Automotive Thermostat Valve Configurations for Enhanced Performance – Control and Testing

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ABSTRACT
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 have been 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.

1. INTRODUCTION

The internal combustion engine has undergone extensive developments over the past three decades with the inception of sophisticated components and integration of electro-mechanical control systems for improved operation (Stence, 1998; Schoner, 2004). For instance, stratified charge and piston redesign offer improved thermal efficiency through lean combustion, directly resulting in lower fuel consumption and higher power output (Evans, 2006). Further, variable valve timing adjusts engine valve events to reduce pumping losses on a cycle-to-cycle basis (Mianzo and Peng, 2000; Hong *et al.,* 2004). However, the automotive cooling system has been overlooked until recently (Couëtouse and Gentile, 1992; Wagner *et al.,* 2002a). The conventional spark and compression ignition engine cooling systems can be improved with the integration of servo-motor based actuators (Melzer *et al.,* 1999). Replacement of conventional thermal management components (i.e., wax thermostat mechanical water pump, and mechanical radiator fan) with updated electric and/or hydraulic versions offer more effective operation (Redfield *et al.,* 2006). In particular, the main function of the thermostat valve (Wanbsganss, 1999) is to control coolant flow to the radiator. Traditionally, this is achieved using a wax-based thermostat which is passive in nature (Allen and Lasecki, 2001) and cannot be integrated in an engine management system (Wagner *et al.,* 2002b). A smart thermostat valve offers improved coolant flow control since it can be controlled to operate at optimal engine conditions (Visnic, 2001).

An electric thermostat valve may be designed with different architectures and control strategies to support a variety of cooling system configurations. For example, a DC motor controlled two-way valve may be utilized at multiple locations in a cooling circuit (Chastain and Wagner, 2006). Similarly, a solenoid controlled three-way valve offers similar functionality to traditional thermostats but could be electrically controlled by the engine control module (ECM). In general, the valve design dictates its placement in the cooling system since valve geometry contributes to the dynamics of the overall cooling system. It should be noted that the thermostat valve may be located on the engine block with internal passages for coolant flow or external to the block with supporting hoses. The next generation of internal combustion engines should be designed to facilitate advanced thermal management concepts.

A series of automotive cooling system architectures may be created using different thermostat valve scenarios as shown in Figure 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 without radiator baffles (Case 4); and valve absent with radiator baffles (Case 5).

The factory configuration has the mechanically driven water pump and radiator 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 and/or radiator loop. The proper utilization of a variable speed pump potentially allows the thermostat valve 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.

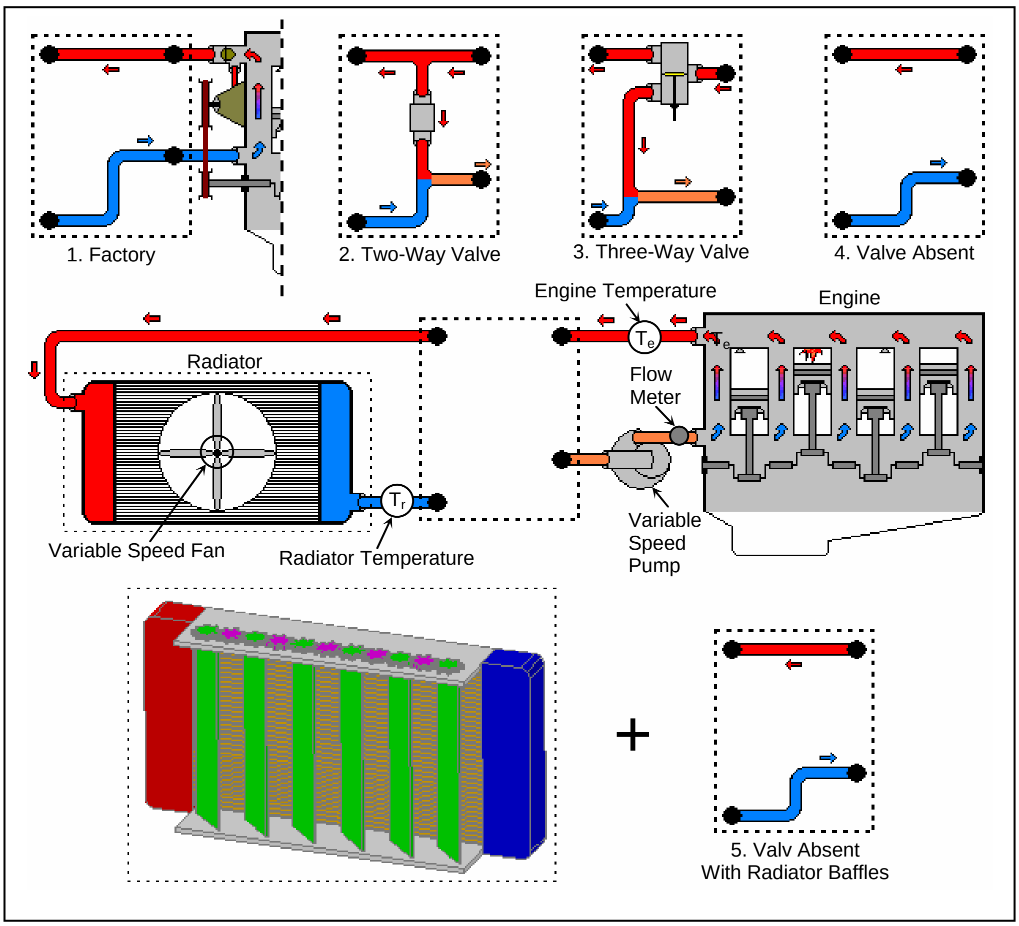

Figure 1: Five valve configurations to enhance fluid flow control; note the two thermocouples, $T_e(t)$ and $T_r(t)$, and fluid flow meter after the pump

Hypothesis: *The automotive thermostat valve's primary role is to route the coolant flow between the bypass and radiator branches during warm-up conditions. The best thermostat valve cooling system configuration utilizes a computer controlled three-way valve since it offers the most precise coolant flow regulation for warm-up scenarios.*

In this paper, the thermostatic valve's functionality will be investigated in ground vehicle advanced thermal management systems. In Section 2, an overview of the predominant cooling system configurations and the thermostatic valve's operation will be discussed. A model-based nonlinear control law, with underlying system thermal model, will be introduced in Section 3 to regulate the coolant pump and radiator fan servo-motor actuators. Two valve control strategies will also be introduced. In Section 4, the experimental test bench which creates a repeatable testing environment will be reviewed. Representative experimental results will be presented and discussed in Section 5 to evaluate each configuration's effectiveness in terms of warm-up time, temperature tracking, temperature overshoot, and power consumption. Finally, the summary is presented in Section 6. The Appendix contains a complete Nomenclature List.

2. COOLING SYSTEM CONFIGURATIONS AND VALVE OPERATION

The typical automotive cooling system has two main thermal components: engine and radiator. The coolant flow through the engine loop transports excess combustion heat to the radiator loop which dissipates this heat. Controlling and directing coolant flow between these two loops is the main function of most thermostat valves. This functionality may be accomplished through different valve configurations and system architectures.

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 (refer to Figure 2). In operation, when the engine is cold, the thermostat is closed and coolant is forced to flow through an internal engine bypass (usually a water passage parallel to the engine water jackets). Once the coolant reaches the desired operating temperature, the thermostat begins to open and allow coolant to flow through the radiator where excess heat can be rejected. Coolant flowing through the radiator is further cooled by the radiator fan pulling air across the radiator. When the coolant has dropped below the thermostat temperature rating, the valve closes (via spring force) directing the coolant again through the bypass. 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 (Choukroun and Chanfreau, 2001). 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.

