High-pressure EGR line condensation model of an IC diesel engine operating at cold conditions

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Abstract

To fulfill the future NOx emissions regulations, the use of exhaust gas recirculation (EGR) could be necessary from the engine starting at cold conditions. In this context, condensation inside the EGR line could appear, affecting in several ways to the engine components and their life span. In this paper, a mathematical model is developed with the aim of predicting the conditions that produce the condensation phenomenon to appear inside the EGR circuits of an internal combustion engine working at low ambient temperatures (-7ºC). In particular, the humidity ratio and the internal engine conditions that characterize the appearance of this phenomenon are estimated by the model. The model is validated experimentally visualizing the condensation behavior along a representative driving cycle by means of cameras fitted on the EGR rail. This validation process shows that the predictions made by the model are in good agreement with the results obtained from the experimental tests.

Keywords

Condensation, IC engine, EGR, Cold conditions, Warm up, NOx Reduction

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NOMENCLATURE

Acronyms

EGR	Exhaust Gas Recirculation
HP	High Pressure
LP	Low Pressure
ICE	Internal Combustion Engine
NOx	Nitrogen Oxides
CO_2	Carbon Dioxides
HC	Hydrocarbons
CFD	Computational Fluid Dynamics
1D	One Dimensional
ECU	Electronic Control Unit
DPF	Diesel Particulate Filter
FPS	Frames per Second
DPI	Dots per Inch

Notation

Latin

$\dot{m}$	Mass flow	kg/s
$mass$	Unified atomic mass unit	u
p	Pressure	bar
T	Temperature	°C
W	Humidity ratio	kg_{H2O}/kg_{dryair}

Greek letters

φ	Relative Humidity	--

Subscripts

exh	Exhaust gases side
in	Inlet conditions
out	Outlet conditions
$fuel$	Fuel
air	Air

1. Introduction

The type approval regulations applied to vehicles equipped with internal combustion engines (ICE) are becoming more stringent with the years. In this sense, the maximum pollutant emissions levels allowed are being reduced, together with the addition of new testing conditions as per example realistic and extreme driving conditions at very low temperatures [1]. By this reason, it is expected that this kind of approval procedures will take into account the engine cold start period in the future regulations. Apart from the imperative reduction in CO_2 emissions [2], the NOx emissions reduction in diesel engines will be a major concern in future regulatory stages. By this reason, strategies to reduce these emissions will continue taking precedence. Exhaust gas recirculation (EGR) is a widely used technique to fulfill the current NOx emissions regulations [3] [4], even at cold conditions [5].

The use of EGR to reduce the NOx levels presents some issues, as per example the condensation that can appear inside the components when the engine is running at low temperatures. The gas coming from the fuel combustion has important water content as well as other products of the combustion process [6]. Thus, if the gas temperature is reduced below the dew point, the partial condensation of the water vapor will occur. In this sense, condensation deposits may appear in the engine components through which the EGR flows, contributing to the fouling processes on these elements. Even more, with the moisture, some acids and hydrocarbon (HC) species coming from the fuel composition may condensate corroding these components and reducing their life [7] [8] [9]. Furthermore, these hydrocarbon species could affect in-cylinder processes during the combustion [10]. For instance, Furukawa et al. [11] studied the deposits resulting from high concentration of hydrocarbons due to the exhaust gas recirculation on control valves. The author identified the chemical components of each type of hydrocarbons present and their dew points in order to determine the conditions in which valve sticking occurs in engines. On the other hand, the liquid water injected into the cylinders can improve the engine behavior considering different injection-combustion strategies [12] [13].

The condensation phenomenon has been widely studied by many authors in the past, however the detailed models proposed in the literature are difficult to parameterize and validate [14]. This fact makes interesting the study of semi-empirical models for different cases.

In automotive applications, it is not usual that the EGR temperature falls below the dew point when a high pressure (HP) EGR layout is used. However, the condensation of the moisture contained in the burnt gas can occur during cold start phases [15]. As soon as the coolant temperature warms-up (≈80ºC) the condensation becomes almost unreachable [16]. By contrast, with a low pressure (LP) EGR arrangement, the condensation may occur even after engine warm-up, mainly when the humid stream of EGR is mixed with a cold (typically below 0ºC) stream of fresh air before the compressor [17] [18].

Some researchers have studied the condensation phenomenon inside the heat exchangers used to cool down the EGR in internal combustion (IC) engines applications. Yang et al. [19] presented a CFD model coupled with a 1-D heat and mass transfer model to study the most important physical aspects of the condensation process inside a typical EGR cooler in a heavy-duty diesel engine operating under off-road conditions. The model may be used to give accurate predictions of the water vapor, sulfuric acid and nitric acid corrosion in equipment with complicated geometry such as the EGR coolers. Warey et al. [20] developed a 1-D model to simulate the soot deposition, soot removal and condensation of several HC species inside EGR coolers operating at normal conditions. This model predicts the mass distribution in the deposit layer on the tube walls and the condensation of different HC species. Furthermore, these authors visualized the condensation of the water vapor and hydrocarbons at the outlet of the EGR cooler over a range of coolant temperatures in the EGR cooler, concluding that the condensation of water or hydrocarbons is not observed for coolant temperatures above 40 °C [21]. However, temperatures below 40 ºC produce the condensation of water and hydrocarbons, besides, soot formation inside the EGR cooler.

Many authors have investigated the soot formation inside the EGR coolers used in diesel engines, specially characterizing and analyzing its composition and features [22] [23]. , Few studies have been carried out regarding the condensation provoked by the EGR because of its complexity and magnitude of the work [24]. A recently work performed by Qiu et al. [25] present the in-cylinder condensation processes observed in optically accessible engine experiments and a phase transition model to determine when the mixtures coming from low temperature combustion become unstable and a new phase is formed. The experiments reveal the importance of fuel condensation on the emission characteristics of low temperature combustion.

According to the previous paragraphs, the study of the conditions in which condensation appears becomes especially interesting when a high-pressure EGR system is used in a diesel engine operating at cold conditions (-7 ºC). The objective of this work deals with this purpose, introducing a condensation model in order to estimate whether or not there is condensation in the HP EGR line. The mathematical description in which the model is based and the experimental results of the model validation will be presented.

2. Experimental setup and methodology

2.1. Test bench description and configuration

In order to perform this theoretical-experimental work, an in-line 4 cylinders, 1.6 liter, turbocharged, diesel engine was used. Table 1 summarizes the technical features of the engine used. To carry out the experiments at cold conditions, the engine was installed in a climatic test bench, where the temperatures of the test bench air, fuel and coolant are under control. The test bench is instrumented to measure the torque, speed, temperatures and pressures at different engine points. The injected fuel mass and the air mass flow through the intake line are also measured.

Table 1. Engine Specifications

Number of Cylinders	4
Number of Valves	16
Bore x Stroke (mm)	80 x 79.5
Total Displacement (cc)	1598
Maximum Power (kW/rpm)	96/4000
Maximum Torque (N m/rpm)	320/1750
Compression Ratio	15.4 : 1
Fuel Injection System	Common Rail Direct Injection
EGR System	HP and LP Cooled EGR
Intake Cooling System	Water Charge Air Cooler (WCAC)

The engine has two EGR circuits. The first one is the LP EGR circuit, in which the exhaust gas pass through the catalyzer and the diesel particulate filter (DPF), and then are redirected into the turbo compressor. The second circuit is the HP EGR. In this case, the exhaust gas is directly cooled in the cylinder head and mixed with the fresh air that comes from the intake line. Fig. 1 shows the HP EGR line and its instrumentation. This circuit consists of an internal duct that guides the exhaust gases coming from the exhaust manifold through a compact section of the cylinder head (without EGR cooler) to reduce its temperature, a HP EGR valve controlled by the ECU of the engine, and a HP EGR rail, where the exhaust gases are driven and distributed to the engine intake manifold and mixed with the air at the intake of the four cylinders.

