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A Model Predictive Controller for the Cooling System of Internal Combustion Engines

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Abstract

The paper presents some results of the Model Predictive Controller (MPC) methodology applied to the case of the cooling system of an Internal Combustion Engine. To this end, a small spark ignition engine, about 1.2 dm^3 displacement volume, is equipped with an electric pump, which is actuated by the controller, independently of engine speed. The goal of the proposed control is to achieve a faster engine warm-up and an effective engine cooling with a much lower coolant flow rate than the one usually adopted, by bringing the cooling system to operate around the onset of nucleate boiling. The developed Model Predictive Control application makes use of a lumped-parameter model, which predicts the heat transfer both in the case of a single-phase forced convection condition and in the presence of nucleate boiling. The performance of the proposed controller is evaluated during the city driving part of the NEDC homologation cycle, which was replicated at the engine test rig. The results show that the proposed controller is robust in terms of disturbance rejection and is effective in reducing warm-up time.

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Keywords: Internal Combustion Engine, Model Predictive Control, Engine warm-up, Cooling System, Nucleate Boiling.

1. Introduction

The reduction of fuel consumption and the increase of efficiency of Internal Combustion Engines is nowadays one of the most important goals for car manufacturers', owing to the severe requirements of the regulatory agencies on CO2 emissions [1, 2]. The engine thermal management is one of the most promising and low-cost solutions for achieving this goal [3]; the cooling system is therefore widely investigated both by modelling and by experimental

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tests [4-8]. Today, production engines are characterized by poor regulation capability as coolant flow rate is fixed by the engine speed, so that they are over-cooled for about 95% of their operating time [9].

Several strategies were investigated and proposed over the years to regulate coolant flow rates by using electric pumps [9-12]. However, the proposed approaches and control strategies were all based on empirical methods.

This paper presents a more systematic approach to the engine cooling, based on electric pump and low coolant flow rates, which may be working in the presence of nucleate boiling. It is known, in fact, that reducing the coolant flow rate in some operating conditions has the advantages of a faster engine warm-up, lower fuel consumption and CO_2 emission during warm-up and lower pump power under fully warmed conditions. The real question is how much and when the coolant flow rate can be reduced safely. It is commonly accepted that some nucleate boiling is allowed, but severe boiling must be avoided. So, the main goal of this work is the presentation of a methodology for controlling the coolant flow rate.

The proposed methodology is based on Robust Model Predictive Control and on the possibility of predicting the heat transfer regime in the cooling system for given values of engine speed and fuel flow rate. To this aim, a lumped parameter model of the cooling system was developed [13]. The model predicts the occurrence and the extent of the nucleate boiling and calculates the spatial-averaged metal temperature and the average coolant temperature, by the use of simple input data. By taking advantage of the in-line results of the predictive model, a control algorithm, widely described in [14, 15], adjusts the coolant flow rate in order to fulfill the control strategy requirements. Both the engine model and the control algorithm were developed by using the Matlab-Simulink® platform, and were widely validated through test-rig experiments [16].

It is worth pointing out that the proposed methodology has a wide validity and allows the definition of different control strategies for the cooling system according to manufacturers' requirements. In this case, the control strategy aims to gain a fast rise of engine wall temperatures at cold start and, in order to evaluate the advantages of the novel control strategy with respect to the traditional cooling system, the low part of the New European Driving Cycle (NEDC) is imitated in laboratory experiments. The experimental set-up includes an electric pump instead of the standard crank-shaft driven one. Therefore, in order to assess the advantages of the proposed control strategy with respect to the mechanical pump, this last one was 'imitated' by the electric pump by enforcing the coolant flow rates recorded at the roll test bed under the same operating conditions.

2. Cooling system model

For determining the heat transfer mechanism between the engine walls and the coolant, the model makes use of Chen's approach [17], which includes the forced convection and nucleate boiling contributions according to the following equation:

$$\dot{Q}_c = h_{mac} A (T_w - T_c) + h_{mic} A_{nb} (T_w - T_{sat}) \tag{1}$$

In Eq.1, h_{mac} is the forced-convection heat transfer coefficient and is computed through the *Dittus-Boelter* correlation [18], h_{mic} is the nucleate boiling heat transfer coefficient [17], A is the total heat exchange area and A_{nb} is the part of the engine walls involved in the nucleate boiling phenomenon. T_w, T_c and T_{sat} are wall average, coolant average and saturation temperature, respectively. Nucleate boiling occurs only if the total heat flux q_w is higher than the needed one q_{ONB}:

