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Modeling and fuzzy control of the engine coolant conditioning system in an IC engine test bed †

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Abstract

Mechanical and thermodynamical performance of internal combustion engines is significantly affected by the engine working temperature. In an engine test bed, the internal combustion engines are tested in different operating conditions using a dynamometer. It is required that the engine temperature be controlled precisely, particularly in transient states. This precise control can be achieved by an engine coolant conditioning system mainly consisting of a heat exchanger, a control valve, and a controller. In this study, constitutive equations of the system are derived first. These differential equations show the second- order nonlinear time-varying dynamics of the system. The model is validated with the experimental data providing satisfactory results. After presenting the dynamic equations of the system, a fuzzy controller is designed based on our prior knowledge of the system. The fuzzy rules and the membership functions are derived by a trial and error and heuristic method. Because of the nonlinear nature of the system the fuzzy rules are set to satisfy the requirements of the temperature control for different operating conditions of the engine. The performance of the fuzzy controller is compared with a PI one for different transient conditions. The results of the simulation show the better performance of the fuzzy controller. The main advantages of the fuzzy controller are the shorter settling time, smaller overshoot, and improved performance especially in the transient states of the system.

Keywords: Fuzzy control; IC engine; Coolant temperature; Simulation

1. Introduction

Nowadays, experimental tests and examinations of the performance of mechanical systems through experimental models are broadly under attention. The engine test bed is the heart of the automotive research centers working on improving the performance of the engines. In a test bed, the internal combustion engines are tested by a dynamometer under different operating conditions. In this case, the engine temperature must be controlled precisely, particularly in transient states [1]. This precise control can be achieved by engine coolant conditioning systems (CCS), mainly consisting of a heat exchanger, a control valve, and a controller. Most of the CCSs have been controlled by conventional controller techniques, especially conventional PID controllers because of their easy implementation and simple structure [2]. Because of the nonlinear and time-varying nature of these systems, conventional PID control schemes will not attain a high degree of control performance in all operating conditions [3]. On the other hand, more precise and less sensitive controllers are needed in engine test beds especially for transient conditions.

Over recent decades, there have been many improvements in the design theory of fuzzy logic controllers, and they have been widely used in thermal systems. In this paper a fuzzy logic controller is designed and simulated for the engine coolant conditioning system, which shows appropriate performance for different steady and transient conditions of the engine. In the next section the engine coolant conditioning system characteristics will be presented.

2. System description

The engine coolant conditioning system is designed to control the engine coolant temperature in steady and transient conditions. According to the literature, the cooling capacity of the system is approximately equal with engine output power [2]. The rated maximum specific power of a passenger car gas engine is typically 20-50 kW/L of the engine displacement [4]. Therefore, the service module is designed to match the thermal characteristics of the engine. The dynamic model derived in this paper is valid for different engines; however, for each engine the values of the model parameters should be deter-

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mined after a complete setup. The CCS consists of the following elements:

- Shell and tube heat exchanger
- Electro- pneumatic control valve
- PT100 temperature sensor
- Controller (PI or fuzzy)

The heat exchanger is the heat transfer unit of the system and an energy balance should be considered to choose the appropriate one. The heat exchangers are 1st order dynamic systems with large time constants. This means that they are thermally inertial systems and can affect overall system performance [5, 6]. Some notes are essential about the heat exchanger:

(1) The time constant of a heat exchanger depends on inlet and outlet temperatures, and cold and hot coolant flow rates. In our study the cold water flow rate is adjusted by the control valve and its inlet temperature is assumed to be constant.

(2) The hot flow is the engine coolant flow and its flow rate depends on the engine speed.

The electro-pneumatic control valve and PT100 temperature sensor response time are 1 s and 3 s, respectively. Since we are dealing with the slow dynamics of a thermal system with time constants as high as in the 100s, the effect of these elements on the system response is negligible.

2.1 Experimental setup

The experiments were conducted on a four cylinder 1800 cc gasoline engine. The engine specifications are given in Table 1.

The engine is coupled with a 130 kW eddy current dynamometer. The engine test bed, coolant conditioning system and their connections are shown in Figs. 1 and 2. The CCS shown in Fig. 2 is designed and built by the authors of the paper in Fuel and Environment Research Institute at University of Tehran.

Table 1. The engine specifications.

