Control of spray evaporative cooling in automotive IC engines

Article (Accepted Version)

Jafari, S, Dunne, J, Langari, M, Yang, Z, Pirault, J, Long, C and Thalackottore Jose, J (2018) Control of spray evaporative cooling in automotive IC engines. *Journal of Thermal Science and Engineering Applications*, 10 (4). pp. 1-11. ISSN 1948-5085

This version is available from Sussex Research Online: <http://sro.sussex.ac.uk/id/eprint/73266/>

This document is made available in accordance with publisher policies and may differ from the published version or from the version of record. If you wish to cite this item you are advised to consult the publisher's version. Please see the URL above for details on accessing the published version.

Copyright and reuse:

Sussex Research Online is a digital repository of the research output of the University.

Copyright and all moral rights to the version of the paper presented here belong to the individual author(s) and/or other copyright owners. To the extent reasonable and practicable, the material made available in SRO has been checked for eligibility before being made available.

Copies of full text items generally can be reproduced, displayed or performed and given to third parties in any format or medium for personal research or study, educational, or not-for-profit purposes without prior permission or charge, provided that the authors, title and full bibliographic details are credited, a hyperlink and/or URL is given for the original metadata page and the content is not changed in any way.

ASME Accepted Manuscript Repository

Institutional Repository Cover Sheet

First

Last

ASME Paper Title: CONTROL OF SPRAY EVAPORATIVE COOLING IN AUTOMOTIVE IC ENGINES

Authors: S. Jafari, J. F. Dunne, M. Langari, Z. Yang J-P Pirault, C. A. Long, and J. Thalackottore Jose

ASME Journal Title: Journal of Thermal Science and Engineering Applications

Volume/Issue 10/4 Date of Publication (VOR* Online) 7th May 2018

ASME Digital Collection URL: <https://doi.org/10.1115/1.4039701>

DOI: 10.1115/1.4039701

*VOR (version of record)

CONTROL OF SPRAY EVAPORATIVE COOLING IN AUTOMOTIVE IC ENGINES

by

S. Jafari, J. F. Dunne*, M. Langari, Z. Yang\$
J-P Pirault, C. A. Long, and J. Thalackottore Jose

Department of Engineering and Design
School of Engineering and Informatics
University of Sussex, Falmer, Brighton, BN1 9QT, UK.

\$Department of Engineering, College of Engineering
and Technology, University of Derby, Derby DE22 3AW, UK.

* Corresponding author: E-mail: j.f.dunne@sussex.ac.uk

Abstract

A novel approach is proposed for precise control of two-phase spray evaporative cooling for thermal management of road vehicle internal combustion engines. A reduced-order plant model is first constructed by combining published spray evaporative cooling correlations with approximate governing heat transfer equations appropriate for IC engine thermal management. Control requirements are specified to allow several objectives to be met simultaneously under different load conditions. A control system is proposed and modelled in abstract form to achieve spray evaporative cooling of a gasoline engine, with simplifying assumptions made about the characteristics of the coolant pump, spray nozzle, and condenser. The system effectiveness is tested by simulation to establish its ability to meet key requirements, particularly concerned with precision control during transients resulting from rapid engine load variation. The results confirm the robustness of the proposed control strategy in accurately tracking a specified temperature profile at various constant load conditions, and also in the presence of realistic transient load variation.

Keywords: engine cooling, road vehicles, evaporative sprays, precision control, robustness.

17 main-section pages (double spaced) 58 references Figures 1 – 13 No appendices

Nomenclature

A	area
Bo	boiling number (defined in Equation 3)
C	specific heat capacity
D	diameter
d_{32}	Sauter mean diameter
h_{fg}	specific enthalpy of vaporisation
H	nozzle exit to wall distance
Ja	Jakob number (defined in Equation 5)
k	thermal conductivity
K	three-way mixing valve constant
m	mass
p	pressure
q	heat flux
Q	heat flow
t	time
T	temperature
U	velocity
w	plate thickness
We	Weber number (defined in Equation 4)

α , β and γ constants in Equation 2

Δp	pressure difference
ΔT	temperature difference
μ	dynamic viscosity
ρ	density
σ	surface tension

Subscripts

c	pertaining to the coolant
L, v	pertaining to the liquid and vapour phases, respectively
in, out	pertaining to inlet and outlet conditions, respectively
o	pertaining to the nozzle exit
w,c, w,g	pertaining to wall coolant side and gas side, respectively

