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Unsteady Contribution of Water Vapor Condensation to Heat Losses at Flame-Wall Interaction

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Abstract. This study addresses experimentally the influence of a second-order phenomenon concerning wall heat losses in combustion chambers, namely water vapour phase change, that is likely to occur subsequently to flame-wall interaction. For this purpose, flame quenching experiments are carried out on methane-air mixtures in a constant volume combustion chamber, using heat gauges based on surface thermometry. The transient effect of condensation on wall heat flux, that was already predicted using numerical simulation, is evidenced in our experiments. Its evolution and magnitude are quantified and compared to the relevant literature. Although the time evolution of condensation heat flux was poorly predicted by numerical simulation, its magnitude is well estimated thanks to mass diffusion across the quenched layer at the wall. In the conditions investigated, the contribution of condensation to wall heat flux reaches about 0.1 MW/m², which represents around 6% of conductive wall heat flux at quenching and up to 23% during the cooling phase after combustion.

1. Introduction
In most engine combustion chambers, flame-wall interaction (FWI) is ruled mainly by heat losses to the cold walls. During flame quenching, several processes are involved simultaneously: as the flame approaches the wall, heat flux is yielded by the flame to the wall until chemical extinction occurs. The thickness of the quenched layer of fresh gases, namely quenching distance, is also a key parameter of FWI. Indeed, wall heat flux and quenching distance have an impact on engine performance and emissions, especially at cold start conditions.

This situation is encountered as long as wall temperature does not exceed a temperature of 400–600K [1] or 623K [3], beyond which the wall can no longer be considered as an inert medium. Below a wall temperature of about 600K, flame quenching can be addressed without taking into account surface chemistry, which is usually referred to as “cold wall quenching”. Above a wall temperature of about 600K, surface reactions play a role in FWI through radicals and absorption-desorption processes.

In the past, the behavior of FWI has been investigated usually through experiments [3],[4],[6], numerical simulation [1],[2],[5],[7] or theoretical modeling [9]. However, predicting accurately wall heat flux and quenching distance in practical devices remains a challenge in most cases. This is noteworthy in engines, where the high levels of pressure and temperature increase the wall heat flux up to several MW/m², while decreasing the quenching distance down to a few tens of µm. Additional knowledge is therefore required to address FWI respectively in cold and hot wall conditions.

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As far as cold wall quenching is concerned, the flame is quenched mainly by thermal losses, because the extinction proceeds from a quenching of the thermo-diffusive zone by conductive losses to the wall. Therefore it is important to point out the separate contributions of combustion to wall heat flux, namely: heat conduction, radiation, and water vapor phase change. The first two contributions are well characterized during combustion, in terms of modeling and experimental measurements. This is not the case for the heat flux contribution due to water vapor condensation. This contribution has been quantified previously [7] using one-dimensional numerical simulation of propane-air combustion. The results of such simulation indicate a slight increase of wall heat flux due to condensation (+5%), that is simultaneous to flame extinction. Although this increase is limited in magnitude, it is necessary to identify its effect experimentally, because it could validate the comparison between the experiments carried out in cold wall conditions [4],[6], and the numerical simulations carried out without computing water vapor phase change [1],[5].

The aim of our study is to quantify experimentally the time-resolved contribution of water vapor condensation on a cold wall of temperature 20–120°C, during the transient combustion of quiescent methane-air mixture.

2. Experimental setup and measurements

2.1. Experimental setup
The experimental device is a constant volume vessel dedicated to the combustion of a gaseous charge of air-fuel mixture. The internal volume of this 246 cc combustion chamber has a cubic shape with a pent-roof (see Figure 1). Indeed, this internal geometry is not specific for this study, but the chamber is a versatile device that allows different configurations for the study of constant volume combustion.

The chamber is fitted with quartz windows (HOQ310) that provide a whole visualization of the inner volume. In order to reach the desired wall temperature, the chamber is heated electrically by cartridge heaters, and the wall temperature is controlled by a PID regulator using a K-type thermocouple.

As mentioned above, the experimental setup and diagnostics are designed to evidence the effects of water vapor phase change on heat flux at FWI. For this purpose, combustion must be performed in targeted conditions.

