DETECTING THE ONSET OF NUCLEATE BOILING IN INTERNAL COMBUSTION ENGINES

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ABSTRACT

The use of an electric pump instead of the standard crankshaft-driven one in Internal Combustion Engines, gives the possibility of controlling the coolant flow rate independently of engine speed, allowing therefore, the use of much lower coolant flow rates than usually adopted and the development of nucleate boiling flow regimes within the engine cooling system. In order to take advantage of nucleate boiling and of the associated high heat transfer coefficients, the onset of this heat transfer regime must be correctly identified. This work presents the results of an experimental campaign, which was carried out on a small displacement spark ignition engine (1.2 dm^3, 60 kW) with the aim of detecting the occurrence of nucleate boiling within the engine cooling system. The test rig was properly instrumented in order to measure coolant temperatures at engine inlet and outlet, coolant pressure at several locations in the circuit, coolant flow rate and engine metal temperatures. Operating conditions involving different coolant flow rates were selected in order to enforce both the usual single-phase heat transfer regime and nucleate boiling conditions. Several experimental quantities were analyzed with the aim of establishing the coolant flow rates ranges where the nucleate boiling occurs. The agreement in the trend of coolant temperature and pressure and engine wall temperature provides hints to identify experimentally the onset of nucleate boiling.

HIGHLIGHTS

- Cooling system of an internal combustion engines equipped with an electric pump.
- Onset of Nucleate Boiling experimentally detected by coolant flow rate regulation.
- Different experimental techniques tested.
- Effects of coolant pressure on Nucleate Boiling Onset investigated.

KEYWORDS

Nucleate boiling; thermal management; cooling system; experimental test; electric pump; internal combustion engines.

1. INTRODUCTION

Thermal management plays an important role for Internal Combustion Engines (ICE) performances, emissions, fuel consumption and reliability. Nowadays many works deal, in particular, with the behavior of the cooling system during the engine warm-up. As is known, in fact, high engine emissions and fuel consumption occur at low engine temperatures owing to high frictional losses and combustion inefficiency. Will et al. [1,2] estimated an increase in frictional losses amounting to 2.5 times the ones obtained under fully warmed conditions, for lubricant temperatures around 20°C, while engine efficiency drops to about 9% after a cold start [1,3]; Samhaber et al. [4] predicted an increase of about 13.5% in fuel consumption for lubricant temperatures around 0 °C. A fast warm-up is therefore desirable and can be achieved by an optimal thermal management control strategy.

The control of the engine cooling system today is still very simple and the possibility of regulation is limited: the rotational speed of the cooling pump is imposed by the engine speed and the cooling system is designed to ensure engine reliability by providing adequate coolant flow rate under high-power conditions. Under low-load or low coolant temperature conditions, the coolant flow direction is managed by using a wax thermostatic valve, which by-passes the radiator and redirects part of the coolant to the pump, while the coolant flow rate is regulated by varying the flow resistance that is determined by the thermostatic valve. In most engine operating conditions, therefore, the coolant flow rate is higher than strictly necessary and the coolant is maintained in liquid phase. Brace [5], for instance, estimated that an engine is over-cooled for 95% of its operating time.

Different cooling management strategies were proposed in order to overcome the standard cooling system limitations and most of them are based on the possibility of using an electric pump, whose speed can be controlled independently of engine speed. Ap and Golm [6] proposed a reduction of the coolant flow rates by the use of a small electric water pump. Brace [5,7] proposed the use of an electric pump in cooperation with a throttle valve and developed a control strategy based on the use of empirical look-up tables; 2% improvement in bsfc was obtained. Bent [8] proposed a cooling strategy where the pump was switched on and off cyclically, in order to obtain a faster warm-up, while Gardiner [9] investigated the effects of coolant circulation rate on fuel consumption reduction during engine warm-up.

Regulation of coolant flow rates by means of electrically driven pumps, in conjunction with a proper control strategy, can be conveniently used to allow the cooling system to

operate under controlled nucleate boiling flow regime [10,11], both during engine warm-up and under fully warmed conditions. The possibility of adopting new components for the cooling system of internal combustion engines and of taking advantage of the nucleate boiling regime was investigated by several researchers for decades [12-16] and is still under investigation [17]; these contribute to identify the advantages of operating under nucleate boiling: smaller coolant mass, smaller radiator, lower pump power requirements, faster warm-up time, and lower friction during the warm-up period; all these conditions allow the reduction of fuel consumption and, consequently, of CO_2 emissions [18], which is currently sought by these and other advanced internal combustion engine technologies [19,20]. In addition, a coolant pump working independently of the engine would eliminate the risk of unwanted phenomena, like after-boiling [21]. Recently, Pizzonia et al. [22] proposed and developed a control strategy which sets the coolant flow rate at much lower values than usually adopted, allowing a certain level of nucleate boiling. Under fully warmed conditions, a coolant flow rate just below that which determines nucleate boiling, ensures appropriate cooling with low coolant flow rate, still preserving engine reliability.

However, at present, the difficulty of obtaining on-board information about the heat transfer regime, which occurs under the various operating conditions, is the main obstacle to setting up a practical nucleate boiling cooling system. In order to overcome this limitation and to enable the development of such an innovative system, a deeper understanding of the behavior of a cooling system as a result of the different operating conditions is needed. At the moment, no systematic experimental analysis of Onset of Nucleate Boiling (ONB) in ICE exists.

The present work summarizes the results of an extensive experimental campaign, which was carried out at the engine test-rig, with a Spark Ignition (SI) engine properly instrumented, with the purpose to detect the occurrence of nucleate boiling and to identify the engine variables which cause nucleate boiling. Although the test rig is well instrumented, the experimental identification of nucleate boiling is not simple: the observation of vapor bubbles through transparent windows placed along the cooling circuit is not reliable, owing to the fact that in the early stages of nucleate boiling bubbles form but implode before arriving at the window; they become visible only in the fully developed stage of the phenomenon. Therefore, several experimental techniques to identify the Onset of Nucleate Boiling have been developed and are summarized in the present paper. A zero-dimensional model of the cooling system of an SI Engine, which was developed in [10] and is able to predict

dynamically the occurrence and the extent of nucleate boiling flow regime, was used to help to explain better the recorded data.

