Automotive Thermostat Valve Configurations: Enhanced Warm-Up Performance

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The automotive cooling system has unrealized potential to improve internal combustion engine performance through enhanced coolant temperature control and reduced parasitic losses. Advanced automotive thermal management systems use controllable actuators (e.g., smart thermostat valve, variable speed water pump, and electric radiator fan) that must work in harmony to control engine temperature. One important area of cooling system operation is warm-up, during which fluid flow is regulated between the bypass and radiator loops. A fundamental question arises regarding the usefulness of the common thermostat valve. In this paper, four different thermostat configurations were analyzed, with accompanying linear and nonlinear control algorithms, to investigate warm-up behaviors and thermostat valve operations. The configurations considered include factory, two-way valve, three-way valve, and no valve. Representative experimental testing was conducted on a steam-based thermal bench to examine the effectiveness of each valve configuration in the engine cooling system. The results clearly demonstrate that the three-way valve has the best performance as noted by the excellent warm-up time, temperature tracking, and cooling system power consumption. [DOI: 10.1115/1.3117183]

1 Introduction

The internal combustion engine has undergone extensive developments over the past 3 decades with the inception of sophisticated components and integration of electromechanical control systems for improved operation [1,2]. However, the automotive cooling system was overlooked until recently [3,4]. The conventional spark and compression ignition engine cooling systems can be improved with the integration of servo-motor-based actuators [5]. The replacement of conventional thermal management components (i.e., wax thermostat mechanical water pump, and mechanical radiator fan) with updated electric and/or hydraulic versions offers more effective operation [6]. In particular, the main function of the thermostat valve [7] is to control coolant flow to the radiator. Traditionally, this is achieved using a wax-based thermostat, which is passive in nature [8] and cannot be integrated in an engine management system [9]. A smart thermostat valve offers improved coolant flow control since it can operate at optimal engine conditions [10].

A series of automotive cooling system architectures may be created using different thermostat valve scenarios, as shown in Fig. 1. The valve and radiator baffle configurations considered include factory mode (Case 1), two-way valve (Case 2), three-way valve (Case 3), valve absent (Case 4), and valve absent with radiator baffles (Case 5). The factory configuration has the mechanically driven water pump and fan emulated by an electric variable speed pump and fan. The two-way valve operates by regulating coolant flow in either the bypass or radiator branch of the cooling circuit. The three-way valve proportionally directs the flow through either the bypass or the radiator loop. The proper utilization of a variable speed pump potentially allows the thermostat to be removed since the coolant flow rate may be predominantly controlled by the pump. The introduction of radiator baffles in the valve absent configuration provides external radiator airflow control (due to vehicle speed) further enhancing effectiveness.

In this paper, the thermostatic valve’s functionality will be investigated in ground vehicle advanced thermal management systems. In Sec. 2, an overview of the predominant cooling system configurations and the thermostat valve’s operation will be discussed. A model-based nonlinear control law, with underlying system thermal model, will be introduced in Sec. 3 to regulate the pump and fan servo-motor actuators. In Sec. 4, the experimental test bench will be reviewed. Representative experimental results will be presented and discussed in Sec. 5 to evaluate each configuration’s effectiveness. Finally, the summary is presented in Sec. 6.

2 Cooling System Configurations and Valve Operation

The automotive cooling system thermostat can be configured in several different manners to maintain engine temperature through combustion heat discharge in the radiator.

2.1 Traditional Thermostat Valve Fluid Control (Case 1).

The common cooling system has three key components working to regulate engine temperature: thermostat, water pump, and radiator fan. Conventional thermostats are wax based; their operation depends on the material properties of the wax in the thermostat housing and the coolant temperature surrounding it [11]. Traditional water pumps and radiator fans are generally mechanically driven by the engine’s crankshaft. Specifically, the water pump is driven as an accessory load while the radiator fan is often connected directly to the crankshaft with a clutch. Factory cooling systems typically present two problems [12]. First, large parasitic losses are associated with operating mechanical components at high rotational speeds due to their mechanical linkages. This not only decreases the overall engine power but increases the fuel consumption. Second, over/undercooling may occur since the water pump speed is directly proportional to the engine speed (again due to the mechanical linkages).