In order to reach a well-defined composition of the combustion products, a stoichiometric mixture has been chosen, so that only water vapor, carbon dioxide and nitrogen are produced. This condition is referred to as a unity equivalence ratio, $\phi = 1$. The complete combustion of the stoichiometric methane-air mixture leads to the following equation at standard temperature ($T = 25^\circ$C):

\begin{align*}
\text{Heat} & = \text{g} + 2\text{H}_2\text{O} + \text{CO}_2 + \text{N}_2
\end{align*}
\[ CH_4 + 9.52(0.21O_2+0.79N_2) = 2H_2O + CO_2 + 7.52N_2 \]  

(1)

In such conditions, the mole fraction of H\textsubscript{2}O is 0.19. Moreover, this reaction is nearly conservative of the mole number from reactants to the final products: thus, the initial and final pressures are expected to be equal.

Referring to the phase equilibrium diagram of water, the initial pressure of the mixture is chosen so that the transition between condensation and vaporization can be observed in the middle of the wall temperature range, i.e. 20–120°C. The initial pressure of the mixture is fixed at 0.1 MPa, which leads to a partial pressure of H\textsubscript{2}O equal to 0.019 MPa [8]. The saturation pressure of water is close to 0.02 MPa at 60°C, thus the transition is expected to occur nearby this level of wall temperature.

2.2. Experimental measurements

The chamber is instrumented by a capacitance pressure transducer (MKS 722B, 0.5% uncertainty), a piezoelectric pressure transducer (Kistler 601A, 0.5% uncertainty), and a couple of heat flux gauges based on surface thermometry. The first one (gauge A) is well-suited to resolve the dynamics of FWI [4],[9],[10], and consists of a surface thermocouple placed on a body made of steel. However, this device is no longer manufactured, nor commercially available. This is the reason why another gauge is presented in this study, the features of which are believed to be close to that of gauge A. This second gauge (gauge B) is made of a thin film thermocouple formed by erosion at the surface of a stainless steel body. This type of "eroding thermocouple" is also reported as a fast-response sensor [11],[12]. Compared to gauge A that is made of steel, gauge B is of the same material as the chamber wall (see Table 1). The heat gauges are placed symmetrically in the chamber, so that FWI occurs simultaneously on both with the same pressure, gas temperature, and flame geometry (see Figure 1). Experimental signals are recorded at a rate of 100 kHz thanks to a digital oscilloscope (Lecroy Waverunner 104Xi).

| Table 1: Properties of the heat gauges |
|---------------------------------------|
| Heat gauge   | A          | B          |
| Reference    | CFTM       | Nanmac E12-3 |
| Thermocouple type | J      | E          |
| Body material | Steel H11  | Stainless steel 304 |
| Diameter     | 4 mm       | 3.2 mm     |

In this study, each experiment consists of the following: the chamber is heated up to the desired temperature, it is filled up with stoichiometric methane-air mixture at the desired pressure, then the charge is spark-ignited at the center of the chamber. In this study, experiments have been carried out at initial pressure 0.1 MPa and wall temperature ranging in 20–120°C.

For each experiment, the following signals are recorded: dynamic pressure \( P \), wall temperature \( T_w \) respectively from the J-type and from the E-type thermocouples. The computation of wall heat flux from surface temperature is done by considering the one-dimensional conduction of the heat wave in the wall, that is assumed to be a semi-infinite medium. The solution of this inverse conduction problem leads to a Duhamel integral that allows computing wall heat flux \( Q_w \) from surface temperature:

\[
Q_w(t) = \sqrt{\frac{\lambda \pi}{\rho c_p}} \int_{t=0}^{t} \frac{dT_x(\tau)}{\sqrt{t-\tau}} d\tau
\]

(2)

This formulation points out the influence of the thermal effusivity of the wall, \( \varepsilon=(\lambda \rho c_p)^{0.5} \), in the dynamics of unsteady heat conduction; in this equation \( \lambda \), \( \rho \) and \( c_p \) denote respectively the wall...
conductivity, density and specific heat. Therefore, computing wall heat flux requires a good knowledge of the wall properties inside the heat gauge. This is of significant interest because gauge A and B are not made of the same material: as far as effusivity is concerned, the steel of gauge A is characterized in only one study [13], whereas the stainless steel of gauge B is well known. The comparison of their thermal properties shows a higher conductivity for gauge A, hence a higher effusivity (see Figure 2).