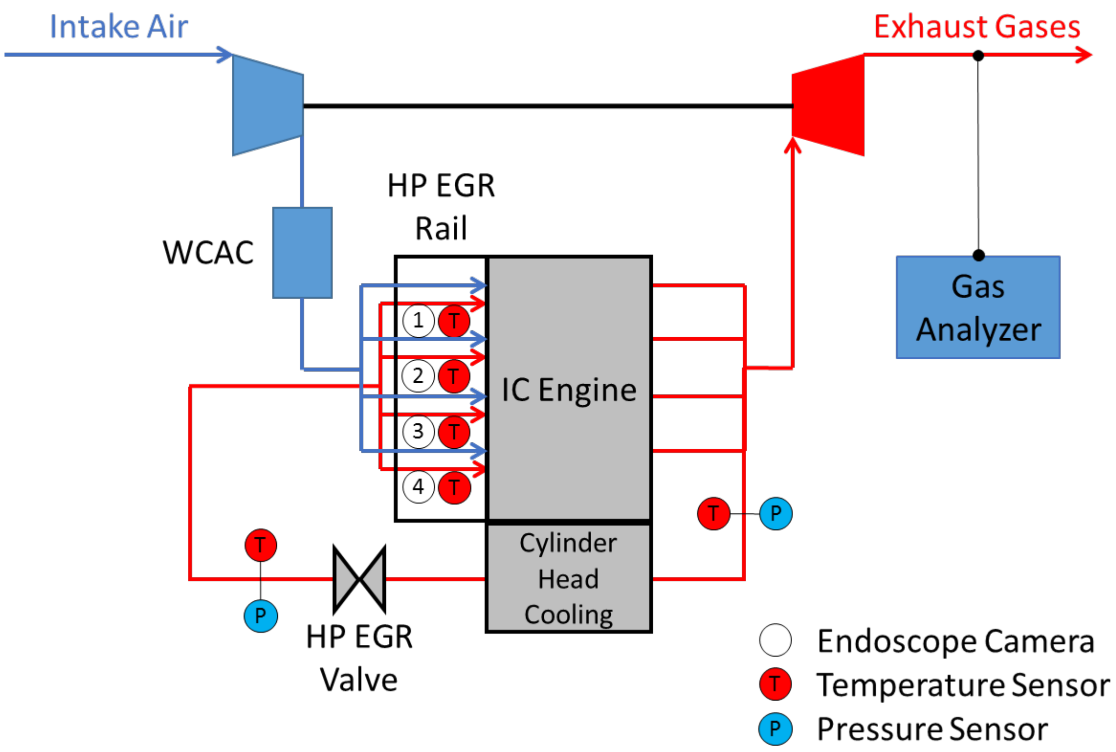

Fig. 1. HP EGR circuit

In order to perform the data acquisition for the condensation model, the HP EGR line is instrumented to measure the pressure and temperature before the turbine, at the outlet of the HP EGR valve and on the HP EGR rail, just at the inlet of each of the four cylinders of the engine. The thermocouples used to measure the temperature at the inlet of the four cylinders of the engine have been installed in two different configurations. First, introducing the thermocouples to the central area of the inlet ducts to the cylinders (gas side), and second, placing the thermocouples just in the wall surface of these ports (wall side).

The ambient temperature inside the climatic chamber has been set at -7ºC, the air mass flow through the intake line of the engine was measured by means of a hot wire anemometer with a measurement error of 1%. The fuel consumption during the cycle was measured with an AVL fuel balance, which has measurement error of 0.2%. An Horiba Mexa 7100 DEGR was used to measure O_2, CO_2, CO, using a non-dispersive infrared analyzer, and unburned hydrocarbons with a chemiluminescent detector. The error of the gas analyzer is in the range of 2%. The measurement point is located downstream the turbine and upstream after treatment systems.

In order to validate the condensation model experimentally, four endoscope cameras were placed on the HP EGR rail cover to visualize inside the rail at the inlet side of the cylinder ports. Fig. 2 shows the locations of these cameras on the engine experimental set up as well as the field of view that is observed and recorded during the experiments. These cameras allow to record videos and pictures at high quality (30 fps and 96 dpi).

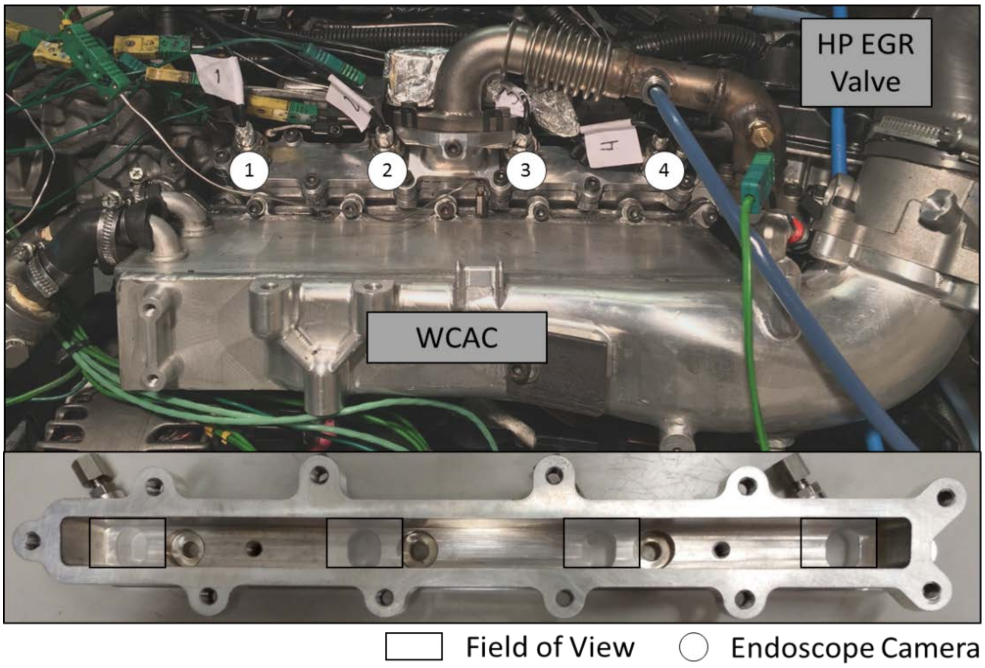

Fig. 2. Cameras Configuration and Field of View

2.2. Methodology and Strategies

A transient engine cycle with special characteristics in terms of NOx generation, soot formation and condensation inside the EGR components of a diesel engine running at cold conditions was conducted [26]. Fig. 3 shows the profile of the cycle performed for these experiments. This cycle is characterized by low-middle load points with important EGR rates and sudden accelerations.

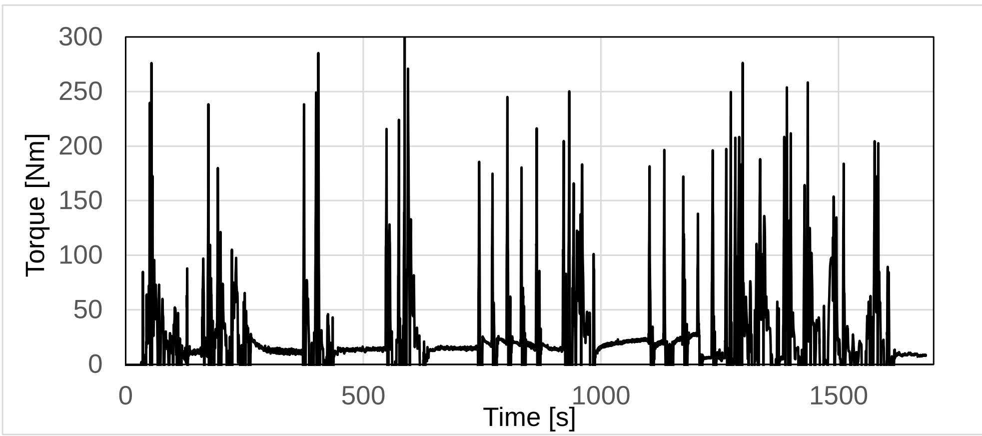

Fig. 3. Profile of the cycle performed

For this study, it was necessary to modify the standard engine calibration in order to activate the HP EGR during the complete cycle, from the beginning of the warm up process in a cold starting of the engine at -7ºC. The standard engine calibration is not configured to perform HP EGR under these particular conditions. Fig. 4 shows the activation of the HP EGR by showing the EGR valve position during the entire cycle. This modification was done with the purpose of introducing a similar EGR rate to the standard engine calibration when the warm-up process finishes (approx. 40%) and to check under which conditions during the warm up process at -7ºC the condensation is produced.

