

A Novel Cooling System Control Strategy for Internal Combustion Engines

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ABSTRACT

An innovative control strategy, which is based on the Robust Model Predictive Control (MPC) methodology, was developed with the purpose of optimizing the engine thermal management; the proposed control strategy adjusts the coolant flow rate by means of an electric pump, in order to bring the cooling system to operate around the onset of nucleate boiling. In the present paper, the advantages of the proposed cooling approach are evaluated along the NEDC homologation cycle, which was both simulated and replicated by means of laboratory tests; the latter include coolant, lubricant and wall temperature measurements. Special attention was reserved to the warm-up period. The case considered herein is that of a Spark Ignition engine, about 1.2 dm³ displacement, and a comparison with standard crankshaft driven pump is included. The proposed strategy makes use of a dynamic model of the cooling system of an ICE that is able to predict the heat transfer both under single-phase forced convection and in the presence of nucleate or saturated boiling. The model, which was widely validated by experimental tests, also defines a metrics for establishing the heat transfer mechanism inside the engine and for estimating the distance of the engine thermal state from the onset of nucleate boiling. Results show that the developed MPC algorithm is robust in terms of disturbance rejections, respects the defined system constraints and it is effective in decreasing the warm-up time and in reducing the coolant flow rate under fully warmed conditions as compared to the standard mechanical pump.

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INTRODUCTION

An optimal thermal management for Internal Combustion Engines constitutes one of the most promising and low-cost solutions for the reduction of fuel consumption and for increasing engine efficiency [1]. The requirements of regulatory agencies are, in fact, more and more severe [2,3] and innovative technological solutions, aimed to reduce vehicle pollutant and CO₂ emissions, were proposed. A comprehensive review of influences of thermal management on internal combustion engines efficiency can be found in Roberts et al. [4]. The proposed solutions aim to reduce the warm-up time at engine cold-start: this phase constitutes, in fact, the major source of pollutant emissions and of engine inefficiency owing to the low cylinder temperatures, which increase frictional losses and cause combustion inefficiencies. For lubricant temperatures around 20°C, for instance, Will et al. [5,6] estimated an increase in frictional losses amounting to 2.5 times the ones obtained under fully warmed conditions.

By a proper management of the coolant flow rate, an advantageous control of the mid-stroke cylinder liner temperature can be achieved. The traditional engine cooling systems offer a poor regulation capability: the coolant flow rate is fixed by the engine speed, as the standard mechanical pump is connected to the engine crankshaft. The coolant flow rate is, consequently, regulated by the use a wax thermostat valve. Under these conditions, Brace [7] estimated that engines are over-cooled for about 95% of their operating time.

In order to overcome the above limitations, innovative systems, which make use of electrically driven pumps in conjunction with different control strategies, have been investigated and proposed. Ap and Golm [8] proposed a reduction of the coolant flow rates by the use of a small electric water pump. Brace in [7] proposed a control strategy, which was based on the use of empirical look-up tables and in [9] found a 2% improvement in bsfc by reducing the coolant flow rate in a plant, which included an electrically driven pump in cooperation with a throttle valve. A pulsed coolant flow rate strategy was proposed by Bent [10]: an electric pump was switched ON and OFF in order to circulate the coolant only when it was required. The adopted solutions, however, are all based on empirical methods.

The paper presents a more systematic approach to engine cooling and proposes a novel Robust Model Predictive Control. This is based on the use of an electric pump and on the possibility of predicting the heat transfer regime, which takes place in the cooling system for given values of engine speed, fuel flow rate, coolant pressure, temperature and flow rate. To this purpose, a lumped parameter model of the cooling system was developed [11]. This is able to detect dynamically the occurrence and the extent of the nucleate boiling phenomenon and calculates the spatial-averaged metal temperature and the average coolant temperature, by the use of simple input data. Most of them (fuel flow rate, engine speed and engine-out coolant temperature) are already available in the ECU while coolant pressure can be obtained from low cost transducers. Calculating correctly the heat transfer, also in the presence of

nucleate or saturated boiling is a fundamental need for a reliable evaluation of coolant and engine walls temperatures and for an efficient and safe engine operation. By taking advantage of the in-line results of the predictive model, a control algorithm adjusts the coolant flow rate. During the engine warm-up, the goal is to gain a fast rise of engine wall temperatures; under fully warmed conditions, the coolant flow rate is regulated in order to maintain nucleate boiling conditions. This ensures reliable cooling with lower coolant flow rates than usually adopted, with reduced pump power consumption.

Both the engine model and the control algorithm were developed by using the Matlab-Simulink® platform, and were widely validated through test-rig experiments [12]. The model, at the moment, does not include engine oil temperature prediction, but measurements of lubricant temperature are available from the experiments; the effect of the control strategy on lubricant temperature is, therefore, evaluated.