![Figure 2: Effusivity of heat gauge materials versus temperature.](image)

2.3. Heat flux processing and uncertainty

The computation of wall heat flux is performed using a numerical discretization that converts the integro-differential equation into a finite difference equation [14]. Obviously, the computation of heat flux is very costly due to the integral shape of Equation (2), that takes into account the whole history of heat flux along time:

$$Q_w(t_n) = \frac{2}{\sqrt{\pi}} \sqrt{\frac{\lambda \rho c_p}{\Delta t}} \sum_{i=1}^{n} \frac{T_{w,i} - T_{w,i-1}}{\sqrt{n-i+1} + \sqrt{n-i}}$$  \hspace{1cm} (3)

In the above expression, the wall temperature $T_w$ is discretized along time $t$, with $n$ time steps of duration $\Delta t$. In the case of our heat gauges, the temperature of surface thermocouples is not converted by an external device, but it is computed directly from the thermocouple voltage $U$, with an amplifier of gain $G$:

$$\delta T_w = G \cdot S \cdot \delta U$$  \hspace{1cm} (4)

Equation (3) then reads:

$$Q_w(t_n) = \frac{2}{\sqrt{\pi}} \frac{\varepsilon G S}{\Delta t} \sum_{i=1}^{n} (U_i - U_{i-1}) \left(\sqrt{n-i+1} - \sqrt{n-i}\right)$$  \hspace{1cm} (5)

As far as uncertainty is concerned, it is clear that this equation can be formulated another way, in order to simplify the uncertainty evaluation:

$$Q_u(t_n) = \frac{2}{\sqrt{\pi}} \frac{\varepsilon G S}{\Delta t} \left[-U_0 \left(\sqrt{n} - \sqrt{n-1}\right) + \sum_{i=1}^{n-1} U_i \left(\sqrt{n-i+1} - 2\sqrt{n-i} + \sqrt{n-i-1}\right) + U_n\right]$$  \hspace{1cm} (6)

Indeed, this highlights the fact that only individual values of voltage measurement contribute to the overall uncertainty measurement.
This formulation allows assessing the experimental uncertainty of heat flux measurement, using the quadratic method:

\[
\frac{\delta Q_w}{Q_w} = \frac{\delta \varepsilon}{\varepsilon} + \frac{\delta G}{G} + \frac{\delta S}{S} + \frac{\delta \Delta t}{\Delta t} + \left( \frac{\delta (\Sigma U)}{\Sigma U} \right)
\]

(8)

This introduces the individual uncertainty of each voltage measurement, \(\delta U\):

\[
\frac{\partial Q_w}{Q_w} = \frac{\delta \varepsilon}{\varepsilon} + \frac{\delta G}{G} + \frac{\delta S}{S} + \frac{\delta \Delta t}{\Delta t} + (n + 1) \left( \frac{\delta U}{\Sigma U} \right)
\]

(9)

It is clear that, through \(n\), the heat flux uncertainty depends mainly on the integration time that is required in the experiments. In our case, this time is linked to the duration of two phenomena, i.e. the combustion of fresh gases and the cooling of burned gases. Nevertheless, all sources of uncertainty are quantified below.

| Symbol | Parameter | Order of magnitude: \(X\) | Uncertainty: \(\delta X\) |
|--------|-----------|--------------------------|--------------------------|
| \(\Delta t\) | Oscilloscope time step | \(10^{-3}\) s | 10 ppm |
| \(\varepsilon\) | Thermal effusivity | \(10\) kJ/K.m².s\(^{\frac{1}{2}}\) | 2 % |
| \(G\) | Amplifier gain | 500 | 5 |
| \(S\) | Seebeck coefficient | 50 µV/K | 1 µV/K |
| \(U\) | Oscilloscope voltage | 800 mV | 0.4 mV |