Displacement	1800 cc
Bore × Stroke	83 × 81.3 mm^2
Compression Ratio	9.31
Rated Power	110 HP (77 kW)

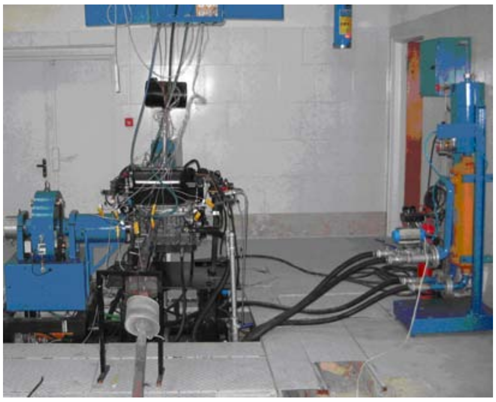

Fig. 1. Dynamometer, engine and coolant conditioning system.

Fig. 2. Coolant conditioning system in engine test bed.

3. Modeling and constitutive equations of the system

In this section the open-loop behavior of the system is modeled by an analytical-experimental approach. The control input of the system is the current exciting the valve which determines the proportion of the cooling tower water flow rate passing through the heat exchanger. In open-loop tests, this value is set manually and the system response is investigated. The process value of this system is the output temperature of the engine coolant. This variable is affected by the controlling input, and engine load and speed. The effect of these variables on the system performance is exactly considered in the thermal model. Fig. 3 shows the open-loop schematic of the system and Fig. 4 shows the thermal-fluid inputs and outputs of the combined engine and coolant conditioning system circuit. Now we discuss more details of the thermal-fluid dynamic modeling of the system.

The following energy conservation equation for a control volume is mainly used to derive the system differential equations.

$$\frac{dE_{c.v.}}{dt} = \dot{Q} + \dot{m}(h_i - h_e). \tag{1}$$

This equation is applied to two separate control volumes; engine block and engine coolant. For the engine block control volume the following equation is obtained.

$$m_b c_{pb} \frac{dT_b}{dt} = -\dot{Q}_1 - \dot{Q}_2 + \dot{Q}_3 \tag{2}$$

where the variables of this equation have been defined in the nomenclature section.

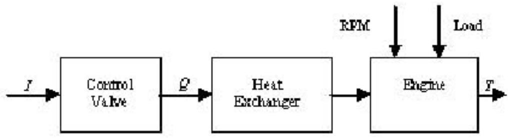

Fig. 3. Open-loop schematic of the system.

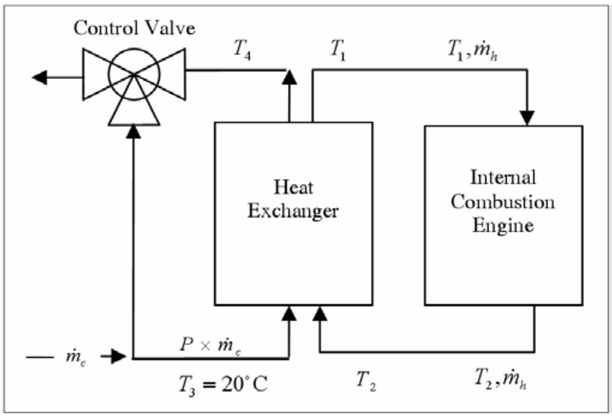

Fig. 4. Thermal-fluid schematic of the engine coolant conditioning circuit.

The heat transfer rate from the engine block to the engine coolant is calculated from the following equation—an average temperature is assumed for the engine block (T_b) and the temperature of the coolant is considered as T_1 ,

$$\dot{Q}_1 = h_i A_i (T_b - T_1). \tag{3}$$

In this equation, h_i is the convective heat transfer coefficient between the block and coolant. It is assumed that there is an overall average coefficient for different parts of the engine block. This coefficient is obtained from Eq. (4). This equation is based on the Dittus-Boelter equation and some analytical results from literature [7, 8]. The constant of this equation is set to be 3.76 to yield a heat transfer coefficient of $2500\ \mathrm{W/(m^2K)}$ in 5000 rpm and 87 °C .

$$h_i(T_1, \omega) = 3.76 \frac{\mathrm{Pr}^{0.4}\, \dot{m}_h^{\ 0.8}}{\mu^{0.8}}. \tag{4}$$

The values of Prandtl number and viscosity depend on the coolant temperature and are obtained from the following equations.

$$\mathrm{Pr} = \frac{168.3}{T_1 - 3.576}, \quad \mu = \frac{0.0268}{T_1 - 3.129} \quad \text{for} \quad 20^{\circ}\mathrm{C} < T_1 < 130^{\circ}\mathrm{C}. \tag{5}$$

These equations are derived experimentally from thermo-physical properties of saturated water listed in [7]. The engine coolant flow rate depends on the engine speed. A linear approximation is assumed to determine its value as follows.

$$\dot{m}_h \text{ (kg/s)} = \frac{0.92}{6000} \omega \text{ (rpm)}. \tag{6}$$

In Eq. (3) the inner heat transfer surface between the block and coolant is determined for the engine under study and its value is $0.7\ \mathrm{m}^2$. This is obtained by calculating the approximate contact surface area of the cylinders and other parts of the engine with the coolant.