2.2 Two-Way Valve Fluid Control (Case 2).

The two-way smart valve controls flow by blocking the coolant from entering an external bypass. When the valve is oriented in the bypass mode, some coolant will always flow through the radiator, which is a major drawback when trying to rapidly warm the engine to operating temperature. Further, the amount of coolant flow through the bypass and radiator is determined by the valve’s geometry and location within the cooling circuit. It is possible to place two-way valves in many locations for an advanced cooling system that would alter the thermal dynamics. For instance, the valve could be shifted to the inlet of the radiator, preventing flow from entering the radiator (when fully closed) to aid in engine
warm-up times. However, a pressure drop was added in series with the radiator and some fluid will always flow through the bypass.

2.3 Three-Way Valve Fluid Control (Case 3). The operation of a smart three-way valve is very similar to the two-way valve. However, a three-way valve controls coolant flow through the bypass and radiator loops. Unlike the two-way valve, the coolant flow can be completely blocked from entering the radiator or bypass, which aids in engine warm-up time [13]. This is the primary advantage of utilizing a three-way valve in the cooling circuit. Although increased control is achieved, the introduction of hardware with greater functionality can be expensive. In addition, valve geometries can become complicated when designing a three-way valve that proportionally controls coolant flow while minimizing the pressure drop.

2.4 No Valve Fluid Control (Cases 4 and 5). When control over the coolant pump speed (and therefore flow rate) can be achieved, the possibility exists to eliminate the thermostat valve completely. As mentioned earlier, the thermostat’s main role is to regulate the coolant flow rate and direction. Therefore, the valve loses one of its primary purposes due to active pump speed control. The valve is now reduced to controlling fluid flow between the bypass and radiator loops, which is only required during warm-up conditions. However, the valve could potentially be eliminated if the pump circulates coolant as required by the engine. Note that coolant must be circulated at all times since hot spots may develop, leading to engine damage. Temperature control is handled by varying the pump speed (or flow rate). During warm-up conditions, the pump speed is minimized to reach operating temperature quickly. Once the engine reaches its operating temperature, the pump speed would then be adjusted according to the heat load. The radiator fan becomes active when the pump alone cannot control the thermal input from the engine and is adjusted to match the necessary amount of heat rejection. A further improvement of warm-up times, without a thermostat, may be achieved with servo-motor driven radiator baffles to control ram-air effects.

3 Thermal Models and Operating Strategy

To describe an engine’s in-cylinder thermal behavior, detailed multiple node lumped parameter thermal models were proposed by Bohac et al. [14] with application to automotive coolant flow control by Wagner et al. [4]. However, a reduced order mathematical model can describe the engine’s thermal management system transient response for controller design needs. The thermal dynamics for the engine and radiator nodes [15], $T_e(t)$ and $T_r(t)$, in Fig. 1 may be written as

$$C_e(T_e - T_i) = Q_o - c_{pc}m_i(T_e - T_i) + c_{pc}m_i(T_e - T_o)$$

$$C_r(T_r - T_i) = c_{pc}m_i(T_r - T_i) - e, c_{pc}m_i(T_e - T_o) - Q_o$$

The variables $Q_o(t)$ and $Q_o(t)$ denote total net heat generated by the combustion process including losses and radiator heat loss due to uncontrollable air flow, which may be estimated or obtained through testing.

In the two-way valve configuration, a flow rate exists through the radiator branch, $m_1(t)$, at all times so that $m_1 = (1-e)pHT_m_1 + e_c pHT_m_c$. The coolant mass flow rate through the bypass branch, $m_2(t)$, becomes $m_2 = (1-e_c)(1-H)m_c$. Note that the parameters $e$ and $e_c$ are temperature-dependent on the pressure drop across the radiator and bypass branches. The variable $H(x)$ represents the normalized valve position, which is dependent on the valve position, $x(t)$. Finally, the overall coolant mass flow rate is $m_1 = m_1 + m_2$. The cooling circuit dynamic behavior varies slightly when a three-way valve is introduced. The three-way valve may be modeled using a linear relationship between the normalized valve position, $H(x)$, and the coolant flow rate through the radiator branch, $m_1(t)$, for a given water pump speed. Thus, the flow rates through the radiator and bypass branches become $m_1 = Hm_1$ and $m_2 = (1-H)m_2$. If the valve and bypass are completely removed from the cooling system, then the flow rates will be equivalent, $m_1 = m_2$.

