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Control-Oriented Engine Thermal Model

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Abstract

The optimization of modern internal combustion engines and vehicles led several researchers to investigate the effects of the coolant system on overall efficiency losses. Electric water pumps have been proposed as a solution to decrease the high power consumption that typically affects mechanically-driven water pumps at high engine speed. Furthermore, decoupling the coolant flow from engine speed allows achieving a better warm-up behavior. The coolant system components, however, also impact vehicle efficiency: the radiator area affects the overall aerodynamic drag coefficient, especially for race vehicles and motorcycles.

A thermal model can be used to assess the effects of the components characteristics (pump size, efficiency, speed; radiator surface, fan size, etc.) both on the coolant system capability to reach and maintain the target temperature, and the power it requires. The same model-based approach can be used for optimal thermal management, to control the coolant system actuators (electric pump and valves, fan). The paper shows how the thermal behavior of the engine can be represented by means of a concentrated parameters model, taking into account the main coolant system components features. The model has been calibrated on a set of data referring to a high-performance motorcycle engine, including both idling and high vehicle speed conditions. The good agreement of the model output with experimental data both in static and dynamic conditions confirms that the model is able to catch a large part of the phenomena influencing the coolant temperature.

Keywords: cooling system model; heat exchange; centrifugal pump; radiator; effectiveness-NTU

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1. Introduction

The coolant system plays a key role on the reliability of internal combustion engines, keeping the engine temperature under control, to avoid overheating. However, coolant system dramatically affects the overall vehicle

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aerodynamics, because the coolant radiator area increases the front section of the vehicle and this is a remarkable disadvantage, especially for motorcycles [1, 2].

Moreover, a mechanical water pump directly coupled to the engine is responsible of high power consumption at high engine speed [3]. Therefore, a meticulous study is devoted to the development of coolant systems, adopting technical solutions to tackle the minimization efficiency losses [4] and aerodynamics worsening [2], ensuring at the same time a proper engine cooling.

This work presents a concentrated parameters engine thermal model aiming at the description of thermal exchange mechanisms between the engine and the coolant system components: the scope of the model is to represent the thermal dynamics dependence on the components characteristics and engine running conditions. Such a model will reduce drastically the time effort spent on the cooling system design during the development phase, by assessing the impact of engine components characteristics on the coolant temperature control, allowing to test solutions well before the vehicle is ready for on-road tests. Cooling system simulation models are also useful for thermal management strategies that are becoming increasingly important [5-9].

The main coolant system components [10] have been analytically modelled: the model has been calibrated on a set of data, related to a high-performance motorcycle V4 engine running in different conditions (idle and high-speed), and then other simulations have been carried out and compared to experimental data not used for the calibration.

The paper is subdivided into three sections. The first one briefly shows the coolant system layout. The second section describes the thermal model, focusing on the analytical representation of the most significant elements. Finally, the simulation results are presented in the last section, where simulated and experimental temperature are compared for different tests. Moreover, the same section reports the results of a comparison between electrical water pump and mechanically-driven pump, highlighting differences in terms of efficiency.

Nomenclature

T_{Eo}, T_{Ei}	[°C]	temperature of the coolant at the engine outlet/inlet
T_{Ro}, T_{Ri}	[°C]	temperature of the coolant at the radiator outlet/inlet
Q_P, Q_R, Q_{BP}	[l/min]	coolant volumetric flow on the pump/radiator/by-pass branches
$\dot{m}_P, \dot{m}_R$	[kg/s]	coolant mass flow on the pump/radiator branches
$\dot{m}_{air\_tot}, \dot{m}_{air\ sp}, \dot{m}_{air\ f}$	[kg/s]	radiator air flow rate: total, vehicle motion, fans activation
sp_V	[km/h]	vehicle speed
T_{air}	[°C]	air temperature
ρ_{air}	[kg/m³]	air density
A_{rad}	[m²]	radiator front area
n_{ref}	[rpm]	reference engine speed
K_{tot}	[bar·(min/l)²]	total hydraulic resistance of the circuit
K_{V1}, K_{V2}	[bar·(min/l)²]	hydraulic resistances simulating the thermostatic valve
K_R	[bar·(min/l)²]	radiator hydraulic resistance
K_E	[bar·(min/l)²]	engine hydraulic resistance
$\Delta p_P, \Delta p_R, \Delta p_{BP}$	[bar]	pressure drop across the pump/radiator/by-pass branches
C_E, C_R	[J/K]	engine/radiator thermal capacity
C_W	[J/kg·K]	water coolant specific heat
$\dot{Q}_{Ei}$	[kW]	power supplied to the coolant by the combustion process
$\dot{Q}_{Eo}$	[kW]	power extracted from the engine by the coolant
$\dot{Q}_{Ri}$	[kW]	power supplied to the radiator by the coolant
$\dot{Q}_{Ro}$	[kW]	power extracted from the radiator by the air
$\dot{Q}_{Ro\_max}$	[kW]	maximum power theoretically extractable from the radiator by the air
$\dot{C}_{min}, \dot{C}_{max}$	[W/K]	minimum/maximum heat capacity rate between water and air
u	[W/m²K]	overall heat transfer coefficient
A	[m²]	heat transfer surface
ε	[]	radiator heat transfer efficiency
ε_{rad}	[]	experimental radiator aerodynamic efficiency

