\bar{\varphi}(x)$ is the mean term and $\tilde{\varphi}(x,t)$ is the fluctuating one. This last does not have a physical significance because its integral over the period is null:

$$\int_0^\tau \tilde{\varphi}(x,t)\,\mathrm{d}t = 0 \tag{3}$$

The coefficient $h(x,t)$ is also periodic and present the same characteristics:

$$h(x,t) = \bar{h}(x) + \tilde{h}(x,t) \tag{4}$$

where $\bar{h}(x)$ is the mean term and $\tilde{h}(x,t)$ is the fluctuating one. Starting from this consideration most of the authors are interested by the mean term $\bar{h}(x)$, which is used to define a mean Nusselt number such as:

$$Nu = \frac{\bar{h}D}{\lambda_f} \tag{5}$$

The definition of $\bar{h}(x)$ remains the same but can take another form. Hemida [8] proposes the following expression:

$$\bar{h}(x) = (\lambda_f/\tau)\int_0^\tau \varphi\,\mathrm{d}t/(\bar{T}_w - \bar{T}_g) \tag{6}$$

where:

$$T_g = \frac{1/\tau}{\dot{m}}\int_0^\tau \varphi\left(\int_S \rho_f \vec{n} u T\,\mathrm{d}S\right)\mathrm{d}t \tag{7}$$

on the basis of such a definition, $\bar{h}(x)$ is consistent with Newton's law and with the resistance analogy developed for steady state flows.

2.2. Model's hypothesis

In order to estimate accurately $h(xt)$ from temperature measurements obtained on an experimental device that we have designed and constructed, we consider a theoretical model that describes the heat transfer in an exhaust pipe seat of an intermittent or continuous hot gas flow. We retained the following assumptions that are discussed farther:

1. The variations of the gas density are only due to the variations in its temperature.
2. The flow mixed-speed is the same in the whole tube.
3. The flow and heat transfers present an axial symmetry.
4. We neglect axial conduction in the flow.
5. We neglect axial conduction in the solid part of the model. The temperature part θ depend on x only because $h = h(x)$.
6. All materials present homogeneous and isotropic characteristics.

Assumptions 1 and 2 were verified in [9,10] by solving a coupled thermo-mechanic model. The 1D instantaneous compressible Euler equations in the fluid and the energy equation in both the fluid and the solid are solved. The solution of this problem shows that the pressure variation along the pipe represents 0.4% of the mean pressure value. Thus, the axial pressure gradient can be neglected. Hence, gas density variations are only due to temperature variations. In the same way, it is shown that the phase shift of the axial velocity is almost zero, and the amortisement along the pipe is 1.8%. At any time, the axial velocity along the pipe can be considered equal to the inlet velocity.

The experimental device has been conceived in order to create an axisymmetric flow in the tube. So the third assumption is experimentally verified.

Assumption 4 is consistent since the Peclet Number value is greater than 200. That is the case when we consider the mean value of the velocity to define the Reynolds number. The range value of Reynolds number is between 17 000 and 30 000. Prandlt number of air remains equal to 0.71. One can hold another reasoning which consists in saying: when the valve is closed, this assumption is still valid since the characteristic diffusion time D^2/a is greater than the intermittency period τ: $D^2/a \gg \tau$.

Assumption 5 was validated in previous work. Bourouga et al. [11] showed that in thermal entrance zone, in the restrictive case of a lumped pipe insulated on the outside face, axial conduction is negligible as soon as the distance from the flow entry is greater than one diameter. In the particular case of the present study, this hypothesis remains valid: the test of validity was also studied by Sorin and is presented in Ref. [9].

2.3. Model description

A general diagram of the model is represented on Fig. 1. Considering the previous assumptions, the heat transfer in the exhaust pipe can be modeled by the one-dimension conduction equation in the solid part and by a one-dimension energy transport equation in the gas flow. Considering the periodic regime and these sets of equations are the following ones:

For $r_i \leqslant r \leqslant r_o \cap 0 \leqslant x \leqslant L$

$$\frac{1}{r}\frac{\partial}{\partial r}\left(\lambda(\theta)\frac{\partial\theta(r,x,t)}{\partial r}\right) = C\rho(\theta)\frac{\partial\theta(r,x,t)}{\partial t} \tag{8a}$$

$$\lambda_s(\theta)\frac{\partial\theta(r,x,t)}{\partial r}\bigg|_{r=r_i} = h_i(x,t)\big(\theta(r_i,x,t) - T(x,t)\big) \tag{8b}$$

$$-\lambda_s(\theta)\frac{\partial\theta(r,x,t)}{\partial r}\bigg|_{r=r_o} = h_o(x)\big(\theta(r_o,x,t) - T_{\mathrm{out}}\big) \tag{8c}$$

$$\theta(r,x,t+\tau) = \theta(r,x,t) \quad \text{Periodic condition} \tag{8d}$$

And for $0 \leqslant r \leqslant r_i \cap 0 \leqslant x \leqslant L$

$$\rho_f\frac{\partial T(x,t)}{\partial t} + \rho_f u_{\mathrm{inlet}}(t)\frac{\partial T(x,t)}{\partial x} = \frac{4h_i(x,t)}{D(C_v)_f}\big(\theta(r_i,x,t) - T(x,t)\big) \tag{8e}$$

$$T(0,t) = T_{\mathrm{inlet}} \tag{8f}$$

$$T(x,t+\tau) = T(x,t) \quad \text{Periodic condition} \tag{8g}$$

The inlet velocity $u_{\mathrm{inlet}}(t)$ imposes the periodicity of the problem. It is determined with the knowledge of the cross section between the valve and its seat. The cross section depends on