It is difficult to quantify the precision of thermal effusivity, due to the lack of experimental discussion in the literature; this is the reason why 2% was chosen. The other uncertainties are quantified from the specifications of experimental devices. Finally, the experimental uncertainty of both heat gauges depends mainly on the number of time steps required for heat flux computation, \(n\). In our experiments, the combustion duration is of about 25 ms and wall heat flux is computed during 10 ms before FWI, which leads to \(n \sim 1000\):

\[
\left| \frac{\delta Q_w}{Q_w} \right| = 0.045
\]

(10)

This value of 4.5% is a theoretical uncertainty. It proceeds from our data processing method that accumulates the intrinsic uncertainty of each temperature measurement, under assumption that all measurements are independent from each other. Nevertheless, it should be noticed that the actual uncertainty is really close to 5%, because our measurement methodology for heat flux has been validated previously with a good agreement versus Direct Numerical Simulation (within 10%) in the case of transient premixed combustion [4].

3. Results and discussion

3.1. Comparison of heat gauges
As a preliminary test, the heat gauges are used simultaneously in order to check their ability to capture the dynamics of flame extinction. For this purpose, the evolution of wall heat flux from both heat gauges is compared during an experiment, in the conditions selected for this study (see Figure 3).

As a result, both gauges provide the usual evolution of heat flux during flame extinction. Firstly, the pressure increases under the effect of flame expansion, so that the temperature of unburned gases increases, hence a slight increase of wall heat flux up to 0.1 MW/m² until \( t \sim 20 \text{ ms} \). Secondly, the heat wave yielded by the flame to the wall during FWI generates a steep increase of wall heat flux, hence a peak of about 1 MW/m² at \( t = 25 \text{ ms} \). Thirdly, wall heat flux decreases strongly down to 0.1 MW/m² due to the slow cooling of burned gases.

This typical behavior is observed for both heat gauges. However, compared to gauge A, the peak of wall heat flux is more elevated and narrow in the case of gauge B. Out of the peak, both heat gauges are roughly in agreement. Therefore, there is a difference in their dynamic response in the case of steep heat fluxes: gauge B seems to capture better a transient heat wave, whereas gauge A would rather smooth it. To check this assumption, the response of gauge B was artificially filtered using a moving average over 250 µs (see Figure 4). As a result, the heat flux trace of both gauges compares quite well, as if gauge B had reached numerically the same response time as gauge A. This comparison highlights the benefit of gauge B in terms of dynamic response time, which may be due to a thinner surface thermocouple compared to gauge A. Indeed, the surface junction of gauge A is a “classical” thermocouple mounted flush at the wall surface, with a dimension a few tenths of mm, whereas the junction formed by erosion at the tip of gauge B may be really sensing the surface. However, the comparison also points out the weakness of gauge B in terms of signal to noise ratio: for the same heat wave due to flame quenching, gauge B leads to a measurement noise that reaches twice that of gauge A. This prevents us from being confident in the peak value at FWI, for instance. For this reason, in the following the study is carried out using gauge A, because gauge B is too much noisy despite a better response time.

3.2. Measurement of water vapor condensation heat flux
Experiments have been carried out in the conditions defined to capture the condensation phenomenon: it consists of a spark-ignited combustion of quiescent methane-air mixture of equivalence ratio \( \Phi=1 \), and initial pressure 0.1 MPa. The initial temperature of the unburned mixture is fixed by the wall
temperature, that is controlled in 20–120°C. For each condition, a set of 8 experiments is recorded. As stated earlier, heat flux measurements are carried out using gauge A.

As an example, an experimental recording is presented (see Figure 5). The pressure trace shows the development of the premixed flame which expansion heats the unburned mixture through isentropic compression. In the meantime, the wall temperature rises slightly during the pressure growth, but increases dramatically when FWI occurs, at $t \sim 19$ ms. When the maximum pressure is reached, at $t \sim 21$ ms, the combustion phase is followed by a cooling phase. Although the wall temperature goes on rising, the wall heat flux decreases monotonically. Overall, during such an experiment the pressure reaches around 0.5 MPa, the wall temperature rises by around 5°C, and the wall heat flux peak is of around 1 MW/m².