$$q_w \geq q_{ONB} = h_{mac}((\Delta T_{sat})_{ONB} + \Delta T_{sub}) \tag{2}$$

Therefore, a metrics has been defined [14], NB_Index, which measures the distance of the system thermal state from the Onset of Nucleate Boiling:

$$\mathrm{NB}_{\mathrm{Index}} = \frac{q_w - q_{ONB}}{q_{ONB}} \tag{3}$$

According to this formulation, the following model was developed [13]:

$$\dot{T}_w C_w = \dot{Q}_g - \dot{Q}_c \tag{4.a}$$

$$\dot{T}_c C_c = \dot{Q}_c - \dot{Q}_r \tag{4.b}$$

It is a lumped parameter model with two state variables, which are the spatial average engine wall temperature, T_w, and the average coolant temperature T_c. In the above equations, C_w and C_c are the engine and coolant thermal capacities, respectively. The thermal power transferred from the wall to the coolant, $\dot{Q}_c$, is estimated according to Chen's equation (1). In order to compute the thermal power from the combustion gases to the engine walls, $\dot{Q}_g$, an empirical literature formulation [19] is used; however, this formulation was modified in [13] in order to correlate the fuel thermal power not only to fuel flow rate but also to engine speed and coolant flow rate, according to the following:

$$\dot{Q}_g = cN^{n_1}\dot{m}_c^{n_2}\dot{m}_f^{n} \tag{5}$$

where c, n, n_1, and n_2 are tunable coefficients. Finally, $\dot{Q}_r$ is the thermal power gained by the coolant from engine inlet to engine outlet:

$$\dot{Q}_r = \dot{m}_c c_p \left(T_{out} - T_{in}\right) \tag{6}$$

Typically, one of these two temperatures (T_{out}) is known from measurements. Therefore, by integrating the two equations (4), the wall and the coolant temperatures can be obtained. The model needs inputs, which are usually available on board a vehicle: fuel flow rate, coolant inlet or outlet temperature and engine speed. Coolant pressure must be measured and coolant flow rate is given by the controller. Model outputs are T_w, T_c and NB_Index.

The model was validated both under steady-state and transient conditions; more details can be found in [13,16].

3. Control Method and Control Strategy

The proposed model is used in a scheme developed to control the state of the cooling system. The control scheme is based on the robust Model Predictive Control (MPC) method. As an engine is a multi-input-multi-output system, the classical PID controllers cannot be used. For slow dynamical systems, the MPC is instead widely adopted.

The model runs in parallel with the engine (Fig.1, left), receives from it the needed inputs (engine speed, fuel flow rate, coolant temperature and pressure) and predicts the state of the system (T_w and T_c). In an equilibrium condition, with given inputs ($\dot{m}_{f\_eq}$, N_{eq}, P_{in_eq}, T_{in_eq}), one can obtain the desired (equilibrium) values of wall and coolant temperature (T_{w_eq}, T_{c_eq}) with a certain value of coolant flow rate ($\dot{m}_{c\_eq}$). Then, in a general situation, with other inputs, if T_c and T_w are the actual coolant and wall temperatures, as predicted by the model, these temperatures can be brought to their desired (equilibrium) values by adding to the equilibrium coolant flow rate a correction, $\Delta\dot{m}_c$:

$$\Delta\dot{m}_c = K_1\left(T_w - T_{w\_eq}\right) + K_2\left(T_c - T_{c\_eq}\right) \tag{7}$$

The constants K_1 and K_2 are calculated by solving a min-max (minimization of the worst case) optimization problem with constraints and are estimated following the approach presented in Kothare et al. [20].

The constraints are the minimum and maximum flow rate value that the pump can provide, the minimum and maximum allowed values for the state and output variables and the minimum and maximum expected values of the inputs, which can be measured, but are not known in advance, so from the automatic control point of view they are disturbances or perturbations. The optimization problem with the given constraints, of course, may not have a solution; in fact, a solution does not exist if the range of the expected input variables is too wide or if the interval of the allowed values for the output or state variables is too tight. In this case, it is necessary to reduce the range of the input variables. To this aim, the engine speed–torque plane was split into sub-regions; for each of them, the optimization problem was solved and a set of controllers K_1 and K_2 was computed.

