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Energy Procedia 81 (2015) 907 – 917

Energy Procedia

69th Conference of the Italian Thermal Engineering Association, ATI 2014

Experimental investigation and lumped-parameter model of the cooling system of an ICE under nucleate boiling conditions

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Abstract

The work presents the results of experimental tests, which were carried out on a small-displacement spark ignition engine, where a low flow rate electric pump was used to substitute the standard crankshaft-driven one. The engine was then operated both under usual single-phase heat transfer regime and under nucleate boiling conditions. The engine was properly instrumented in order to record coolant pressure, temperature and flow rate as well as metal temperatures.
The experimental investigation was coupled with the development of a dynamic lump-parameter model of the engine cooling system. The model calculates the spatial averaged metal temperature, the engine-out coolant temperature and the fraction of metal heat transfer area which is involved in nucleate boiling as a function of engine-in coolant flow rate, pressure and temperature, fuel mass flow rate and engine speed.
The experimental data and the model results show a good agreement and the model is suitable to develop a coolant flow rate control system. This facilitates faster engine warm-up, lower fuel consumption and lower CO_2 emissions, which can be significant under low-load and cold-start conditions.

Peer-review under responsibility of the Scientific Committee of ATI 2014

Keywords: Cooling system; electric pump; nucleate boiling; internal combustion engines; experimental data; model.

1. Introduction

Reliable models are today required to describe the behavior of the different interconnected sub-systems, which form the structure of a modern powertrain. Fuel consumption and Greenhouse Gas emissions of internal combustion

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Peer-review under responsibility of the Scientific Committee of ATI 2014
doi:10.1016/j.egypro.2015.12.145

engines, which are today the focus of regulatory agencies [1], are determined, in fact, not only by how engines are designed but also on the way they are managed and controlled. In order to obtain more efficient, less-consuming vehicles, car manufacturers therefore address on the one hand new engine architectures (downsized and turbocharged engines, Spark Ignition Direct Injection engines etc.) [2] and, on the other, new methodologies for designing and managing modern powertrains. On-board powertrain control systems, therefore, must change from the current look-up tables approach to the model-based methodology, which must rely on simple, robust models of the physical systems.

A joint research project (PON01-01517) is ongoing, among industries (Fiat Group Automobiles –FGA, today FCA, Teoresi s.r.l.), University and Research Centers (Università di Napoli "Federico II", C.N.R. Istituto Motori, CNISM, Università del Sannio, Università della Calabria,) with the aim of generating new Innovative Development Methodologies For Automotive Powertrain. Within this framework, the University of Calabria is investigating a new cooling system for internal combustion engines.

The basic idea of the new cooling system is to use an electric water pump, which enables the control system to set the coolant flow rate independently of engine speed. The new system generally uses much lower coolant flow rates than the standard one and this fact drives the system toward the nucleate boiling regimes [3-8]. The advantages of such a cooling system have been widely illustrated: smaller coolant mass, smaller radiator, faster warm-up time, lower friction during the warm-up period, lower pump power requirements [4], and no risk of after-boiling [9].

A model of the cooling system, which is able to detect the onset of nucleate boiling is however needed in order to make the system controllable and then to enable the actual development of such an innovative system. Such a model has not been developed yet.

The paper presents an experimental investigation, which was carried out on a small SI engine, which was adequately instrumented and equipped with a small power electric pump, and a lump-parameter dynamic model of the cooling system, which proved to be able to detect dynamically the onset and the extent of the nucleate boiling [10]. As input data, the model needs quantities already available from the on-board instrumentation or that can be obtained from low cost transducers. The comparison of experimental data with model predictions shows a good agreement.

2. Experimental investigation

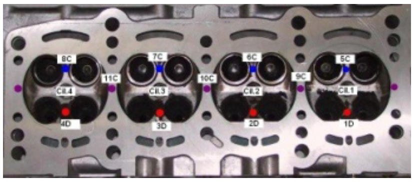
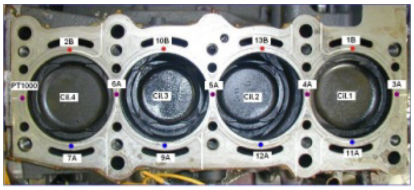
2.1. Set-up

A small-size spark ignition engine (about 1.2 dm^3, 60 kW) was used for experimental tests. The test rig was equipped with an eddy current engine torque dynamometer and with a gravimetric-type fuel consumption metering system. A low flow rate electric pump (127 W at 15 V, 2092 dm^3/h maximum flow rate) was used to substitute the standard crankshaft-driven coolant pump. The cooling circuit layout includes the heater for car passengers comfort and the standard production radiator. The latter, used as condenser/radiator, was immersed in a tank filled with water, whose temperature was controlled by means of flowing cool water: a digital PID regulator was used to keep engine inlet coolant temperature constant within a ±1 °C error band.