The three-way valve dynamics may be applied to evaluate the traditional factory thermostat behavior (Case 1) by adjusting the smart valve’s operation. The valve position, $H(x)$, will respond in a linear manner to the coolant temperature so that [16]

$$H = \begin{cases} 
0, & T_e < T_l \\
T_e - T_l, & T_l \leq T_e \leq T_h \\
1, & T_e > T_h
\end{cases}$$

The parameters $T_l$ and $T_h$ represent the temperatures at which the wax in the thermostat begins to soften and fully melt. In an actual wax thermostat, hysteresis occurs while the wax is changing states such that the valve’s operation is nonlinear. For this paper, the hysteresis was neglected. For on/off (or bang-bang) valve control (Cases 2 and 3), the control authority is

$$H = \begin{cases} 
0, & T_e < T_o - \Delta T \\
1, & T_e \geq T_o - \Delta T
\end{cases}$$

where $\Delta T$ is the boundary layer about the desired engine temperature, $T_o(t)$. The boundary layer was introduced to reduce valve dithering. Note in Eqs. (2) and (3) that $H=1$ corresponds to coolant flow completely through the radiator. Similarly, complete coolant flow through the bypass occurs when $H=0$. Remember that Cases 4 and 5 remove the thermostat.

The main purpose of the engine’s thermal management system is to maintain a desired engine block temperature, $T_{eq}(t)$, while accommodating the unmeasurable combustion process heat input, $Q_o(t)$, and the uncontrollable air flow heat loss across the radiator, $Q_o(t)$. To achieve this goal, a Lyapunov-based nonlinear controller was developed so that the engine’s coolant temperature, $T_e(t)$, tracks the desired temperature, $T_{eq}(t)$, by regulating the system actuators (variable speed electric water pump and radiator fan) in harmony with each other. Note that in Eq. (1), the signals $T_e(t)$, $T_r(t)$, and $T_s(t)$ are measured by thermocouples (or thermostats). The system parameters $c_{pcm}$, $c_{pcm}$, $C_e$, $C_r$, $C_r$, and $e$ are assumed to be completely known and constant throughout the engine’s operation.
The controller objective is to ensure that the actual engine temperature, $T_e(t)$, tracks the desired trajectory, $T_{ed}(t)$, such that $T_e(t) \to T_{ed}(t)$ as $t \to \infty$ while compensating for the system variable uncertainties $Q_o(t)$ and $Q_i(t)$.

To formulate the control law, the dynamics described in Eq. (1) can be rewritten as

$$C_r \dot{T}_r = Q_{in} - u_e,$$  \hspace{1cm}  (4)

where $u_e(t)$ and $u_i(t)$ are the control inputs, which are defined as

$$u_e \triangleq c_p \dot{m}_c (T_e - T_i), \hspace{0.5cm} u_i \triangleq e \dot{c}_{pm} \dot{m}_a (T_e - T_o)$$  \hspace{1cm}  (5)

A Lyapunov-based nonlinear controller can be developed and applied to regulate the engine temperature, similar to that in Ref. [15], so that the control law (which establishes a basis to determine the pump and fan speeds) is designed as

$$u_e = - (K + \alpha) [e - e_o] - \int_{t_o}^{t} [\alpha(K + \alpha)e(\tau) + \rho \text{sgn}(e(\tau))] d\tau$$  \hspace{1cm}  (6)

In this expression, the final term, $\rho \text{sgn}(e)$, compensates for the variable unmeasurable input heat, $Q_{in}(t)$. Finally, the variable $e(t) = T_{ed}(t) - T_i(t)$ and $e_o$ is the initial temperature error.

The radiator’s mathematical description in Eq. (1) states that it operates normally (i.e., as a heat exchanger) if the effort of the radiator fan, denoted by $u_f(t)$ in Eq. (4), is set equal to the effort produced by the water pump, denoted by $u_p(t)$ [17]. Thus, the control input $u_f(t)$ provides the signal $\dot{m}_f(t)$ from the first expression in Eq. (5), and the control input $u_p(t)$ provides the signal $\dot{m}_a(t)$ from the second expression in Eq. (5). The signal $\dot{m}_f(t)$ is unipolar, so a commutation strategy determines the radiator coolant mass flow rate as

$$\dot{m}_f \triangleq \frac{u_f [1 + \text{sgn}(u_f)]}{2c_p(T_e - T_i)}$$  \hspace{1cm}  (7)