![Figure 5. Experimental recording during a single experiment – CH₄-air @ $\phi=1$, 0.1 MPa, $T_w=46°C$.](image)

During this experiment, one could watch some water vapor condensate over the quartz windows of the combustion chamber (see Figure 6). In the conditions of this experiment indeed, the wall temperature (46°C) was below the saturation temperature. In other conditions of wall temperature (80°C), this effect was not observed (see Figure 7).

![Figure 6. Condensation of water vapor during a single experiment – CH₄-air @ $\phi=1$, 0.1 MPa, $T_w=46°C$.](image)

![Figure 7. No condensation during a single experiment – CH₄-air @ $\phi=1$, 0.1 MPa, $T_w=80°C$.](image)
From this observation, 12 series of 8 experiments have been performed with a wall temperature ranging in 20–120°C. Then, for each condition of wall temperature, the experiments are averaged by using a phase averaging, i.e. by shifting the signals over time so that all peaks of heat flux occur at the same relative time.

As the wall temperature increases, the combustion duration decreases due to the thermal increase of flame velocity, and FWI occurs earlier (see Figure 8). This graph highlights a continuous change in the shape of the heat flux curves: for wall temperatures below 50°C, the peak due to FWI is followed by a monotonic decrease of the heat flux curve, with an inflexion point. This is no longer right as soon as the wall temperature exceeds 65°C. This inflexion point is due to an additional amount of heat that is supplied by the combustion products after FWI: this phenomenon is believed to be the result of water vapor phase change, in agreement with the above visualization.

This is an experimental confirmation that in our case, the partial pressure of water vapor (0.02 MPa) is such that the saturation temperature is comprised in 50–65°C, in agreement with the saturation temperature $T_{sat} = 60°C$ predicted by thermodynamic data.

![Figure 8. Time evolution of the phase-averaged wall heat flux versus wall temperature – CH₄-air @ Φ=1, 0.1 MPa.](image)

In order to quantify the fraction of wall heat flux that is due to condensation, the difference between heat flux traces is investigated. For this purpose, two conditions are considered: one condition below the saturation temperature, $T_w = 46°C$, and another one above, $T_w = 74°C$. These two conditions are selected because they offer a sufficient difference with regard to $T_{sat}$, of around 15°C. For the sake of clarity, both curves have been shifted in phase with FWI (see Figure 9). As a result, both curves are the same before FWI, whereas a substantial difference, $ΔQ_w$, is observed at FWI ($t \sim 17$ ms) and during the cooling phase ($t > 20$ ms).
The increase in $\Delta Q_w$ observed during the peak of heat flux is due to the slight variation of pressure between the two conditions: indeed, with the same initial pressure, 0.1 MPa, there is more mixture inside the combustion chamber in the “cold” condition, $T_w = 46^\circ$C, compared to the “hot” condition, $T_w = 74^\circ$C. Therefore, flame quenching occurs at higher pressure in the cold condition, 0.37 MPa, compared to the hot condition, 0.35 MPa. Since wall heat flux at quenching is known to increase with increasing pressure, this is the very behavior of FWI that explains this heat flux difference around the peak ($t \sim 17$ ms).

During the cooling phase, $\Delta Q_w$ has a slowly decreasing behavior, and reaches 52 kW/m² in average, with a maximum value of 69 kW/m². The evolution of $\Delta Q_w$ seems to follow that of the pressure curve: this suggests that the pressure magnitude could have an effect on this phenomenon, through the gas density that enhances the heat and mass transfer between the combustion products and the wall. It is also due to a saturation effect: as water vapor condenses on the wall, there is more H₂O at the wall and less in the burned gases, hence a decay in the condensation heat flux. In terms of magnitude, the amount of wall heat flux due to condensation during the cooling phase represents 5.8% of the peak of heat flux due to FWI. This value is close to that found in the 1-D numerical simulation of a stoichiometric propane-air mixture [7]: approximately 5%. The most significant aspect of condensation heat flux is its magnitude with respect to the conduction heat flux: it reaches up to 15% of the conduction heat flux during the cooling phase. Indeed, this ratio proceeds from a difference between two heat flux measurements, thus its intrinsic uncertainty is of around 9%, versus 4.5% for each heat flux measurement. Although its magnitude is not very precise, its order of magnitude is much more reliable.