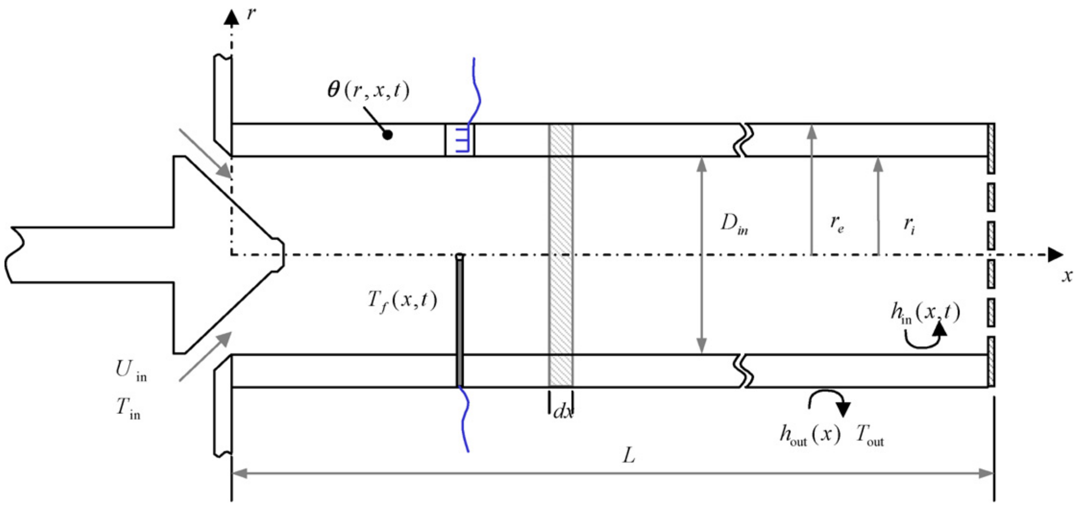

Fig. 1. Mathematic model.

the valve lift and the valve's geometric parameters and can be computed by formulas proposed by Heywood [3]. The valve lift is measured in order to compute the cross section, and the gas flow rate is considered proportional to this cross section.

The direct problem (8) is solved numerically using the finite volume method. The numerical resolution needs an initial condition that can be unspecified and problem (8) is solved as a transient problem. Periodicity conditions (8d) and (8g) are used as stop criteria for the computation of the solution.

The model needs the knowledge of the convective coefficient $h_o(x)$ that prevails on the outer side of the tube. Thus it has to be previously estimated from temperature measurements in the wall by means of a one-dimension conductive model in steady state.

2.4. The measurement principle

The measurement principle is founded on the analysis in established periodic regime of the measured temperature fields in the channel part and in the flow. This analysis is based on the resolution of the inverse problem (8) describing the coupled heat transfer in the exhaust pipe. The parameter to estimate is the function $h_i(x,t)$. This estimation needs temperature measurements in the solid that are carried out by means of heat flux sensors at some well-chosen locations along the channel. It needs also the temperature in the flow that is in measured at the inlet and in the channel at the same right sections where are implanted the heat flux sensors.

The inverse problem used to estimate $h_i(x,t)$ present in Eqs. (8b) and (8e) is solved thanks to the conjugated gradient method [13]. This last is a global method that considers the measurement of inlet temperature of the gas and measurements of temperature wall in different points of the entrance zone. The inversion procedure is tested and validated in the case of a continuous flow [12]. In the case of the intermittent flow, we observed that the solution of the inverse problem is very sensitive to the noise of measurement. Therefore we have to well design the experimental device and particularly the temperature sensors in order to get the highest measurement quality.

2.5. Measurement errors

Our estimation of the heat transfer coefficient, as any experimental measurement, meets all of its interest only if it is accompanied by an error analysis. We do not develop here the whole methodology that led to the results of this analysis. We just present its main ideas. For this calculation we make the assumption that all the error sources have the same probability of altering the estimation of the heat transfer coefficient. The error on the estimated parameter Δh can then be written in the following form:

$$\Delta h = \sqrt{\sum_{i=1}^{n}\left(\frac{\partial h}{\partial P_i}\Delta P_i\right)^2} \tag{9}$$

In the expression (9), P_i is the ith parameter on that h depends, ΔP_i is its uncertainty and n the number of parameters.

It is necessary to distinguish two categories in these parameters. The first one gathers the sources of error related to measurement. They are mainly the error sources related to temperature measurement. The second one gathers the sources of error on data such as error on thermal conductivity, on specific heat and on sensors position. This last category is associated to the error due to the indirect character of the estimation.

3. Experimental device

The experimental device was designed in order to meet the following conditions: to check the basic assumptions of the theoretical model, to allow an accurate control of the inlet conditions and to recreate the working conditions of a real engine. This experimental device is divided in two parts. In the first one, we create and control a source of hot compressed air that is used as the thermal excitation when we carry out measurements. The second one is an exhaust pipe that is instrumented with heat flux sensors, gas thermocouple probes and pressure sensors. A general view of the experimental device is shown on Fig. 2.