1. INTRODUCTION

The development of new cooling strategies across a range of different application areas has resulted in a high degree of functionality for both component hardware and cooling systems. This includes applications within the automotive sector, aerospace, electronics, and the nuclear industry. In the electronics industry for example, the ever-demanding requirements of the 'lab-on-a-chip', has made thermal management a major challenge. And of particular relevance in the automotive sector, thermal management of 'highly-boosted' internal combustion engines is placing a serious limit on the ability to improve engine efficiency by 'downsizing'. Temperature control of complex systems in these sectors plays a vital role in achieving consistent system performance. This may mean either tracking a prescribed temperature profile or maintaining values within very carefully specified thermal limits. And although further advances in cooling technology may overcome the current limitations of existing cooling methods, precise thermal management may not be possible with existing control strategies. The focus of this paper is to find an appropriate control strategy to realise the full benefits of spray evaporative cooling for combustion engines that will mainly be used for automotive vehicle propulsion in the light-duty sector.

The design of any cooling system requires careful consideration of several interrelated factors: e.g. the maximum permissible heat flux and temperature, thermally-induced stresses, tolerances, reliability, parasitic energy consumption (of the cooling system), and the operating environment. As shown in figure 1 [1], spray evaporative cooling offers great potential for heat removal because the heat transfer coefficient is an order of magnitude greater than in nucleate boiling, and two orders of magnitude greater than single phase forced-convection with water. This enhancement in heat transfer over more-conventional cooling systems is a result of the complete evaporation of small (sub-millimetre diameter) droplets which impinge onto the heated surface.

The first experimental evidence of spray cooling on solid surfaces was reported in 1966 [2][3] - the result of carefully examining the spray cooling mechanism and the associated boiling curves generated in [4]. The first empirical spray cooling correlations were obtained 4 years later [5-7]. But it took another 20 years to understand the principal concepts involved and to derive correlations for single (liquid) phase spray cooling. The milestones for this (single-phase) period are summarised in Table 1.

Table 1 – Milestones in spray cooling research (1966-1985)

	Main achievement	Year
1	Surface temperature and coolant temperature play are important in spray cooling effectiveness [2].	1966
2	Specific water impact density implies a specific boiling curve [3].	1966
3	There is a strong relationship between the heat transfer coefficient and the spray mass flux [5-7].	1970
4	Different regions needs different correlations to analyse [8-9].	1972 1974
5	Strong dependency of the heat transfer coefficient on the coolant temperature [10-11].	1979 1980
6	The heat transfer effectiveness is a function of the inverse of the square root of droplet diameter [12].	1982
7	The spray cooling behaviour on a flat surface is hypothesized [13].	1984

In the late 1980s, detailed studies on two-phase spray cooling were undertaken to understand and establish the effects on heat transfer of droplet size and velocity, mass flow rate, injector nozzle geometry, and the amount of sub-cooling [13-17]. It was in this period that a maximum heat flux of around 12 MW/m² was shown to be achievable with ‘superheating’ of only 20°C (where here the term ‘superheat’ refers to the difference between the temperature of the target surface and the coolant saturation temperature). This confirmed the dramatic improvement in heat transfer possible using spray evaporative

cooling [19-21]. But it also confirmed that it is possible to avoid the undesirable transition boiling phenomenon that occurs for high excess temperature differences in both the pool and flow boiling regimes.

The first practical application of spray evaporative cooling was reported in 1994 in an application to a computer processor [22], where miniature atomisers were used to spray a dielectric coolant. Other notable achievements in the development of spray cooling technology were the experimental studies on generating single-phase spray correlations [23-24]; the work on two-phase spray correlations [25-28]; and the spray parameter correlations (i.e. the 'Sauter Mean Diameter', and the 'critical heat flux') [29-30]. Finally, the notable achievements in studying the sensitive effect that spray parameters have on the heat transfer characteristics are reported in [31-35].

In the past decade, research has focused mainly on the challenges of implementing spray cooling in practical applications. The main challenge is that of control, in particular how the system parameters (coolant temperature and nozzle pressure difference) can be regulated to effectively control surface temperatures. Rybicki and Mudawar [36] undertook experimental studies to assess the effects of spray orientation on cooling performance, developing general correlations for single-phase heat transfer, nucleate boiling, and the critical heat flux. They showed that regulation of spray mass flow rate, and Sauter mean diameter, are the key hydrodynamic parameters that influence spray cooling performance. The results of the numerical modelling of multiphase flow spray cooling by Selvan *et al.* [37], showed that regulating spray parameters (in order to maintain a thin film over the heated surface, and its interaction with impinging liquid droplets) is very important for heat removal at high heat flux values [37].