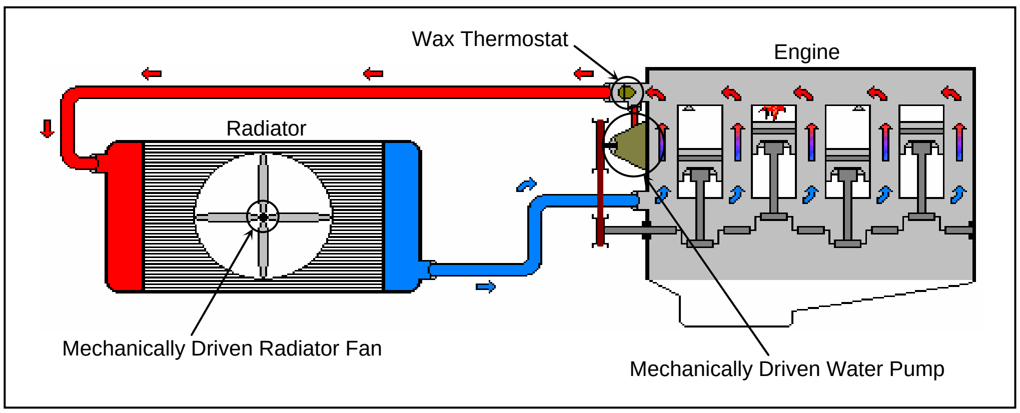

Figure 2: Factory cooling system configuration demonstrating the use of mechanically driven water pump and radiator fan with a wax thermostat (Case 1).

Factory cooling systems typically present two problems (Chalgren and Barron, 2003). 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. Additionally, these parasitic losses are compounded since the traditional cooling system components are designed for maximum (and often infrequent) cooling loads. Second, over/under cooling may occur since the water pump speed is directly proportional to the engine speed (again due to the mechanical linkages). At low engine speeds, the water pump may not be circulating coolant fast enough to properly cool the engine at higher loads. Similarly, the water pump may be circulating the coolant too fast, causing the engine to be overcooled and lose efficiency at higher speeds. Fundamentally, the traditional cooling system is passive and there is no direct control over its operation.

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 as shown in Figure 3. When the engine is cold, the valve is open and coolant flows through the bypass at a rate proportional to the pressure drop across the bypass and valve. Therefore, the pressure drop can be partially controlled by the valve position. During this time, the radiator is also receiving a portion of the coolant flow. Once the engine has reached operating temperature, the valve begins to close and coolant is routed through the radiator only.

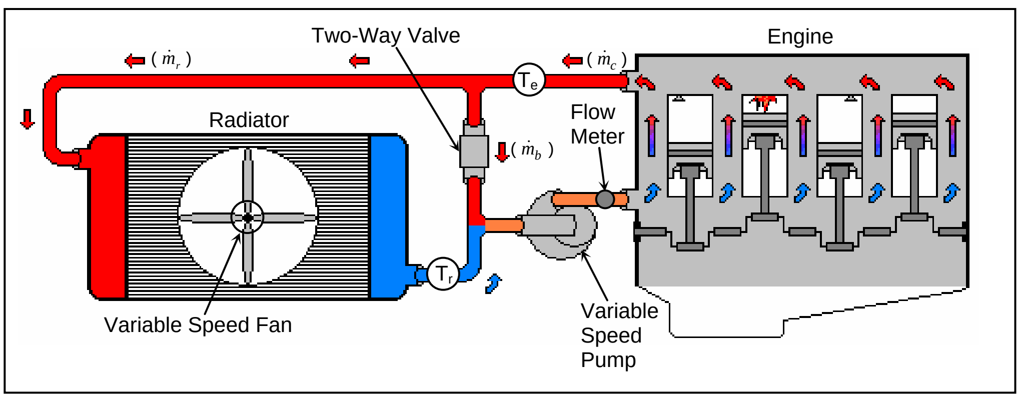

Figure 3: Advanced thermal management system with two-way valve configuration which emphases design simplicity while providing non-precise fluid flow control between radiator and bypass loops (Case 2).

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. Therefore, a two-way valve would be specific to a particular cooling system and would most likely not be interchangeable between vehicles. 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, completely preventing flow from entering the radiator (when fully closed) to aid in engine warm-up times. However, a pressure drop has been 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 as shown in Figure 4. 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 (Chalgren, 2004). 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.

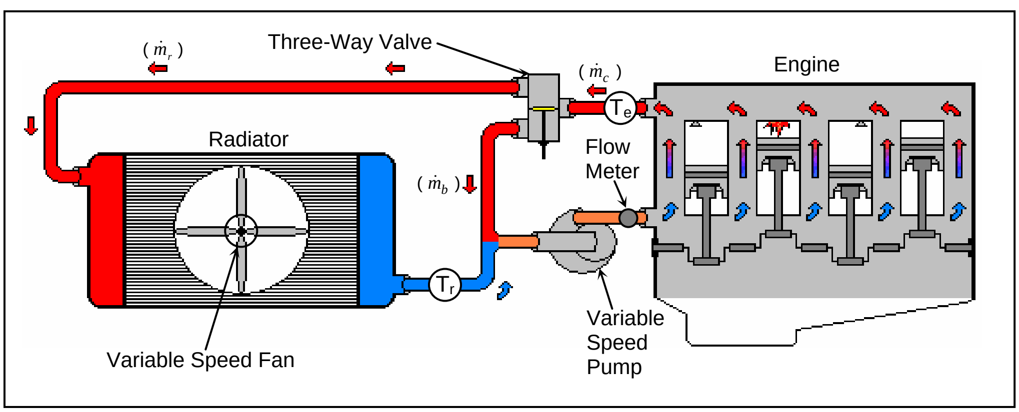

Figure 4: Advanced thermal management system with three-way valve configuration which offers precise fluid flow regulation (Case 3).

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 roll 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. That is, when the engine is cold, coolant is routed through the bypass via valve position to reduce warm-up times. However, the valve could potentially be eliminated if the pump circulates coolant as required by the engine (refer to Figure 5). Note that coolant must be circulated at all times since hot spots may develop, leading to engine damage.

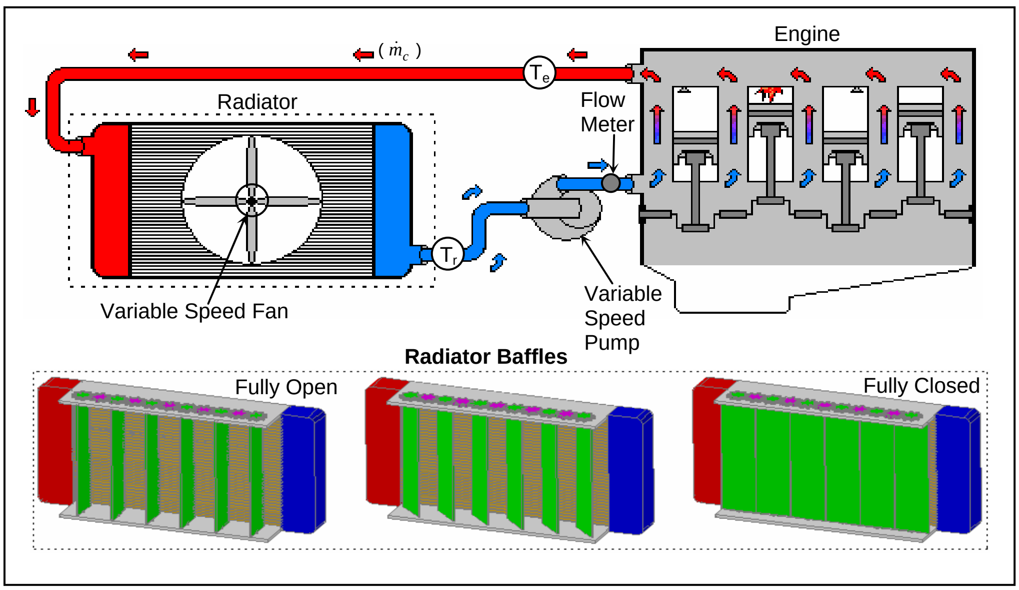

Figure 5: The thermostat valve is removed which eliminates the need for a bypass; temperature control achieved by the coolant pump and radiator fan. Note that the radiator baffles range from fully open to fully closed (Cases 4 and 5).