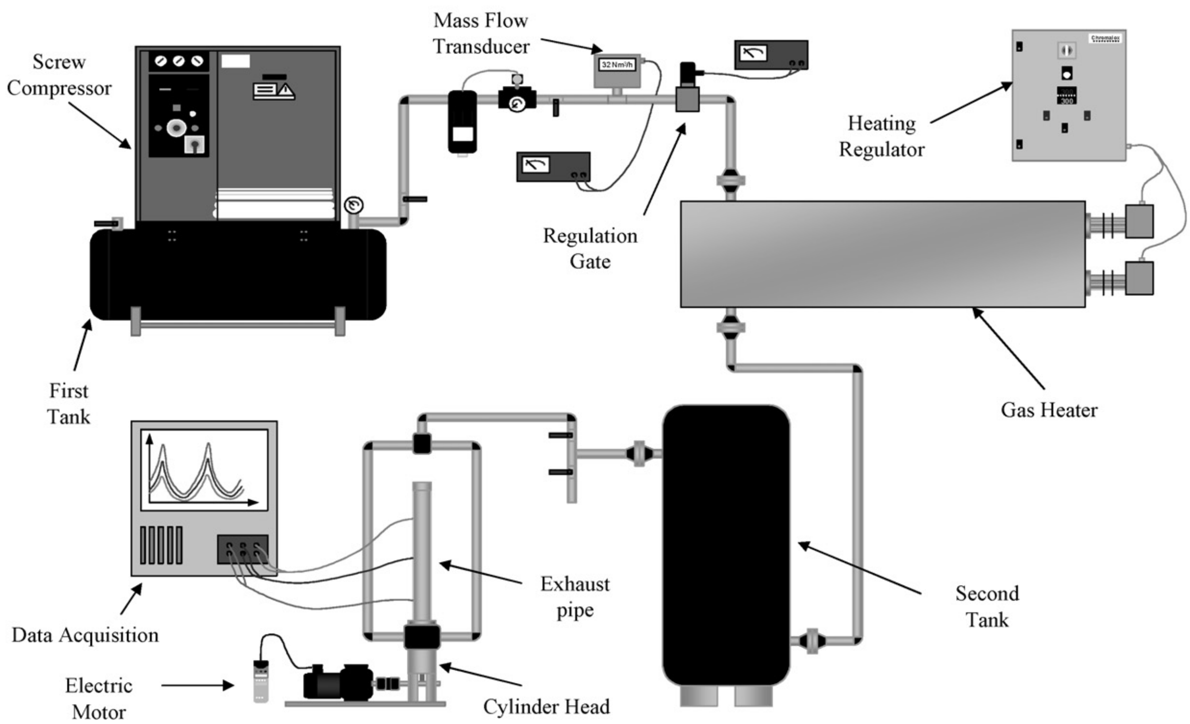

Fig. 2. General view of the experimental device.

The first part is mainly composed of an air compressor, a gas heater and a tank. The gas flow is realized with a pressure tank equipped with a valve dragged by a camshaft. The volume of the tank was calculated in order to have negligible variations of pressure during experimental process. It was also insulated to have a constant entry temperature of the gas flow. Note that it is really important to control all inlet conditions of the gas flow in order to be able to estimate their influence on heat transfers on the inner tube interface. Therefore we use two regulation systems in order to control the temperature and the pressure level at the entry of the tube.

We present on Fig. 3, the second part of the experimental device that consists of a hollow cylinder made of aluminium that has been instrumented. The instrumentation covers the entrance region of the exhaust pipe over a length of ten diameters. The geometric characteristics and properties of the pipe are given in Table 1. Fluid properties are given in Table 2.

In order to estimate the heat transfer coefficient several sensors and transducers are used to get the thermal information. First, we use a mass flow transducer to measure the average mass flow in the tube over a period. Next, we set up a temperature and a pressure sensor in the cylinder in order to measure the inlet conditions. A second pressure sensor next to the outlet of the exhaust tube gives the measure of the average pressure in the tube. A thermocouple is set up at the end of the tube next to the diaphragm.

Both heat flux sensor and gas temperature probe have been implanted in several suitable selected positions along the exhaust pipe. The measurements of the temperature in the solid part and in the fluid flow are really fundamental. So we have to choose the best way to realize these measurements. Therefore we improve the temperature measurement in the solid part by carrying out a theoretical study on the measurement error due to the intrusive character of the thermal instrumentation by thermocouple [9,10].

3.1. *Heat flux and surface temperature measurement*

A scheme of the heat flux sensor is given on Fig. 4. The sensor is composed of two half cylinders made of the same material as the channel. In order to estimate the heat flux that is assumed to be one-dimensional according to the radius of the channel, each sensor is equipped with three 50 μm diameter K-type thermocouples. Hot junctions of thermocouples are located on the axis of the sensor that follows the local radial direction of the tube. The wires of thermocouples are in grooves of 120 μm of width and depth. In order to minimize constriction effects, the groove receiving the first thermocouple (nearest from the surface to characterize) is machined at a distance of 8 groove diameters away from the inner surface, as recommended by Sorin et al. [14]. To reduce fin effects, the wires follow isothermal directions normal to the radius of the tube on a distance of about 40 thermocouple diameters or more.