Three comprehensive, theoretical and experimental studies on spray evaporative cooling (and its application) were published in 2008 and 2009 by Mudawar and Visaria [38-40]. They showed that the onset of nucleate boiling, critical heat flux, the evaporation efficiency, and

the magnitude of heat flux could all be changed by varying the amount of sub-cooling and mass flow. In 2009, Mudawar *et al.* [41] explored the viability and implementation of spray cooling for hybrid vehicle electronics, demonstrating that spray cooling is feasible for thermal management of hybrid vehicle electronics. However, despite proposing a viable system design, the control problem was not discussed. In 2010, Tilton and Tilton [42] patented a design for a thermal management system using spray evaporative cooling of microprocessors and other electronic devices. They did not propose any particular type of control strategy but suggested a passive solution involving sloped-surfaces and drain-channels to direct coolant across the surface in order to manage the surface temperature. In 2016 Cheng *et al.* [43] published a comprehensive review of both spray and flash evaporation cooling, in which the challenges for future development and applications were discussed. Different applications of spray cooling were examined for supercomputers [44], spacecraft [45], automotive engineering [41], and reactor pressure vessels [46]. The challenges identified in these publications were: i) development of a suitable control strategy, ii) the adoption of good spray tactics to avoid droplet rebound from the heated surface, and iii) the improvement of droplet distribution. ‘Good tactics’ ultimately means: i) the proper match between the frequency and duration of consecutive injection cycles needed to control heat transfer, ii) the homogeneous dispersion of droplets, and iii) control of the liquid deposited to avoid excessive secondary atomization or pre-impingement-evaporation.

In this paper, a control strategy and control system are proposed for a spray evaporative cooling of highly-boosted light-duty automotive engines. This control strategy and system, potentially offer a highly-effective cooling method accruing benefits of reduced fuel consumption and carbon emissions, reduced engine mass, better knock control, and more uniform cylinder head temperatures. The proposed control strategy has been developed to meet two main requirements: performance and robustness.

The paper is structured as follows: in Section 2 a plant model is constructed to describe the physics associated with spray evaporative cooling of an engine. This combines a lumped-parameter conductive and convective heat transfer modelling approach with published spray correlations. Section 3 describes the development of the control strategy which is then implemented in the form of a simulation model. Section 4 gives the results of three different types of simulation test involving a i) compatibility test, ii) a robustness test, and iii) steady and transient engine tests at part- and full-load. These simulated tests are designed to demonstrate the effectiveness of the proposed methodology in satisfying system requirements. The objective of the paper is to confirm the potential of the proposed cooling control methodology for spray evaporative cooling of highly-boosted automotive engines.

2. A MODEL FOR SPRAY EVAPORATIVE COOLING

This section describes the construction of a reduced-order mathematical model to represent the physical ‘plant’ associated with spray evaporative cooling of an internal combustion engine. This plant model will ultimately be controlled to demonstrate its effectiveness in thermal management. First, the empirical equations needed to model spray evaporative heat transfer physics are adapted, and the key dimensionless parameters established. A reduced-order model is then obtained in the form of a lumped-parameter heat-balance model which is followed by an iterative parameter selection procedure to enable a specific plant model to be constructed. This is achieved by selecting a particular nozzle type, and obtaining associated nozzle parameters. An overall (specific) plant model is thus obtained which has sufficient accuracy to allow a controlled version of the system model to be simulated. The controller, and the simulation model, will be developed in Section 3,

2.1 Empirical correlations and nozzle selection for spray evaporative cooling

To model temperature control of conventional IC engine cooling systems, several experimentally-validated, lumped-parameter models have been cited in the literature, such as Wagner *et al.* [47-49], Eberth *et al.* [50], Henry *et al.* [51], Setlur *et al.* [52], and Page *et al.* [53]. The governing equation used in all of these studies stems from a simple heat balance which assumes that the engine can be represented as a single mass m , with sections of spatially-uniform (but time-varying) temperature T :

$$m C \dot{T} = \dot{Q}_{in} - \dot{Q}_{out} \quad (1)$$

where $\dot{Q}_{in}$ is the rate of heat-release from the combustion process to the engine block (a function of engine load), $\dot{Q}_{out}$ is the heat dissipation rate to the cooling system, C is the engine block specific heat capacity. The heat input: $\dot{Q}_{in}$, and the temperature T , are set by design, since the engine wall temperatures depend on the duty cycle. A cooling control system must therefore not only be able to deal with a prescribed variation in heat load but must also be able to track variations in engine temperature (specified as set-points).