2. Coolant System Layout

The present section describes the structure of a motorcycle coolant system, highlighting those components whose analytical modelling is showed in the next paragraph.

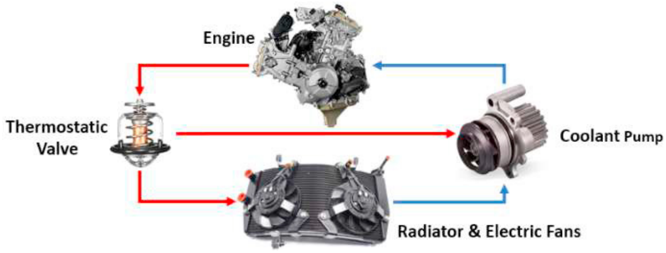

Fig. 1. Scheme of the coolant system layout

Usually a coolant system is a set of components where a coolant liquid (a mixture of free mineral water and Ethylene Glycol) flows. As shown in figure 1, it consists of:

- the thermostatic valve, managing the coolant flow at the outlet of the engine. The valve can allow or impede the coolant flow through the radiator, depending on the coolant temperature. In fact, during the engine warm up, the thermostatic valve is closed, forcing the coolant to by-pass the radiator, in order to quickly increase the water temperature;
- the radiator, where the heat exchange between air and coolant takes place. Usually the radiator is equipped with electric fans, useful to speed up the airflow, which is necessary to ensure a proper cooling of the engine when the vehicle is idling;
- the pump, pushing the coolant through the cooling circuit. Usually it is a centrifugal pump mechanically-driven by the engine, but lately the use of electric water pumps is spreading.

Table 1 shows the accuracy of sensors used to carry out tests and collect experimental data useful for the calibration and the evaluation of the proposed model.

Table 1. Sensors accuracy

Measure	Sensor type	Sensor accuracy
Air flow	Pitot	0.4 m/s
Water flow	Vortex	1.5 l/min
Water temperature	Pt100 (type A)	0.15 °C
Water pressure at pump inlet/outlet	Piezoresistive	0.006 bar

3. The Coolant System Model

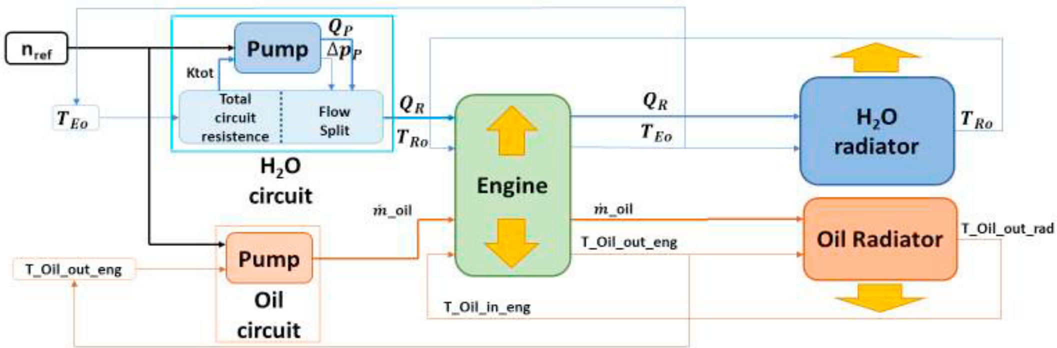

Fig. 2. Inputs and outputs of the thermodynamic model

The engine thermal behaviour is represented in Simulink by means of a zero-dimensional dynamic model, maintaining an accurate simulation capability even during transients, without bearing the complexity and the long computational time of multi-dimensional models. The engine thermal model is divided into sub-models corresponding to the main components: engine, water circuit, and oil circuit: indeed, part of the engine heat is transferred to the lubricant. However, this paper will focus on the coolant system model.