2. EXPERIMENTAL SETUP

The experimental tests were carried out on a small-size four-stroke SI engine. The engine displaces about 1.2 dm^3 in four in-line cylinders with four-valve per cylinder aluminum head and about 60 kW between 5000 and 6000 rpm rated power. The standard crankshaft-driven coolant pump is substituted by a small power electric pump (127 W at 15 V, 2092 dm^3/h maximum flow rate). A schematic of the test-rig is plotted in Figure 1.

Special attention was dedicated to the cooling circuit instrumentation (Fig. 2). A differential pressure gauge is installed at the pump ends, while the coolant pressures at engine inlet and outlet are measured with miniature piezoresistive pressure transducers, 3 kHz bandwidth. PT100-type temperature sensors are used to obtain the coolant temperatures at engine inlet and outlet. Coolant volumetric flow rate is measured at engine inlet and outlet using turbine type flowmeters, with a repeatability of ±0.05%. In addition, an optical access is also installed in the cooling circuit near the engine outlet, in order to observe the presence of bubbles in the coolant flow pattern during the experimental tests (Fig. 1). The standard radiator is immersed in a tank filled with water, which also contains the coolant expansion tank. A digital PID regulator is used to control the cool water flow rate, which is needed in the tank in order to keep engine inlet coolant temperature constant (85 °C) within a ±1 °C error band. Finally, k-type thermocouples are located in the cylinder block (head gasket side) and in the cylinder head at various locations, as reported in Figure 3. The main measuring devices specifications are reported in Table 1.

All tests are performed with a 50/50 (by mass) mixture of water and commercially available ethylene glycol.

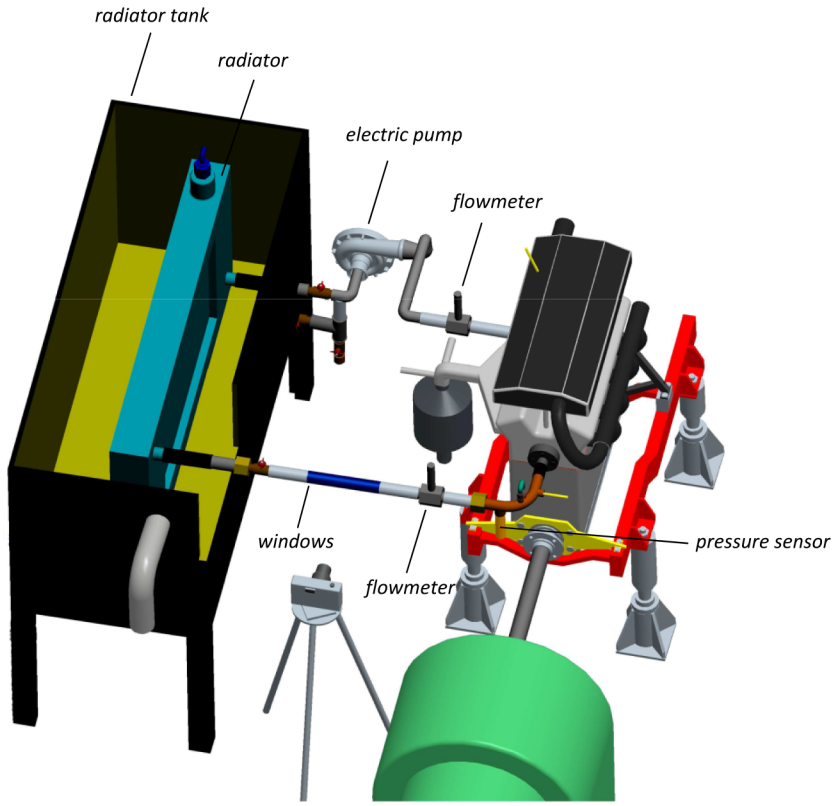

Fig. 1. Schematic of the test rig.

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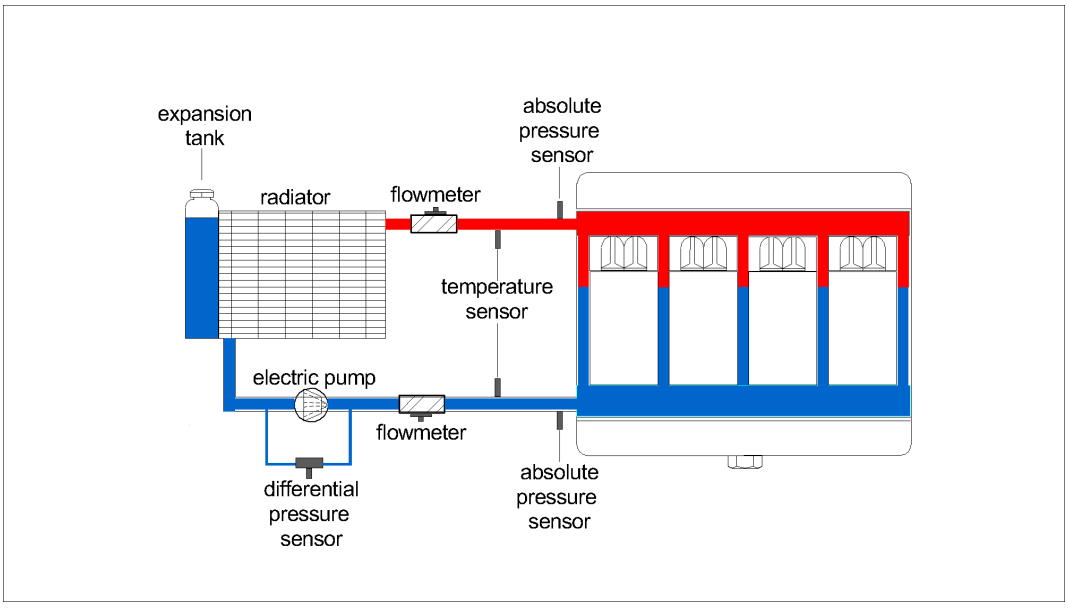

Fig. 2. Schematic of the cooling circuit.