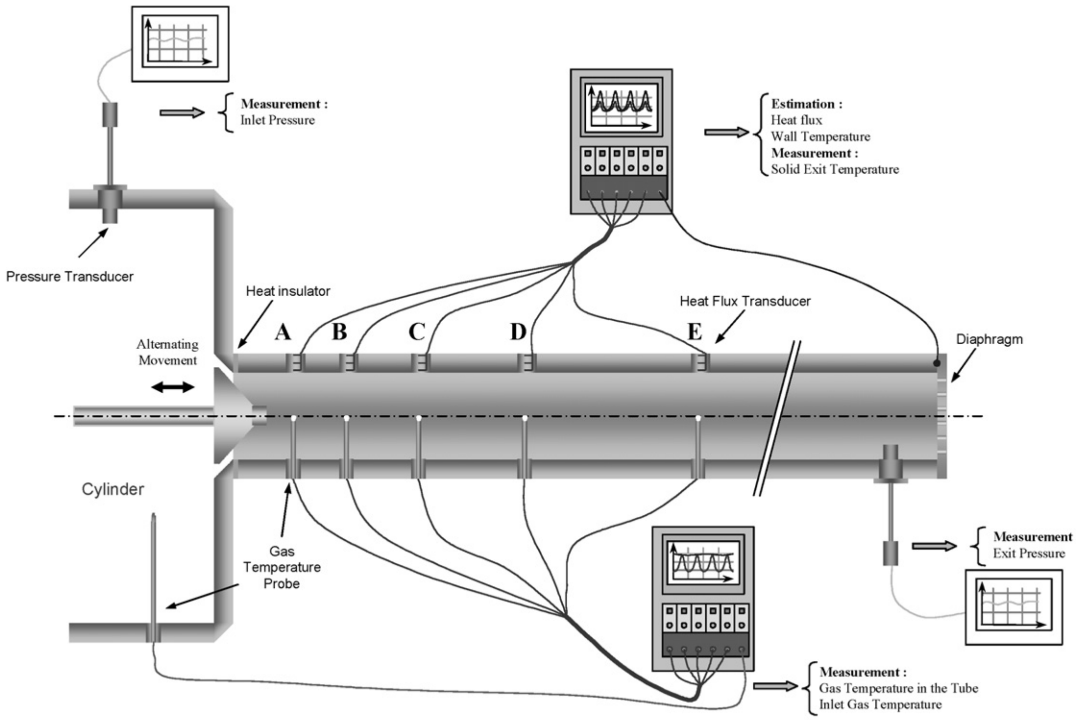

Fig. 3. Instrumentation of the exhaust tube.

Table 1
Pipe properties

Diameter D (m)	Length L (m)	Thermal conductivity λ ($\mathrm{W\,m^{-1}\,K^{-1}}$)	Density ρ ($\mathrm{kg\,m^3}$)	Thermal diffusivity a ($\mathrm{m^2\,s^{-1}}$)
0.03	0.6	200	2500	$8.867\mathrm{e}^{-5}$

Table 2
Gas properties

Thermal conductivity λ ($\mathrm{W\,m^{-1}\,K^{-1}}$)	Density ρ ($\mathrm{kg\,m^3}$)	Thermal diffusivity a ($\mathrm{m^2\,s^{-1}}$)	Kinematic viscosity υ ($\mathrm{m^2\,s^{-1}}$)
0.04	0.73	$5.4\mathrm{e}^{-5}$	$3.8\mathrm{e}^{-5}$

3.2. Temperature flow measurement

In the same section where a heat flux sensor is implanted, the mixed-temperature of the gas flow is measured on the axis of the tube with a two-thermocouple probe. The temperature measurement of the gas flow given by thermocouple is out of phase and amortized compared to the real gas temperature. In absence of radiation, the response of the thermocouple is a first order filter that can be expressed by:

$$T_g = T_c + \tau \frac{\mathrm{d}T_c}{\mathrm{d}t} \tag{10}$$

with

$$\tau = \frac{\rho_c c d^2}{4 Nu \lambda_g} = \frac{\rho_c c d}{4h}$$

In order to compute the real gas temperature T_g, the time constant τ must be estimated. τ varies with time throw h, and its measurement remains difficult. Different techniques have been set-up to evaluate τ, by means of two thermocouples probe [15–17]. If two thermocouples with different sizes measure the same temperature, their respective responses differ. For both thermocouples it can be written that:

$$\begin{cases} T_{g1} = T_{c1} + \tau_1 G_1 \\ T_{g2} = T_{c2} + \tau_2 G_2 \end{cases} \tag{11}$$

with

$$G_i = \frac{\partial T_{ci}}{\partial t} \quad (i = 1, 2)$$

Considering relation (11), we used Tagawa's method that computes the time constants of both thermocouples by minimizing the standard deviation between the two estimated temperature $e = \overline{(T_{g2} - T_{g1})^2}$. This can be expressed by:

$$\begin{cases} \frac{\partial e}{\partial \tau_1} = 0 \\ \frac{\partial e}{\partial \tau_2} = 0 \end{cases} \tag{12}$$

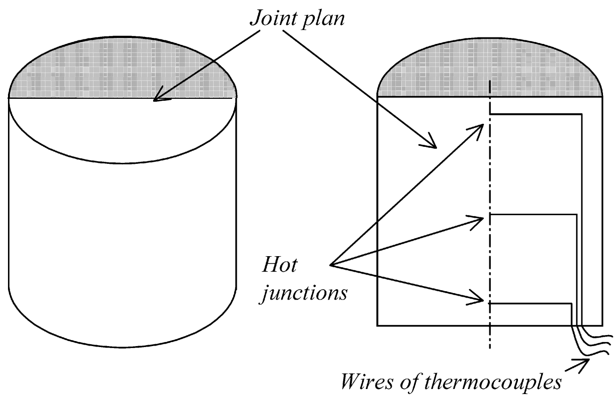

Fig. 4. Schema of the heat flux sensor.