Figure 2 shows a schematic and simplified representation of the engine thermal model, highlighting the components I/O.

3.1 The coolant Pump

Starting from a set of experimental data reporting the head Δp supplied by the pump as a function of flow Q for different engine speed n, the coolant pump is modelled fitting the data with a 3rd grade polynomial expression. Each operating parameter is normalized (parameters with "red suffix" in the following equations) by means of a reference engine speed according to the similarity theory [11], thus collapsing the curves referring to different speed to a single pump characteristic, expressed by equation 2

$$n_{red} = \frac{n}{n_{ref}};\ Q_{red} = \frac{Q}{n_{red}};\ \Delta p_{red} = \frac{\Delta p}{{n_{red}}^2} \quad (1)$$

$$\Delta p_{red} = a \cdot {Q_{red}}^3 + b \cdot {Q_{red}}^2 + c \cdot Q_{red} + d \quad (2)$$

3.2 The coolant circuit

As shown by figure 3a, the water circuit has been modelled through the hydraulic-electrical analogy [12]:

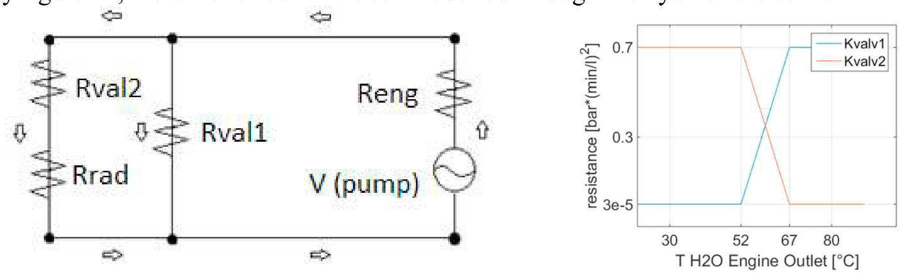

Fig. 3. (a) Electrical analogy of the coolant circuit; (b) characteristics of equivalent resistance modelling the thermostatic valve

The relationship $\Delta p = K \cdot Q^2$, is assumed to be valid for the description of the relationship between flow and differential pressure, for each component: engine, radiator, thermostatic valve. K is the hydraulic resistance, that is equivalent to the electric resistance R. Based on experimental data, the engine and radiator resistances as considered constant, while the thermostatic valve, is modelled by means of two inversely proportional resistances on the by-pass and radiator branches. Thermostatic valve resistances characteristics depend on the coolant temperature at the engine outlet, as described by figure 3b. On both branches the minimum value of resistance is set to ensure a leakage of 1% of the total water flow on the circuit. The pump is considered as an electrical generator. As shown by figure 5a, R_{val2} and R_{rad} are in series with each other and in parallel with R_{val1}. Then the equivalent resistance is in series with R_{eng}. So, thanks to analogy between hydraulic and electric resistance, the total resistance of the circuit can be simply calculated with equation 3:

$$K_{tot} = \frac{K_{V1} \cdot (K_{V2} + K_R)}{K_{V1} + K_{V2} + K_R} + K_E \quad (3)$$

The coolant pump operating point can be found matching the pump characteristic with the total resistance of the hydraulic circuit, which means for this point equation 4 holds. This allows finding the value of Q_{red}; then, thanks to the hydraulic similitude it is possible to find Q_P and Δp_P starting from the reduced quantities:

$$\Delta p_{red} - K_{tot} \cdot Q_{red}{}^2 = 0; Q_P = Q_{red} \cdot n_{red}; \Delta p_P = \Delta p_{red} \cdot n_{red}{}^2 \tag{4}$$

Once again, thanks to the electrical analogy, flows on the bypass and radiator branches can be evaluated, using equation 5:

$$\Delta p_R = (K_{V2} + K_R) \cdot Q_R{}^2 = \Delta p_P - K_E \cdot Q_P{}^2 = \Delta p_{BP} = K_{V1} \cdot Q_{BP}{}^2; Q_R = \sqrt{\frac{\Delta p_P - K_E \cdot Q_P{}^2}{(K_{V2} + K_R)}}; Q_{BP} = \sqrt{\frac{\Delta p_{BP}}{K_{V1}}} \tag{5}$$