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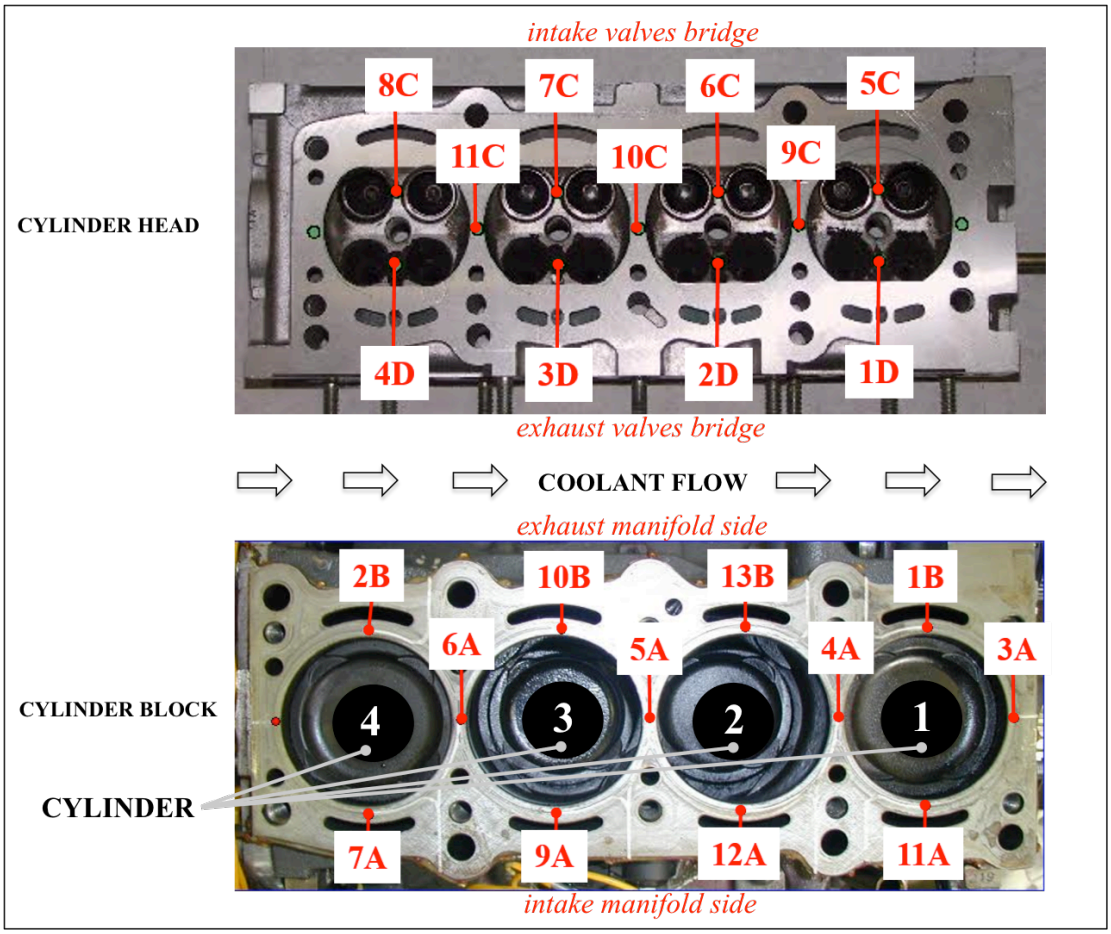

Fig. 3. Thermocouples location in the engine metal and cylinder number identification.

Table 1
Main measurement devices specifications.

Device type	Model	Specifications
AVL, fuel meter	733 S	FS 150 kg/h Sensitivity and linearity deviation ± 0.12% of the fuel mass withdrawn
BORGHI&SAVERI, eddy current dynamometer	FE 260 S	FS 190 kW @ 12000 rpm Speed measurement: FS 20000 rpm, resolution 1 revolution, accuracy ± revolution Torque measurement: FS 2000 Nm, resolution 1 digit, accuracy ±0.1% FS
EG&G FLOW TECHNOLOGY, flow meter	FT-12	FS 75 l/min Repeatability ±0.05% of reading Linearity ±0.5% over normal
EG&G FLOW TECHNOLOGY, flow meter	FT-16	FS 190 l/min Repeatability ±0.05% of reading Linearity ±0.5% over normal
Pt100 temperature sensor	TRC#P1A1X	FS 120 °C Precision ± 0.15 °C
K-type thermocouples		FS 1000 °C Precision ± 1.5 °C
KULITE, pressure sensor	ETQ-12-375	FS 10 bar Resolution infinitesimal Bandwidth 3kHz

2.1 Experimental activity

Specific experimental tests were carried out to understand better the behavior of the cooling circuit. First of all, the effects of temperature on coolant pressure were investigated. The coolant pressure was measured within the sealed cooling circuit for different coolant temperatures going from 15°C to 80°C, while both engine and pump were switched-off. Subsequently, the pump head was recorded at switched off engine and coolant temperature of about 15°C. During the tests, the pump voltage was fixed and the coolant flow rate in the circuit was varied through a gate valve. The circuit characteristic was also measured: in such case, the gate valve was left completely open and the coolant flow rate was varied by varying

the pump speed. Finally, in order to identify the onset of nucleate boiling several experimental campaigns were carried out, with different couples of engine speed – bmep values, both under steady state and transient conditions. Here, a specific test is reported, where the engine was operated under stable conditions at fixed engine speed (2000 rpm) and bmep (2 bar), while the coolant flow rate was varied from a maximum value of 2000 dm^3/h down to 500 dm^3/h as a sequence of steady state conditions. The coolant temperature at the engine inlet was kept constant at 85± 1°C. Tests were replicated on different days to assess the day-by-day repeatability [10]. The experimental tests are summarized in Table 2.

Table 2
Summary of the experimental tests.

Goal	Coolant Temperature	Operating conditions
Coolant thermal expansion analysis	15-85 °C	Engine off/pump off
Pump internal characteristic quantification	15°C	Engine off/ fixed pump speed, gate valve regulation
Circuit characteristic quantification	15°C	Engine off/ wide open gate valve, pump speed regulation
ONB detection	85°C	2000 rpm 2 bar bmep; 2000 to 500 dm^3/h flow rate

3. RESULTS

The analysis of the experimental data was focused on three main physical quantities: coolant pressure, coolant temperature and metal temperature. Other quantities like the coolant pressure pulsations or the engine-in/engine-out volumetric flow rate comparison, which proved useful for detecting the onset of the nucleate boiling in [11], where an atmospheric expansion tank was used, are not presented here. In the presence of a standard closed radiator expansion tank their indications are, in fact, not clear enough.