The heat transfer rate from the engine block to the lab environment consists of convection and radiation heat transfer terms (Eq. 7):

$$\dot{Q}_2 = \underbrace{h_o A_o (T_b - T_\infty)}_{Convection} + \underbrace{e\sigma A_o (T_b^4 - T_\infty^4)}_{Radiation}. \tag{7}$$

The exact values of h_o and A_o are difficult to determine directly, so an assumption based on the literature is used. It is known that $\dot{Q}_2$ is about $\frac{1}{15}$ of the engine output power [2]. Therefore, the values of h_o and A_o are set to yield this known value of heat transfer (Table 2). The last term in Eq. (2) is the heat transfer rate from the combustion chamber to the engine block and is obtained from Eq. (8),

$$\dot{Q}_3 = T \times \omega. \tag{8}$$

The heat transfer rate to the engine coolant in steady state conditions from combustion chambers is equal to the engine output power [2]. This power is the product of engine load (T) and engine angular velocity (ω). In transient conditions of the engine, the heat transfer rate is related to the engine power as in Eq. (2).

In a similar approach we consider the engine coolant in the engine as a control volume and obtain the conservation equation for it,

$$m_e c_p \frac{dT_1}{dt} = \dot{Q}_1 + \dot{m}_h c_p (T_1 - T_2). \tag{9}$$

Finally, the governing differential equations of the system are summarized as follows:

$$\begin{cases} m_e c_p \dfrac{dT_1}{dt} = h_i A_i (T_b - T_1) + \dot{m}_h c_p (T_1 - T_2) \\ m_b c_b \dfrac{dT_b}{dt} = -h_i A_i (T_b - T_1) - h_o A_o (T_b - T_\infty) - e\sigma A_o (T_b^4 - T_\infty^4) + T \times \omega. \\ T_1 = (1 - \varepsilon Cr) T_2 + \varepsilon Cr T_3 \end{cases} \tag{10}$$

The last equation comes from the heat exchanger conservation equation stated in a form of $\varepsilon - NTU$ method. The heat capacity ratio Cr is the ratio of the minimum to the maximum

Table 2. The values of the system variables used for the simulation.

Symbol	Variable	Value
$\dot{m}_c$	Cooling water flow rate (from cooling tower)	0.6 Kg/s
T_3	Input temperature of cooling water flow	20°
m_e	Coolant mass in engine	2.8 Kg
m_b	Engine block mass	100 Kg
h_o	Convection heat transfer coefficient between engine and lab environment	$5\ \frac{\text{W}}{\text{m}^2\text{K}}$
A_o	Engine outer heat transfer surface	0.8 m^2
A_i	Engine inner heat transfer surface	0.7 m^2

Table 3. Experimental tests of system performance.

Test#	Engine rpm	Engine load (N.m)	Valve exciting current (%)
1	2500	40	77
2	2500	40	65
3	2500	40	80
4	2500	40	70
5	2500	80	70
6	2500	20	70
7	3500	80	70
8	1500	80	70

heat capacity and in our case is defined as follows:

$$Cr(P,\omega) = \frac{C_{\min}}{C_{\max}} = \frac{P\dot{m}_c}{\dot{m}_h} \tag{11}$$

where P is the opening percent of the control valve which is related to the percent of exciting current of the valve I according to the following equation:

$$P = \sqrt{2}\sin(\frac{\pi}{4}(1-I)). \tag{12}$$

Therefore, when the current is zero, the valve is fully open and when the current is one, the valve is closed. The effectiveness of the heat exchanger ε is calculated from Eq. (13). This formula is valid for shell and tube heat exchangers with one shell pass and even (2, 4 …) tube passes [7]. The heat exchanger used in the system is a shell and tube heat exchanger with one shell pass and one tube pass and there is no specific formula for this type of heat exchanger. Therefore, we have used the following formula as a substitutive one.

$$\varepsilon(P,T,\omega) = 2\times\left\{1 + Cr + (1+Cr^2)^{1/2}\times\frac{1+\exp[-NTU(1+Cr^2)^{1/2}]}{1-\exp[-NTU(1+Cr^2)^{1/2}]}\right\}^{-1}. \tag{13}$$

The number of transfer units (NTU) is determined from Eq. (14).

$$NTU(P,T,\omega) = \frac{nUA}{Pc_p\dot{m}_c} \tag{14}$$

where n is the number of the tubes and UA is the overall heat transfer coefficient for a tube. There is no exact formula to compute the value of UA for the one shell pass-one tube pass heat exchanger [9] and we need to find an experimental formula to estimate its value. First, a dimensional analysis is performed to determine the suitable relationship among the variables.