The effect of water vapor phase change depends also on the magnitude of wall temperature. The evolution of $\Delta Q_w$ is shown in the condition of wall temperature equal to room temperature, $T_w = 20^\circ$C (see Figure 10). In this case, the difference between both curves is amplified: $\Delta Q_w$ reaches 73 kW/m² in average, with a maximum value of 92 kW/m². However, the pressure level is no longer the same (0.46 MPa versus 0.41 MPa), so these two conditions should be compared with caution. In terms of magnitude, the condensation heat flux reaches up to 23% of the conduction heat flux, which is very
significant. Thus, attention should be paid to the wall temperature when flame quenching is considered: heat flux measurements may depend heavily whether the wall temperature is below or above the saturation temperature of water vapor. This statement applies mainly for the cooling phase, i.e. after the heat flux peak due to FWI.

3.3. Theoretical estimation of water vapor condensation heat flux

Theoretically, the heat flux transferred to the wall during condensation can be evaluated simply by considering a simple model: the condensation heat flux would proceed from the water vapor that flows from burned gases towards the wall across the layer of unburned gases, under the effect of molecular diffusion [7]. Following this method, the condensation heat flux, \( Q_{w,H_2O} \), is estimated as the product of the mass flux of water vapor with the latent heat of vaporization, \( h_{lv} \):

\[
Q_{w,H_2O} = \rho_u \cdot \frac{D \cdot \Delta Y}{\delta_q} \cdot h_{lv} \tag{11}
\]

According to this theoretical approach, it is clear that the water vapor mass flux is ruled mainly by the difference of water vapor mass fraction between the burned gases and the wall, that is evaluated theoretically as \( \Delta Y = 0.12 \). The other parameter that governs \( Q_{w,H_2O} \) is the quenching distance that has been measured in similar conditions in a previous study [9], \( \delta_q = 85 \text{ \micro m} \). These two parameters allow determining the mass fraction gradient that produces the mass flux, by using the diffusion coefficient for water vapor in the quenched layer, \( D = 9.0 \cdot 10^{-6} \text{ m}^2/\text{s} \), and the density of unburned gases, \( \rho_u \).

The corresponding estimation is performed in the case of \( T_w = 20^\circ \text{C} \). It leads to a condensation heat flux of \( Q_{w,H_2O} = 133 \text{ kW/m}^2 \) according to Equation (11), which is 44% higher than the experimental value of 92 kW/m². Similarly in the case of \( T_w = 46^\circ \text{C} \), a condensation heat flux of \( Q_{w,H_2O} = 117 \text{ kW/m}^2 \) is computed from Equation (11), which is 33% higher than the experimental value of 69 kW/m². This can be explained as follows: the estimated heat flux is based on the assumption that all the water vapor transferred through mass diffusion would condense at the wall,
whereas several effects interfere. Indeed, the mass diffusion of water vapor is linked to the heat diffusion that modifies the properties of the quenched layer at the wall. As water vapor diffuses across the quenched layer, the composition and diffusivity of the quenched layer are modified by the water vapor concentration. Moreover the water droplets deposited at the wall contribute to the modification of conduction at the wall. Due to these phenomena, the mass fraction gradient across the quenched layer, $\Delta Y$, decreases along time. In the meantime, the length scale of this gradient, represented by the quenching distance, $\delta_q$, increases as the quenched layer gets thicker under the effect of cooling. The cumulated effects of these two parameters during the cooling phase result in a decrease of the condensation heat flux, $Q_{w,H2O}$, along time.

The condensation heat flux computed along time using Equation (11) remains almost constant during the cooling phase. A refined estimation should take into account the transient increase in water vapor inside the quenched layer, as well as the thickening of the thermal boundary layer during the cooling phase. In the aforementioned numerical simulation of flame quenching [7], the maximum of $Q_{w,H2O}$ was obtained simultaneously to the peak due to FWI. This was not realistic, because time is required for water vapor to diffuse across the quenched layer. In our case, experiments show that the phenomenon occurs a few milliseconds after FWI. To that extent, the predictions of such numerical simulation are less accurate than both the theoretical model and the present experiments.