3.1 Coolant pressure

Coolant pressure not only gives useful information regarding the presence of nucleate boiling, but also directly and strongly influences its onset. Coolant pressure is determined by two main contributions.

First, it is determined by coolant expansion caused by temperature increase. In fact, as the coolant volume increases while the total volume of the circuit is fixed, the air volume within the expansion tank (Fig. 2) decreases and the circuit pressure increases. Figure 4 presents two sets of experimental data of coolant pressure vs. coolant average temperature (symbols), which were collected on different days, and which were obtained by keeping both the engine and coolant pump off. The experimental data show a linear trend in the 60-85 °C range of temperature. Data, however, differ considerably day-by-day even though the coolant level within the cooling system was kept constant; this is due to the temperature of the water within the radiator tank (Fig. 1), which controls coolant engine inlet temperature, and is therefore not constant. This affects the air temperature within the radiator expansion tank (Fig. 2) and consequently its pressure. Out of this arises a warning on the repeatability of the nucleate boiling phenomenon in the engine under real operating conditions. A difference in the initial coolant level within the cooling circuit or an air leakage of the radiator cap (a component costing few euro) would cause a significant discrepancy in the operating pressure of the cooling circuit and consequently an important variation in the tendency to the onset of nucleate boiling.

A second contribution to coolant pressure is determined by the pump head. The pump speed increases with voltage and, for any fixed voltage, the pump head decreases as the flow rate increases. Figure 5 shows the characteristic of the pump at 13V and the pressure-flow rate characteristic of the hydraulic circuit. It is interesting to observe that, as the circuit is sealed, the pump action develops partly by determining an increase in the absolute pressure at the delivery side and for the remaining part by determining a decrease in the pressure at the suction side. This second contribution can be even greater than the first one and changes as the coolant temperature varies (Fig. 6).

During the tests for ONB detection (Table 2 row 4), the coolant temperature was measured; the coolant pressure was then estimated by adding the contribution due to the thermal expansion, which results from this temperature on the basis of Fig. 4 correlation, to the contribution due to pump head. Pump head variations due to the fluid temperature increase would only provide a second order correction. The results are presented in Figure 7.

The effect of coolant expansion is computed in the two different situations of Figure 4. This determines a variation of 0.4-0.5 bar in the coolant total pressure of Figure 7. However, the trend is similar in both cases and this pressure rises as the coolant flow rate increases. In Figure 8, this calculated total coolant pressure has been made equal to the experimental pressure at engine inlet in the range of coolant flow rate 1400-1700 dm^3/h and compared with the experimental value of coolant pressure measured at engine inlet and engine outlet as a function of coolant flow rate.

Independently of the absolute values, it is evident that the trend is the same for coolant flow rates higher than 1200-1300 dm^3/h, where the heat transfer regime is that of single-phase forced convection. For lower coolant flow rates, the measured engine-in and engine-out pressure increases as the coolant flow rate diminishes and the reason is the onset of nucleate boiling within the cooling circuit. In fact, the presence of bubbles makes the coolant thermal expansion greater than the one due to temperature increase only. This difference in pressure trend can therefore be used as an indication of the onset of nucleate boiling. For coolant flow rates of about 1200-1300, however, bubbles are not yet visible at the transparent window, as they condense and collapse within the bulk flow short after they form; their presence can be noted for much lower flow rates, about 950-1050 dm^3/h. As the coolant flow rate further diminishes the fully developed sub-cooled boiling condition establishes at a value of about 950 dm^3/h, where the bulk liquid temperature reaches the saturation temperature (saturated boiling), and a net vapor production is visible at the transparent window (*FBD*). These two flow rates values are indicated by the two vertical dashed lines in Fig. 8, where a few pictures of the coolant, taken through the transparent window, are also presented.

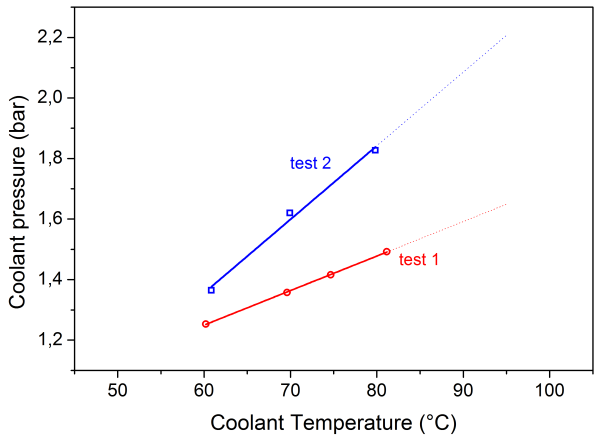

Fig. 4 Influence of the coolant temperature on coolant pressure within the sealed circuit at switched-off engine and pump.

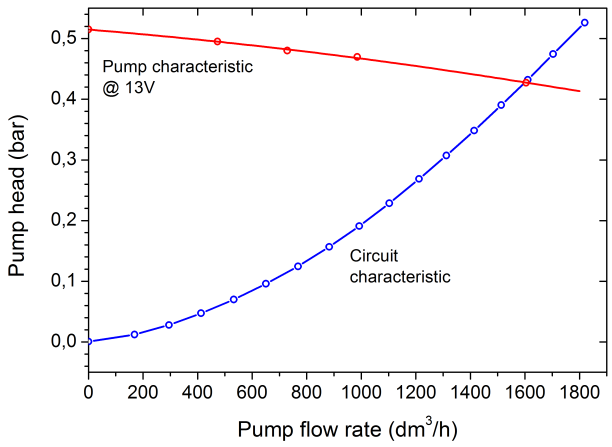

Fig. 5 Pump and circuit characteristics.