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. Overall, this configuration simplifies the cooling system by eliminating the thermostat valve.

A further improvement of warm-up times, without a thermostat valve, may be achieved with servo-motor driven radiator baffles. Radiator baffles control ram-air effects acting on the radiator (due to the vehicle speed). In essence, the baffles serve as a valve for external air flow. In warm-up conditions, the baffles would be closed to block airflow across the radiator and minimize the amount of heat rejected. Once the engine has reached its operating temperature, the baffles may be opened and the radiator functions normally. The coolant pump operation would be similar to the no baffle case.

3. THERMAL MODELS AND OPERATING STRATEGY

A reduced order lumped parameter thermal model may be used to describe the transient response of the engine thermal management system. The thermal dynamics for the engine and radiator nodes, $T_e(t)$ and $T_r(t)$, in Figure 1 may be written as (Setlur *et al.*, 2005)

$$C_e\dot{T}_e = Q_{in} - c_{pc}\dot{m}_r(T_e - T_r), \qquad C_r\dot{T}_r = c_{pc}\dot{m}_r(T_e - T_r) - \varepsilon c_{pa}\dot{m}_a(T_e - T_\infty) - Q_o. \qquad (1)$$

In the two-way valve configuration (refer Figure 3), a flow rate exists through the radiator branch, $\dot{m}_r(t)$, at all times so that $\dot{m}_r = (1-\varepsilon)H\dot{m}_c + \varepsilon\dot{m}_c$. The coolant mass flow rate through the bypass branch, $\dot{m}_b(t)$, becomes $\dot{m}_b = (1-\varepsilon)(1-H)\dot{m}_c$. Note that the parameter $\varepsilon(\Delta p)$ depends on the pressure drop, $\Delta p(t)$, across the radiator and bypass branches. The variable $H(x)$ represents the normalized valve position which is dependant on the actual valve position, $x(t)$. Finally, the overall coolant mass flow rate is $\dot{m}_c = \dot{m}_r + \dot{m}_b$.

The cooling circuit dynamic behavior varies slightly when a three-way valve is introduced as shown in Figure 4. 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, $\dot{m}_r(t)$, for a given water pump speed. In this case, the flow rates through the radiator and bypass branches become $\dot{m}_r = H\dot{m}_c$ and $\dot{m}_b = \left(1-H\right)\dot{m}_c$, respectively. If the valve and bypass are completely removed from the cooling system (refer to Figure 5), then the flow rate through the radiator branch and water pump will be equivalent, $\dot{m}_r = \dot{m}_c$.

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, $H\left(x\right)$, will respond in a linear manner to the coolant temperature so that (Zou *et al.,* 1999)

$$H = \begin{cases} 0; & T_e < T_l \quad (bypass\ only) \\ \dfrac{T_e - T_l}{T_h - T_l}; & T_l \le T_e \le T_h \\ 1; & T_e > T_h \quad (radiator\ only) \end{cases} \tag{2}$$

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 has been neglected. For on/off (or bang-bang) valve control (Cases 2 and 3), the control authority is

$$H = \begin{cases} 0; & T_e < T_{ed} - \Delta T \quad (bypass\ only) \\ 1; & T_e \ge T_{ed} - \Delta T \quad (radiator\ only) \end{cases} \tag{3}$$

where ΔT is the boundary layer about the desired engine temperature, $T_{ed}(t)$. The boundary layer was introduced to reduce valve dithering. Note in equations (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 valve.

The main purpose of the engine's thermal management system is to maintain a desired engine block temperature, $T_{ed}(t)$, while accommodating the un-measurable combustion process heat input, $Q_{in}(t)$ and the uncontrollable air flow heat loss across the radiator, $Q_o(t)$. To achieve this goal, a Lyapunov-based nonlinear controller has been developed so that the engine's coolant temperature, $T_e(t)$, tracks the desired temperature, $T_{ed}(t)$, by regulating the system actuators (variable speed electric water pump and radiator fan) in harmony with each other. Note that in equation (1), the signals $T_e(t)$, $T_r(t)$ and $T_\infty(t)$ are measured by thermocouples (or thermistors). The system parameters c_{pc}, c_{pa}, C_e, C_r, and ε are assumed 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 $Q_{in}(t)$ and $Q_o(t)$.

To formulate the control law, the thermal system dynamics described in equation (1) can be rewritten as

$$C_e \dot{T}_e = Q_{in} - u_e, \qquad C_r \dot{T}_r = u_e - u_r - Q_o \tag{4}$$

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

$$u_e \triangleq c_{pc} \dot{m}_r (T_e - T_r), \qquad u_r \triangleq \varepsilon c_{pa} \dot{m}_a (T_e - T_\infty). \tag{5}$$

A Lyapunov based nonlinear controller can be developed and applied to regulate the engine temperature (similar to Setler *et al.*, 2005) so that the control law (which establishes a basis to determine the pump and fan speeds) is designed as

$$u_e = -\left(K + \alpha\right)\left[e - e_o\right] - \int_{t_o}^{t} \left[\alpha\left(K + \alpha\right)e(\tau) + \rho \operatorname{sgn}(e(\tau))\right] d\tau. \tag{6}$$

In this expression, the final term, $\rho \operatorname{sgn}(e)$, compensates for the variable un-measurable input heat, $Q_{in}(t)$. The error, $e(t)$, is the difference between the desired and actual engine temperatures, $T_{ed}(t) - T_e(t)$. Finally, the variable, e_o, is the initial temperature error.

The radiator's mathematical description in equation (1) states that it operates normally (i.e., as a heat exchanger) if the effort of the radiator fan, denoted by $u_r(t)$ in equation (4), is set equal to the effort produced by the water pump, denoted by $u_e(t)$. Therefore, the control input $u_e(t)$ provides the signal $\dot{m}_r(t)$ and the control input $u_r(t) = u_e(t)$ provides the signal $\dot{m}_a(t)$ as shown in equation (5). The signal $\dot{m}_r(t)$ is uni-polar, so a commutation strategy determines the radiator coolant mass flow rate as

$$\dot{m}_r \triangleq \frac{u_e\left[1 + \operatorname{sgn}(u_e)\right]}{2c_{pc}\left(T_e - T_r\right)}. \tag{7}$$

The coolant mass flow rate, $\dot{m}_c(t)$, or pump effort, is now determined using equation (7) and the valve configuration with its normalized position, H. For Cases 2 and 3, the coolant flow rates become $\dot{m}_c = \dfrac{\dot{m}_r}{H - \varepsilon(H+1)}$ and $\dot{m}_c = \dfrac{\dot{m}_r}{H}$, respectively. If a valve does not exist for Cases 4 and 5, then $\dot{m}_c = \dot{m}_r$. Note that the coolant pump command voltage is determined by an *a priori* empirical relationship (e.g., Chastain and Wagner, 2006). From equation (7), if $u_e(t)$ is bounded for all time, then $\dot{m}_r(t)$ is bounded for all time.