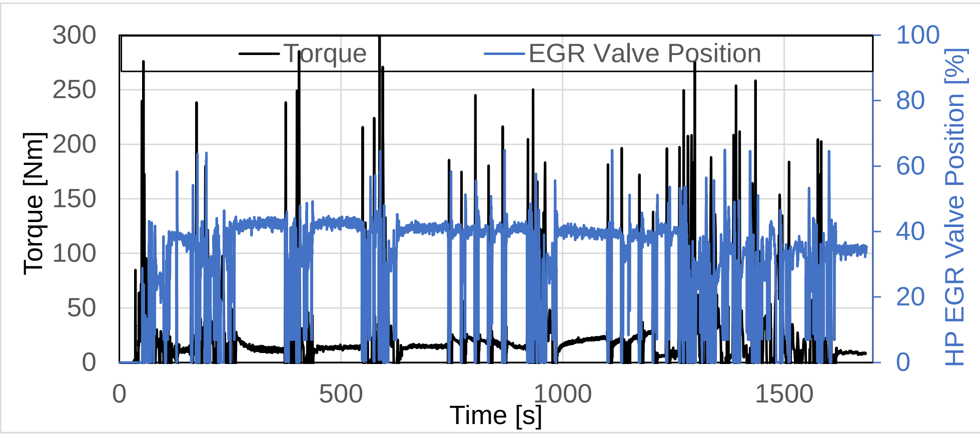

Fig. 4. HP EGR activation since engine cold start

In order to estimate the EGR rate, it is necessary to estimate the engine volumetric efficiency for this engine configuration. Equation 1 shows this efficiency as a function of the air mass flow, the engine parameters, the boost pressure and the intake temperature. When the volumetric efficiency is assessed, we consider this value as a constant value and it is possible to obtain the EGR rate as a relation between the air estimated theoretically by using the volumetric efficiency and the air mass flow measured. Equation 2 shows the EGR rate estimation.

$$\eta_{vol} = \frac{\dot{m}_{air}}{\frac{P_{boost}}{R\,T_{intake}}\,.\,V_{eng}.\,n\,.\,i} \tag{1}$$

$$EGR_{rate} = \frac{\eta_{vol}\cdot\frac{P_{boost}}{R\,T_{intake}}\,.\,V_{eng}.\,n\,.\,i - \dot{m}_{air}}{\eta_{vol}\cdot\frac{P_{boost}}{R\,T_{intake}}\,.\,V_{eng}.\,n\,.\,i} = \frac{\dot{m}_{theor} - \dot{m}_{air}}{\dot{m}_{theor}} \,*\, 100 \tag{2}$$

3. Condensation Model

In order to estimate whether or not there is condensation in the EGR line, a simple condensation model has been defined. The condensation model has as input parameters the temperature, pressure and humidity at ambient conditions, the air mass flow through the intake line of the IC engine and the amount of fuel injected into the cylinders (air-fuel ratio). Based on these data, the model calculates the amount of water (humidity ratio) that is available at the cylinders outlet. With this value of humidity ratio, and considering the local temperature and pressure at different points of the HP EGR line, the relative humidity at these points is estimated as the output parameter of the model.

In the following section, the model is described and the hypotheses on which it is based are established.

3.1 Mathematical description

Four main hypothesis are considered to determine the condensation model:

- Quasi-steady conditions in intake line, IC engine and HP EGR line.

- Although condensation conditions could appear, it is assumed that the condensed water is entrained by the exhaust gas mass flow and it is not accumulated at different points of the IC engine outlet or the HP EGR line.
- All the water present in the HP EGR line comes from the ambient, from the combustion process and from the recirculated gases itself.
- The combustion efficiency is 100%, it means that all the injected fuel burns and the combustion process transforms this injected fuel, combined with oxygen, into water and CO_2.

Based on the input variables of the model and the previously defined hypotheses, the following equations allow the estimation of relative humidity values at different points of the EGR line.

First, considering the chemical composition of the fuel, the atomic mass of a hydrogen atom and the atomic mass of a carbon atom, the ratio of Hydrogen mass in a fuel molecule can be calculated following the Equation 3:

$$r_{(H/fuel)} = (nºH * 1.00794)/(nºH * 1.00794 + nºC * 12.0107) \quad (3)$$

Where nºH is the number of hydrogen atoms and nºC is the number of carbon atoms.

Table 2 shows the ratio of hydrogen mass in a fuel molecule for different alkanes of single bonds. If alkane hydrocarbons of single bonds are considered, a particularity of this hypothesis is that this ratio remains practically constant (~0.15) when the number of carbon atoms increase from nine onwards (nºC≥9). In Diesel fuel, the typical hydrocarbon molecule will be higher than this alkane of 9 carbons and lower than a molecule of 25 carbons. For all these cases it is possible to consider the approximation of 0.15 for this ratio as an acceptable value.

Table 2. Ratio of hydrogen mass in an alkane hydrocarbon of single bonds

nºC	*1*	*3*	*6*	*9*	*12*	*15*	*18*	*21*	*24*
nºH	4	8	14	20	26	32	38	44	50
$r_{(H/fuel)}$	0.251	0.183	0.164	0.157	0.154	0.152	0.151	0.150	0.149

Considering the atomic mass of a hydrogen atom, the atomic mass of an oxygen atom is 15 and that a water molecule is made up of two hydrogen atoms and one oxygen atom, the ratio of Hydrogen mass in a water molecule is shown in the Equation 4:

$$r_{(H/H_2O)} = (2 * 1.00794)/(2 * 1.00794 + 15.9994) = 0.11 \qquad (4)$$

This means that 11% of the mass of a water molecule corresponds to the mass of the hydrogen atoms. Considering that all the hydrogen in the fuel reacts with oxygen to produce water, that the mass ratio of hydrogen in water is 0.11 and the mass ratio of hydrogen in the fuel is 0.15, it is possible to obtain the water ratio in the fuel with the Equation 5:

$$r_{(H_2O/fuel)} = r_{(H/fuel)}/r_{(H/H_2O)} = 0.15/0.11 = 1.336 \qquad (5)$$

This means that considering a perfect combustion, as described in the hypothesis, for each gram of fuel burned, 1.336 grams of water are produced. From this expression, and considering the injected fuel mass, it can be obtained the mass flow of water at the IC engine outlet coming from the combustion process as in Equation 6:

$$\dot{m}_{(H_2O/fuel)} = (\dot{m}_{fuel})(r_{(H_2O/fuel)}) \tag{6}$$

On the other hand, to estimate the amount of water coming from the intake air due to the ambient humidity, the "Humidity Ratio" (W_{air}) can be calculated. The air humidity ratio is a relation between the mass of water vapor per mass of dry air [27]. It is possible to calculate this ratio by using a psychometric diagram from the conditions of pressure, temperature and relative humidity in ambient conditions. From these data, and knowing the mass air flow through the intake line, it is possible to obtain the water vapor mass per dry air mass as shown in Equation 7:

$$\dot{m}_{(H_2O/air)} = (\dot{m}_{fuel} * W_{air})/(1 + W_{air}) \tag{7}$$

Adding the two previous terms, that is, the water mass flow coming from the combustion process and the water mass flow coming from the ambient conditions, it is possible to estimate the total water mass flow at the cylinders outlet taking into account the EGR rate. Considering the scheme of the IC engine with an EGR circuit of the Fig. 5, it is possible to estimate the water mass flow at the cylinders through the Equation 8:

$$\dot{m}_{(H_2O/exh)} = 1/(1 - EGR\%) * [\dot{m}_{(H_2O/fuel)} + \dot{m}_{(H_2O/air)}] \tag{8}$$