Eqs. (11) leads to

$$\begin{cases} \tau_1 = \dfrac{\overline{G_1 \Delta T_{21}}\,\overline{G_2^2} - \overline{G_2 \Delta T_{21}}\,\overline{G_1 G_2}}{\overline{G_2^2}\,\overline{G_1^2} - \overline{G_1 G_2}^2} \\ \tau_2 = \dfrac{\overline{G_1 \Delta T_{21}}\,\overline{G_1 G_2} - \overline{G_2 \Delta T_{21}}\,\overline{G_1^2}}{\overline{G_2^2}\,\overline{G_1^2} - \overline{G_1 G_2}^2} \end{cases} \tag{13}$$

with $\Delta T_{21} = T_{c1} - T_{c2}$.

Details of the calculation are given in Ref. [17]. Time constants being known, the real gas temperature can be estimated.

We hence developed a gas temperature probe, composed of two fine wire thermocouples of unequal diameter, which allow us to estimate the real gas temperature without the systematic error due to the thermal inertia of thermocouple. The diameters of these thermocouples are 25 and 50 μm. They are placed inside a ceramic tube with a low thermal conductivity in order to lower any fin effects. They are lead to the centre of the channel by mean of a thin cylinder of 1 mm diameter.

4. Experimental process and first results

4.1. Measurement example

In this section devoted to experimental results, we present an example of both solid and gas temperature measurements.

The following experiment was carried out for an average mass airflow of 11 g/s. The thermal excitation period was set to 0.08 s which corresponds to a camshaft speed of 750 Rpm.

On Fig. 5, we represent the temperature measurements obtained thanks to the thermocouples we have set into the wall at the axial position: $x^* = x/D = 2$. D is the inner diameter of the exhaust tube. These thermocouples are set up respectively at a distance of 0.6, 2.5 and 4.4 mm from the inner surface of the channel.

The fluctuations of temperature on the first thermocouple are about 0.35 °C, one thus expects to note fluctuations of temperature of about 0.5 °C at the surface. There is little noise in these temperature measurements so one should be able to carry out an estimation of the instantaneous heat transfer coefficient under good conditions.

We also represented the position of the valve (opened/closed) for the three periods. The x-axis represents the temporal variable that was put in a dimensionless form as follows: $t^* = t/\tau$ where τ is the period of the excitation. One can note that the deeper the thermocouple is, the more deadened and the more out of phase its response is.

On Fig. 6, we represent the gas temperature we measured on the axis of the tube in the same section as the previous one. These measurements were carried out by means of a temperature probe instrumented with two thermocouples of unequal diameters. This figure shows how important the error due to the thermal inertia of thermocouples is. Indeed, the thermocouple that has the bigger diameter presents fluctuations which seem to be more deadened and more out of phase than the other thermocouple ones. Therefore, these results allow us to check well that the thermocouple behaves as a low-pass filter [18]. We thus use a technique that makes it possible to rebuild the instantaneous gas temperature suggested by Tagawa [17]. This rebuilt temperature is represented with triangular shapes on the previous figure. Therefore we can conclude that if one carries out gas temperature measurements by means of a single thermocouple probe, he will obtain a level of error that can reach 10%.

From these measurements, the surface heat flux density and surface temperature can be estimated. The surface heat flux density and the surface temperature are respectively represented on Figs. 7 and 8. As expected, a high increase in heat flux happens when the valve is open (dimensionless time <0.33). Note that the mean value of heat flux density is about 40 kW/m^2 and present a large peak-to-peak fluctuation going from 15 to 92 kW/m^2. When the valve is closed (dimensionless time >0.33), the heat flux decreases to a nearly constant heat flux. The quality of this estimation is shown by the residuals at the first thermocouple location, represented on Fig. 9. The residuals are below 0.015 °C. This level value is very satisfactory because it is lower than the temperature measurement accuracy.

Firstly, the post-processing of the results of measurement of temperature fields in the wall and the flow is carried out by two different methods. The first that is a global method, considers only temperature measurement the inlet temperature in the fluid in addition to the temperature measurements in the wall. The reference temperature of the flow $T(x_j, t)$ appearing in the formula (1) is then calculated. The inverse technique used is that of the conjugated gradient. In the second method that is local, the real gas temperature $T(x_j, t)$ is rebuilt by Tagawa's method from the recordings obtained by the two thermocouples probe implanted in each instrumented section. The surface heat flux density and temperature are estimated by the inverse method of Beck by considering the recordings of temperature carried out in the wall, on the level of the same cross-section of the channel.

The objective of the comparison is to test the relevance of the estimation results of the coefficient of transfer. It can be checked on Fig. 10. The heat transfer coefficient estimated by means the retained model and considering the inverse technique of the conjugated gradient is represented with square symbols. The continuous line represents the local estimation of the heat transfer coefficient. On observes that the heat transfer coefficient estimated indifferently by one or the other method is presented in the form of a local periodic function. It can be split into a mean value and a fluctuation one. The heat transfer fluc-