For spray evaporative cooling, with a nozzle-to-wall distance H , and a surface-to-coolant temperature difference ΔT , $\dot{Q}_{out}$ can be calculated using established correlations [27-28, 54-55]. To be specific: for a fluid of density ρ_L , dynamic viscosity μ , surface tension σ , and specific enthalpy of vaporisation h_{fg} , the spray cooling heat flux $\dot{Q}_{out} = \dot{q}_{out}$ can be obtained using a non-dimensional correlation [27] expressing the *Boiling Number* Bo as a function of the *Weber Number* We , and the *Jakob Number* Ja . This takes the form:

$$Bo = \alpha (We)^\beta (Ja)^\gamma \quad (2)$$

where the Boiling Number is defined as:

$$Bo = \frac{\dot{q}_{out} H}{\mu_L h_{fg}} \quad (3)$$

the Weber Number defined as:

$$We = \frac{\rho_L u_0^2 d_{32}}{\sigma} \quad (4)$$

and the Jakob Number defined as:

$$Ja = \frac{c_L \Delta T}{h_{fg}} \quad (5)$$

The constant parameters in equation (2) for water are: $\alpha = 15.6$, $\beta = 0.59$ and $\gamma = 1.68$. Also, the exit velocity U_o operating with a pressure difference Δp , and a discharge coefficient of unity, is given by the usual expression:

$$U_o = \left(\frac{2 \Delta p}{\rho_L} \right)^{1/2} \quad (6)$$

The Sauter Mean Diameter (SMD) of the spray droplets (needed in equation (4)) for a nozzle of exit diameter d_o , is obtained from the correlation [26, 29]:

$$\frac{d_{32}}{d_o} = 3.07 \left[\frac{\rho_v^{1/2} \Delta P d_o^{3/2}}{\sigma^{1/2} \mu_L} \right]^{-0.259} \quad (7)$$

where ρ_v is the vapour density.

2.2 Explicit plant model design procedure

To construct an explicit plant model, it is necessary to consider a specific set of conditions because the model must be obtained iteratively. A schematic diagram of the reduced order model is shown in figure 2. The cooled surface is represented by area A , of thickness w , thermal conductivity k , and with respective wall temperatures on the coolant and gas side as $T_{w,c}$ and $T_{w,g}$. By substituting equations (2) - (7) into equation (1) setting $\dot{Q}_{out} = \dot{q}_{out}$, the model for the coolant-side wall temperature is given as:

$$mC\dot{T}_{w,c} = \dot{Q}_{in}(t) - 30.236A \frac{\mu_L h_{fg}}{H} \left[\frac{c_L \Delta T}{h_{fg}} \right]^{1.68} \left[\frac{\rho_L u_0^2 d_o \left[\frac{\rho_v^{1/2} \Delta P d_o^{3/2}}{\sigma^{1/2} \mu_L} \right]^{-0.259}}{\sigma} \right]^{0.59} \quad (8)$$

Considering a heat balance condition from the gas-side to the coolant-side, the heat transfer rate can be expressed either in terms of conductive or convective heat transfer. By using a lumped-mass approximation of conductive heat transfer (from the gas-side to the coolant

wall), and a convective heat transfer model (from the coolant-wall to the coolant) two independent expressions can be constructed allowing elimination of the heat transfer coefficient associated with coolant-wall-to-coolant heat transfer. This results in an expression for the gas-side wall temperature as follows:

$$mC\dot{T}_{w,g} = \dot{Q}_{in}(t) - A \left[\frac{T_{w,g} - T_c}{\frac{w}{k} + A \frac{T_{w,c} - T_c}{\dot{Q}_{out}(t)}} \right] \quad (9)$$

Since the objective is to use heat transfer simulations to assess a proposed engine cooling control system, the design space must be constrained within the boundaries set by the applicability of the correlations used, and also within the limitations of the specified hardware. To meet these design requirements various combinations of control system configuration and nozzle option have been examined (but only the outcome is discussed here). This best spray nozzle selection outcome culminates in a step-by-step iterative procedure as follows:

Step-by-step spray nozzle selection procedure

Step-1: Set the chamber pressure $P_{chamber}$ and coolant temperature, T_c .

Step-2: Obtain tabulated values of coolant thermodynamic properties (liquid phase density ρ , surface tension σ , liquid phase viscosity μ , specific and latent heats, C and h_{fg}).

Step-3: Select values of wall temperature, $T_{w,c}$, nozzle pump pressure P_{nozzle} , and nozzle-to-wall distance, H and diameter d_0 . Obtain the heat flux from the following relationships: $\Delta P = P_{Nozzle} - P_{chamber}$ and $\Delta T = T_{w,c} - T_c$.

Step-4: Obtain SMD, d_{32} , from equation (7).

Step-5: Obtain droplet velocity from equation (6).

Step-6: Calculate Weber number from equation (4).