The temperature measurements of the cylinder block and cylinder head were recorded at various locations in the metal with k-type thermocouples; sensors were installed within a small distance (~1 mm) from the gas/wall interface. Table 1 offers a map of thermocouples installation points. A miniature piezoresistive pressure transducer, 3 kHz bandwidth, was installed in the cooling circuit near the engine outlet to provide coolant pressure measurements.

Coolant temperatures were measured using PT100 type temperature sensors, accuracy ± 0.15 °C, installed at the engine inlet and outlet. Coolant volume flow rate was measured using turbine type flowmeters installed at the engine inlet and outlet, with a repeatability of ±0.05%. An optical access was also installed in the cooling circuit near the engine outlet, to observe visually the coolant flow pattern during experimental tests. A scheme of the engine test-bed and the experimental apparatus is shown in Fig. 1.

Table 1. Thermocouples location in the engine metal

Head and block measurement points	Thermocouples name / metal location
	1D,2D,3D,4D / head, exhaust valves bridge 5C,6C,7C,8C / head, intake valves bridge 9C,10C,11C / head, between cylinders
	1B,2B,10B,13B / block, exhaust manifold side 7A,9A,11A,12A / block, intake manifold side 3A,4A,5A,6A / block, between cylinders

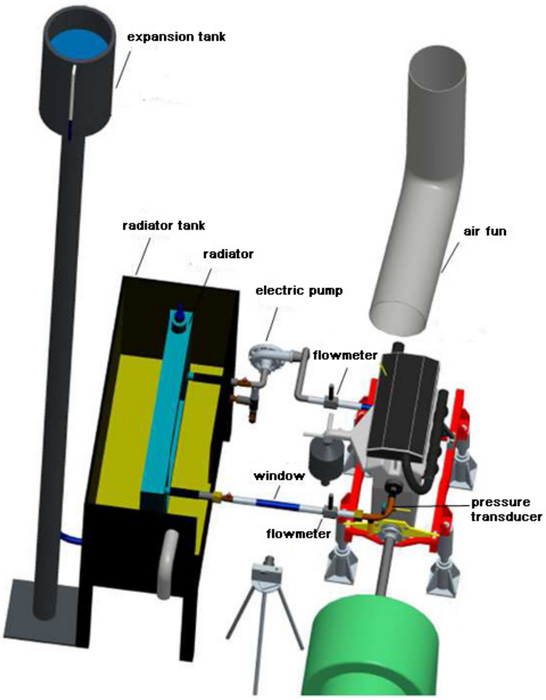


Fig.1 Schematic of the test rig.

2.2. Experimental tests

The experimental activity strategies were established in agreement with the industrial partner FGA as engine speed (rpm) x bmep (bar) couples of values: 1500x1; 2000x2; 2000x4; 2000x8; 3000x3; 3000x5; 4000x5. The aim of the research activity was to detect the onset of nucleate boiling conditions. The coolant flow rate was varied from a maximum value of 1900 dm^3/h down to a minimum value of 500 dm^3/h, while the coolant temperature at the engine inlet was kept constant at 85± 1°C. In this work, attention is focused on the evolution of the recorded engine metal temperatures under two-phase coolant conditions, which is used to validate the prevision of the zero-dimensional model described in the following paragraphs. Details on the other investigated quantities will be shown in a companion paper.

3. Model

The present section briefly describes the zero-dimensional model that was developed for predicting dynamically the behavior the cooling system of an SI Engine. The model identifies the heat transfer mechanism involved (forced convection/nucleate boiling) and predicts the instantaneous coolant temperature at engine exit, T_c, the space-averaged engine wall temperature, T_w, and the amount of wall area where nucleate boiling occurs, A_{nb}. The required input data are coolant flow rate, engine-in coolant temperature and pressure, fuel flow rate and engine speed**.** More details on the model description and development can be found in [10].

3.1. Heat exchange in nucleate boiling flow regimes.

For the mathematical modeling of nucleate boiling thermal exchange, empirical correlations, which are obtained on the basis of experimental observations, are commonly used for practical applications.

In the past few years, several studies have contributed to the development and the description of the heat exchange in the cooling system of ICEs, operating under nucleate boiling flow regimes [11-15].