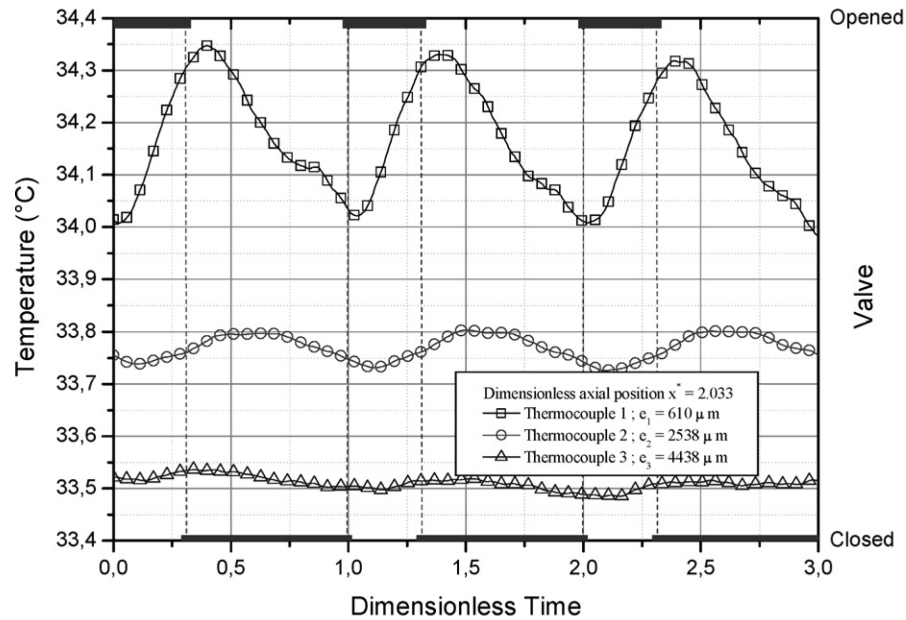

Fig. 5. Temperature measurement example in the solid part of the exhaust tube.

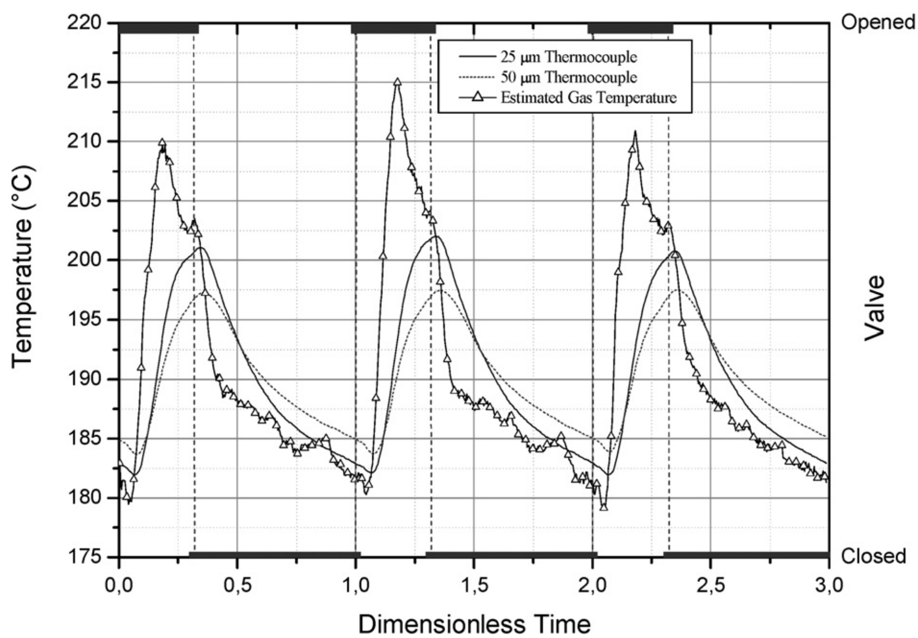

Fig. 6. An example of gas temperature measurement via a two thermocouples probe.

tuations take values from 30% below to 60% above the mean value. This implies a noticeable thermal penetration depth that has a strong influence on thermal wear of the exhaust pipe.

Note that both estimations of $h(x,t)$ coefficient are in good agreements. The two curves presenting the same shape are superimposed. The difference between the mean values the two estimations of the heat transfer coefficient is about 1.7%. This agreement between both methods characterizes a good relevance of the experimental approach. The advantage of the retained global model is that the reference temperature in the flow is estimated, whereas for the local estimation, it is a local temperature at the centre of the measurement section. Moreover, the retained model is time and cost economic in instrumentation since gas temperature probe are easily damaged by cyclic load of intermittent flow.

Beyond the relevance test, the second method appeared heavier to implement because of the difficulty of the measurement of the gas temperature. The reason is that the two thermocouples probe presents a weak resistance to the mechanical loads induced by the intermittency of the flow.

4.2. *Estimation of the heat transfer coefficient for an intermittent gas flow*

In this last subsection, we present the intensity of heat transfer at the interface wall-intermittent flow. To highlight the influ-

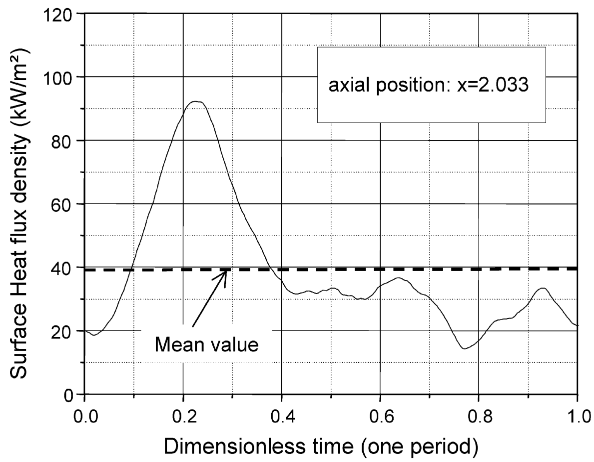

Fig. 7. Surface heat flux density for sensor C.