A second commutation strategy is proposed to compute the uni-polar control input $\dot{m}_a(t)$ so that

$$\dot{m}_a \triangleq \frac{u_r\left[1 + \operatorname{sgn}(u_r)\right]}{2\varepsilon c_{pa}\left(T_e - T_\infty\right)}. \tag{8}$$

As stated earlier, $u_r(t) = u_e(t)$. 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 equations (1) and (4). Again, an *a priori* empirical relationship determined the fan motor voltage. From this definition, if $u_r(t)$ is bounded for all time, then $\dot{m}_a(t)$ is bounded for all time. Note that equation (5) is utilized to develop equations (7) and (8). For further details, the reader is referred to Salah *et al.* (2007).

4. THERMAL TEST BENCH

An experimental test bench was created to investigate the thermostat valve configurations (Cases 1-5) shown in Figure 1. This custom bench offered maximum flexibility and a repeatable environment (refer to Figure 6). 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 shown in Figure 6. 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 Section 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, $H = 1$.

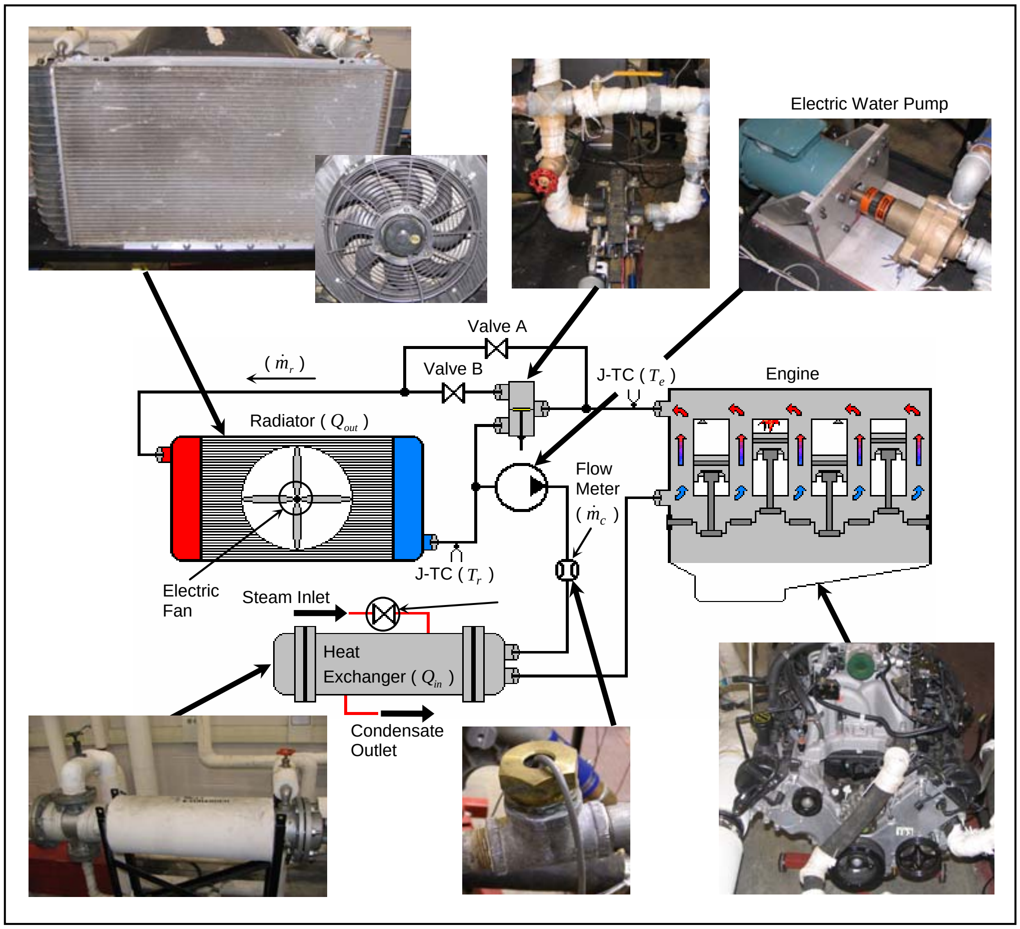

Figure 6: Schematic of thermal test bench with actual cooling system components, engine block, sensors, and steam heat exchanger.

To calculate the rate of heat transfer, $Q_{in}(t)$, condensed steam was collected and measured. It has been assumed that the amount of condenser condensate is proportional to the amount of heat transferred to the circulating coolant (Incropera and DeWitt, 1990). The inlet steam control valve regulates the flow of incoming steam to the condenser which in turn regulates heat transfer. Overall, heat transfer rates of up to 60 kW can be achieved with the current steam heat exchanger. Two J type Omega thermocouples, with Omega OM5-LTC signal conditioners, measure the coolant temperatures at the engine, $T_e(t)$, and radiator, $T_r(t)$, outlets. The coolant mass flow, $\dot{m}_c(t)$, was determined using an Omega FP7001A paddle-wheel mass flow meter placed after the pump.

Data acquisition was performed by a Servo-to-Go controller board which utilizes eight digital-to-analog (DA) outputs and eight analog-to-digital (AD) inputs. This controller board provides signals for the smart valve, variable speed electric pump, and radiator fan. The custom steel body smart valve, with Teflon filled Delrin (Delrin AF) piston, was linearly actuated by a Litton servo-motor driven worm gear. The pump was a standard brass centrifugal pedestal pump (3.2cm outlet and 3.8cm inlet) driven by a Reliance Electric 240VAC three phase electric motor. The electric radiator fan (Summit SUM-G4904) has a diameter d=45.7cm and can handle flow rates up to 850 liters/sec.

5. EXPERIMENTAL RESULTS AND DISCUSSION

Five different valve and radiator baffle configurations were investigated on the steam test bench using the proposed control strategies in Section 3 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. The warm-up time, t_{wu}, is the time required for the engine temperature, $T_e(t)$, to reach its desired set point, $T_{ed}(t)$. The absolute steady-state temperature error, $|E_{ss}|$, represents the difference between $T_e(t)$ and $T_{ed}(t)$ at steady-state operation. The temperature overshoot, O_{sh}, denotes the difference between the engine temperature, $T_e(t)$, at its peak value and the desired engine temperature, $T_{ed}(t)$. The total power consumption, P_{total}, is the average power consumed by the water pump and radiator fan during the test ($0 < t < 40$ minutes). Note that the power consumed by the valve is negligible and has been ignored. All the tests began at $T_e(0) = 305$ ˚K with $T_{ed}(t) = 363$˚K. To simulate a vehicle driving at a constant speed and load, the input heat has been selected as $Q_{in}(t) = 35kW$. Further, the wind speed associated with $Q_o(t)$ was approximately 100 km/hr. The control gains, selected through tuning, were $K = 23$, $\alpha = 1.0e\text{-}04$, and $\rho = 0.1$.

Case	Configuration	Valve Operation	Electric Pump	Electric Fan
1	Traditional factory	T_l=358˚K, T_h=368˚K per (2)	Fixed, half speed	Fixed, full speed
2	Two-way valve	T_{ed}=363˚K, ΔT=0.5˚K per (3)	Variable speed per (6), (7)	Variable speed per (6), (8)
3	Three-way valve	T_{ed}=363˚K, ΔT=0.5˚K per (3)		
4	No valve	n/a		
5	No valve with baffles	n/a		

Table 1: Summary of the valve configuration tests, components, and operating modes.