3.4 Engine thermal model

The aim of the engine thermal model consists in determining the variation of coolant temperature between the engine inlet and outlet. The temperature variation can be evaluated by means of a thermal balance taking into consideration the heat transferred to the walls during the combustion, and the coolant heat absorption capacity. Thermal dynamics are implemented under two simplifying hypothesis: on one hand, the engine block temperature is considered uniform, on the other hand it is considered equivalent to the coolant outlet temperature, as with infinite exchange surface. As equation 6 shows, he system dynamics are dominated by the engine thermal capacity C_E.

$$\frac{dT_{Eo}}{dt} = \frac{1}{C_E}\left[\dot{Q}_{Ei} - \dot{Q}_{Eo}\right] \tag{6}$$

The power supplied to the coolant by the combustion process, $\dot{Q}_{Ei}$, is gathered form experimental data referring to steady speed and load conditions, and constant coolant temperature (T_{EO}=90°C). Experimental values are stored in a 2D look-up table.

The model does not take directly into account for convection with air, which depends on the engine block temperature: during engine warm-up, when the block temperature is in the range of ambient air temperature, this contribution is neglectable, thus a higher portion of the heat transferred to the engine walls will be transferred to the coolant. Then, the output of the lookup table storing $\dot{Q}_{Ei}$ data referring to T_{EO}=90°C will be compensated with a term considering the lower (at lower coolant temperature) or higher convection heat exchange on the air side.

Finally, the power extracted by the coolant can also be calculated with equation 7, based on its temperature variation across the engine:

$$\dot{Q}_{Eo} = \dot{m}_P \cdot C_W \cdot (T_{Eo} - T_{Ei}) = \dot{m}_R \cdot C_W \cdot (T_{Eo} - T_{Ro}) \tag{7}$$

The heat transfer to the lubricant is modelled with the same approach, with obvious differences concerning the hydraulic circuit (pressure limiting valve), the strong effect of temperature on oil viscosity (and thus on oil flow), and the mechanisms of oil heating which is both influenced by friction and combustion.

3.5 Radiator thermal model

As in the previous paragraph, the core of the model is the differential equation based on the balance of heat through the system (the water radiator in this case), and the same hypothesis are considered valid: the temperature of the radiator is assumed to be uniform and equal to the temperature of the fluid exiting the heat exchanger.

$$\frac{dT_{Ro}}{dt} = \frac{1}{C_R} \cdot \left[\dot{Q}_{Ri} - \dot{Q}_{Ro}\right] \tag{8}$$

The heat released by the cooling water to the radiator can be found using temperatures resulting from the previous iteration:

$$\dot{Q}_{Ri} = \dot{m}_R \cdot C_W \cdot (T_{Ro} - T_{Ri}) \tag{9}$$

The heat exchange in the radiator is calculated via *effectiveness - NTU* method [13], using only water and air input temperatures. The heat capacity rates ($\dot{C}$), defined as the product of mass flow by the fluid specific heat, are evaluated both for the cold and the hot fluids, and the ratio between the maximum and minimum capacity rate is then determined. The heat exchange efficiency can finally be found as a function of $\dot{C}_{min}/\dot{C}_{max}$ and $NTU = (u \cdot A)/\dot{C}_{min}$, using dedicated tables for motor-vehicle radiators [13]. The flow with higher heat capacity rate may be on the water side or on the air side, depending on the engine and vehicle operating conditions. For sake of simplicity, the evaluation of the overall heat transfer coefficient is carried out under the hypothesis of a linear relationship between $u \cdot A$ and the minimum heat capacity rate.