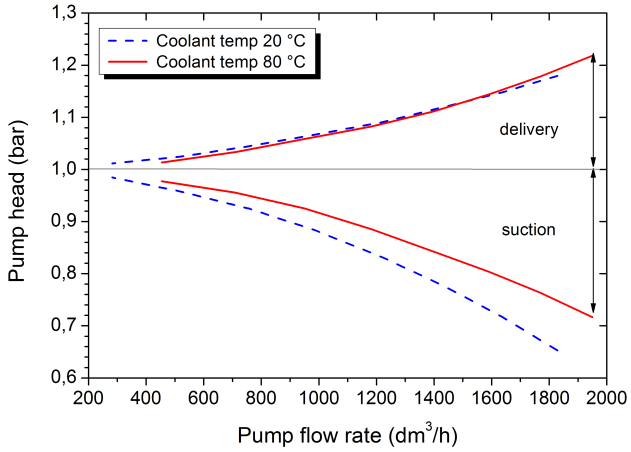

Fig. 6 Fractions of pump head which develops at pump delivery and at pump suction side.

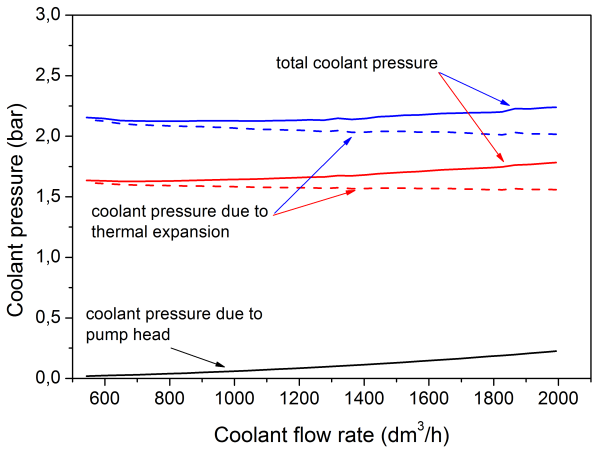

Fig. 7 Coolant pressure owing to the sum of coolant expansion and pump head contributions as a function of volumetric coolant flow rate.

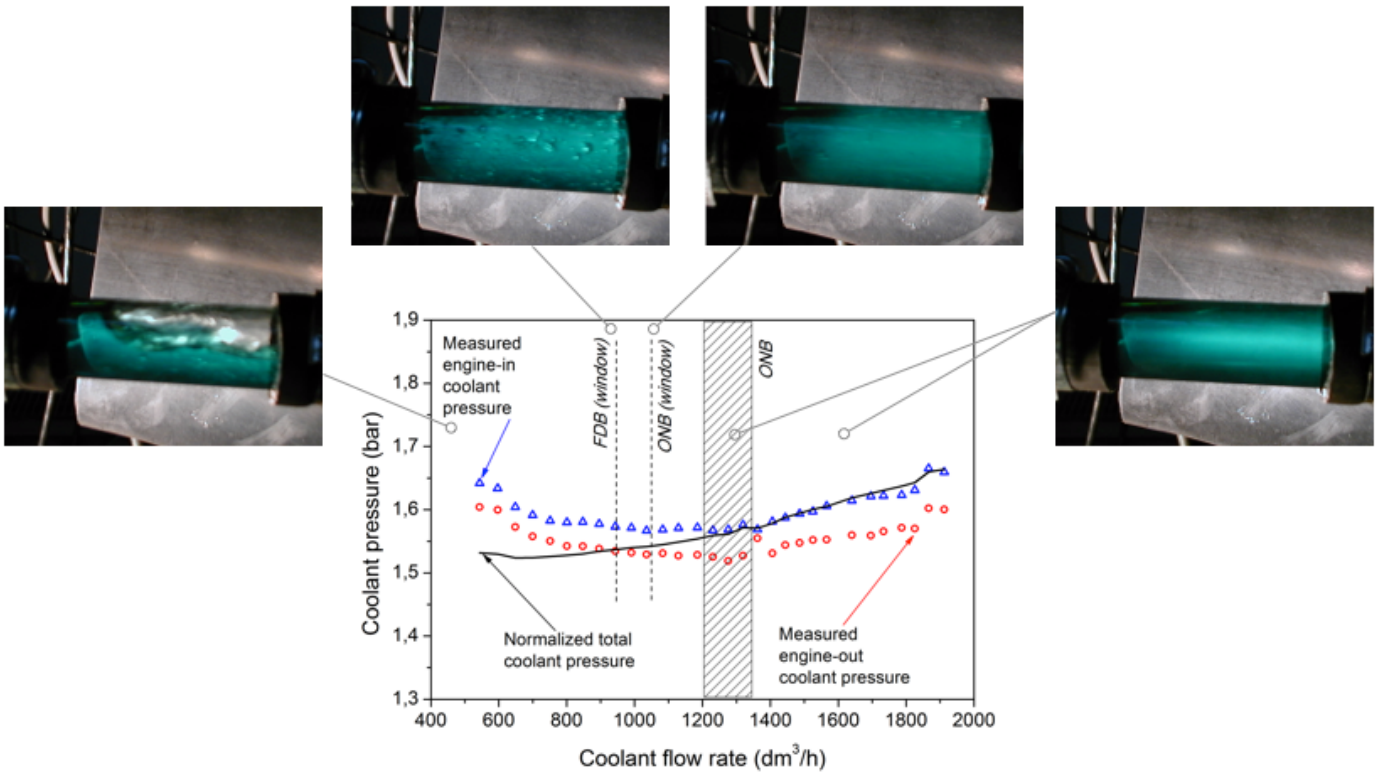

Fig. 8 Comparison of coolant pressure owing to the sum of coolant expansion and pump head contributions and coolant pressure measured at engine inlet and engine outlet.Pictures of the coolant flow through the transparent window are also shown.

3.2 Coolant temperature

Another hint of the presence of nucleate boiling can be found in the increase of coolant temperature from engine inlet to engine outlet (Fig. 9). In fact, if the heat transfer regime is single-phase forced convection, this temperature difference increases almost linearly as the volumetric coolant flow rate is reduced, while the fuel flow rate, engine speed and engine inlet coolant temperature are kept constant. This is due to the fact that coolant density can be considered constant within the temperature range of interest. On the contrary, when nucleate boiling develops, the coolant temperature from inlet to outlet of the engine shows a more rapid increase as the volumetric coolant flow rate is reduced. In this case, in fact, the presence of bubbles causes a significant decrease in coolant density. More details can be found in [10].