5.1 Factory Configuration (Case 1)

In the factory cooling configuration, the engine and radiator temperature responses are shown in Figure 7a with no apparent temperature overshoot since the set point, $T_{ed}(t)$, was not achieved. A steady-state temperature offset of $|E_{ss}| = 0.777$ ˚K was observed. Constant radiator air flow, $\dot{m}_a \cong 1.53$ kg/sec, corresponded to fixed engine speed and clutch operation as displayed in Figure 7b. The water pump flow rate was maintained at approximately $\dot{m}_c \cong 1.50$ kg/sec to emulate constant engine speed (refer to Figure 7c). Operation of the thermostat valve was controlled by equation (2) with T_l=358˚K and T_h=368˚K; the normalized valve position is displayed in Figure 7d. The valve started opening at $t \cong 296$ seconds with steady-state operation achieved at approximately $t \cong 2000$ seconds. Coolant flow rate through the entire cooling circuit decreased when the valve opened and reflects different pressure drops between the radiator and bypass loops.

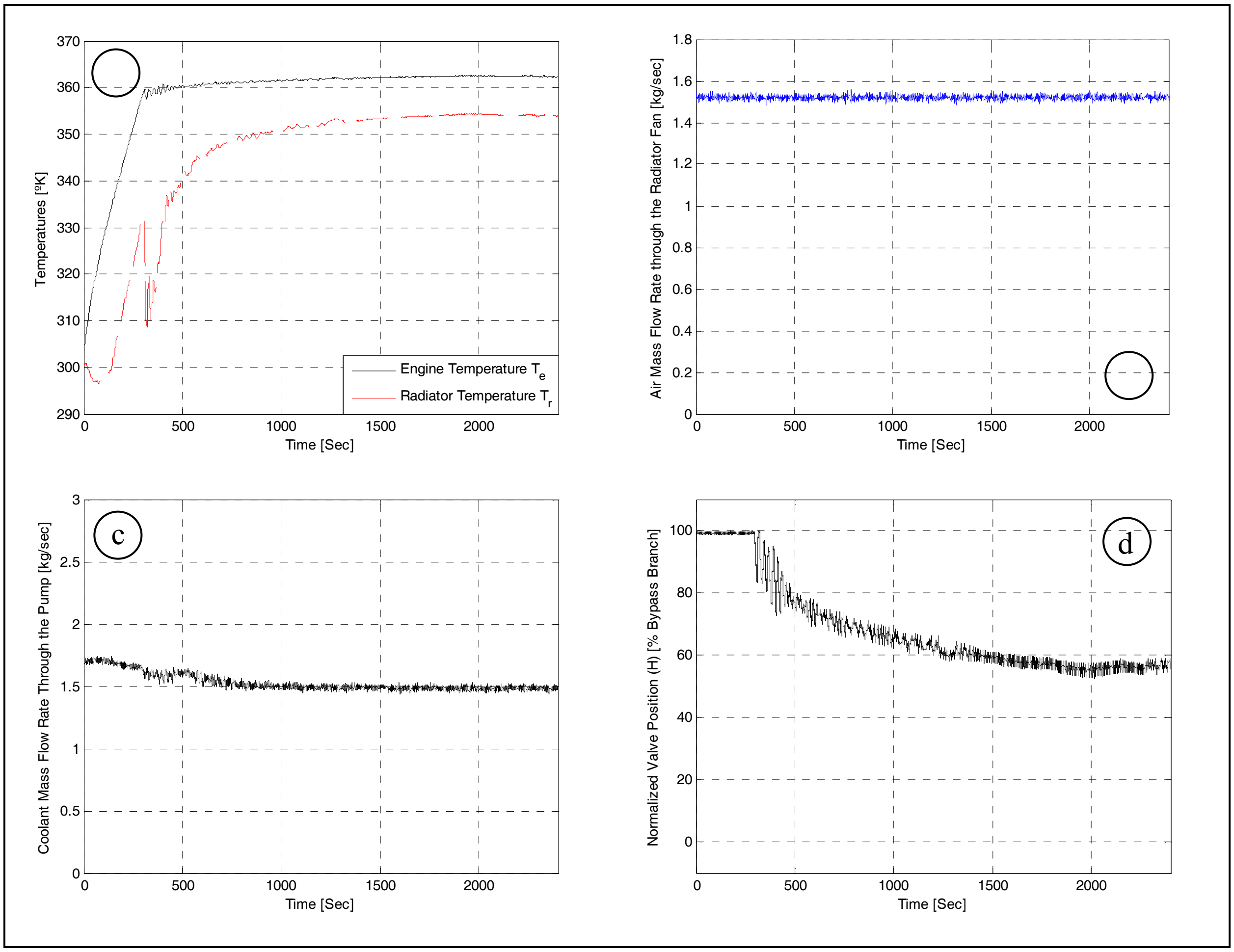

Figure 7: Case 1 - Factory configuration with (a) Engine and radiator temperatures for a desired engine temperature $T_{ed}(t)$; (b) Air mass flow rate through the radiator fan; (c) Coolant mass flow rate through the pump; and (d) Normalized valve position.

5.2 Two-Way Valve Configuration (Case 2)

The two-way valve configuration located the smart valve in the bypass loop as shown in Figure 3. Transient response of the engine and radiator (refer to Figure 8a) display slower temperature dynamics when compared to the factory setting in Figure 7a (e.g., $t \cong 380$ versus $t \cong 270$ seconds for the engine temperature to reach $T_e = 350\,^{\circ}$K). The steady-state temperature error between the engine and prescribed value is $|E_{ss}| = 0.245\,^{\circ}$K. This thermal behavior may be explained by the lack of control over the fluid flow through the radiator. In Figure 8d, the valve changes position only once at $t \cong 548$ seconds, which is attributed to the engine temperature,

$T_e(t)$, remaining above the desired temperature T_{ed} =363˚K when reached. In Figures 8b and 8c, the valve position change (routing from the bypass to the radiator) produces oscillations in the air and coolant mass flow rates. These fluctuations may be attributed to the high gain control of the pump and fan.