It is worth to point out that this process, which can be very time-consuming, is carried-out *off-line*: the optimization code works with the model only. Then, during the usual engine operation, the control parameters are selected *on-line* from a look-up table. This procedure guarantees the stability of the controller and a smooth

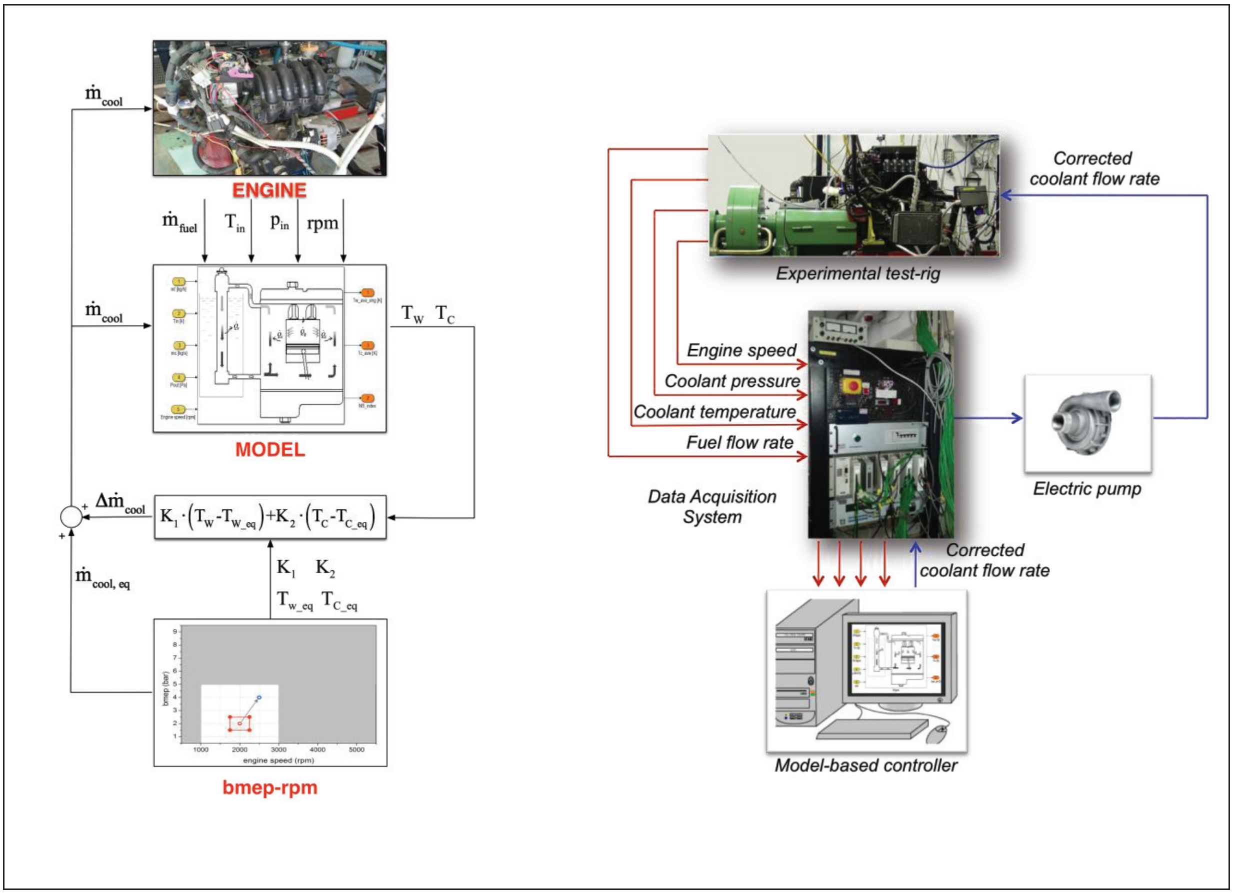

Fig.1. *Left:* Engine Control Strategy; *Right:* Schematic of the integration between the engine test rig and the controller [14].

transition between adjacent sub-regions. By using this procedure, several control strategies can be adopted. In this case, the goal of fast warm-up when coolant temperature is below 80°C, was pursued. Extensive description of the control method can be found in [14,15], which include experimental validation.

The interaction between the test rig and the controller is summarized in Fig. 1 (right): while the engine is running, a data acquisition system records data and passes them to a computer where the model and the controller are running. The calculated coolant flow rate is provided by the electric pump. Data are updated about each second.