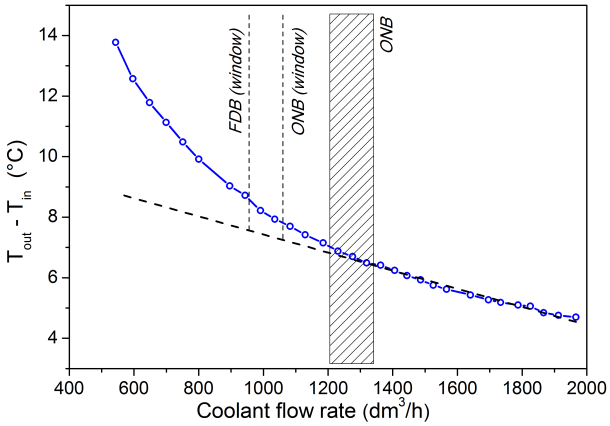

Fig. 9 Difference between coolant temperature at engine outlet and at engine inlet as a function of volumetric coolant flow rate. Dashed area indicates the Onset of Nucleate Boiling (ONB). Operational condition: 2000 rpm, bmep=2 bar.

3.3 Wall temperature

While coolant pressure and engine-in engine-out temperature difference can be explained in terms of engine behavior in the whole, wall temperatures are local values, which can be interpreted only by taking into account the characteristics of the single cylinders. The single cylinders differ in terms of thermal performance for two main reasons: differences in coolant flow rate distribution and differences in cylinder coolant inlet temperature. Cylinder 4, the left-most one in Fig. 3, is cooled by a smaller coolant flow rate at a lower temperature; the subsequent cylinders are cooled by an increasing coolant flow rate, which comes partly from the head of the previous cylinder, and partly from "fresh" coolant rising from the cylinder block. In addition, there is a further diversity due to the different contribution of radiation and natural convection to the surrounding ambient. External cylinders (no. 4 and no. 1, Fig 3) have a much larger heat exchange surface than inner cylinders no. 2 and no. 3. For the engine used in the reported tests, this surface was estimated to be about 35% of the whole engine external surface for cylinders no. 4 and no. 1 and about 15% for cylinders no. 2 and no. 3. However, as the free convection and radiation contributions are small, the coolant must remove a fraction of the whole engine heat transfer for each cylinder, which is only slightly different: it was estimated at 0.26 for cylinders no. 2 and no. 3 and 0.24 for cylinders no. 1 and no. 4.

Data of coolant flow rate distribution among the different cylinders were available from industry confidential reports in the framework of the cooperative research project PON01_01517, CUP: B21H11000400005, with financial support of the European Commission. On the base of these data, a model of the cooling system of an ICE that predicts dynamically the onset of the nucleate boiling phenomenon [10] was used. Inputs of the model are engine speed, fuel flow rate, coolant pressure and temperature at the engine inlet and coolant volumetric flow rate. The model provides engine outlet coolant temperature, spatial averaged metal temperature and the amount of heat exchange area that operates under nucleate boiling conditions. Even though the model [10] was developed for the whole engine, it was used, in this case, to predict the average wall temperature, the cylinder-out coolant temperature and the presence and intensity of nucleate boiling for each cylinder. The *NB_index* is the metrics used to quantify the intensity of nucleate boiling and is defined as:

$$NB\_index = \frac{\left(q_w - q_{ONB}\right)}{q_{ONB}} \qquad (1)$$

where q_w is the actual thermal flux and q_{ONB} is the thermal flux required for the onset of nucleate boiling.

For cylinder no. 4 the inlet coolant temperature was the engine-in coolant temperature; for the subsequent cylinders the inlet temperature was computed as a mass-weighted average of the temperature of the coolant coming from the previous cylinder and of the temperature of the coolant coming up in the cylinder block, which was assumed still equal to the engine-inlet one. Finally the engine-out coolant temperature derives from the mixing of the coolant coming from the head of cylinder no. 1 and the coolant coming up from the right-most passages of cylinder no.1 (Fig. 3). The simulated working conditions are: total engine-in coolant flow rate 1230 dm^3/h; engine-in coolant temperature 85 °C; 2 bar bmep; engine speed 2000 rpm.

The simulation results are reported in Figure 10. Cylinder inlet temperature increases from the engine-in temperature of cylinder 4 (85 °C) to 89, 90, 90 °C at inlet of cylinders 3, 2 and 1 respectively. The predicted cylinder-out temperature is almost constant for the different cylinders (93-94 °C). The predicted engine-out temperature is 90 °C, which is in good agreement with the experimental value (92 °C). As a consequence of coolant temperature and coolant flow rate, which steadily increases from cylinder 4 to cylinder 1, the highest wall temperature is predicted in cylinder 2; wall temperature in cylinders 1 and 3 is only 1-2 °C

lower, while the temperature of cylinder 4 is significantly lower (6 °C). Cylinder 2 is also the one which presents nucleate boiling (*NB_index > 0*), while for cylinders 1, 3 and 4 the model predicts the absence of nucleate boiling (*NB_index* < 0).

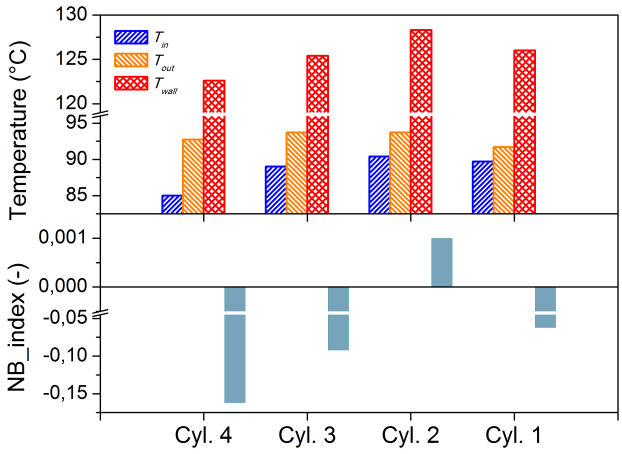

Fig. 10. Predicted values of coolant inlet temperature, coolant outlet temperature, average wall temperature and intensity of nucleate boiling (NB_index) for the different cylinders and for the whole engine. Operational condition: 2000 rpm, bmep=2 bar, 1306 dm^3/h coolant flow rate.