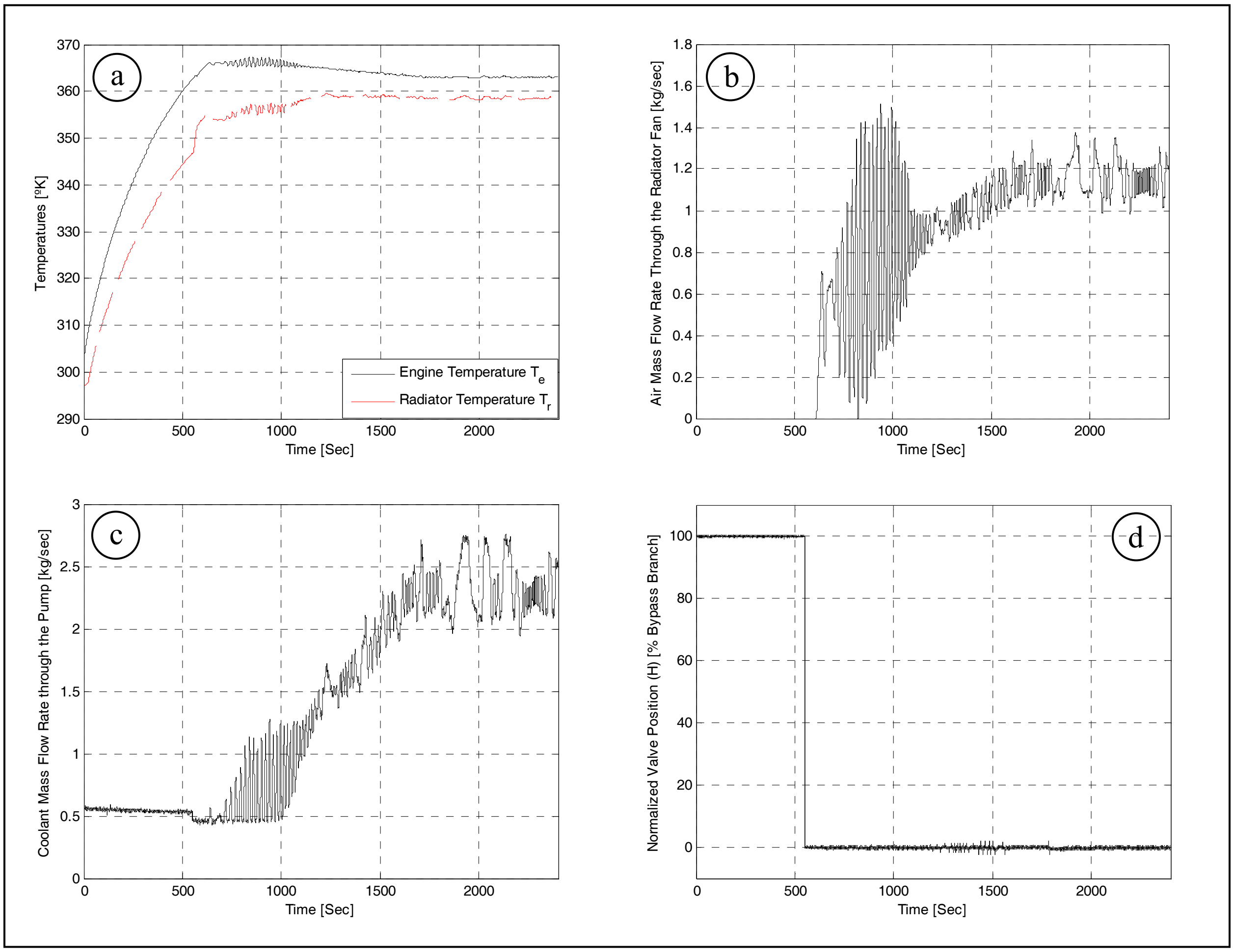

Figure 8: Case 2 - Two-way valve configuration with (a) Engine and radiator temperatures for a desired engine temperature $T_{ed}(t)$; (b) Air mass flow rate through the radiator fan; (c) Coolant mass flow rate through the water pump; and (d) Normalized valve position.

5.3 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 (refer to Figure 4). In Figure 9a, the thermal response of the engine and radiator coolant temperatures is displayed. The three-way valve switches between the bypass and radiator at $t \cong 359$, $t \cong 428$, and $t \cong 471$ seconds (refer to Figure 9d) to realize a short warm-up time of $t_{wu} = 363.9$ seconds. A minimum steady-state error was demonstrated with $\left|E_{ss}\right| = 0.175\ ^{\circ}\mathrm{K}$ which may be attributed to the improved coolant flow control associated with the three-way valve as discussed in Section 2.3. The fan and coolant mass flow rates (refer to Figures 9b and 9c) 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.

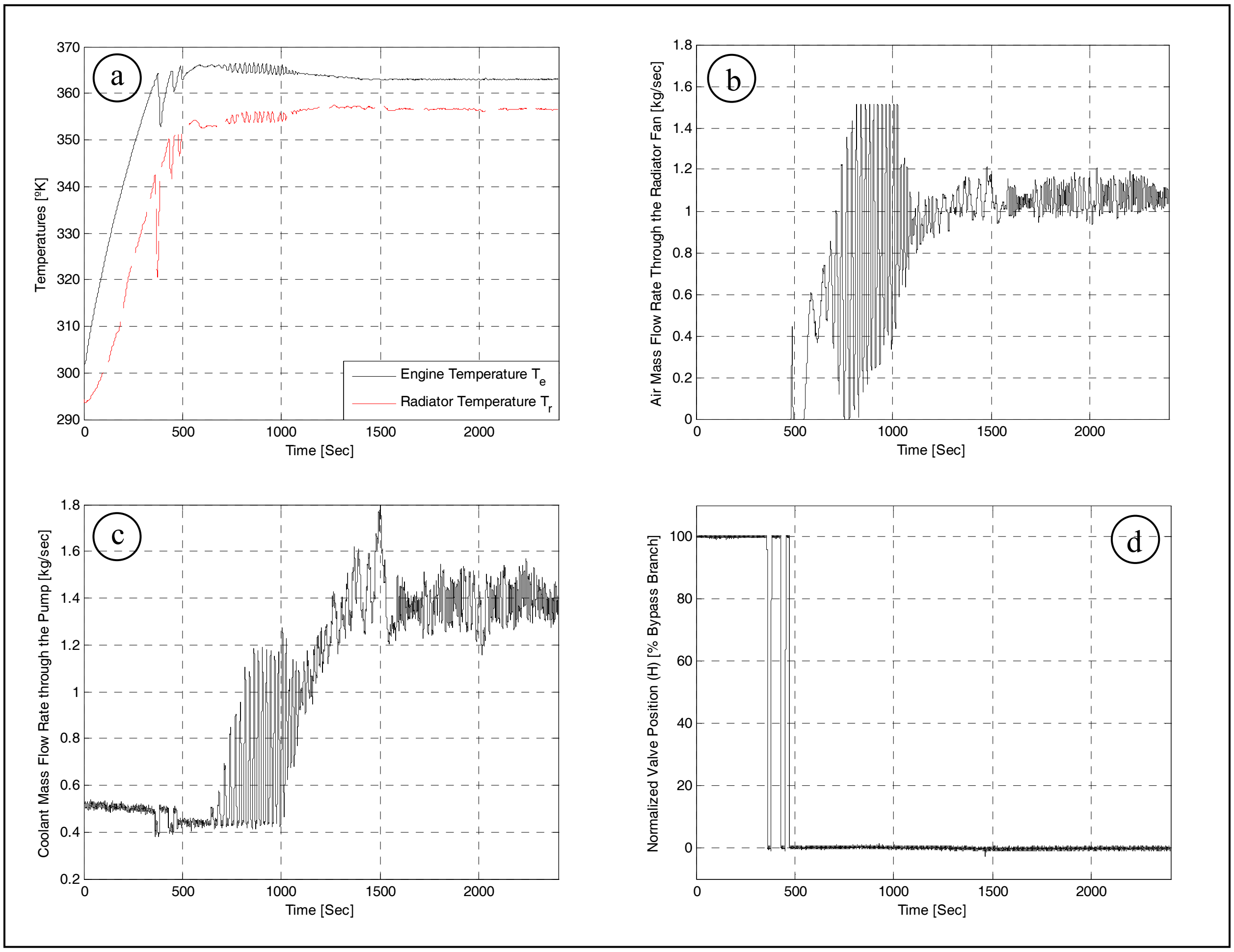

Figure 9: Case 3 - Three-way valve configuration with (a) Engine and radiator temperatures for a desired engine temperature $T_{ed}(t)$; (b) Air mass flow rate through the radiator fan; (c) Coolant mass flow rate through the water pump; and (d) Normalized valve position.

Finally, a correlation exists between the valve's initial position change and the temperature profile which indicates that the restricted radiator flow allowed the radiator's fluid volume to be cooled and waiting to be released into the engine block to quickly lower the engine temperature.