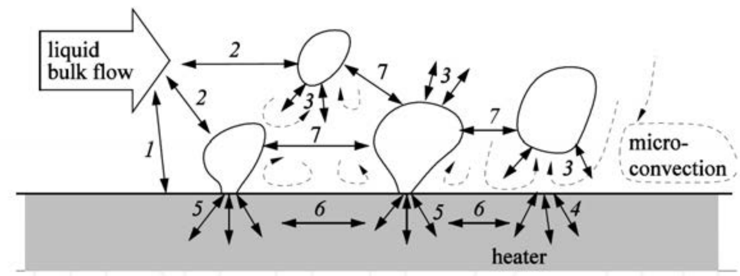


Fig.2 Interactions among liquid bulk flow, vapour and solid heated wall.

According to *Chen*, two basic mechanisms take part in the total heat transfer (Fig.2): ordinary macro-convection (*1*) and micro-convection associated with bubble nucleation, growth and detachment (*2-5*). The heat transfer mechanisms indicated by *6* and *7* are neglected; they correspond to the conductive heat exchange due to the temperature variation within the solid surface and to the bubble-bubble interaction, respectively.

The local heat flux, then, is given by the sum of two contributions:

$$q_w = h_{mac}\left(T_w - T_\infty\right) + h_{mic}\left(T_w - T_{sat}\right) \qquad (1)$$

where T_w is the wall temperature, T_{sat} is the flow saturation temperature and T_∞ is the bulk flow temperature.

The heat transfer coefficient h_{mac} (Eq.1) is computed by the *Dittus-Boelter* heat transfer coefficient for single phase flows and by a correction factor, which takes into account the presence of bubbles in the bulk flow [10, 17].

The micro-convective coefficient, h_{mic}, is obtained from the *Foster* and *Zuber* correlation for pool boiling [18], modified by *Chen* [16] in order to take into account the effects of the flow velocity, through a suppression factor.

3.2. Prediction of the onset of nucleate boiling.

In a typical subcooled flow, the formation of vapor bubbles starts at T_{ONB} (Fig. 3, [19]). The ΔT_{sat} between wall and coolant required for the onset of nucleate boiling is computed following the approach presented by *Frost* and *Dzackowich* [20]:

$$(\Delta T_{sat})_{ONB} = (T_w - T_{sat})_{ONB} = X \,\mathrm{Pr}_l\, q_w^{0.5} \tag{2}$$

where X is a parameter determined by the fluid physical properties and q_w denotes the heat flux through the walls. The heat flux required for the onset of nucleate boiling can be computed by the heat transfer equation for subcooled boiling region as:

$$q_{ONB} = h_{fc}\left((\Delta T_{sat})_{ONB} + (T_{sat} - T_\infty)\right) \tag{3}$$

By solving Eq. (2) and Eq. (3) simultaneously, the heat flux, q_{ONB}, and the wall temperature, T_{ONB}, needed for the onset of nucleate boiling can be computed.

Furthermore, nucleate boiling area, A_{nb} is obtained as a fraction of the total heat exchange surface, A, as:

$$\begin{cases} A_{nb} = \dfrac{(q_w - q_{ONB})}{q_w}\cdot A & q_w > q_{ONB} \\ A_{nb} = 0 & q_w \le q_{ONB} \end{cases} \tag{4}$$

3.3. Zero-dimensional model equations.

Through the energy conservation equations, it is possible to calculate the spatial-averaged wall and coolant temperature, T_w and T_c:

$$\dot{T}_w C_w = \dot{Q}_g - \dot{Q}_c \tag{5}$$

$$\dot{T}_c C_c = \dot{Q}_c - \dot{Q}_r \tag{6}$$

where C_w and C_c are the engine and coolant thermal capacities, respectively, and $\dot{Q}_g$, $\dot{Q}_c$ and $\dot{Q}_r$ are the thermal power transferred by the combustion gases to the engine walls, by the engine walls to the coolant and by the coolant to the radiator respectively.