$$UA_{(\text{W/K})} = f(\omega_{(\text{rad/s})}, T_{(\text{N.m})}, P, T_{1(\text{K})}) \tag{15}$$

which results in Eq. (16),

$$UA = f(T\omega/T_1). \tag{16}$$

The simulations show that the dependency of UA on temperature T_1 is due to the variations of Prandtl number and viscosity values, which can be ignored with a good approximation. Therefore the final form of the Eq. (16) would be

$$UA = aP^b \times T \times \omega \times F. \tag{17}$$

The values of a, b and F are obtained from experiments. Our suggested approach is to find a and b from three experiments in which the load and speed of engine are constant. The correcting factor F is between 0.9 and 1.1, which improves the performance of the equation for different operating conditions. In the absence of this factor an average error of ±4% is observed in output engine coolant temperature.

The accuracy of the model is investigated for eight different experimental tests for various amounts of valve exciting current and engine speed and load. These tests are presented in Table 3. In each test the initial and final values of the engine outlet coolant temperature are in steady state condition. The step change is applied only in one of the system variables and the others are set to be constant.

Fig. 5 shows the results of the simulation for tests 1 to 4. In these tests the valve–exciting current is stepped according to the Table 3. The fitness of the model results to the experimental data is completely acceptable for these tests. The sharp variations in simulation results in the vicinity of the valve current change are due to the abrupt change of the heat exchanger UA which has a very fast dynamic in reality.

Figs. 6 and 7 are related to tests 5 and 6 in which the engine load is suddenly changed. In a real test this step change is not as sharp as in simulation and has an approximately linear trend from initial load to its final state. Therefore some discrepancies in the neighborhood of abrupt change are observed.

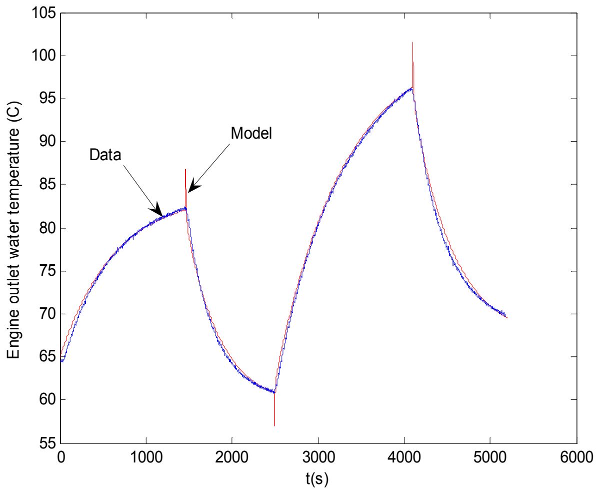

Fig. 5. Comparison between the model output and the experimental data for the engine outlet coolant temperature.

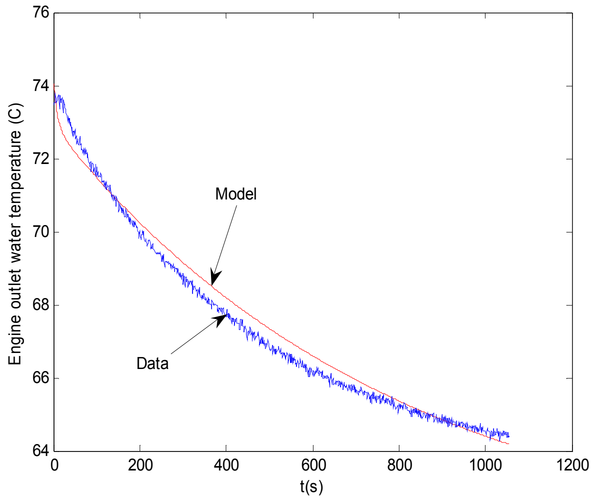

Fig. 6. Comparison between model results and the experimental data for test 5.

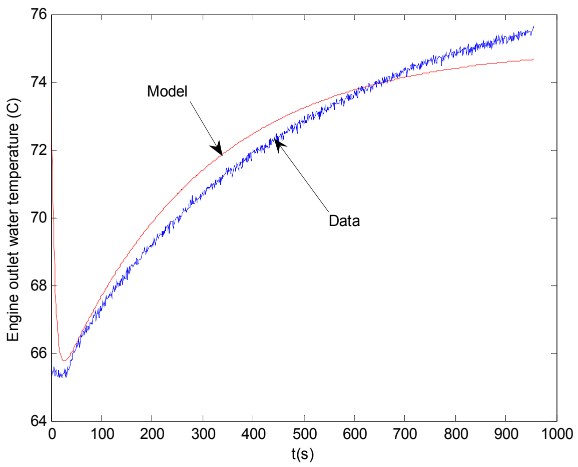

Fig. 7. Comparison between model results and experimental data for test 6.