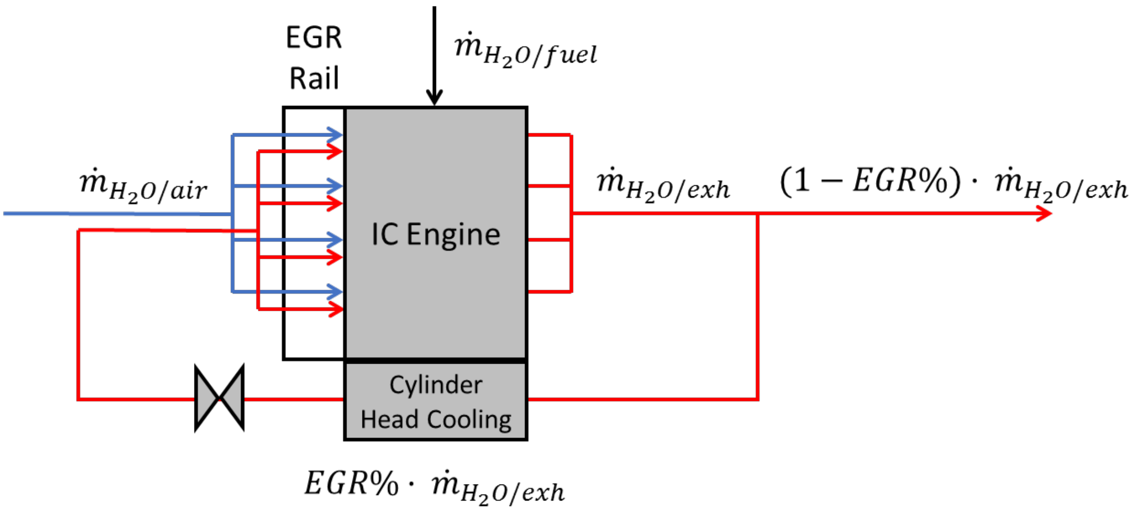

Fig. 5. IC engine scheme with water mass flows

Considering this water mass flow at the cylinders outlet and knowing that the total mass flow at this point is the sum of the intake air mass flow and the mass flow of fuel, the humidity ratio at the cylinders outlet (W_{exh}) can be calculated as shown in Equation 9:

$$W_{exh} = \frac{\dot{m}_{\,(H_2O/exh)}}{\frac{\dot{m}_{air} + \dot{m}_{fuel}}{1 - EGR\%} - \dot{m}_{\,(H_2O/exh)}} \tag{9}$$

Finally, using the psychometric diagram, it is possible to obtain the relative humidity (φ_x) at a local point by using the humidity ratio at the cylinders outlet and the local conditions of pressure and temperature at this local point of the HP EGR line.

3.2 Results and model Validation

The engine used is instrumented to measure pressure and temperature before the turbine, at the outlet of the HP EGR valve and on the HP EGR rail, just at the inlet of each of the four cylinders of the engine. The thermocouples used to measure the temperature at the inlet of the four cylinders of the engine have been installed in two different configurations. Firstly, introducing the thermocouples to the central area of the inlet ducts to the cylinders (gas side). In all these places, the relative humidity has been estimated using the model described in the previous section. On the other hand, the four cameras installed in the EGR rail at the inlet of the cylinders have been used to validate this model by comparing the images of these cameras with the condensation conditions estimated by the model. Fig. 6 shows a sample of the images of the EGR rail obtained by these four cameras. The top of the figure shows the position of the thermocouples in the gas side, while the bottom of the figure present the pictures with the thermocouples positioned in the wall side, just at the middle of the surface. The instrumentation of these sensors was mechanized in the HP EGR Rail used for the experiments and placed to measure the exhaust gases temperature before the mixture at the intake cylinders.

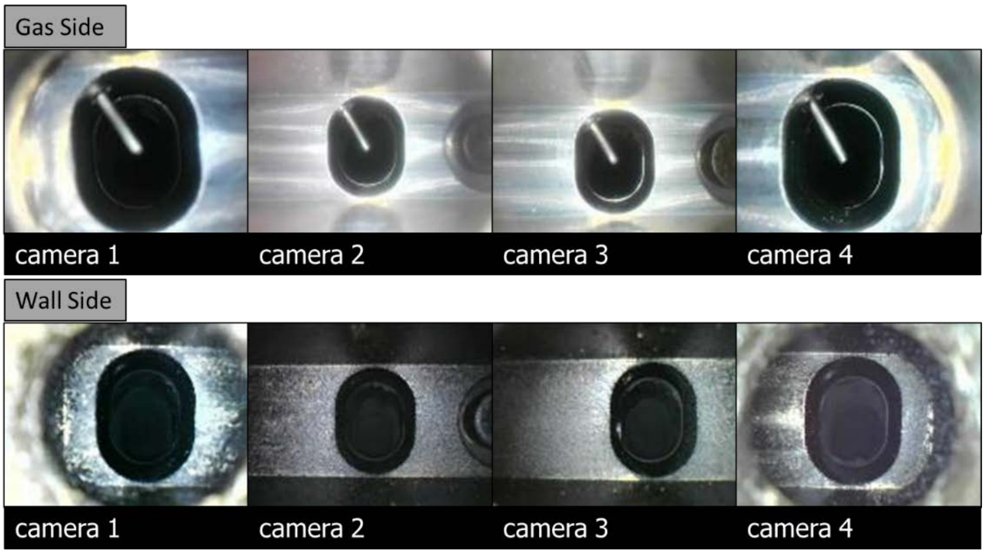

Fig. 6. Sample of the images captured by the cameras at the cylinder intake ports in the EGR

Fig. 7 shows some measured parameters of the IC engine during the first part of the cycle. As can be seen in the graph, the cycle is composed of transient periods with severe accelerations and decelerations and other periods in steady conditions with smoother engine operating conditions. To take into account the operating conditions of the IC engine in new homologation cycles, the ambient temperature in the engine test bench has been set to -7ºC and high EGR rates have been forced from the very beginning of the cycle. The initial temperature of the engine block is -7ºC and the EGR rates in this first stage of the cycle reach values that arrive up to 40%, especially in periods of steady conditions. These conditions at the beginning of the cycle force severe conditions of condensation that will be analyzed in next paragraphs.