$$u \cdot A = k1 \cdot \dot{C}_{min} + k0 \tag{10}$$

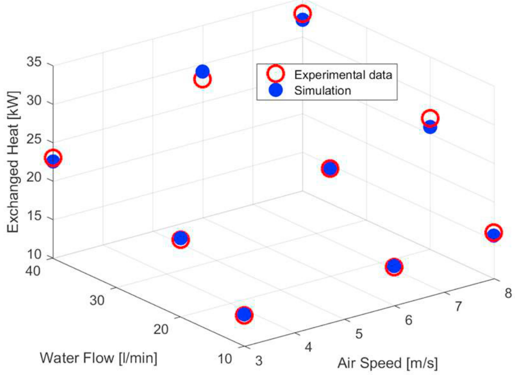

Fig.4 (a) Power extracted from radiator experimental and model

The coefficients $k0$ and $k1$ in the equation 10 have been determined based on experimental data regarding the radiator. Tests have been carried out changing water flow and air speed, while measuring water temperature at the inlet and outlet, allowing to estimate the heat exchanged. Since in these conditions $\dot{C}_{min}$ and $\dot{C}_{max}$ can be easily evaluated, *NTU*, and thus $u \cdot A$ can be determined, finally leading to the optimization of the coefficients in equation 10. Figure 4 shows the superimposition of experimental data and model output, using the optimized values for the coefficients in equation 10. The power extracted from the radiator by air can be finally expressed using equation 11, according to the effectiveness-NTU method:

$$\dot{Q}_{Ro} = \varepsilon \cdot \dot{Q}_{Ro\,max} = \varepsilon \cdot \dot{C}_{min}\,(T_{Ri} - T_{air}) \tag{11}$$

Because of the low value of air specific heat, the main limit to $\dot{C}_{min}$ is on the air side, especially at low vehicle speed; this factor is obviously also affected by fans activation. Regarding the air flow due to the motorcycle motion, it is calculated as shown by equation 12, where ε_{rad} is a reduction coefficient taking into account the air-speed drop from the motorcycle fairing to the radiator.

$$\dot{m}_{air\,sp} = sp_V \cdot \rho_{air} \cdot A_{rad} \cdot \varepsilon_{rad} \tag{12}$$

As fans are activated, a constant term is added to the air flow: the fans dynamics during speed transients are simulated by means of a first order transfer function, while the impact of the air flow due to vehicle motion on the flow generated by the fans is neglected.

$$\dot{m}_{air\,tot} = \dot{m}_{air\,sp} + \dot{m}_{air\,f} \tag{13}$$

The fans are managed by means of a simple hysteresis switch. Finally, the model is completed with variable time delay blocks representing the mass transport from the engine to the radiator and vice versa: the time delay is a function of the coolant flow.

4. Simulation Results

Firstly, the model parameters have been calibrated using the two sets of experimental data presented in figure 5, referring to idling and high vehicle speed conditions. The first set of data has been used to calibrate the engine and radiator thermal capacities, and the thermostatic valve resistances, while the second set allowed calibrating the radiator aerodynamic efficiency. The effect of the airflow produced by the fans is well caught by the model, as it can be inferred from the temperature oscillations at the end of the first test.

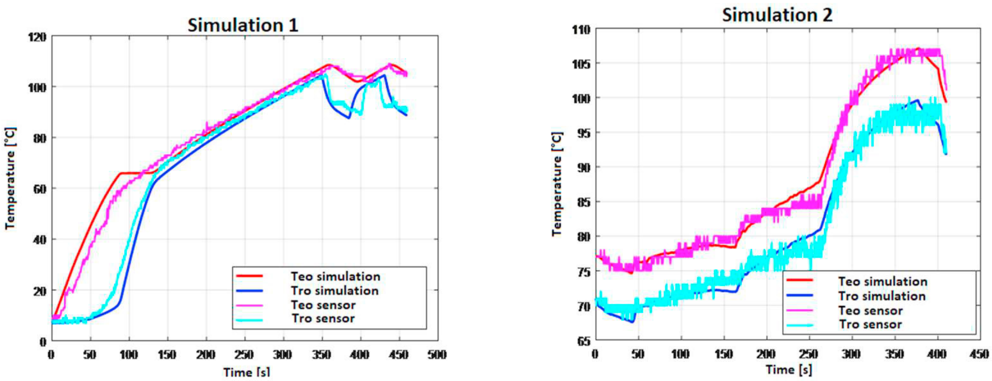

Fig.5 (a)Results of simulation 1; (b) Results of simulation 2

Secondly, other simulations have been executed, using the setup determined with the aforementioned calibration. Figure 6(b) shows the results obtained for the simulation of a lap on a track (the conditions in terms of normalized engine speed, throttle opening and vehicle speed are shown in figure 6(a)): the model capability of catching the coolant temperature average value and its variations can be easily appreciated.