For $\dot{Q}_g$ an empirical expression has been developed, which correlates the fuel thermal power to the fuel flow rate, $\dot{m}_f$, coolant flow rate, $\dot{m}_c$, and engine speed, N, through the coefficients c, n, n_1 and n_2 [10]:

$$\dot{Q}_g = c \cdot N^{n_1} \dot{m}_c^{n_2} \left(\dot{m}_f\right)^n \tag{7}$$

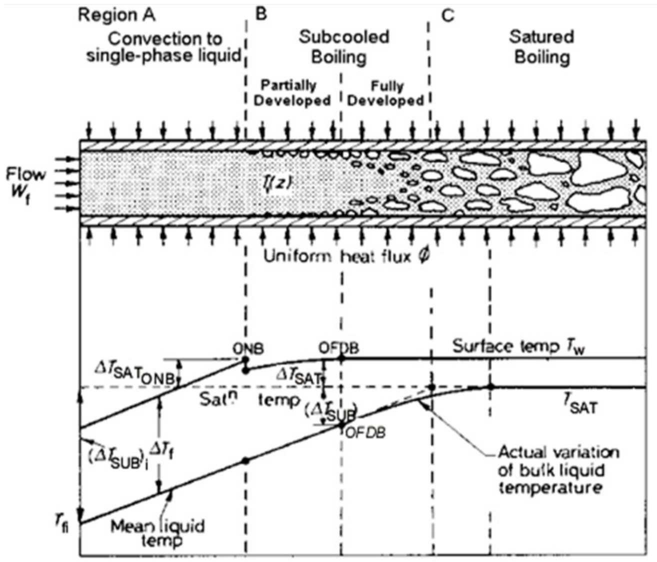


Fig.3 Wall and liquid temperatures variations along the flow direction. Onset of nucleate boiling conditions are marked by ONB [19]

It is worth pointing out that in the present investigation, the use of an electrical pump gives the possibility of varying the coolant flow rate independently of engine speed. Consequently, Eq. (7) takes into account the effects of $\dot{m}_c$ and N on $\dot{Q}_g$ separately. The coefficients c, n, n_1 and n_2 were estimated through an experimental campaign under steady-state conditions.

The thermal power transferred to the coolant by the engine walls, $\dot{Q}_c$, is computed as:

$$\dot{Q}_c = h_{mac} A \left(T_w - T_\infty\right) + h_{mic} A_{nb} \left(T_w - T_{sat}\right) \tag{8}$$

where A_{nb} is given by eq. 4 and A is the total heat exchange area.

4. Results

Several experimental campaigns were carried out to highlight the behavior of the engine under nucleate boiling regimes and to validate the lumped parameter model:

- constant engine speed, constant bmep, with decreasing coolant flow rate;
- constant engine speed, constant coolant flow rate, with step increase of the bmep;
- constant coolant flow rate, constant throttle position (roughly constant bmep) with step increase of the engine speed.

The constant engine speed – constant bmep data will be presented here; more details on the other test data can be found in [10].

Experimental tests were carried out by reducing the coolant flow rate as a sequence of steady state conditions, at fixed engine speed and bmep (2000 rpm, 2 bar). The data acquisition clock was stopped while the coolant flow rate was changed, so that no transient behavior was recorded. Figure 4 plots the time history of the coolant flow rate (top) and of the coolant pressure (bottom). In Fig. 4 also the coolant pressure within the switched-off engine is reported; the coolant temperature was 85 °C and the time-flow rate correspondence is the same as in Fig 4 (top).