The proposed control design procedure has a wide validity and can be easily applied to a variety of different objectives for the cooling system according to manufacturer's requirements. In the present work, the New European Driving Cycle (NEDC) was numerically simulated and replicated in laboratory experiments as test case in order to evaluate the advantages of the novel control strategy with respect to the traditional cooling system.

CONTROL STRATEGY

The operating scheme of the model-based control system on board of a vehicle is summarized in Figure 1. While the engine is running, the model receives the input variables (*engine speed, fuel flow rate, coolant temperature and pressure*) in real time and computes the engine wall average temperature, the average coolant temperature, the thermal power transferred to the coolant and the amount of engine area eventually involved in nucleate or saturated boiling. The computed values of wall and coolant temperatures, T_w and T_c , are then used to calculate the coolant flow rate correction according to the following equation:

$$\Delta \dot{m}_c = K_1 (T_w - T_{w_eq}) + K_2 (T_c - T_{c_eq}) \quad (1)$$

$\Delta \dot{m}_c$ is the correction to be given to the equilibrium (desired) coolant flow rate in order to satisfy the problem constraints on T_{w_eq} and T_{c_eq} , which represent the desired equilibrium engine wall and coolant temperatures. K_1 and K_2 , are the controller values, estimated following the approach presented in Kothare et al. [13].

The proposed strategy, which is currently in a prototype stage, was tested both by means of numerical simulations and by means of experiments at the test rig. In the first case, the engine model reads the input variables (*engine speed, fuel flow rate and coolant pressure*); the coolant engine-in (or engine-out) temperature can be available from experimental data or computed by means of a radiator model; the coolant flow rate is then computed by the controller in order to satisfy the control specifications. In the case of laboratory tests, during engine real operation, a data acquisition system transfers the engine variables (*engine speed, fuel flow rate, coolant pressure*

and coolant engine-in or engine-out temperature) to the model. The model computes the averaged engine wall and average coolant temperatures and passes them to the controller; the controller computes the coolant flow rate correction (Eq.1) and enforces the proper voltage to the coolant electric pump in order to satisfy the thermal management strategy specifications.

Extensive description of the models can be found in Bova et al. [11] and in Pizzonia et al. [14], which include experimental validation. A brief summary is presented in the following sections.

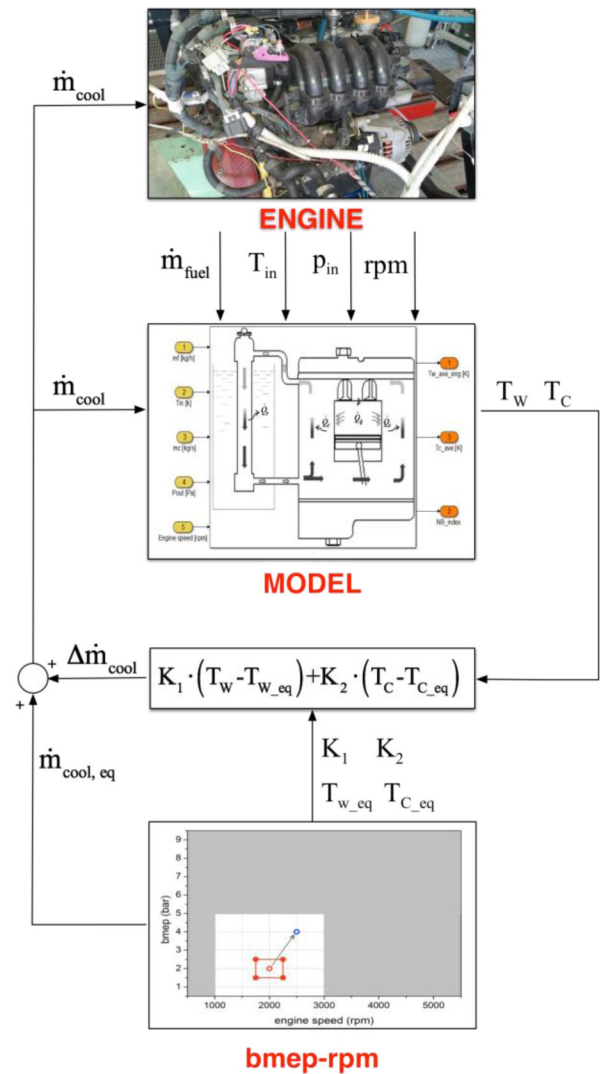


Figure 1. Engine Control Strategy [14]

Engine Model

The energy conservation equations are used to determine the spatial average engine wall temperature, T_w , and the coolant average temperature, T_c :

$$\dot{T}_w C_w = \dot{Q}_g - \dot{Q}_c \quad (2)$$

$$\dot{T}_c C_c = \dot{Q}_c - \dot{Q}_r \quad (3)$$

In the above equations, C_w and C_c are the engine and coolant thermal capacities, respectively. $\dot{Q}_g$ is the thermal power from the combustion gases to the engine walls and is calculated from an empirical formulation derived in [15] and modified in [11] in order to correlate the fuel thermal power not only to fuel flow rate but also to engine speed and coolant flow rate:

$$\dot{Q}_g = cN^{n_1} \dot{m}_c^{n_2} \dot{m}_f^{n_3} \quad (4)$$

$\dot{Q}_c$ is the thermal power from the engine walls to the coolant and includes the forced convection and nucleate boiling contributions:

$$\dot{Q}_c = h_{mac} A (T_w - T_\infty) + h_{mic} A_{nb} (T_w - T_{sat}) \quad (5)$$