The coolant mass flow rate, $\dot{m}_f(t)$, or pump effort, may be determined using Eq. (7) and the valve configuration with its normalized position. For Cases 2 and 3, the coolant flow rates become $\dot{m}_f(t) = \dot{m}_f/(1 - \delta) + \dot{m}_a/\delta$ and $\dot{m}_f(t) = \dot{m}_a/\delta$. If a valve does not exist for Cases 4 and 5, then $\dot{m}_f(t) = \dot{m}_a$. Note that the pump command voltage is determined by an a priori empirical relationship [18]. If $u_i(t)$ is bounded for all time, $\dot{m}_f(t)$ is bounded for all time per Eq. (7).

A second commutation strategy computes the unipolar control input $\dot{m}_a(t)$ so that

$$\dot{m}_a \triangleq \frac{u_a [1 + \text{sgn}(u_a)]}{2c_p(T_e - T_o)}$$  \hspace{1cm}  (8)

As stated earlier, $u_a(t) = u_f(t)$. The radiator fan speed determines the radiator air flow rate, which does not include the ram-air flow due to vehicle speed. The ram-air effects are handled by $Q_o$ in Eqs. (1) and (4). Again, an a priori empirical relationship determined the fan motor voltage. From this definition, if $u_f(t)$ is bounded for all time, then $\dot{m}_a(t)$ is bounded for all time. Note that Eq. (5) is utilized to develop Eqs. (7) and (8). For further details, the reader is referred to Ref. [17].

### 4 Thermal Test Bench

An experimental test bench was created to investigate the thermostaat valve configurations of Cases 1–5. This custom bench offered maximum flexibility and a repeatable environment (refer to Fig. 2). Clemson University Facilities steam was used to rapidly heat engine coolant, which flowed through a double pass shell and tube heat exchanger, to emulate combustion and the engine block. The integration of a 6.0L International V-8 engine block into the...
test bench offered a thermal capacitance similar to actual operation. From the engine, the coolant flowed to the smart thermostat valve, which can be selected to operate in either the traditional (Case 1), two-way (Case 2), three-way (Case 3), or no valve with/without baffles (Cases 4 and 5) through a series of valves. For Cases 1 and 3, Valve A is closed and Valve B is opened. In contrast, Case 2 operation occurs when Valve A is opened and Valve B is closed. This action forces the coolant to flow through either the smart thermostat valve or Valve A as it would in the two-way valve operation per Sec. 2.2. Also, when Valve A is opened and Valve B is closed, Cases 4 and 5 may be explored by positioning the smart valve for flow through the radiator, $H=1$.

To calculate the rate of heat transfer, $Q_{in}$, condensed steam was collected and measured. It was assumed that the amount of condenser condensate is proportional to the amount of heat transferred to the circulating coolant. Overall, heat transfer rates of up to 60 kW can be achieved. Two J type Omega thermocouples measure the coolant temperatures at the engine, $T_{d}(t)$, and radiator, $T_{r}(t)$, outlets. The coolant mass flow, $m_{c}(t)$, was determined using an Omega paddle-wheel mass flow meter placed after the pump. The custom steel body smart valve, with Teflon-filled Delrin piston, was linearly actuated by a Litton servo-motor driven worm gear. A brass centrifugal pedestal pump was driven by a 240VAC Reliance electric motor. The Summit electric radiator fan has a diameter $d=45.7$ cm with flow rates up to 850 lfs.

5 Experimental Results and Discussion

Five different valve and radiator baffle configurations were investigated on the steam test bench using the proposed control strategies to study temperature warm-up time, tracking error, and overshoot, as well as total actuator power consumption. The configuration tests are presented in Table 1 with Case 5 reflecting the radiator blocked by baffles during warm-up. Representative experimental results are presented for Cases 1 and 3; the reader is referred to Ref. [19] for test results on all configurations. The warm-up time, $t_{wu}$, is the time required for the engine temperature, $T_{e}(t)$, to reach its desired set point, $T_{d}(t)$. The absolute state temperature error, $|E_{a}|$, represents the difference between $T_{e}(t)$ and $T_{d}(t)$ at steady-state operation. The temperature overshoot, $O_{sw}$, denotes the difference between the engine temperature, $T_{e}(t)$, at its peak value and the desired engine temperature, $T_{d}(t)$. The total power consumption, $P_{total}$, is the average power consumed by the pump and fan during the test $(0<t<40 \text{ min})$. Note that the power consumed by the valve is negligible and was ignored. All the tests began at $T_{e}(0)=305$ K with $T_{d}(t)=363$ K. To simulate a vehicle driving at a constant speed and load, the input heat was selected as $Q_{in}(t)=35$ kW. Further, the wind speed associated with $Q_{in}(t)$ was approximately 100 km/h. The control gains, selected through tuning, were $K=23$, $\alpha=1.0 \times 10^{-4}$, and $\rho=0.1$.