Figure 11 shows measured head temperatures evolution at exhaust and intake valve bridge locations for an operating condition of 2000 rpm and 2 bar bmep, while the coolant flow rate ranges from 1900 dm^3/h to 500 dm^3/h. Model results and experimental data agree, to indicate that the cylinder 4 is significantly colder than cylinders 1,2 and 3. The difference in average wall temperature predicted by the model is obviously amplified in the experimental exhaust valve bridge temperatures, which are the highest in the engine.

The wall temperature experimental data can also be used to identify the ONB. It was shown in [10] that the thermal power transferred from the hot gases to the engine wall can be expressed as $\dot{Q}_g = cN^{n_1}\dot{m}_c^{n_2}\dot{m}_f^{n}$ where c, n, n_1 and n_2 are constants and $\dot{m}_c$, $\dot{m}_f$ and N are coolant mass flow rate, fuel mass flow rate and engine speed, respectively. Therefore, for constant engine speed and fuel flow rate, $\dot{Q}_g \propto \dot{m}_c^{n_2}$ where n_2 was estimated at 0.15 for the engine used for the reported tests. Under single phase forced convection conditions, the

thermal power transferred from the engine wall to the coolant is $\dot{Q}_c = hA\left(T_w - T_c\right)$ where T_c is the coolant average temperature, T_w is the average wall temperature, A is exchange area and the heat transfer coefficient is $h \propto \dot{m}_c^{0.8}$. It was also shown in [10], that, in the presence of single phase forced convection, the difference between engine-out and engine-in coolant temperature increases linearly as the coolant volumetric flow rate diminishes (Fig. 9); therefore, if the engine-in coolant temperature is kept constant, as in present case, this holds also for the average coolant temperature: $T_c \propto \dot{m}_c$. Under steady-state conditions, $\dot{Q}_g = \dot{Q}_c$ and, therefore, $T_w \propto \dot{m}_c^{-0.65} - \dot{m}_c$. For the values of the involved constants, it results that the first term is one order of magnitude lower than the second one for coolant flow rates greater than about 0.37 kg/s (1300 dm^3/h), so that, if the heat transfer regime is single phase forced convection:

$$T_w = const - K \cdot \dot{m}_c \qquad (2)$$

When ONB occurs, the coolant temperature increases more than linearly as the coolant flow rate diminishes (Fig. 9) and this also happens for the wall temperature.

Figure 12 shows that the nucleate boiling occurs first in cylinder 2, for a coolant flow rate of about 1200-1300 dm^3/h. This value is in agreement with the ones resulting from the analysis of coolant pressure and temperature data for the whole engine (Fig. 8 and 9). Figure 12 also shows that, as the coolant flow rate further diminishes the nucleate boiling occurs in the order in cylinders 3, 1 and 4. This is in agreement with results of the simplified analysis carried out with the zero-dimensional model [10] shown in Fig. 10.

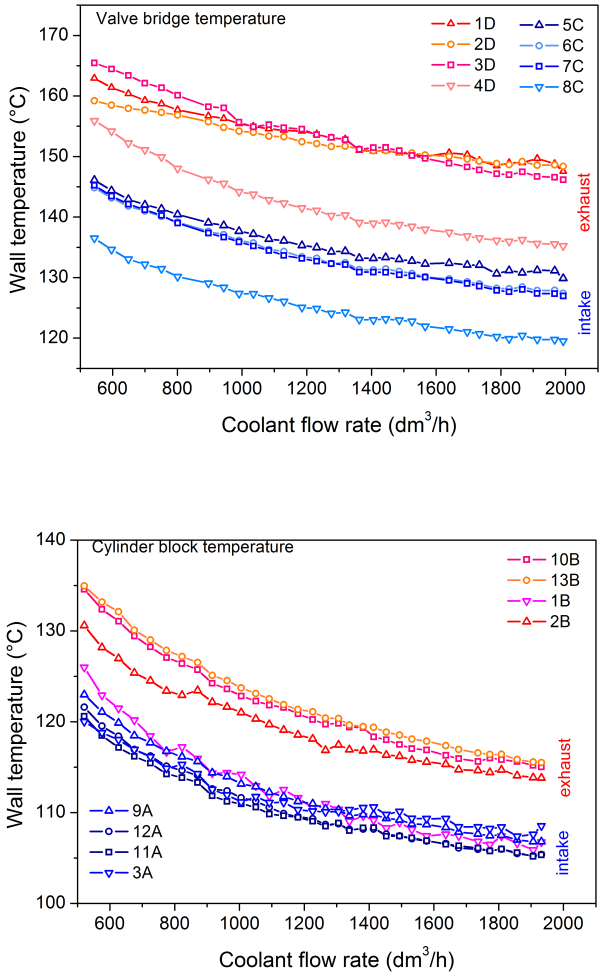

Fig. 11. Wall temperature recorded in the engine head (top) and in the cylinder block (bottom), as a function of volumetric coolant flow rate. Numbers in the legend refer to the thermocouples designation in Fig. 3. Operational condition: 2000 rpm, bmep=2 bar

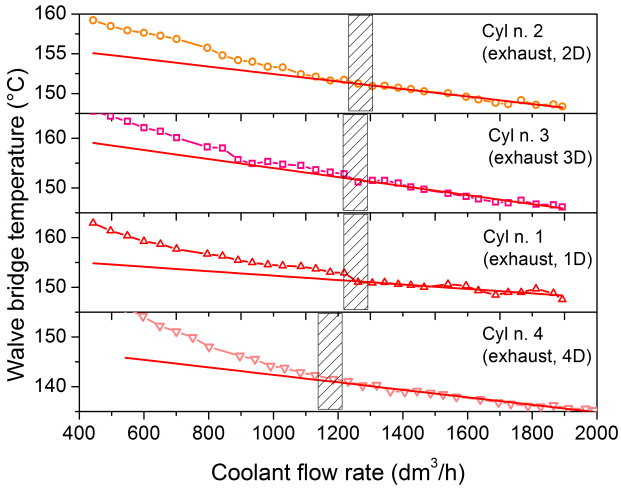

Fig. 12. Wall temperature recorded in the engine head as a function of volumetric coolant flow rate. Dashed area indicates the Onset of Nucleate Boiling (ONB). Operational condition: 2000 rpm, bmep=2 bar.