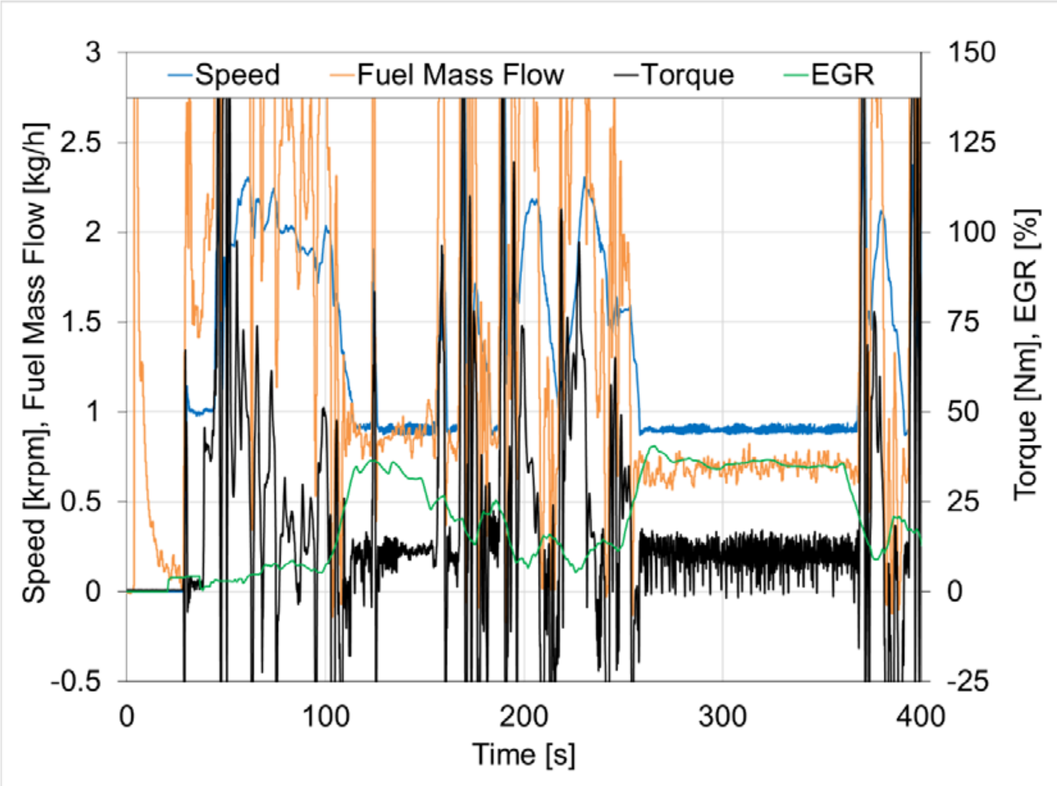

Fig. 7. Measured engine speed, fuel consumption, torque and EGR% during the first 400 sec of the cycle

Fig. 8 shows the pressure measured at the engine outlet and the pressure at the HP EGR line outlet during the first part of the cycle. As can be seen in the figure, the transient periods have pressure peaks which reach up to 4 bar and periods in steady conditions with pressure values close to the ambient pressure. In order to estimate the condensation conditions in the HP EGR line, the engine outlet pressure will be considered a uniform value from the engine outlet to the HP EGR valve inlet, and the HP EGR outlet pressure will be considered a uniform value from the HP EGR valve outlet to the four cylinders inlets.

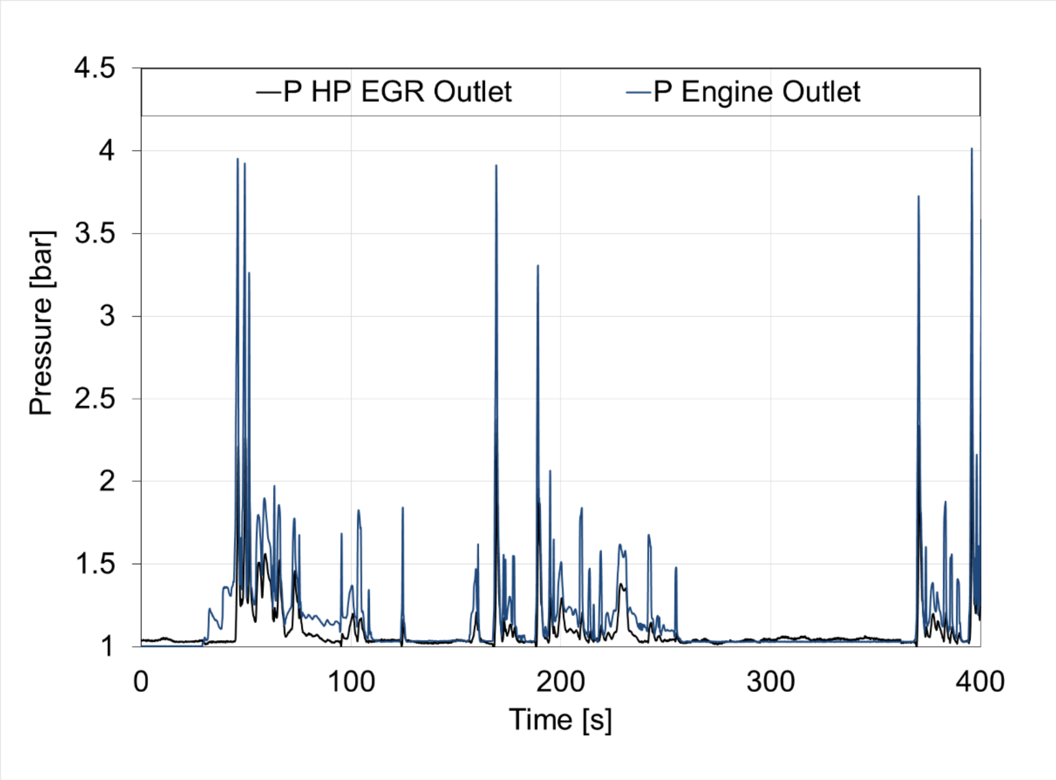

Fig. 8. Measured pressures during the first 400 sec of the cycle

Fig. 9 shows the measured temperature during the cycle at the engine outlet, at the HP EGR valve outlet and at the four cylinders inlet. Temperatures at cylinders inlet have been measured at two different points, the “gas side” by placing the thermocouple approximately in the center of the cylinders inlet ports and the “wall side” by placing the thermocouple just in the wall surface of these ports. As can be seen from Fig. 9, the temperature at the engine outlet increases quickly, avoiding the soon condensation conditions (red line in Fig. 9). However, at the final part of the HP EGR line, these temperatures are lower, specially close to the wall (thick blue and purple lines in Fig. 9) of cylinders inlet ports, where temperatures remain with values lower than 40ºC during the first part of the cycle. Furthermore, due to the heat transfer efficiency in this final part of the EGR line, temperatures in the gas side (thin blue and purple lines in Fig. 9) are considerably higher than close to the wall. These lower temperatures will produce condensation conditions close to these ports during the first part of the cycle. It is interesting to note that the temperatures recorded at the inlet of cylinders 1 and 4 (located at the extremes of the EGR rail) are lower than the temperatures measured in cylinders 2 and 3 (located in the central part of the rail). This is because the conditions for the heating of the EGR rail ducts by the hot exhaust gases and the heat transfer from the engine block are less severe in the ducts located at the extreme of the EGR rail.