5.1 Factory Configuration (Case 1). In the factory cooling configuration, the engine and radiator temperature responses are shown in Fig. 3(a) with no apparent temperature overshoot since the set point, $T_{d}(t)$, was not achieved. A steady-state temperature offset of $|E_{a}|=0.8$ K was observed. Constant radiator air flow, $m_{c}=1.5$ kg/s, corresponded to fixed engine speed and clutch operation, as displayed in Fig. 3(b). The water pump flow rate was maintained at approximately $m_{c}=1.5$ kg/s to emulate constant engine speed (Fig. 3(c)). Operation of the thermostat valve was controlled by Eq. (2) with $T_{e}=358$ K and $T_{d}=368$ K; the normalized valve position is displayed in Fig. 3(d). The valve started opening at $t=296$ s with steady-state operation achieved at $t=2000$ s. The coolant flow rate through the entire cooling circuit decreased when the valve opened reflecting different pressure drops between the radiator and bypass loops.

5.2 Three-Way Valve Configuration (Case 3). The main attribute of the three-way valve resides in the ability to route coolant proportionally through the bypass and the radiator loops. In Fig. 4(a), the thermal response of the engine and radiator coolant temperatures is displayed. The three-way valve switches between the bypass and radiator at $t=359$ s, $t=428$ s, and $t=471$ s (refer to Fig. 4(d)) to realize a short warm-up time of $t_{wu}=363.9$ s. A minimum steady-state error was demonstrated with $|E_{a}|=0.2$ K, which may be attributed to the improved coolant flow control associated with the three-way valve, as discussed in Sec. 2.3. The fan and coolant mass flow rates (Figs. 4(b) and 4(c)) again display flow oscillations during the valve’s operation due to the high control gains. It is important to note that these gains were selected to minimize the temperature tracking error with the understanding that flow oscillations may occur.

5.3 Configuration Performance Comparison. The experimental data presented in Figs. 3 and 4 were summarized in Table 2 plus results for Cases 2, 4, and 5. To aid in the selection of an advanced cooling system architecture focused on the engine warm-up scenario, four observations are discussed.

Observation 1. One of the greatest effects on engine warm-up time (outside of combustion events) is the control of fluid flow across the radiator. The blockage of coolant and/or air flow through the radiator achieves the best warm-up times. The use of a three-way valve or radiator baffles effectively achieves this condition. A two-way valve is not effective at controlling warm-up times when positioned in the bypass circuit and may be removed under this criterion.

The comparison of warm-up times, $t_{wu}$, between the various cooling system configurations reveals that the three-way valve architecture achieved the shortest time followed by the valve absent with baffle configuration. Note that the factory thermostat (Case 1) did not reach the desired temperature due to overcooling so its warm-up time was not reported. A short warm-up time can be primarily attributed to restricting the coolant flow to the bypass allowing minimal heat loss. This effect is especially evident when the two-way and no valve (Cases 2 and 4) warm-up times are considered given that some coolant will flow through the radiator during warm-up. The two-way valve configuration warm-up time, $t_{wu}=558.0$ s, is similar to the valve absent, $t_{wu}=594.1$ s. Hence, the two-way valve is largely ineffective during the warm-up scenario. When air flow across the radiator is blocked (Case 5), the warm-up time is on par with that of the three-way valve design ($t_{wu}=363.9$ versus $t_{wu}=382.9$ s). The blockage of air flow, due to the radiator fan and/or surrounding environment, through the radiator reduces the warm-up time. This phenomenon is evident from the second expression in Eq. (1), which accounts for the air flow.
mass flow rate through the radiator, $m_a$, and external ram air, $Q_o$.