4. NEDC laboratory tests

The impact of the proposed control strategy was evaluated in the city driving part (warm-up) of a typical homologation cycle, the New European Driving Cycle and the engine behavior was replicated at a stationary test rig. The engine is a small S.I. engine, 1.2 liters, 4-cylinders, 4-valve-per-cylinder; the test rig is equipped with a Borghi&Saveri FE 260-S eddy current engine torque dynamometer provided with an actuator for remote control of throttle position and with an AVL 733S metering system for engine fuel consumption measurements. The standard mechanical pump was substituted by a small-power electric unit; the thermostatic valve was removed as it was not compatible with the use of an external electric pump; a fan was installed in front of the radiator, in addition to the production fan. The engine is instrumented in order to record coolant pressure and coolant temperatures at engine inlet and outlet, lubricant temperature, coolant flow rate at engine inlet and wall temperature in a number of points in the engine head and in the cylinder block by K-type thermocouples. Engine speed, torque, fuel flow rate and engine-out coolant temperature are known from industry rolling test bed. As the dynamometer is suitable for stationary

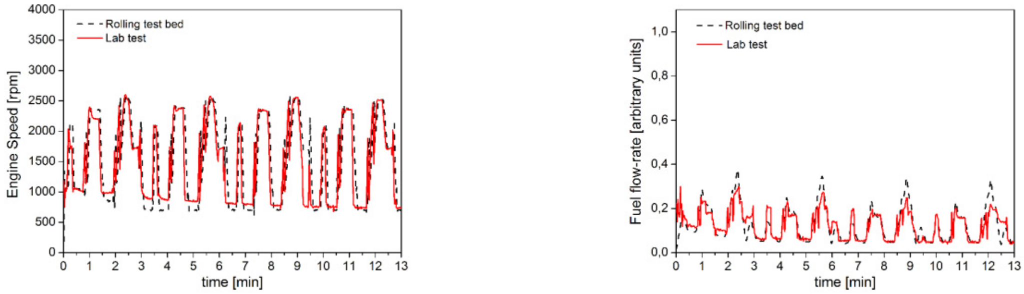

Fig. 2. Engine speed and fuel flow rate during the city part of the NEDC, replicated at the stationary test rig

operation, in order to replicate the NEDC at the test rig, the engine speed-torque sequence of data taken at the rolling test bed was converted by calibration into an engine-speed-throttle sequence of data.

Fig.2 shows that both the engine speed and fuel flow rate are replicated quite well, and only negligible differences can be observed. Despite this is not a real homologation cycle, it is a quite useful tool to get repeatable conditions, similar to the ones of the homologation cycle, in order to test different cooling strategies. In particular, this set-up is used herein to test the behavior of both the standard mechanical pump, imitated by the electric pump, and of controlled coolant flow rate.

The production mechanical pump was "imitated" by the electric pump; the coolant flow rate of the mechanical pump, which is proportional to the engine speed, N, Eq.8, was calculated from data at the rolling test bed given by the industry. This flow rate was then enforced to the electric pump.

$$\dot{m}_c = K \cdot N \tag{8}$$

During the laboratory test for controlled coolant flow rate, the model receives the values of engine speed, fuel flow rate, coolant temperature and pressure and the controller calculates the coolant flow rate (Eq. 7). This value is then enforced to the electric pump

5. Results and discussion

Figure 3 (left) shows the coolant flow rate supplied by the production mechanical pump and by the electric pump when it replicates the mechanical pump during the experimental test previously described. The two curves overlap quite well and demonstrate therefore the reliability of the experimental procedure. Figure 3 also shows the flow rate given by the controller: it is significantly lower, about one-quarter of the standard average flow rate. In fact, for a production pump, at cold engine conditions, its value oscillates approximately between a minimum of 500 dm^3/h at idle and a maximum of 1700 dm^3/h, in the low 7' of the cycle. When the wax thermostatic valve opens, the coolant flow rate increases, ranging from about 630 dm^3/h at idle to about 2000 dm^3/h. In the case of the controller, the measured coolant flow rate was about 200-300 dm^3/h in the low 10'. This lower coolant flow rate determines a faster warm-up of engine walls and engine lubricant.

Figure 3 (right) shows the comparison between oil temperature with mechanical pump and with the controller enabled: in the case of the controlled coolant flow rate, red line, the lubricant reaches the reference value of 80°C level about 1 minute earlier than in the case of the mechanical pump. With the lower coolant flow rates, in fact, a lower heat is transferred to the coolant and more energy is stored into the engine walls; consequently, the heat transfer to the lubricant is higher. The coolant temperature shows negligible differences in the two cooling strategies. However, it is significantly lower than the coolant temperature measured at the rolling test bed, where the thermostatic valve was used.