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331 **3.4 Nucleate Boiling Area**

332 The fraction of cooling circuit, which operates under nucleate boiling, cannot be
333 obtained directly from experimental data. The model which predicts the onset of the nucleate
334 boiling [10] was therefore used. Figure 13 shows the ratio of this quantity to the total heat
335 exchange area and the corresponding value of *NB_index*. The results of the model are well in
336 agreement with the conclusions of the experimental observation. The calculated Nucleate
337 Boiling Area is, in fact, zero for coolant flow rates higher than about 1250 dm^3/h and
338 increases as the coolant flow rate is reduced below this value.

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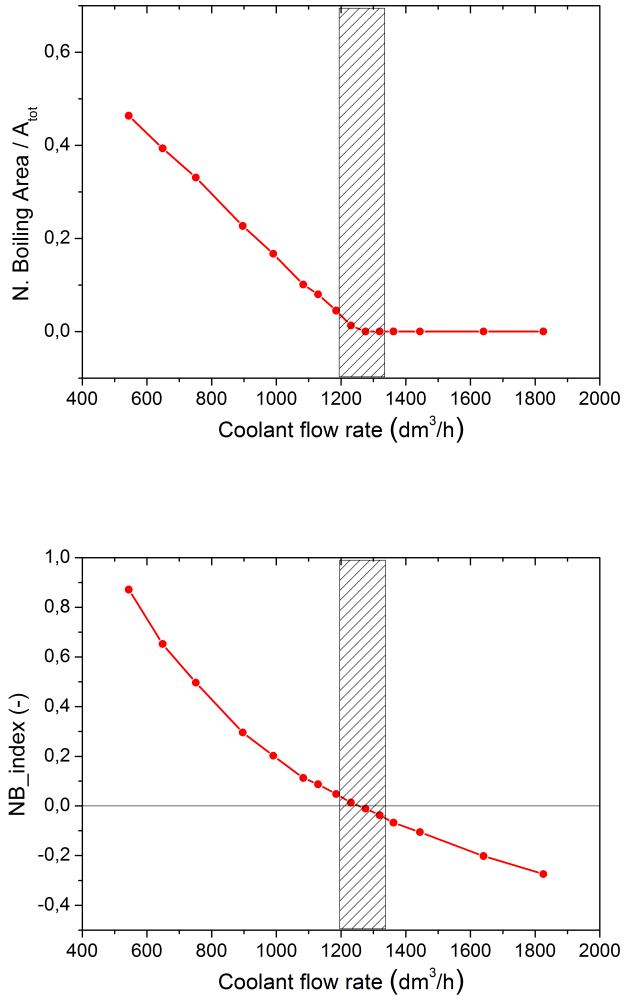

Fig. 13. Fraction of cooling circuit area which operates in nucleate boiling regime (top) and *NB_index* (bottom) as a function of coolant flow rate.

4. SUMMARY AND CONCLUSIONS

Several experimental campaigns were carried out to investigate the onset of nucleate boiling condition: the coolant flow rate was varied in a sequence of steady state conditions, while engine speed, bmep and inlet coolant temperature were maintained constant. Attention was paid to coolant pressure and temperature behavior and to wall temperature, which result from coolant flow rate variations. The indications of other quantities like the coolant pressure pulsations or the engine-in engine-out volumetric flow rate comparison are not clear enough when a standard closed radiator expansion tank is used.

The trends of these quantities provide clear hints to identify the coolant flow rate range where the onset of nucleate boiling occurs, which are in good agreement among them. However, it appears rather problematic to assess the occurrence of nucleate boiling from one single measurement of one or more of these quantities; only the analysis of their trend seems to make it possible. This is especially true when engine operating conditions are continuously varying or when the measurements have to be taken on board a vehicle. In addition, a warning must be flagged on the repeatability of the nucleate boiling phenomenon in the engine under real operating conditions. A difference in the initial coolant level within the cooling circuit or an air leakage of the radiator cap would cause a significant discrepancy in the operating pressure of the cooling circuit and, consequently, an important variation in the tendency to the onset of nucleate boiling.

The use of a lumped parameter dynamic model of the cooling system, which incorporates the description of the nucleate boiling regime [10], seems, on the contrary, adequate to determine whether a given engine operating condition causes nucleate boiling or not and is therefore suitable for the on-board identification of the heat transfer mechanism.

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FIGURES CAPTIONS

Fig.1 Schematic of the test rig.

Fig.2 Schematic of the cooling circuit.

Fig.3 Thermocouples location in the engine metal and cylinder number identification.

Fig.4 Influence of the coolant temperature on coolant pressure within the sealed circuit at switched-off engine and pump.

Fig.5 Pump and circuit characteristics.

Fig.6 Fractions of pump head which develops at pump delivery and at pump suction side.

Fig.7 Coolant pressure owing to the sum of coolant expansion and pump head contributions as a function of volumetric coolant flow rate.

Fig.8 Comparison of coolant pressure owing to the sum of coolant expansion and pump head contributions and coolant pressure measured at engine inlet and engine outlet.Pictures of the coolant flow through the transparent window are also shown.

Fig.9 Difference between coolant temperature at engine outlet and at engine inlet as a function of volumetric coolant flow rate. Dashed area indicates the Onset of Nucleate Boiling (ONB). Operational condition: 2000 rpm, bmep=2 bar.

Fig.10 Predicted values of coolant inlet temperature, coolant outlet temperature, average wall temperature and intensity of nucleate boiling (NB_index) for the different cylinders and for the whole engine. Operational condition: 2000 rpm, bmep=2 bar, 1306 dm^3/h coolant flow rate.

Fig.11 Wall temperature recorded in the engine head (top) and in the cylinder block (bottom), as a function of volumetric coolant flow rate. Numbers in the legend refer to the thermocouples designation in Fig. 3. Operational condition: 2000 rpm, bmep=2 bar.

Fig.12 Wall temperature recorded in the engine head as a function of volumetric coolant flow rate. Dashed area indicates the Onset of Nucleate Boiling (ONB). Operational condition: 2000 rpm, bmep=2 bar.

Fig.13 Fraction of cooling circuit area which operates in nucleate boiling regime (top) and *NB_index* (bottom) as a function of coolant flow rate..

TABLE CAPTIONS

Table 1 Main measurement devices specifications.

Table 2 Summary of the experimental tests.