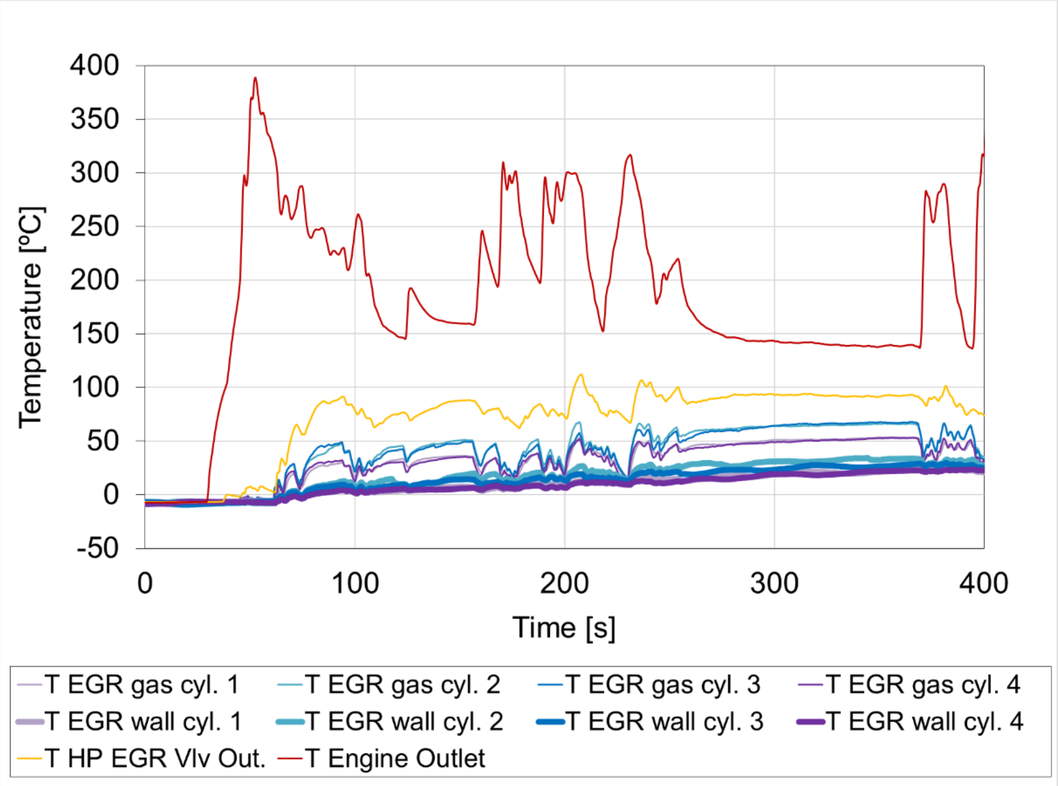

Fig. 9. Measured temperatures during the first 400 sec of the cycle

Figure 10 shows the mass flow of water and hydrocarbons during the first seconds of the cycle. Hydrocarbons mass flow was measured through the gas analyzer and the total water mass flow at the cylinders outlet was estimated through the Equation 8 described in the mathematical description. As can be seen in the graph, most of the products coming from the combustion process and from the ambient conditions can be considered as water and approximately a 2% of this mass flow corresponds to hydrocarbon species.

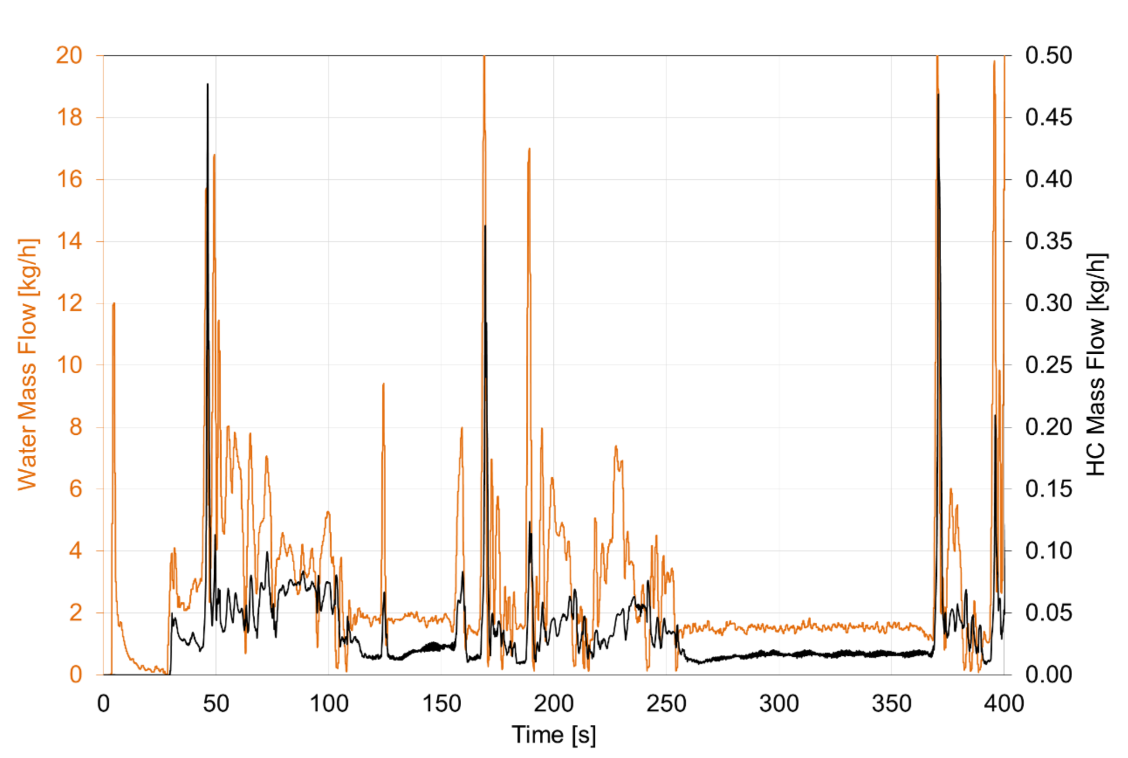

Fig. 10. Measured mass flows during the first 400 sec of the cycle

The condensation model was used to calculate the relative humidity at different points of the HP EGR line, considering the pressures and temperatures of previous figures, air and fuel mass flows and the ambient conditions. Fig. 11 shows the results of the model to estimate relative humidity at different points of the HP EGR line. As can be seen in this figure, the relative humidity at the engine outlet and at the HP EGR valve outlet drops to values below 50% before the second 80 of the cycle, avoiding condensation conditions at these points. However, at the cylinders inlet, these values remain well above 100% due to the low temperatures at these points. It is necessary to wait until 250 seconds to avoid the condensation conditions (relative humidity less than 100%) in the central part of the inlet ducts to the cylinders. However, this does not prevent it from being followed in conditions of condensation near the walls.

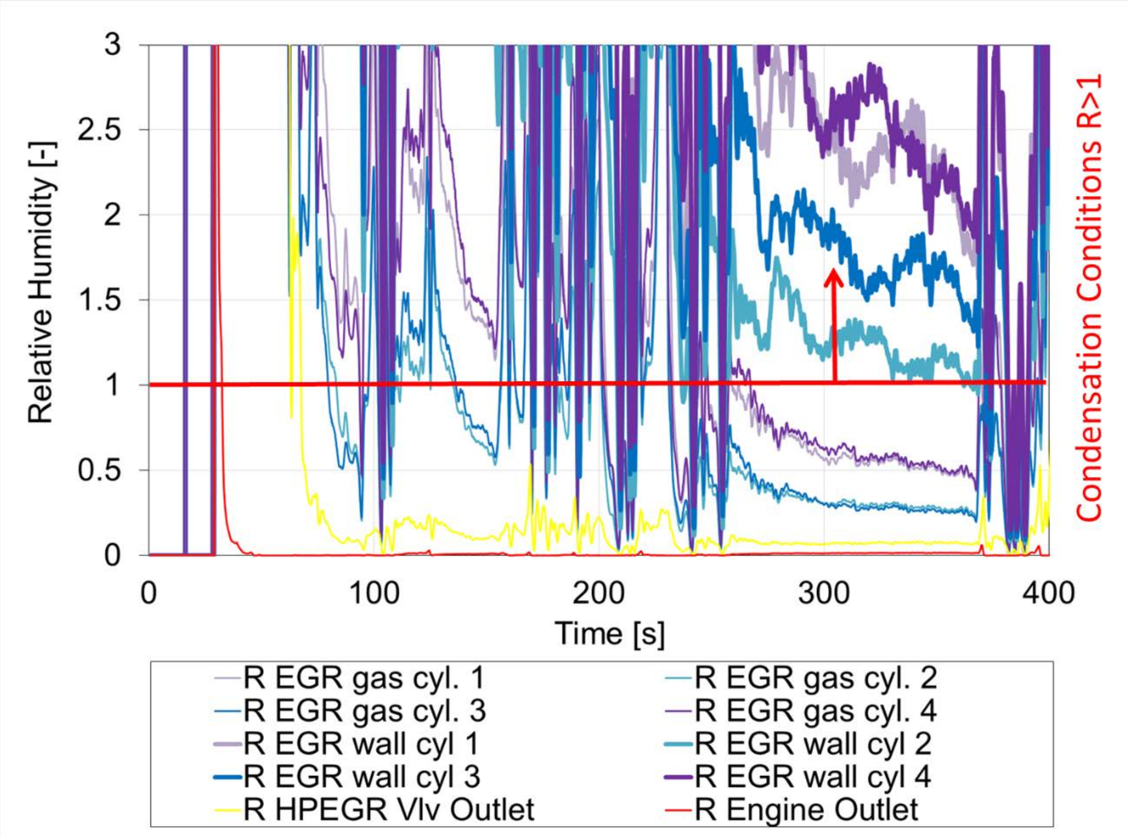

Fig. 11. Estimated relative humidity during the first 400 sec of the cycle

In order to validate the condensation model, these results are compared with the videos recorded. Fig. 12 shows a sequence of frames of the four videos recorded at the inlet of the four cylinders. The first frame (top of the figure) corresponds to a frame of the first second of the videos showing the initial conditions after the cycle begins. The second frame of the figure shows the images of the four ports at 100 seconds. At this point in time, a severe fog appears on the four ports and first drops precipitate on the EGR rail surface, as shown the bright spots on the surface of the camera 2 and 3. The third frame shows the images at 200 seconds.