Figure 4 reports the engine wall temperature values recorded at the intake valve bridge of the coldest cylinder and at the exhaust valve bridge of the hottest cylinder. The wall temperature increases more rapidly in the case of the reduced flow rate. By considering a reference level of 120°C, the advantage obtained by adopting the proposed control strategy is about 3 minutes.

On the basis of the recorded coolant temperature and flow rate, the thermal power transferred by the engine walls to the coolant was computed. To this aim, Eq.4 (b) was used. $\dot{Q}_r$ and $\dot{T}_c$ were computed from the recorded values of T_{out} and T_{in}, and $\dot{Q}_c$ was consequently calculated on the basis of the experimental data. This "experimental" value was then compared with the one obtained from the model by Eq. (1): Fig.5 (left) shows that the agreement is satisfactory. The same approach was used for the mechanical pump. Fig.5 (right) shows that the agreement is also good.

The thermal power for the two flow rates is compared in Fig.6: in the case of the controlled flow rate, the power transferred to the coolant is significantly lower. In fact, the average value is 2.5 kW in the case of the mechanical pump and 1.7 kW in the case of the controlled flow rate: 1/3 less thermal power as a result of 75% less coolant flow rate. The power transmitted from the combustion gases to the wall is about the same, both in the case of the mechanical pump and in the case of the controlled coolant flow rates; therefore, the lower thermal power to the coolant determines a higher thermal power retained in the metal and, consequently, a higher heat transfer to the lubricant.

The lubricant and wall temperature are consistent with these results, in fact, higher values are registered in the case of the proposed control. Finally, the metrics for the distance from the onset of nucleate boiling is evaluated. Fig. 6 (right) shows that in the case of the controlled flow rate the system is closer to nucleate boiling but it still stays in the single-phase regime. By using the thermostatic valve, a higher value of the NB_Index and an even faster warm-up could be obtained.

A final consideration regards the fuel consumption. As the tests were carried out by enforcing (engine speed – throttle) series of values, the fuel consumption is the same in the two different cooling strategies. However, warm-up tests reported in [14] at constant engine speed (2000 rpm) and constant brake torque (20 Nm) for a time of 700 s proved a fuel consumption of about 1-3 % lower in the case of the controlled flow rate.

Summary and Conclusions

This work presented the results of an innovative control strategy developed for the thermal management of internal combustion engines.

The aim of the proposed strategy is to vary the coolant flow rate in order to obtain a faster engine warm-up and to exert the cooling action with the lowest possible coolant flow rates, while preserving the engine reliability.

This was realized by substituting the standard engine-driven mechanical pump with an electrical pump properly managed by the proposed control algorithm, which is based on a Robust Model Predictive Control (MPC) approach and requires the use of a model of the cooling system.

The model of the cooling system, in particular, predicts quantities which are not directly measurable on board a vehicle, and is useful to define a metrics for estimating the distance of the engine thermal state from the onset of nucleate boiling. The MPC algorithm is characterized by the following aspects: the heavier computations are executed off-line; the controllers are calculated for small sub-regions of engine speed-fuel flow rate plane and the local control parameters are then picked-up from a map on-line; by following this approach the stability is guaranteed.

The proposed control methodology was evaluated by replicating the city driving part of the NEDC homologation cycle at the test rig. The results demonstrated that the control algorithm is robust in terms of disturbance rejections and that the wall temperature, coolant temperature and NB_Index were maintained within the defined limits. Furthermore, the benefits of the proposed control strategy consist in a 75% reduction of coolant flow rate in the early part of the homologation cycle, in a faster warm-up of lubricant temperature, which reaches 80 °C about 1 min earlier, and finally in a faster warm up of metal temperature, which reaches 120 °C about 3 min earlier, with respect to the standard mechanical pump.

The results are satisfactory, even though improvements can be obtained by modifying the constraints of the controller and/or by a better calibration of the model constants. The methodology can be adopted with different cooling system layouts, and it is expected that the warm-up advantages are more evident if the thermostatic valve is kept in the cooling system layout.