Observation 2. To minimize temperature tracking error, precise fluid control must be maintained by the thermostat valve, coolant pump, and/or radiator fan. In each case, the engine temperature can be regulated per Eq. (1) by controlling the radiator coolant mass flow rate, $m_r(t)$, through the water pump speed or valve position (if applicable). The steady-state temperature error, $|E_{ss}|$, ranking can be stated as Cases 3, 2, 5, 4, and 1, respectively. The three-way valve offered a 28.6% improvement over the two-way valve. Similar observations may be made for Case 3 when compared with the valve absent Case 4 and the valve absent with baffles Case 5 with 36.6% and 31.1% improvements. An explanation for this behavior may be attributed to the improved fluid control in the three-way valve configuration. The factory configuration Case 1 had the highest relative error measure due to the fact that the engine temperature was realized by overcooling the system through elevated coolant pump and radiator fan operation. The emulated wax thermostat valve only allowed 42% of maximum radiator flow to control the coolant temperature to a neighborhood of $T_{ed}$. 

Observation 3. Power consumption may be minimized using a smart cooling system (Cases 2–5) in comparison to the factory architecture (Case 1). Further, the three-way valve configuration (Case 3) consumed less power than the valve absent design (Case 4) without the associated increase in warm-up time and error measure.

The power measure

$$P_{pump} = \frac{1}{\Delta t} \int_{t_0}^{t_1} \left[ \frac{1}{2 \rho c_p A_c} \right] \frac{dp_c}{d\tau} d\tau$$

and

$$P_{fan} = \frac{1}{\Delta t} \int_{t_0}^{t_1} \left[ \frac{1}{2 \rho c_p A_f} \right] \frac{dp_c}{d\tau} d\tau$$

calculates the average power consumed (refer to column 6 in Table 2) by the pump and fan, respectively, over the time period $\Delta t = t_1 - t_0 = 40$ min. As stated earlier, power consumed by the valve is minimal and was neglected. For Cases 2 and 3, the valve position algorithm presented in Eq. (2) was utilized only for the warm-up condition. Once the engine was sufficiently warm, the engine temperature was regulated using the control law in Eq. (6). The pump and fan speeds were chosen based on the commutation strategies of Eqs. (7) and (8) plus the control signal from Eq. (6). Observing the power consumption values, the three-way configuration (Case 3) consumed the least power, $P_{total} = 24.3$ W, followed closely by the valve absent design (Case 4) at $P_{total} = 24.7$ W. However, the trade-offs between the lower power consumption for the valve absent configuration were longer warm-up times and increased error measure. The factory configuration consumed the most power during testing, $P_{total} = 109.4$ W, due to the
constant pump and fan operation. In the valve absent with baffle design (Case 5), the total power consumption does not consider the power required to operate the baffles, which would increase the reported value in Table 2.

**Observation 4.** The three-way valve configuration provides the most benefits, with very few drawbacks in system design, and outperforms all other valve configurations. If simplicity is desired, completely removing the valve negatively impacts the cooling system performance. The addition of baffles, while removing the thermostat valve, provides similar performance to the three-way valve configuration at the cost of additional radiator hardware.

From a design perspective, the valve absent configuration (Case 4) would be ideal for cooling system simplicity. Although hardware is eliminated by removing the valve, the coolant warm-up time increased (rank 3). The addition of baffles (Case 5) improved the warm-up time when the valve was removed (rank 2). For maximum control and flexibility, the three-way valve design (Case 3) provided lower warm-up times and tracking error with little power consumption (rank 1). The two-way valve configuration (Case 2) did not compare favorably with the three-way valve’s performance due to longer warm-up times and carries the same relative cost and time for cooling system integration (rank

![Fig. 4 Case 3—Three-way valve configuration with (a) engine and radiator temperatures for a desired engine temperature $T_{\text{req}}$, (b) air mass flow rate through the radiator fan, (c) coolant mass flow rate through the water pump, and (d) normalized valve position.](image)

**Table 2** Summary of the valve configuration test performance in terms of warm-up time, absolute steady-state error, temperature overshoot, average power consumption, and relative rank. Case 5 does not consider energy required to operate the baffles.