5.4 Valve Absent Configuration (Cases 4 and 5)

The elimination of the bypass and thermostat valve results in a simplified, single loop cooling circuit composed of the engine block, radiator fan, and coolant pump (refer to Figure 5). In Figure 10a, the temperature profiles demonstrate that the warm-up time, $t_{wu} \cong 594.1$ seconds,

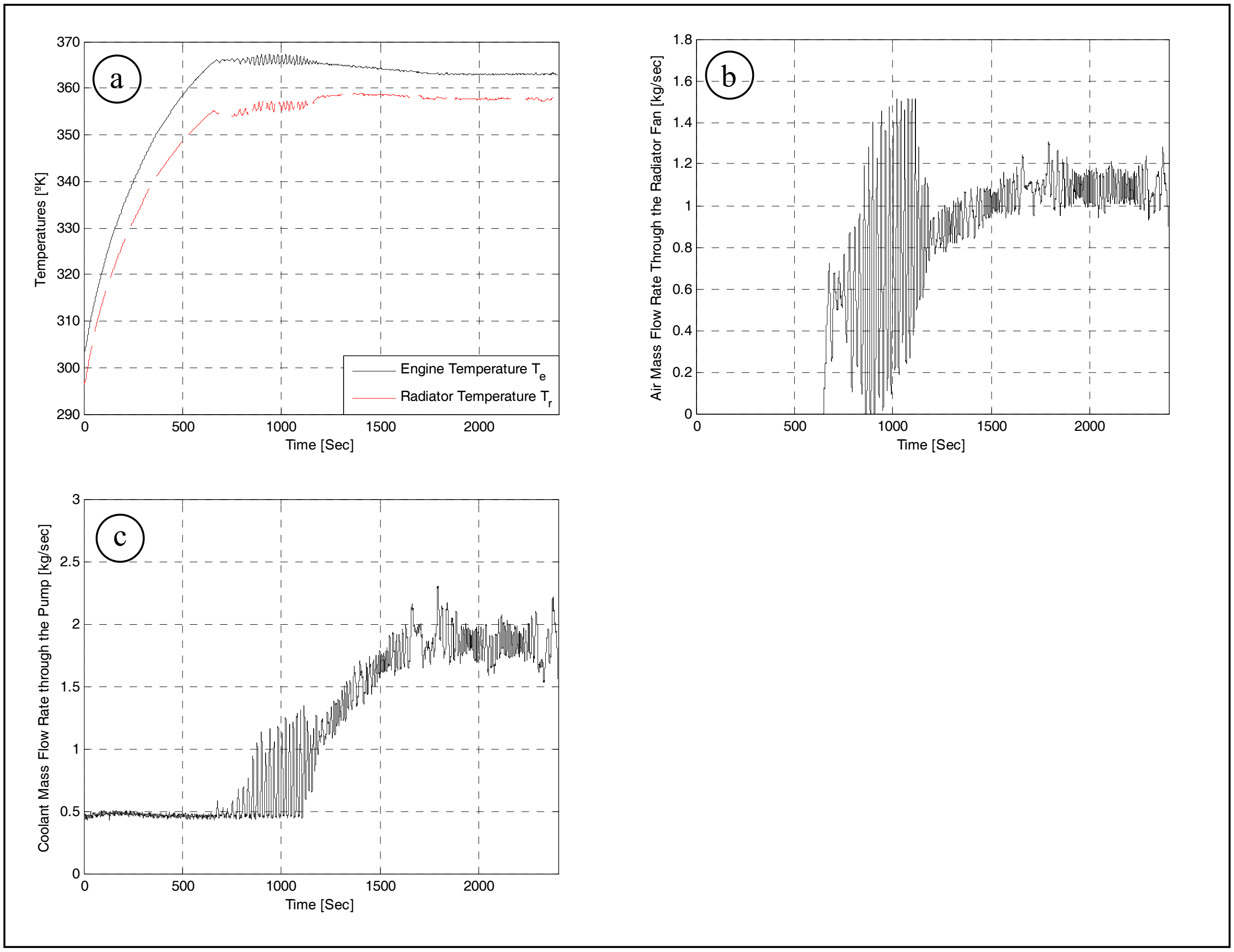

Figure 10: Case 4 - Valve absent configuration with (a) Engine and radiator temperatures for a desired engine temperature $T_{ed}(t)$; (b) Air mass flow rat through the radiator; and (c) Coolant mass flow rate through the pump.

is larger when compared to the three-way valve configuration (Cases 1, $t_{wu} \cong 363.9$ seconds) but very similar to the two-way valve architecture (Case 2, $t_{wu} \cong 558.0$ seconds). Note that the steady-state temperature error is $|E_{ss}| = 0.276$ ˚K. The radiator fan and water pump responses are shown in Figures 10b and 10c. Once again the signal oscillations may be attributed to the high control gains for the pump and radiator fan.

In an attempt to minimize the warm-up time with this configuration, baffles blocked the air flow through the radiator until the engine coolant temperature reached the desired temperature. As shown in Figure 11a, the warm-up time, $t_{wu} \cong 382.9$ seconds, was significantly