In Eq. 5, A is the total heat exchange area and A_{nb} is the part of the engine walls involved in the nucleate boiling phenomenon; h_{mac} is the forced-convection heat transfer coefficient and is computed through the *Dittus-Boelter* correlation [16]; finally, the *Chen* approach [17] is used to determine the nucleate boiling heat transfer coefficient h_{mic} . The onset of nucleate boiling is strongly determined by local wall temperatures and by the presence of hot spots. As the model is zero-dimensional, this phenomenon was taken into account by properly calibrating the model parameters, which enter the calculation of h_{mac} .

$\dot{Q}_r$ is the thermal power released by the coolant to the atmosphere through the radiator; it may be obtained from a radiator model, whose equations will be described in the next section, or by measurements of engine-in and engine-out temperature.

One metrics used for establishing the heat transfer mechanism inside the engine is the *NB_Index*, defined as:

$$NB_Index = \frac{q_w - q_{ONB}}{q_{ONB}} \quad (6)$$

The model, in fact, computes both the heat flux needed for the onset of nucleate boiling, q_{ONB} , and the actual heat flux through the engine walls, q_w . Positive values of the *NB_Index* indicate that nucleate boiling occurs, owing to the fact that the actual thermal flux is higher than the needed one; on the contrary, negative values of the *NB_Index* indicate that the actual thermal flux is lower than the needed one and the cooling action is due to forced convection only.

Radiator Model

The equations of the thermal exchange at the radiator define a relationship between engine-in and engine-out temperatures; they, therefore, in conjunction with eq. 2, 3 and 4, allow the computation of the coolant temperature at engine inlet and of the thermal power $\dot{Q}_r$ released by the coolant to the radiator wall. They include the thermal exchange between the coolant and the radiator walls (coolant side) and between the radiator walls and the external air flow (air-side) [18].

The first set of equations is given by:

$$\dot{Q}_r = \dot{m}_c \cdot c_c \cdot (T_{c,in_r} - T_{c,out_r}) \quad (7)$$

$$T_{c,in_r} - T_{c,out_r} = \frac{h_c (T_c - T_{w_r})}{w_c d_e \rho_c c_c} \cdot H \quad (8)$$

In the equation above, T_{c,in_r} and T_{c,out_r} are the coolant temperature at the radiator inlet and exit respectively; h_c is the coolant-wall convection coefficient; T_c is the average coolant temperature in the radiator and T_{w_r} is the average temperature of the radiator wall. w_c is the coolant speed inside the radiator ducts, d_e is the radiator pipes diameter, ρ_c and c_c are coolant density and specific heat, respectively, and H is the whole duct length. Eq. 8 is obtained by the integration of the elemental thermal power from the coolant to the radiator over the whole radiator duct length.

The thermal exchange from radiator air-side (wall-air) is described by the following equations:

$$\dot{Q}_{air} = \dot{m}_{air} \cdot c_{air} \cdot (T_{air,in} - T_{air,out}) \quad (9)$$

$$T_{air,out} = T_{w_r} - (T_{w_r} - T_{air,in}) \cdot e^{-\left(\frac{h_{air} S}{w_{air} d_e \rho_{air} c_{p,air}}\right)} \quad (10)$$

Eq. 10, similarly as Eq. 8, is obtained by the integration of the elemental thermal power from the radiator walls to the air over the radiator fins depth. $T_{air,in}$ and $T_{air,out}$ are the air temperature at radiator inlet and exit, respectively. S is the radiator surface area invested by the air stream.

Finally, the following relation completes the set of equations, which allow the solution of the thermal exchange at the radiator:

$$\dot{Q}_r - \dot{Q}_{air} = \dot{m}_c \cdot c_c \cdot \dot{T}_{w_rad} \quad (11)$$

Controller

The reduction of engine warm-up time is the main goal of the control strategy proposed in the present work; a second goal is to reduce the pump power consumption at fully warmed engine. This can be achieved by properly varying the coolant flow rate.

The controller parameters, K_1 and K_2 , in Eq. (1) are used to compute the correction $\Delta \dot{m}_c$, to be given to the equilibrium coolant flow rate in order to satisfy the problem constraints on T_{w_eq} and T_{c_eq} (Fig. 1). The desired equilibrium engine wall and coolant temperatures are selected in order to provide forced convection or nucleate boiling. T_w and T_c are the actual engine wall and coolant temperatures. The values of the controllers K_1 and K_2 are determined following the empirical procedure presented in Kothare et al. [13]. The mathematical model of the