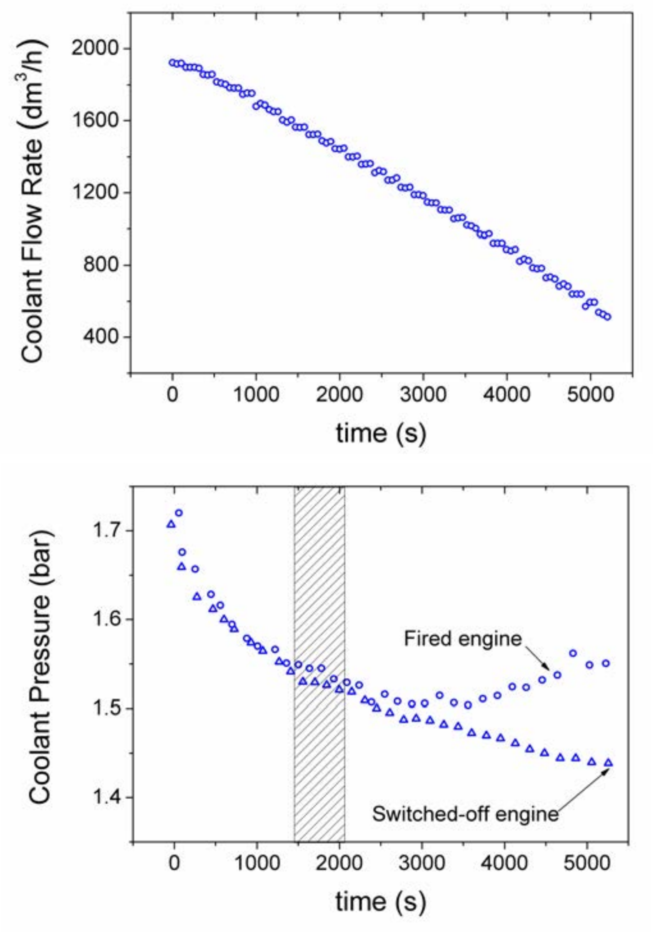


Fig 4. Experimental engine-out coolant pressure. Engine speed 2000 rpm; 2 bar bmep

As long as the flow regime is that of forced convection (flow-rates higher than about 1450-1550 dm^3/h), the behavior of the coolant pressure within the fired engine and within the switched-off engine is very similar: the pressure decreases as the flow rate diminishes with a parabolic low, and this is true until the time 1500-2000 s, (coolant flow rate of 1450-1550 dm^3/h). After this time, as the coolant flow rate diminishes, vapor bubbles start to form in the fired engine and the coolant pressure starts to increase, whereas the pressure within the switched-off engine still decreases with the usual parabolic law. Therefore, the region of 1500-2000 s (1450-1550 dm^3/h coolant flow rate) can be identified as the Onset of Nucleate Boiling.

Figure 5 presents the nucleate boiling area computed by the model; it shows that up to a time about 1500 s (flow rate about 1550 dm^3/h) the heat transfer is purely convective (zero Nucleate Boiling Area). After this time, as the flow rate still diminishes, the nucleate boiling Nucleate Boiling Area increases.

The total heat transfer coefficient computed by the model in the same operating conditions is plotted in Fig. 6. In the single-phase regime, the heat transfer coefficient diminishes as the coolant flow rate is reduced; it suddenly increases during the transition from single phase to nucleate boiling and then reaches a value (7-8000 $W/m^2/K$), which is significantly higher than the ones available in the single-phase regime.

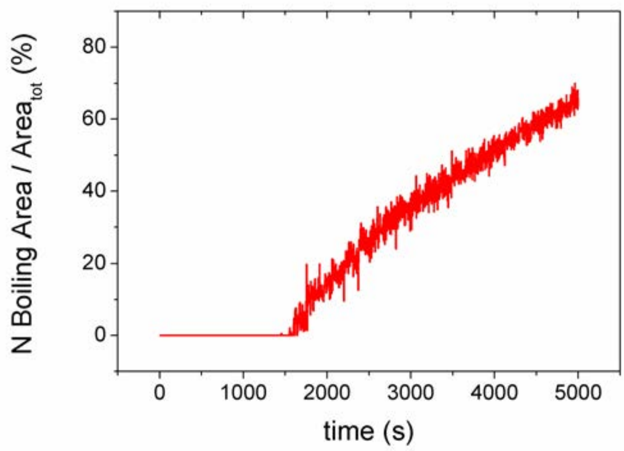


Fig 5. Fraction of the heat exchange area subjected to nucleate boiling (model). Engine speed 2000 rpm; 2 bar bmep.

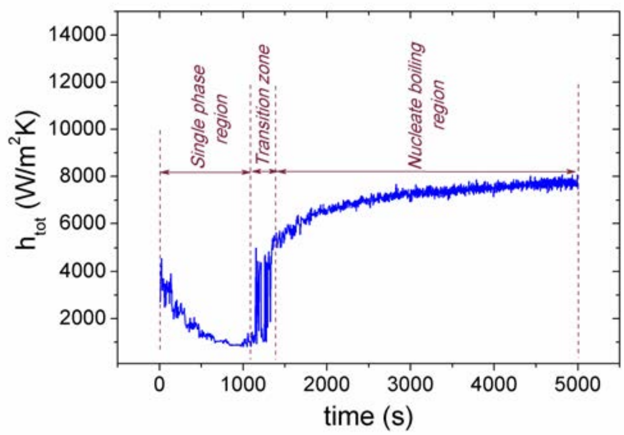


Fig 6. Total heat exchange coefficient as a function of coolant flow rate. The three heat exchange regimes (single phase, transition zone, nucleate boiling) can be clearly distinguished. Engine speed 2000 rpm; 2 bar.

Figure 7 presents the experimental temperatures measured near the exhaust and the intake valves (dots) and the average engine wall temperature, as obtained by the model (continuous line). The spatial-averaged wall temperatures model prediction lies between the experimental curves.