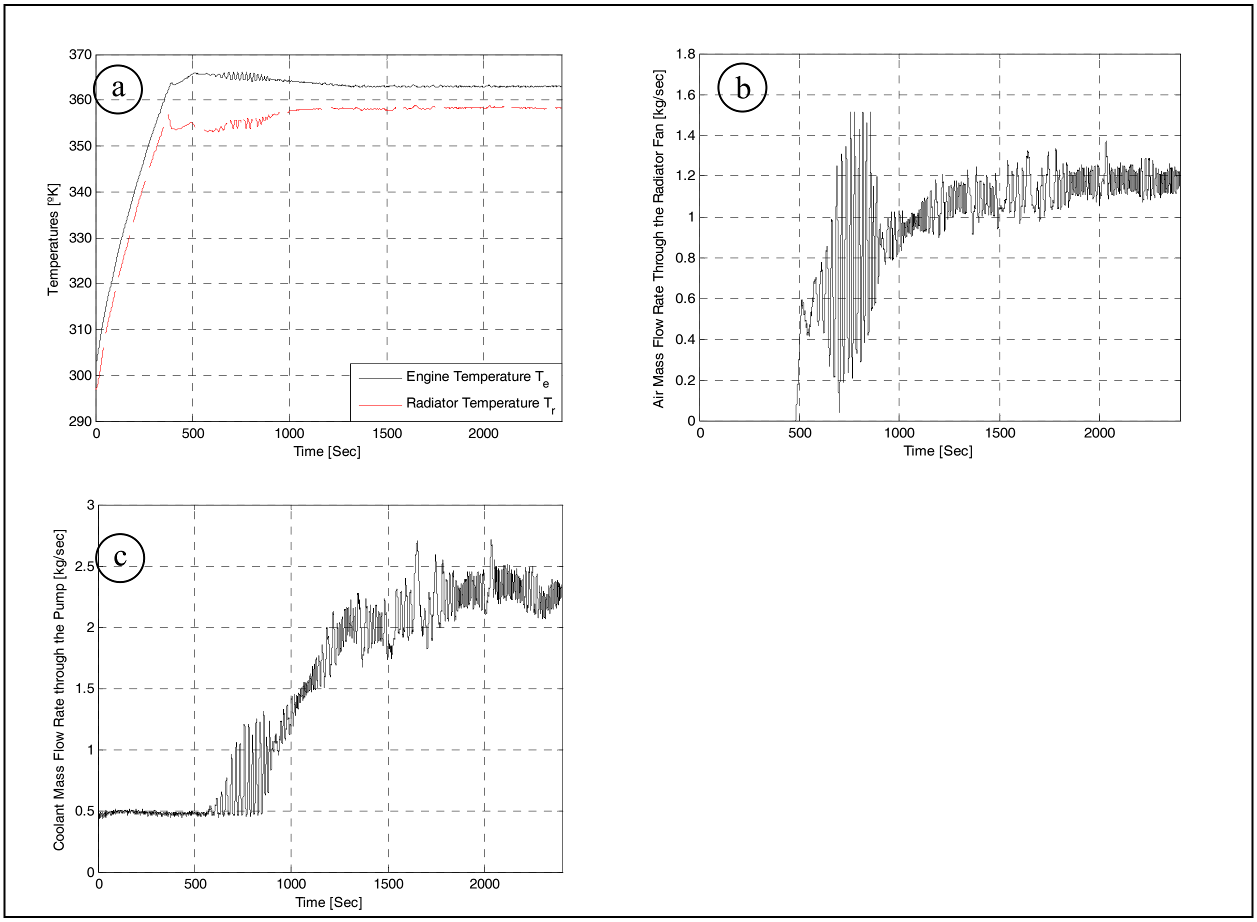

Figure 11: Case 5 - Valve absent and baffles present configuration with (a) Engine and radiator temperatures for a desired engine temperature $T_{ed}(t)$; (b) Air mass flow rate through the radiator; and (c) Coolant mass flow rate through the pump.

improved by a 35.5% decrease in comparison to Case 4. However, the integration of servo-actuated baffles into the radiator would add to system complexity and power consumption. Note that these experimental results do not reflect the power required to drive the radiator baffles so the power consumption value reported in Table 2 should be slightly larger. For this test, the baffles were removed at $t \cong 406$ seconds as noted by the decrease in radiator temperature due to the enhanced cooling process via available ram air flow.

5.5 Configuration Performance Comparison

The experimental data presented in Figures 7 through 11 has been summarized in Table 2. 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. 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 completely 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 baffles 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. It is interesting to observe that the two-way valve configuration warm-up time, $t_{wu} = 558.0$ seconds, is similar to the valve absent, $t_{wu} = 594.1$ seconds. 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 configuration ($t_{wu} = 363.9$ versus $t_{wu} = 382.9$ seconds).

***Observation 2**: To minimize temperature tracking error, precise fluid control must be maintained by the thermostat valve, coolant pump, and/or radiator fan.*

The temperature tracking error is an important metric when discussing advanced thermal management systems. Engine coolant temperature should be controlled as close as possible to the set point temperature to facilitate combustion efficiency. The steady-state temperature error, $\left|E_{ss}\right|$, ranking can be stated as Case 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 to 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 an advanced thermal management 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 architecture (Case 4) without the associated increase in warm-up time and error measure.*

The power measure $P_{pump} = \frac{1}{\Delta t}\int_{t_o}^{t}\left[\frac{1}{2\rho_c^2 A_c^2}\dot{m}_c^3(\tau)\right]d\tau$ and $P_{fan} = \frac{1}{\Delta t}\int_{t_o}^{t}\left[\frac{1}{2\rho_a^2 A_f^2}\dot{m}_a^3(\tau)\right]d\tau$ calculates the average power consumed by the water pump and radiator fan, respectively, over the time period $\Delta t = t - t_o = 40$ minutes. The total average power consumption has been reported in column 6 of Table 2. As stated earlier, power consumed by the valve is minimal and has been neglected. Observing the power consumption values, the three-way configuration (Case 3) consumed the least power, $P_{total} = 24.31$ W, followed closely by the valve absent configuration (Case 4) at $P_{total} = 24.72$ W. However, the trade-off 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, $P_{total} = 109.37$ W, due to the constant pump and fan operation. In the valve absent with baffles configuration (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. Extra hardware is eliminated by removing the valve which reduces cost and engineering time associated with its development. However, this architecture comes with the penalty of increased warm-up times, and therefore, results in the valve absent configuration being ranked third. The addition of baffles helped to improve the warm-up time when the valve was removed (Case 5) resulting in the configuration being ranked second. For maximum control and flexibility, the three-way valve configuration (Case 3) was ideal and ranked the highest in Table 2. During testing, it performed very well by providing low warm-up times and tracking error while consuming little power compared to the other cases. The two-way valve configuration (Case 2) may be labeled obsolete when compared to the three-way valve's performance and carries the same relative cost and time for integration into the cooling system.

Further, the two-way valve configuration warm-up time was very similar to the valve absent configuration, leading to the fifth rank. The fact that the factory configuration (Case 1) had a favorable relative warm-up time compared to the other cases at the critical cost of total power consumption and steady-state temperature error resulted in the fourth rank.

Case	Configuration	t_{wu} [sec]	$\lvert E_{ss} \rvert$ [˚K]	O_{sh} [˚K]	P_{total} [W]	Rank
1	Traditional factory	n/a [1]	0.777	n/a	109.37	4
2	Two-way valve	558.0	0.245	4.3	33.14	5
3	Three-way valve	363.9	0.175	3.6	24.31	1
4	No valve	594.1	0.276	4.3	24.72	3
5	No valve with baffles	382.9	0.254	3.2	36.56	2

Table 2: Summary of the valve configuration tests 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.

6. SUMMARY

Thermostat valve configuration is an important topic in advanced thermal management systems as it pertains to engine coolant temperature warm-up. Four valve configurations were examined and tested for effectiveness: factory, two-way valve, three-way valve, and valve absent. Summarizing the findings, the three-way valve configuration provides excellent temperature tracking, power consumption, and warm-up time when compared to 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 form the cooling system. Finally, a tradeoff 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. Overall, the results and observations made in Section 5 support the hypothesis formulated in Section 1.

[1] The engine coolant temperature never reached the desired value of T_{ed} = 363˚K.

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APPENDIX: Nomenclature List

A_c	pump outlet cross section area [m^2]
A_f	fan blowing area [m^2]
C_e	engine block capacity [kJ/ºK]
c_{pc}	coolant specific heat [kJ/kg.ºK]
c_{pa}	air specific heat [kJ/kg.ºK]
C_r	radiator capacity [kJ/ºK]
d	diameter [cm]
e	engine temperature tracking error [ºK]
e_o	initial engine temperature tracking error [ºK]
E_{ss}	steady state error [ºK]
H	normalized valve position [%]
K	control gain
$\dot{m}_a$	fan air mass flow rate [kg/sec]
$\dot{m}_b$	bypass coolant mass flow rate [kg/sec]
$\dot{m}_c$	pump coolant mass flow rate [kg/sec]
$\dot{m}_r$	radiator coolant mass flow rate [kg/sec]
O_{sh}	temperature overshoot [ºK]
P_{fan}	fan power [kW]
P_{pump}	pump power [kW]
P_{total}	total power [kW]
Δp	pressure drop accros the radiator [bar]
Q_{in}	combustion process heat [kW]
Q_o	uncontrollable air flow heat loss [kW]
sgn	standard signum function
T_e	engine outlet coolant temperature [ºK]
T_{ed}	desired engine coolant temperature [ºK]
T_h	liquid wax temperature [ºK]
T_l	wax softening temperature [ºK]
T_r	radiator outlet coolant temperature [ºK]
T_∞	surrounding ambient air temperature [ºK]
t	test time [sec]
t_{wu}	warm-up time [sec]
u_e	control input
u_r	control input
x	valve position [cm]
α	control gain
ρ	control gain
ρ_a	air density [kg/m^3]
ρ_c	coolant density [kg/m^3]
ε	valve constant
τ	constant of integration
Δt	sample time [sec]
ΔT	valve operating temperature deviation [ºK]