As Fig. 11 shows, at this time the temperatures in the gas side of the EGR rail have increased enough to avoid condensation conditions. However, close to the EGR rail surface, temperatures are lower and condensation conditions appear. As the two central cylinders present higher temperatures, it is possible to observe a reduction in the fog in these two central cylinders and a dense fog in the two cylinders at the ends of the EGR rail. Also, an increment in the quantity of drops on the EGR rail surface can be observed in this frame. The fourth and fifth frames show the images at 300 seconds and at second 400. At these moments, the fog practically disappear and only a little foggy appears on camera 4 because the glass of the camera was fogged up. As Fig. 11 shows, , although the temperature of the gas has increased considerably at these moments and no fog is formed in the gas side, temperatures in the EGR rail wall continue to be low and the dew conditions cause a film of water on the surface of the rail.

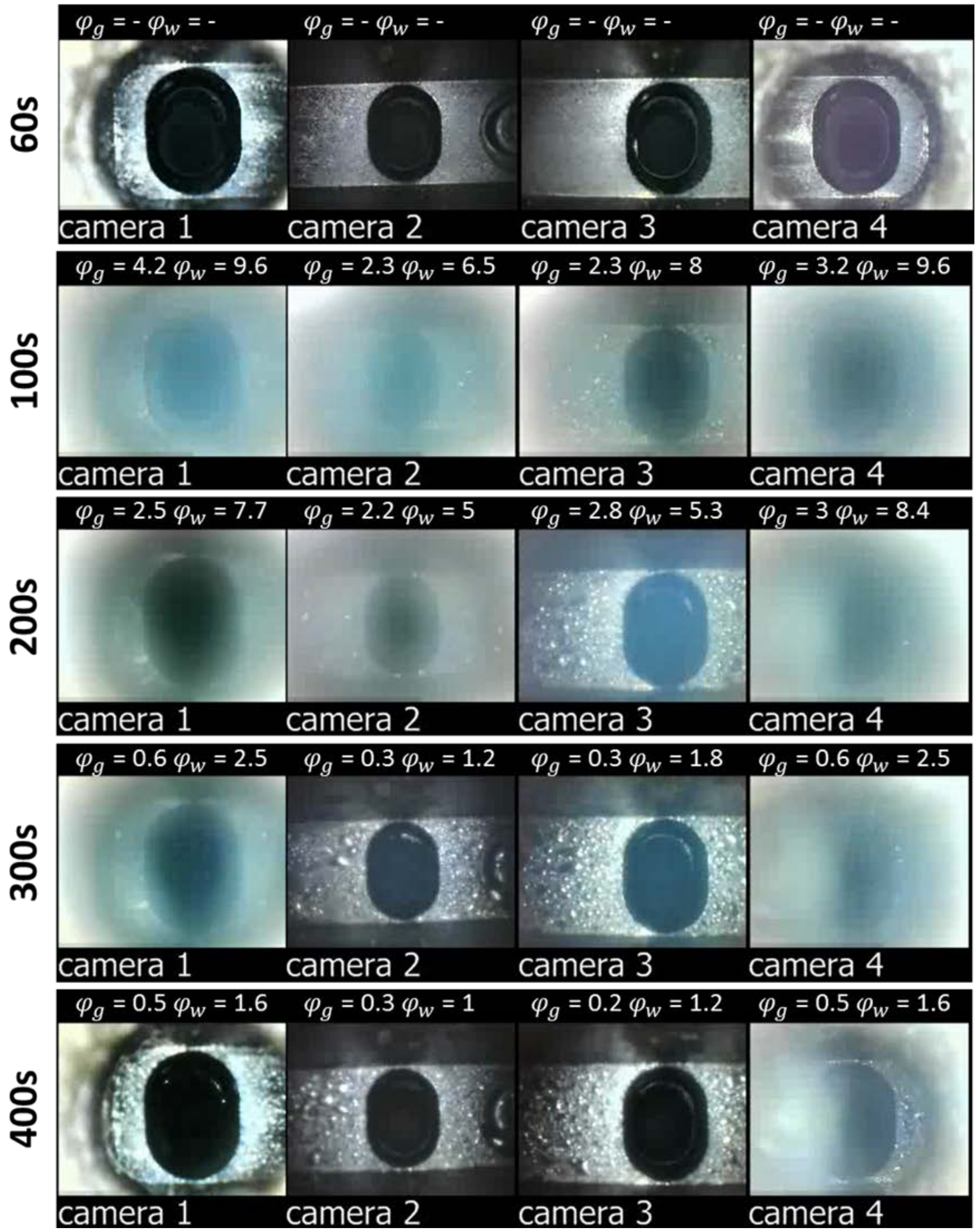

Fig. 12. Video frames of the 4 cylinder cameras in the second 1, 100, 200, 300 and 400. In these pics φ_g is the relative humidity in the gas side and φ_w is the relative humidity in the wall side.

Fig. 13 shows the results of the model to estimate the relative humidity from the 400 to the 1200 seconds of the cycle. As can be seen in this figure, the relative humidity in the gas side is lower than 50% avoiding condensation conditions in the central part of the ducts. However, the EGR wall remains at low temperatures and dew conditions continue in the two central cylinders before the 800 seconds and the two ends cylinders remain in dew conditions practically until the 1200 seconds.

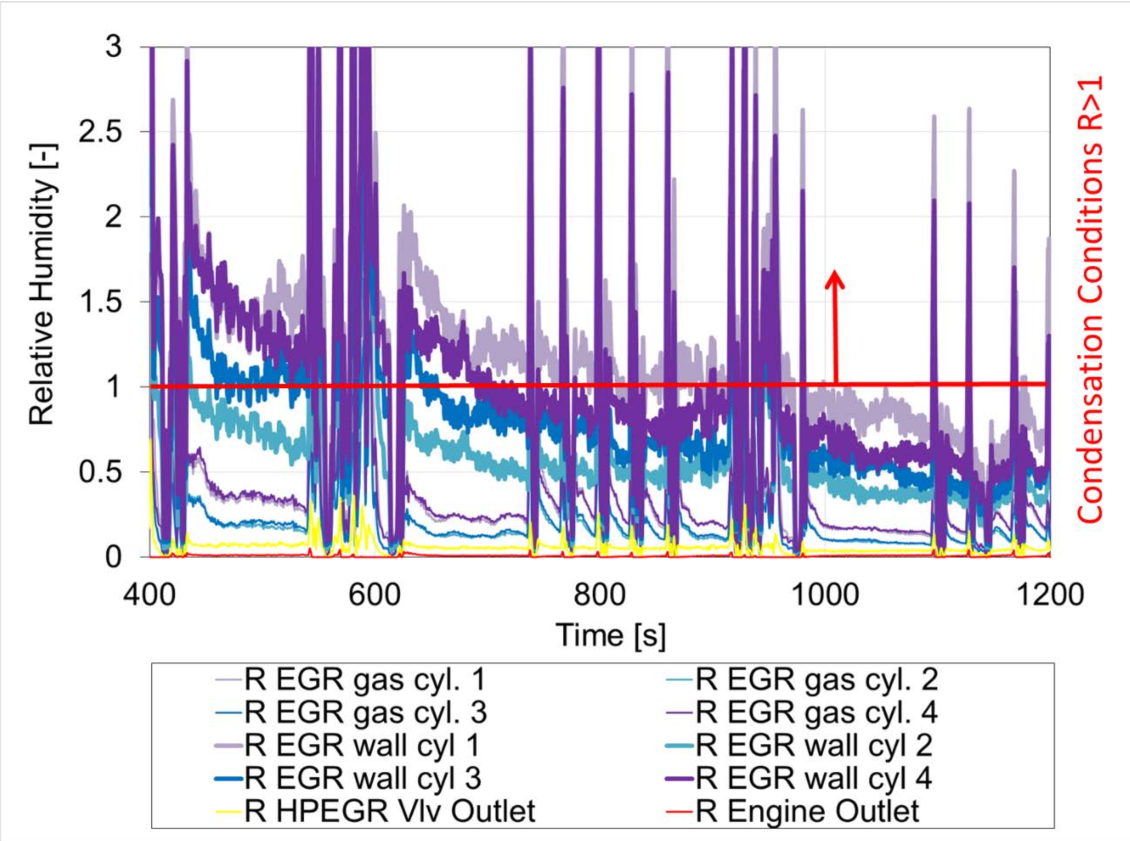

Fig. 13. Estimated relative humidity from the second 400 to the second 1200 of the cycle