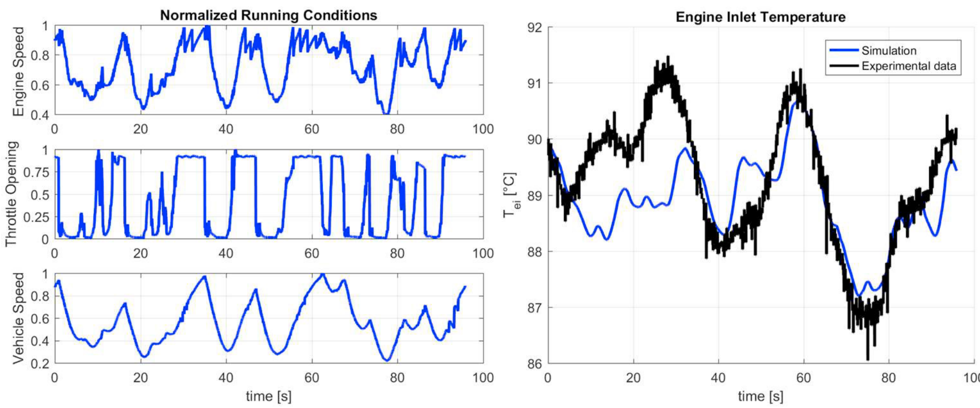

Fig.6 (a) Results of simulation 3; (b) Results of simulation 4

Figure 7 shows an example of how the model can be used to compare different cooling system designs: the results obtained with a mechanical pump in the scenario of Figure 6 are compared to those obtained with a feedback control of the pump speed, based on engine out coolant temperature. Two simulations are carried out with the speed controller: the first one with a target temperature equivalent to the average temperature obtained with the mechanical pump scenario (92°C); the second one with a higher target temperature (94°C). The benefits of a solution managing the coolant temperature based on the pump speed control clearly emerge, especially when the target temperature is increased: the lower coolant flow and head required to the pump imply a massive reduction in the power absorbed by the pump. A comparison in terms of power, however, should take into account the actual pump efficiency (both for the mechanical and electrical pump), the alternator efficiency and the battery charge/discharge efficiencies. Furthermore, the additional weight and space required by the electrical pump are not always compatible with the considered application, especially using 12V electrical systems.

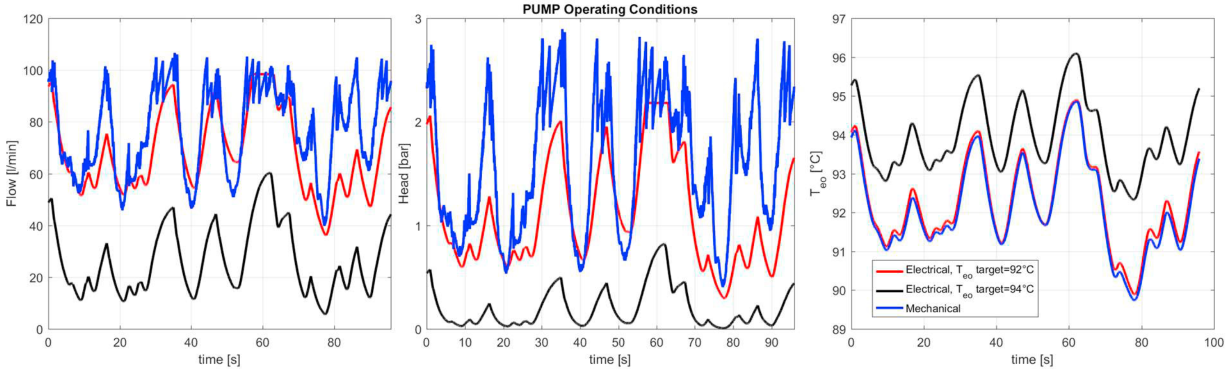

Fig.7 (a) Simulated Flow; (b) simulated Head; (c) simulated T_{EO} with electrical pump (target T_{EO}=92°C and 94°C) and mechanical pump

5. Conclusions

The paper concerns the implementation of a zero-dimensional dynamic model of the engine cooling system. The model aim is to assess the effects of thermal management strategies and cooling system components characteristics on the engine thermal state (namely, coolant temperature).

The paper describes the model structure and the simplifying hypotheses allowing an accurate yet low-computational power representation of the plant. The model calibration parameters are also introduced: a set of experimental data is used to set the model parameters, and a simulation carried out in different dynamic conditions is compared to a different set of data, with satisfying results, both in terms of coolant temperature average value and variations over the duration of a track lap.

Finally, the model is used to compare two different cooling system layouts (mechanical vs. electrical pump), highlighting the potential benefits of the second solution.

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