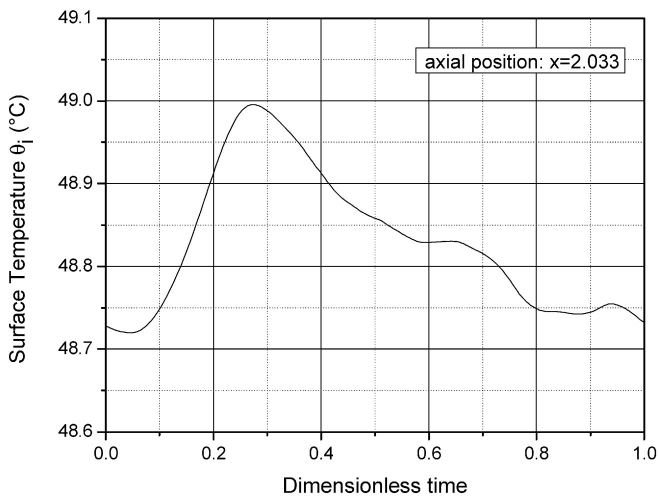

Fig. 8. Inner surface temperature for sensor C.

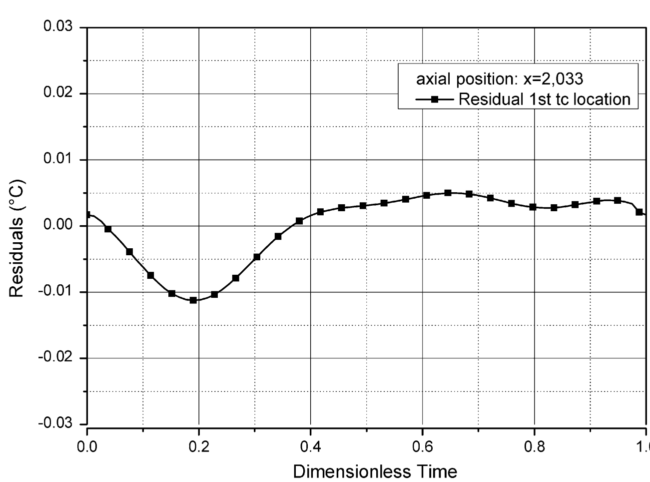

Fig. 9. Residuals at the location of the first thermocouple of sensor C.

ence of the flow intermittency on heat transfer coefficient, we carry out a measurement, for the same experiment conditions on a continuous flow: same inlet temperature and mass flow. The comparison of the values Nusselt number values allows appreciating the role of the flow intermittency.

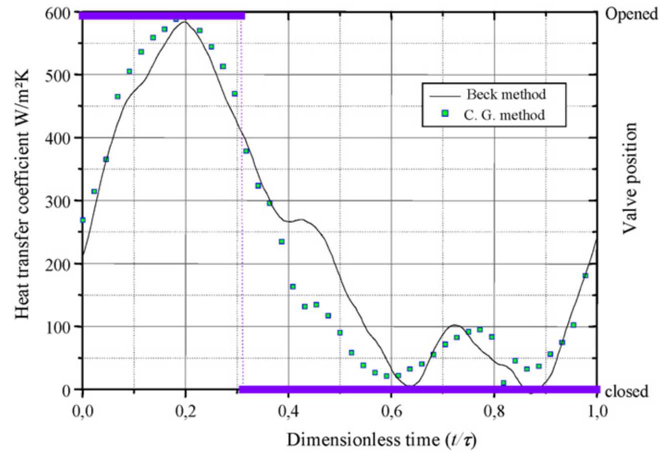

Fig. 10. Instantaneous heat transfer coefficient estimated by both global and Beck methods.

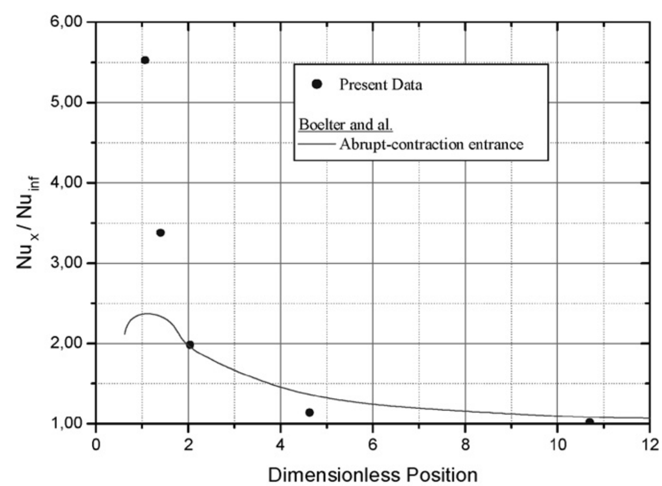

Fig. 11. Comparison of the estimated heat transfer coefficient for both continuous and intermittent flows.

First, the reliability of the experimental device is tested by comparing on Fig. 11 some obtained results with those obtained by Boelter et al. [19] for similar experimental conditions.

We note that our results are close to those referring to the abrupt-contraction entrance. However, it is important to note that the studied configuration is different from all of those studied by Boelter. Indeed, in present case, we have an obstacle at the inlet of the flow: the valve. We observe that the results we obtained for $0 < x/D_{\text{int}} < 1$ are higher than those of Boelter. This can be explained by the particular position of the measurement points and by the dynamics of the flow. Indeed, the valve that we use has a seat angle of 45°. Consequently the inlet velocity vector forms an angle close to 45° with the x-axis. The impact zone of the flow with the wall is thus around the axial position $x = D_{\text{int}}$ where we have set up two heat flux sensors. It is well known that heat transfers increase around the impact area of a flow. This explains the higher values of the heat transfer coefficient we obtained. To check this assumption, it would be necessary to get a representation of the instantaneous speed field in the tube by means of PIV or LDV.