Table 4. The ranges of the inputs and outputs of the fuzzy controller.

	e (°C)	P (kW)	VC
Range	-40-40	1-70	0-1

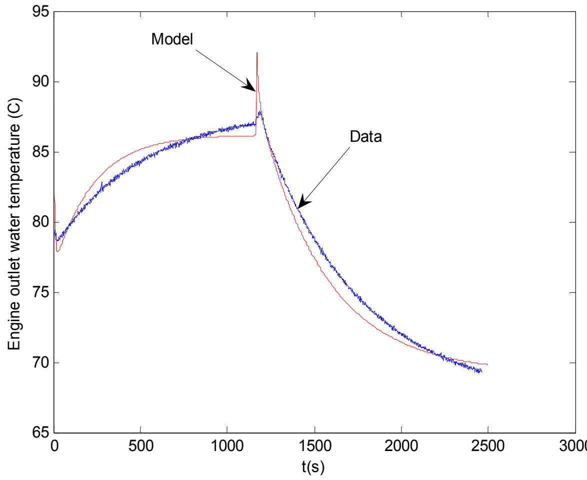

Fig. 8. Comparison between model results and the experimental data for engine outlet coolant temperature for tests 7 and 8.

Fig. 8 shows the same results for engine speed step change. The simulations show the acceptable accuracy of the derived model for different operating conditions of the engine.

In the next section, the design procedure of a fuzzy controller for this system will be discussed.

4. Fuzzy controller design

After the system is modeled, it is used to design a fuzzy controller to control the coolant temperature. The fuzzy controller is set up in Matlab as a two-input, one-output structure. The input variables are coolant temperature error (e) and engine power (P). The temperature error is the difference between temperature set-point and the actual temperature of the coolant. The engine output power is the product of the engine load and speed and it includes the variations of both quantities. The output variable is the valve current (VC). The ranges of the inputs and outputs are presented in Table 4.

The real valve current range is 0-20 mA and it has been normalized to 0-1. Appropriate selection of the inputs is a crucial step in designing the fuzzy controller. In our design, firstly the rate of temperature error (de/dt) was considered as an input to the system. But further works showed that this input had no additional effect on the controller performance and two mentioned inputs were sufficient for this system. Temperature error (e) should be considered because the closed-loop control requires the feedback of the process variable. Actually, the main performance of the controller is based on this input. Some notes are necessary about the second input (engine power). In engine test beds the values of the engine load and speed can be set independently with the existing controllers. This is essential to conduct different road and city

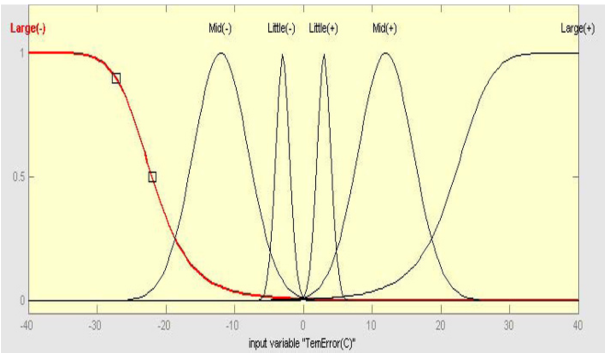

Fig. 9. Membership functions of the engine coolant temperature.

tests of the engines. In a special test cycle the load and speed of the engine are controlled by dynamometer. Therefore, the output power of the engine is controlled and changed via the controlling systems during a test. The role of this input in fuzzy controller will be discussed more in the next section.

4.1 Membership functions of fuzzy controller

A trial-and-error approach is used to determine the membership functions (MF) of the controller. The response of thermal systems is usually very slow and they have long time constants and delays. Therefore, in control efforts the future trend of the system response should be considered. The temperature error range is divided to six regions and an MF is set for each of them. We have considered the "high", "middle" and "little" MFs for both low and high temperatures. Fig. 9 shows the MFs of the temperature error. The Gaussian membership functions are chosen because of their soft switching nature. The marginal MFs are bell- shaped because the value of the MFs in these regions should be constant for an appropriate range.

The MFs of the engine power are chosen to be bell-shaped in two "Low" and "High" regions (Fig. 10). It is observed that the performance of the fuzzy controller for the low powers is not appropriate for the powers over 35 kW. Therefore, a membership function was selected for high power to ensure the better performance of the controller. Two regions were selected, which reduced the number of the rules and provided suitable results in simulations.