Fig. 14 shows a sequence of frames of the four videos at the four cylinders inlet. The first frame (top of the figure) corresponds to a frame at the 400 seconds. Although at this time there are no condensation conditions on the gas side, dew conditions are produced in all the EGR wall and water drops appear in the EGR surface. The second frame of the figure shows the images of the four ports at the 600 seconds. At this time, the warm-up process of the two central cylinders reduce the relative humidity and dew conditions begin to disappear. It is possible to observe in the two central images a reduction in the quantity of liquid in the EGR wall, especially in camera 3. The last three frames, third, fourth and fifth, show the images at the 800, 1000 and 1200 seconds respectively. These images show a progressive reduction of the liquid drops in the EGR rail surface. This reduction is clear in the two central cylinders and it is less clear in the two ends cylinders where the relative humidity remains at high values practically until the last frame at the 1200 second. The results observed in the cameras have an important relationship with the values of relative humidity in the wall estimated by the model shown in the Fig. 13.

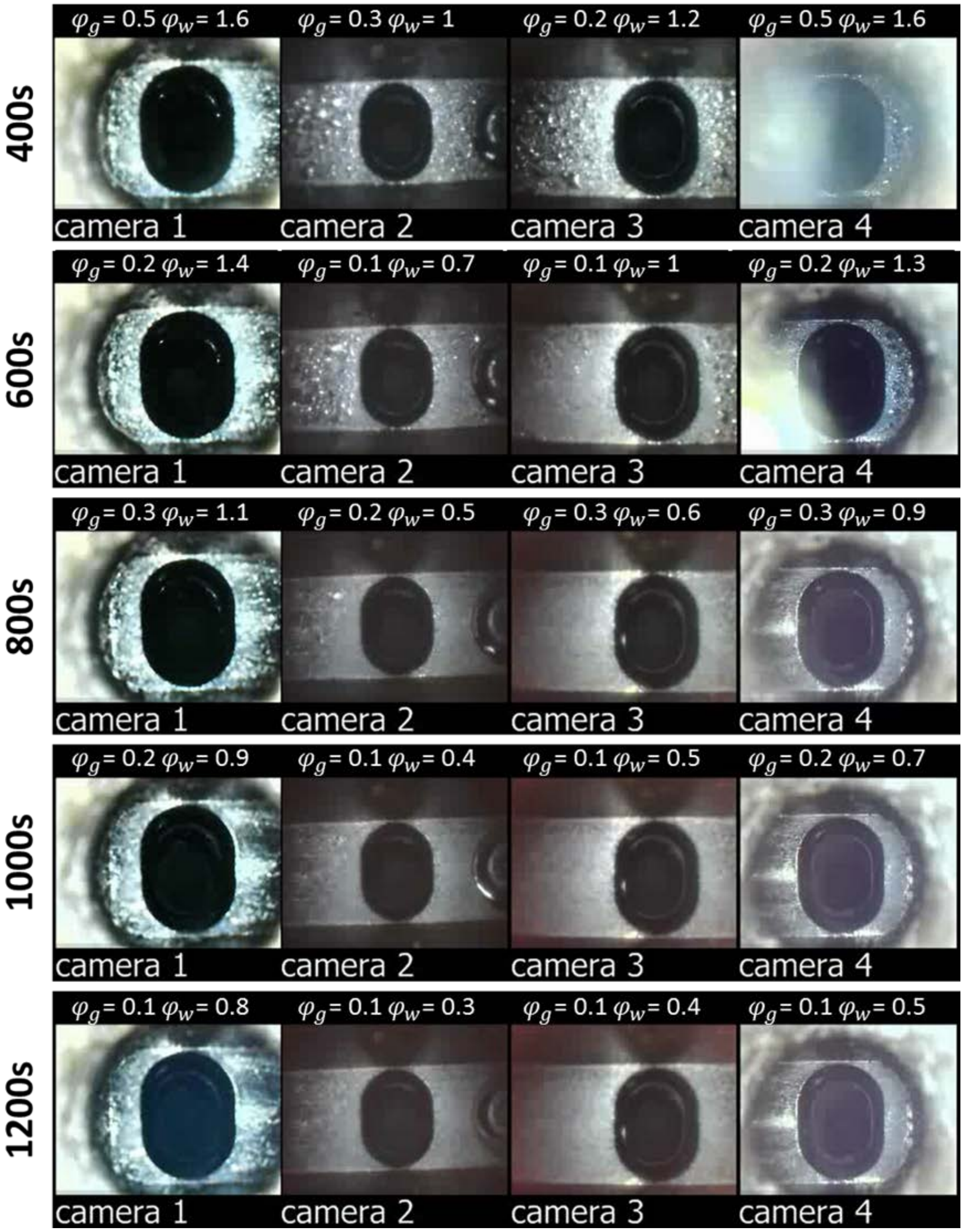

Fig. 14. Video frames of the 4 cylinder cameras in the second 400, 600, 800, 1000 and 1200. In these pics φ_g is the relative humidity in the gas side and φ_w is the relative humidity in the wall side.

4. Conclusions

In the present paper, a simple model to predict condensation conditions in EGR flow and in the duct walls of the EGR line has been developed and validated experimentally. The model predicts in which states the condensation conditions may appear or disappear. Besides, an experimental measurement technique has been developed to check the condensation.

The condensation conditions depend on the engine working conditions (air to fuel ratio, EGR rate…), ambient humidity and local conditions of temperature and pressure at the point where this condensation phenomena is analyzed. For this work, two different positions for the EGR thermocouples were used, one at the center of the EGR rail ports (gas side) and a second position, close to the EGR rail wall (wall side). The main conclusion of this work is that a simple model, as the model described in this article, can be used to easily predict at different points of the EGR line, the mist conditions in gases and also the drops attached to the walls in the ducts.

This research work could be used to establish strategies to avoid or cause the appearance of the condensation in EGR circuits of IC engines. The model is enough simple to be used combined with control strategies to avoid or cause these condensation conditions in the gas side and in the duct walls. This is very interesting from the point of view of the new approval emissions regulations that will require more aggressive EGR strategies at low temperatures where condensation could appear.

It could be stated that for this particular working conditions (for an engine warm up at -7ºC, a symmetric rail to distribute the EGR and measuring temperatures at the HP-EGR outlet), the condensation conditions in the gas side are present until the gas temperature reaches 50ºC at cylinders 1 and 4 and 60ºC at cylinders 2 and 3. In our particular test, the EGR flow reaches these conditions in the second 300 approximately. In the wall side case, the condensation conditions are present until the wall temperature reaches values around 40ºC at cylinders 1 and 4, and 30ºC at cylinders 2 and 3. The walls reach these conditions in the second 800 approximately. Once these conditions have been reached, the relative humidity is below one and the dew point, which is a parameter that depends on pressure and humidity, indicates that no condensation phenomenon occurs in these parts of the EGR line.

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