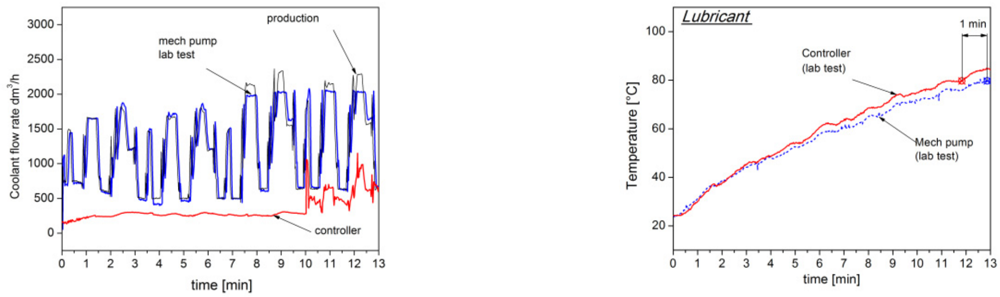

Fig. 3. Coolant flow-rate: Production mechanical pump, replicated mechanical pump in lab tests and controlled coolant flow rate. Lubricant temperature for the mechanical pump and in the case of the controller enabled.

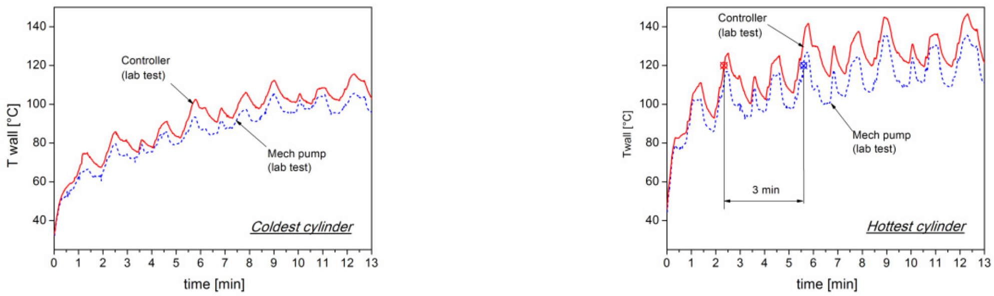

Fig.4 Wall temperature (valve bridge): intake valve of the coldest cylinder and exhaust valve of the hottest cylinder.

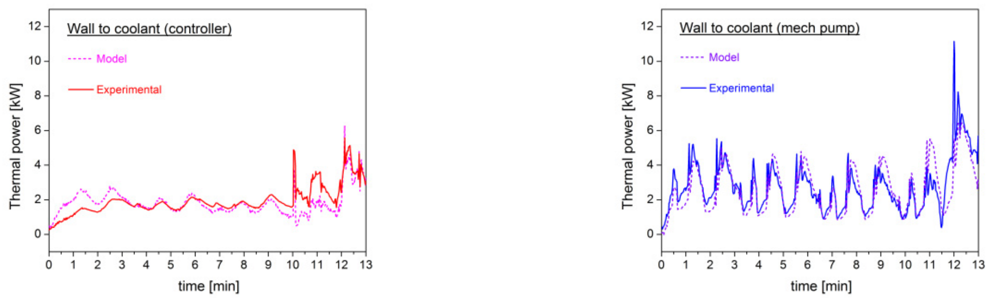

Fig.5. Thermal power transferred from engine walls to the coolant, in the case of the controlled coolant flow rate.

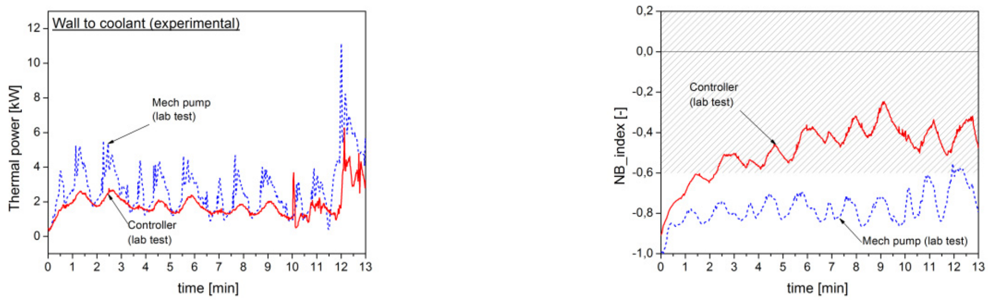

Fig.6 Thermal power transferred from engine walls to the coolant, comparison between controlled coolant flow rate and mechanical pump; time history of the *NB_Index* (eq. 3).

The methodology can also be employed to obtain different goals, as, for example, a larger thermal power transferred to the cabin heater or for controlling the cooling of other devices, like electric motors or power electronic units.

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