The selection of MFs for the temperature error e and the valve current VC is of a great importance. The MFs of the valve current should ensure the proper performance of the controller for different operating conditions, specifically the transient ones. Five MFs are selected for VC which span the whole range of the output from 0.3 to 1. The lower limit is set to 0.3 to prevent from undesirable actions of the controller which create great changes in system response. As shown in Fig. 11, the bell-shaped MFs are chosen for extremes and Gaussian MFs for the other regions. The label of the MFs are "Very Open", "Open", "Middle", "Closed", "Very Closed", respectively from low to high values.

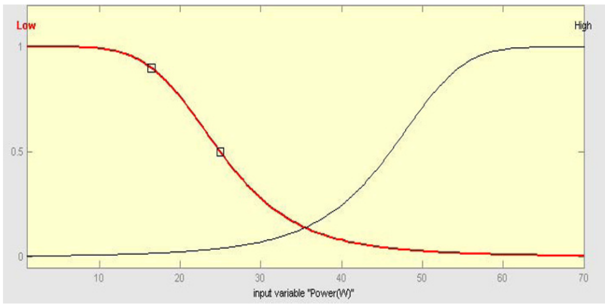

Fig. 10. Membership functions of the engine power.

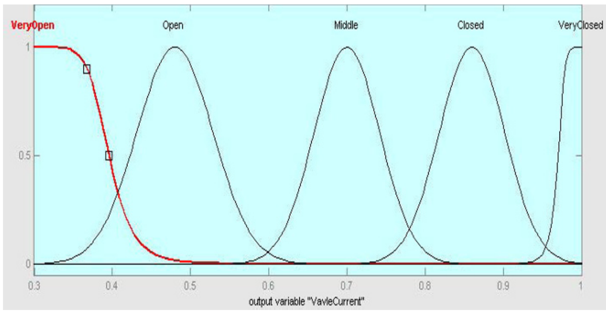

Fig. 11. Membership functions of the valve current as the output of the controller.

4.2 Rules of fuzzy controller and fuzzy inference

The assembled fuzzy system is of Mamdani type and the centroid defuzzification method is used [10].The derivation of the fuzzy controller rules is based on the trial and error approach and prior knowledge about the physics of the system [11]. Some open-loop simulations were run to determine the main characteristics of the system (Fig. 12). This nonlinear thermal system holds two main features. First, the time constant of the response in temperature increase (dT/dt>0) is larger than temperature decrease (dT/dt<0). Second, the response of the system is faster for higher engine powers in comparison to lower ones. The second feature reveals the significance of using especial MFs for low and high powers. These facts are shown in Fig. 12 and Table 5 for three different low, middle and high engine powers. Fig. 12 shows the step responses of the system for the valve current step from 0.7 to 0.8 and vice versa.

The control effort should correspond to this nature of the system. As an example when the power is low and temperature is increasing, the control effort should be sharp to push the system. In contrary, when the power is high and the temperature is decreasing the control action should not be too sharp.

Based on our knowledge about the dynamic behavior of the system, the fuzzy rules are determined as described in Table 6. In a fuzzy inference system, the “And” and implication methods are product and aggregation method is maximum.

The set of fuzzy rules, tuned on the numerical simulations, has been compiled by the compromise between fast attaining

Table 5. Comparison between settling time of the system Open-loop responses for different powers.

Power (kW)	dT/dt	Settling time (s)
70	Positive	900
70	Negative	700
40	Positive	1000
40	Negative	800
10	Positive	2000
10	Negative	1500

Table 6. Rules of the fuzzy controller.

#Rule	Temp. error (*e*)	Engine power	Valve current (*VC*)
1	Large(+)	none	VeryClosed
2	Large(-)	none	VeryOpen
3	Mid(-)	none	VeryOpen
4	Mid(+)	Low	VeryClosed
5	Mid(+)	High	Closed
6	Little(-)	Low	VeryOpen
7	Little(-)	High	Open
8	Little(+)	Low	Closed
9	Little(+)	High	Middle

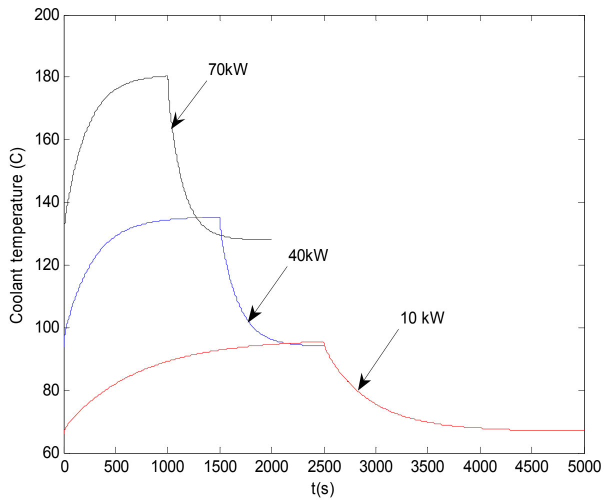

Fig. 12. Open-loop step responses of the system for different powers and valve current step from 0.7 to 0.8 and vice versa.

of the desired temperature and avoiding its overshoot. The following performance characteristics are considered in designing the controller:

(1) Small settling time of the closed-loop response of the system.