The model also predicts the engine-out coolant temperature, which is shown in figure 8 in comparison with the experimental data. The agreement is satisfactory and the average error amounts to 0.4 %.

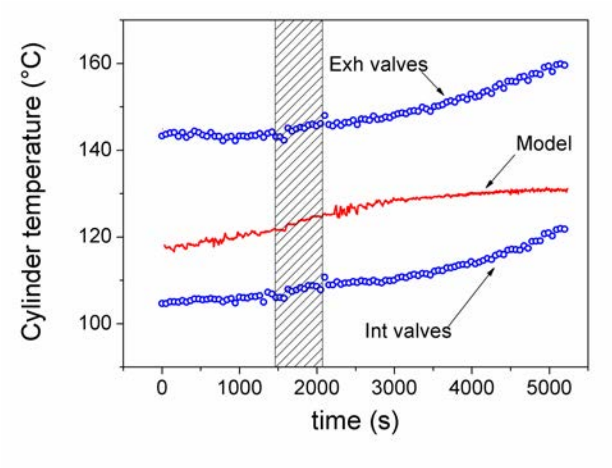


Fig 7. Engine wall temperature as a function of coolant flow rate. Symbols: experimental data; line: model result (spatial averaged temperature of the metal). Engine speed 2000 rpm; 2 bar.

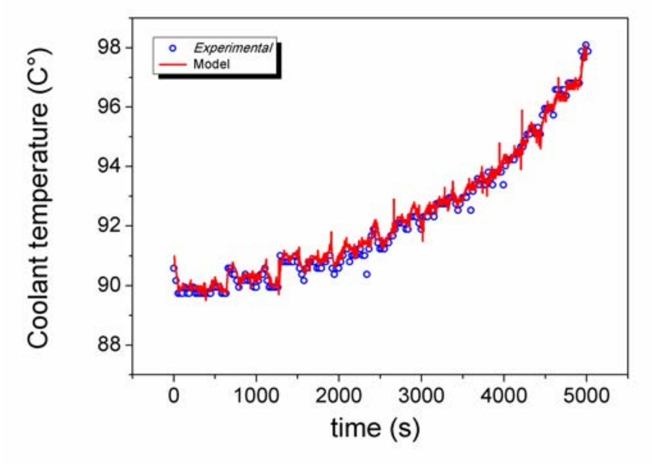


Fig 8. Engine out coolant temperature as a function of coolant flow rate; comparison of experimental data (symbols) and model results (continuous line).. Engine speed 2000 rpm; 2 bar bmep.

Finally, Figure 9 includes the thermal power entering with the fuel, the brake power and the thermal power delivered to the coolant. This last quantity is compared with the thermal power delivered to the engine wall by the fuel, as predicted by the proposed model. It can be observed that, as the fuel mass flow rate is kept constant, the total power entering the engine and brake power are constant. On the contrary, the thermal power delivered by the fuel to the wall diminishes as coolant flow rate decreases, as a result of wall temperature increase.

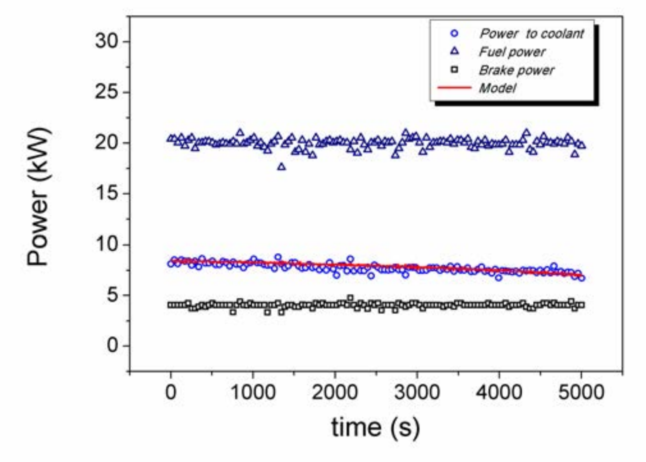


Fig 9. Global power balance as a function of coolant flow rate: power of entering fuel (triangles), thermal power transferred to the coolant (experimental, circles, and model result, line) and brake power (squares). Engine speed 2000 rpm; 2 bar bmep.

Acknowledgements

The investigation was funded by the Italian Ministry of University and Research under the project PON01_01517, CUP: B21H11000400005, with financial support of the European Commission.

Technical support was provided by Mr. Ernesto Ramundo, whose cooperation is gratefully acknowledged.

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