3.4. Estimation of the final pressure

This study enables us to estimate the contribution of water vapor to wall heat flux when the wall temperature is lower than the saturation temperature, thus indicating that water vapor no longer contributes to the gas phase but condenses definitely at the wall. From this statement, it is clear that the final pressure inside the combustion chamber should reflect this decrease of the gas phase due to condensation. The final pressure, $P_f$, depends on the partial pressures of the combustion products:

$$P_f = P_{CO2} + P_{N2} + P_{H2O}$$  \hspace{1cm} (12)

The partial pressure of water vapor, $P_{H2O}$, is imposed by the saturation pressure of water vapor for a given wall temperature, $P_{sat,H2O}$. The partial pressures of the other products, $P_{CO2}$ and $P_{N2}$, can be expressed as a function of their mole fractions, respectively $x_{CO2}$ and $x_{N2}$. Under assumption that the combustion equation is conservative of the mole number, the final pressure depends on the initial pressure, $P_0$, and the saturation pressure of water vapor, $P_{sat,H2O}$:

$$P_f = (x_{CO2} + x_{N2}) P_0 + P_{sat,H2O}$$  \hspace{1cm} (13)

The final pressure measured inside our combustion chamber is reported along with the computation of Equation (13) for the conditions investigated:

| $T_w$ (°C) – initial | $T_w$ (°C) – final | $P_f$ (MPa) – experimental data | $P_f$ (MPa) – Equation (13) |
|----------------------|---------------------|--------------------------------|-----------------------------|
| 20                   | 26                  | 0.097                          | 0.084                       |
| 38                   | 45                  | 0.098                          | 0.090                       |
| 46                   | 53                  | 0.099                          | 0.095                       |
| 51                   | 58                  | 0.100                          | 0.099                       |

As the wall temperature increases, the final pressure at the end of combustion gets closer to the initial pressure, $P_0 = 0.1$ MPa. It is important to notice that the wall temperature increases during FWI and during the phase of gas cooling after combustion. Thus, the final pressure computed from Equation (13) takes into account the saturation pressure at final wall temperature, because it is the relevant condition after FWI (see Figure 5). The comparison of the measured and computed values of final pressure indicates that the experimental pressure is closer to $P_0$ than the final pressure computed.
from Equation (13). This may be due to an underestimation of the saturation pressure of water vapor, which may proceed from an underestimation of the average wall temperature throughout the chamber. Due to the heating systems and to different wall materials, the wall temperature and properties may not be homogeneous, which results in a non-uniform temperature distribution. Thus, the wall temperature indicated by the heat gauge may not represent at best the whole chamber, whereas the saturation pressure is given for the whole inner surface: this probably explains the difference between the experimental and computed final pressures.

4. Conclusion
As a conclusion, the main results of this study are summarized as follows:

- Two surface thermocouples proved to be well-suited for the time-resolved study of wall heat flux in transient combustion. In spite of a better bandwidth, the thin film thermocouple (Nanmac Co.) has a lower signal-to-noise ratio than the surface thermocouple (CFTM Co.). Both provide a physical description of unsteady heat transfer during flame-wall interaction, in the case of a stoichiometric methane-air flame.
- When the wall temperature decreases below the saturation temperature of water vapor, condensation appears on the walls of the combustion chamber; in the meantime, a change in the time evolution of wall heat flux is observed. The peak due to flame-wall interaction is not modified, but an increase in wall heat flux is observed a few milliseconds after it during the phase of heat transfer between the burned gases and the wall.
- This excess of wall heat flux is attributed to water vapor phase change at the wall; its magnitude is estimated experimentally to 69 and 92 kW/m² respectively for wall temperatures of 46°C and 20°C, for a pressure of about 0.4 MPa.
- This is the experimental validation of a phenomenon that had been predicted numerically and quantified theoretically using a simple mass diffusion approach. The theoretical model leads to a computed value overestimated by around 40% compared to the experimental measurement.
- In water vapor condensing conditions, the final pressure is inferior to the non-condensing final pressure, because the partial pressure of water vapor in the combustion products is limited by the saturation pressure, that depends itself from the wall temperature. The magnitude of this final pressure is computed and proves to be slightly inferior to the experimental measurements.

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