(2) Small overshoot of the response in transient conditions. When the engine load and speed change, coolant temperature should not vary in a wide range.

(3) Small tolerance of the temperature in steady-state conditions. In practice, a value of ±1°C is considered to be suitable.

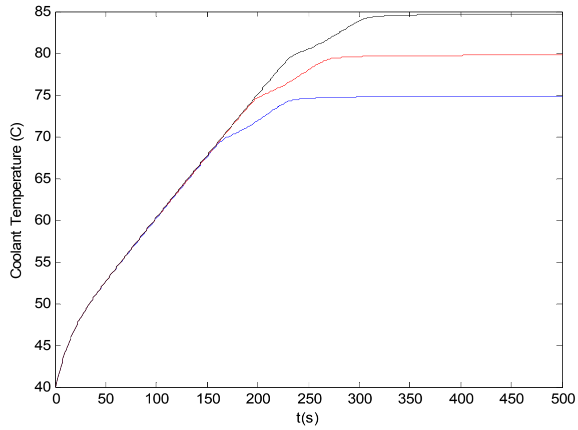

Fig. 13. Fuzzy controller performance for different temperature set-points and 15kW power.

The next section is the summary of the simulation results and the comparison of the fuzzy and PI controller performances.

5. Simulation results and fuzzy and PI controllers comparison

The simulations are conducted for different powers and temperature set-points. The performance of the fuzzy controller is compared with a PI controller. The controller gains are chosen to meet the abovementioned performance requirements. Proportional gain is 60 and the integrating gain is chosen to be 0.4. The set point value is considered to be constant during the engine tests and is about 75-85°C. The main objective of the controller is to maintain the outlet temperature of the engine in vicinity of the set point, in response to the variations of engine speed and load. The embedded control modules in the test bed allow us to change the values of the engine load and speed independently.

Fig. 13 shows the performance of the fuzzy controller to control the coolant temperature in three different temperature set-points. The initial coolant temperature is 40°C and the engine power is 15 kW. It is obvious that the settling time is smaller for lower set-points.

Fig. 14 is the results of the simulation for different engine powers. The set point is 80°C. The settling time is smaller for lower powers.

In few cases it is required to change the temperature set-point. Therefore, the controller should act appropriately in these cases. Fig. 15 shows the performance of the fuzzy and PI controllers for temperature control in T=85°C from the warm-up condition and also the temperature set-point step from 85°C to 75°C at *t*=700 s. The engine power is 30 kW for this simulation. In these cases, the response of the system with the fuzzy controller is faster than that of the PI controller.

Fig. 16 shows the simulation results of the engine power steps. The first step is from 40 kW to 50 kW at *t*=600 s and the second one is from 50 kW to 30 kW at *t*=900 s. The deviations of the coolant temperature from set-point, in response to

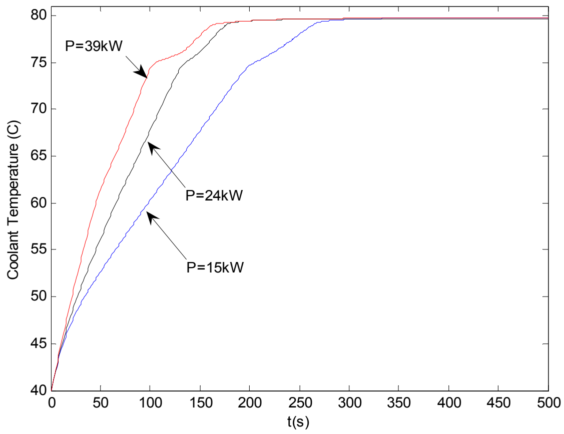

Fig. 14. Fuzzy controller performance for different engine powers.