| Case | Configuration     | $t_{wu}$ (s) | $|E_a|$ (K) | $O_{\text{sh}}$ (K) | $P_{\text{total}}$ (W) | Rank |
|------|------------------|--------------|----------|-------------------|------------------------|------|
| 1    | Traditional factory | n/a          | 0.8      | n/a               | 109.4                  | 4    |
| 2    | Two-way valve     | 558.0        | 0.2      | 4.3               | 33.1                   | 5    |
| 3    | Three-way valve   | 363.9        | 0.2      | 3.6               | 24.3                   | 1    |
| 4    | No valve          | 594.1        | 0.3      | 4.3               | 24.7                   | 3    |
| 5    | No valve with baffles | 382.9      | 0.3      | 3.2               | 36.6                   | 2    |

*aThe engine coolant temperature never reached the desired value of $T_{\text{req}}=363\text{ K.}*

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5). Finally, the factory configuration (Case 1) had a favorable relative warm-up time compared with the other cases at the critical cost of total power consumption and steady-state temperature error (rank 4).

6 Summary

The automotive thermostat valve’s primary role is to route the coolant flow between the bypass and radiator branches during warm-up conditions. In this paper, four valve configurations were examined and tested for effectiveness: factory, two-way valve, three-way valve, and valve absent. A reduced order lumped parameter thermal model was presented to facilitate the design of linear and nonlinear control algorithms. Some of the model limitations include neglecting heat transfer from the engine block to ambient surroundings and the quick identification of model parameters, which will be addressed in future studies. To summarize the findings, the three-way valve configuration provides excellent temperature tracking, power consumption, and warm-up time when compared with the other cases. The two-way valve and valve absent configurations were very similar in performance, leading to the conclusion that a two-way valve can possibly be eliminated entirely from the cooling system. Finally, a trade-off exists between the three-way valve and valve absent configurations. The inherent cost of designing and implementing a three-way valve must be weighted against the improved performance.

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Nomenclature

\[ A_p = \text{pump outlet cross section area (m}^2) \]
\[ A_f = \text{fan blowing area (m}^2) \]
\[ c_{pa} = \text{air specific heat (kJ/kg K)} \]
\[ c_{pe} = \text{coolant specific heat (kJ/kg K)} \]
\[ C_e = \text{engine block thermal capacitance (kJ/K)} \]
\[ C_r = \text{radiator thermal capacitance (kJ/K)} \]
\[ d = \text{initial cylinder (cm)} \]
\[ e = \text{engine temperature tracking error (K)} \]
\[ e_o = \text{initial engine temperature tracking error (K)} \]
\[ E_o = \text{steady-state error (K)} \]
\[ H = \text{normalized valve position (%)} \]
\[ K = \text{control gain} \]
\[ m_f = \text{fan air mass flow rate (kg/s)} \]
\[ m_{bat} = \text{bypass coolant mass flow rate (kg/s)} \]
\[ m_p = \text{pump coolant mass flow rate (kg/s)} \]
\[ m_r = \text{radiator coolant mass flow rate (kg/s)} \]
\[ Q_{bat} = \text{temperature overshoot (K)} \]
\[ P_{fan} = \text{fan power (kW)} \]
\[ P_{pump} = \text{pump power (kW)} \]
\[ P_{total} = \text{total power (kW)} \]
\[ Q_{in} = \text{combustion process total heat (kW)} \]
\[ Q_{u} = \text{uncontrollable air flow heat loss (kW)} \]
\[ sgn = \text{standard signum function} \]

\[ T_e = \text{engine outlet coolant temperature (K)} \]
\[ T_{ed} = \text{desired engine coolant temperature (K)} \]
\[ T_h = \text{liquid wax temperature (K)} \]
\[ T_f = \text{wax softening temperature (K)} \]
\[ T_r = \text{radiator outlet coolant temperature (K)} \]
\[ T_a = \text{surrounding ambient air temperature (K)} \]
\[ t = \text{time (s)} \]
\[ t_w = \text{warm-up time (s)} \]
\[ u, u_p = \text{control inputs} \]
\[ x = \text{valve position (cm)} \]
\[ \alpha = \text{control gain} \]
\[ \Delta p = \text{pressure drop across the radiator (bar)} \]
\[ \Delta T = \text{valve operating temperature deviation (K)} \]
\[ \Delta t = \text{sample time (s)} \]
\[ e_b = \text{bypass valve effectiveness (%)} \]
\[ e_r = \text{radiator fan effectiveness (%)} \]
\[ \rho = \text{control gain} \]
\[ p_a = \text{air density (kg/m}^3) \]
\[ p_c = \text{coolant density (kg/m}^3) \]

References