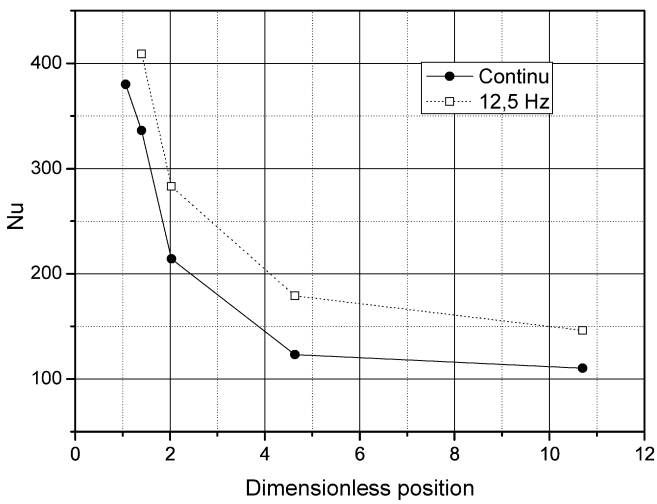

Fig. 12. Comparison of continuous flow results with Boelter et al. ones [13].

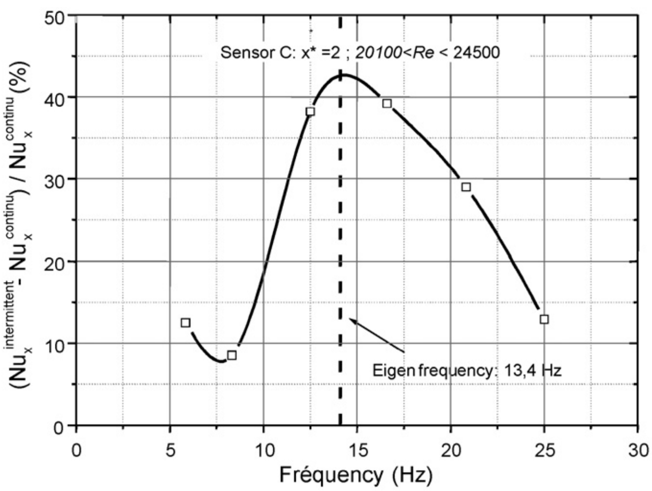

Fig. 13. Comparison of the estimated Nusselt numbers for both continuous and intermittent flows.

On Fig. 12, we present the heat transfer coefficient estimated successively for a continuous flow and for an intermittent flow. For this last, we consider the mean value. These two estimations were carried out under the same inlet conditions and with a same mean mass flow. The two curves present globally the same form. Therefore we conclude that the axial variations of the heat transfer coefficient only depend on the geometry of the entry. However, we note that the Nusselt number values obtained in both cases are rather different. Indeed, there is an increase of approximately 30% on the average value of the heat transfer coefficient when the flow is intermittent. When the valve is open, the intermittent flow has a higher speed than the one of its equivalent continuous flow. Therefore, the thickness of boundary layer is weaker in this case and there are higher heat transfers towards the wall. When the valve is closed, it is difficult to predict the behavior of the fluid. Does the closing of the valve induce an oscillation of the fluid in the channel? It is not easy to answer to this question. But if the velocity vanishes, the boundary layer vanishes also and heat transfers would be, in this case, due to conductive phenomenon. Considering the estimation results, one can note, as Heywood [3] did, that these latter are not negligible.

On Fig. 13, one observes that the curve of the intensification of the heat transfers related to the intermittency of the flow presents very significant variations according to the frequency of the intermittency. In the considered field of value, if one excludes the result corresponding to 8.33 Hz, one observes that intensification by intermittency in the flow grows, passes by a maximum that is around 14 Hz then decrease until a frequency of 25 Hz. This maximum of the increase of the heat transfer coefficient due to the intermittency in the flow is about 42% of the value of the convection coefficient, estimated in the case of the continuous flow.

This result confirms those of the bibliography concerning heat transfer in intermittent flow [3,4] studied on real engines. The original result of the present study is that the peak of increase happens at the device mechanical resonance frequency. This result differs from the results known for pulsating flow. Indeed, the maximum of the increase of heat transfer for pulsating flow happen at the acoustic pulsating flow. This may be also the case for intermittent flow, but the frequency range of interest is far from the pipe acoustic resonance. Anyway, one can thus suppose that if we use a thermal excitation which has a profile no far from a crenel, we will excite almost all the spectrum of the frequencies and, in particular, the device mechanical resonance one. Finally, the higher the amplitude of the excitation associated to the device resonance frequency is, the more the heat transfers increase.

5. Conclusion

We presented an original experimental approach aimed to study the solid–fluid interaction in the case of an intermittent gas flow. The experimental device is designed and constructed to re-create the main features of the compressible flow in engines.

The direct model describing the coupled problem makes it possible to develop an inverse problem in order to estimate heat transfer coefficient without any extra temperature sensor in the gas flow. We have carried out some experiments in order to check the reliability of our experimental device. Our first estimation for a continuous flow shows agreements with Boelter data. We have checked the relevance of our estimation by the means of a second inverse method developed by Beck.

The present results show that the intermittent feature of the flow does not change the repartition of the mean heat transfer coefficient in the entry region. However, it begets an increase of about 30% in its value. Measurements show that, for a given flow rate, the intensification of the heat transfer due to the flow intermittency corresponds to the eigen frequency of the exhaust pipe.

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