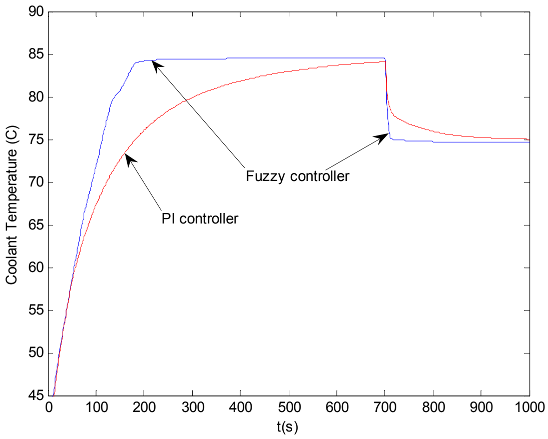

Fig. 15. Fuzzy and PI controller performances for control from warm-up condition and temperature set-point step.

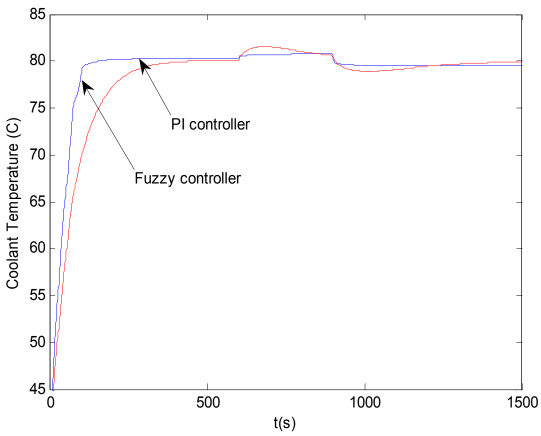

Fig. 16. Fuzzy and PI controller performances for control of warm-up condition and engine power steps.

the power steps, are smaller for fuzzy controller.

Fig. 17 shows the control effort of the fuzzy and PI controllers for described simulation conditions. The outputs are similar for a wide range, but the fuzzy controller adjusts the output in a smarter way to attain better performance.

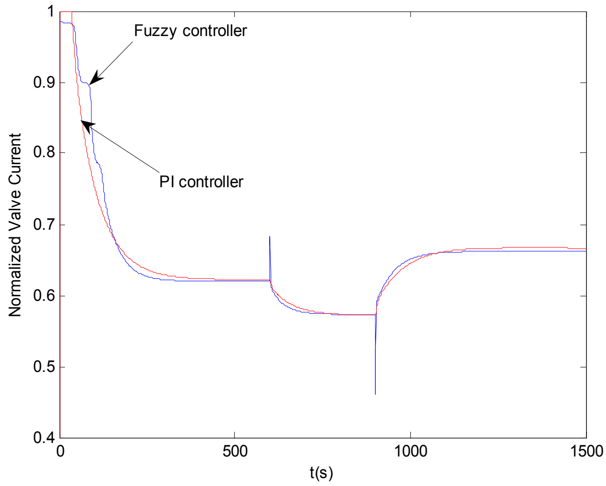

Fig. 17. Fuzzy and PI controller outputs for warm-up condition and engine power steps.

6. Conclusion

An analytical model of the engine coolant conditioning system in the engine test bed was derived. The model was verified with experimental data for different operating conditions of the system. This model was used as a suitable base for the fuzzy controller design. The membership functions and fuzzy rules of the controller were determined by a trial and error approach and based on our knowledge about the dynamics of the system. After the controller was designed, the closed-loop responses of the system were investigated in the presence of different operating conditions of the engine and satisfactory results were obtained. Finally, the fuzzy controller was compared with a conventional PI controller. The faster response to variations of the engine power, and less deviation from temperature set point especially in transient conditions were among the better performance characteristics of the fuzzy controller.

Nomenclature

CCS : Coolant conditioning system
E_{cv} : Energy of control volume
$\dot{Q}$: Net heat transfer rate of control volume
h_i , h_o : Input and output enthalpies of control volume
$\dot{m}_h$: Engine coolant flow rate (0.92 kg/s in 6000 rpm)
T_1 : Engine coolant inlet temperature (°C)
T_2 : Engine coolant outlet temperature (°C)
T_b : Engine block average temperature (°C)
P : Opening percent of the control valve
T : Engine load (N.m)
ω : Engine crankshaft angular velocity (rad/s)
h_i : Convective heat transfer coefficient between block and water (W/m^2K)
UA : Overall heat transfer coefficient for a tube
Cr : Heat capacity ratio
ε : Effectiveness of heat exchanger

$\dot{Q}_1$: Heat transfer rate from engine block to the engine coolant (W)
$\dot{Q}_2$: Heat transfer rate from engine block to the lab environment (W)
$\dot{Q}_3$: Heat transfer rate from combustion chambers to the engine block (W)
T_b : Engine block average temperature (°C)
m_e : Coolant mass inside the engine block (kg)
m_b : Engine block mass (kg)
c_{pb} : Specific thermal capacity of block material (J/kg-K).

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