Step-7: Calculate Jacob number from equation (5).

Step-8: Calculate the boiling number from equation (3).

Step-9: Obtain heat flux q , from Bo . If this value is unacceptable, go back to Step 3 and iterate.

The acceptable designed value for $\dot{q}$ is set to 1.6 MW/m^2 , which is considered to be the largest value of local heat flux in a state-of-the-art boosted gasoline engine cylinder head at full-load. The optimum spray-to-the-wall distance (i.e. parameter H in figure 2) is calculated for the condition where the spray just spans the heated area to achieve the largest heat flux as confirmed in [30]. As an example, the heated surface, as shown in figure 2, is modelled as a circular block with diameter $D = 30 \text{ mm}$, thickness $w = 10 \text{ mm}$, and thermal conductivity $k = 205 \text{ W/(mK)}$.

Following through this step-by-step spray nozzle selection procedure, the parameters associated with the selected nozzle (i.e.: a *Bete JP8* [56]), are shown in Table 2. The calculated coolant mass flow rate of 0.77 gm/s corresponds to a mass flow rate of 81.41 gm/s for spray cooling of a 100 kW engine (which represents just 3.5% of the required 2280 gm/s coolant flow needed for a conventional cooling system). And although this is achieved at the expense of a much higher coolant pump discharge pressure (i.e. 11.5 bar instead of the 1.0 bar gauge pressure in a conventional system), spray evaporative cooling of an IC engine still offers a 60% reduction in the cooling power consumption. This is expected to bring significant benefits in improved fuel economy and reduced CO_2 emissions. However, these benefits can only be realised with a precise control and management strategy. The development of an appropriate control and management strategy is therefore discussed in Section 3.

Table 2. Spray nozzle specification

Spray cone angle	Nozzle injector diameter (mm)	Droplet SMD, d_{32} (μm)	Mean impact velocity (m/s)	Nozzle to Surface distance (mm)	ΔP (bar)	Coolant Temp (C)	Wall Temp (C)	Weber number We	Jacob number Ja	Boiling number Bo	Mass flow (g/s)
90	0.15	33	47	15	11.5	90	140	2338	0.093	28.28	1.46

3. CONTROL SYSTEM DESIGN FOR SPRAY EVAPORATIVE COOLING OF ENGINES

The ultimate objective of an engine cooling control system is to maintain the gas-side wall temperature $T_{w,g}(t)$ within pre-defined limits under all operating conditions. An engine cooling controller should therefore be able to precisely track engine temperature set-points while compensating for both cyclic and progressive variations in the combustion heat release which is effectively a heat input $\dot{Q}_{in}(t)$ to the cooling jacket of the engine block. Equations 8 and 9 show that the control parameters in a spray evaporative cooling system are: the coolant temperature T_L , and the pressure difference Δp across the spray nozzle.. It is practically impossible to measure gas-side metal temperatures of a running engine but the coolant-side wall temperature, and the vapour chamber pressure, can be easily measured (for example, respectively using a thermocouple and a pressure sensor). Assuming the system parameters, such as the engine jacket mass m , the specific heat capacity C of the jacket metal, and relevant dimensions, are known constants, the control problem can be considered in terms of regulating the coolant-side metal temperatures rather than the gas-side metal temperatures. To be specific, the requirement is to precisely track a desired coolant-side wall temperature with the aim of keeping the gas-side wall temperature within a reasonable pre-defined range for all operating conditions. The operating conditions for the highest heat flux regions of the cylinder head are intended to replicate engine behaviour under both part- and full-load conditions corresponding to respective heat flux levels of $\dot{q}_{in} = 0.2 \text{ MW/m}^2$ and $\dot{q}_{in} = 1.5 \text{ MW/m}^2$. A control structure is proposed in figure 3, which will be shown by simulation to precisely achieve the desired temperature tracking for these load conditions. This contains two control loops and a regulating mechanism as now explained in the following sub sections:

The spray coolant pump pressure control loop

First, a coolant pump pressure PID control loop is designed (i.e. the dashed line shown in figure 3). The purpose is to regulate the coolant supply pump pressure upstream of the spray nozzle to enable the pressure difference Δp in equation (8) to be set. The difference between the coolant-side wall temperature, and the desired wall temperature, is used as the error signal in the controller to regulate the nozzle pressure.

The spray coolant temperature control loop

The coolant temperature control loop is shown as a dotted line in figure 3. This loop regulates the coolant temperature using the chamber pressure, and a pre-defined degree of 'sub-cooling' (where the term 'sub-cooling' here refers to the difference between the coolant temperature, and the coolant-side metal temperature. This loop calculates the coolant saturation temperature from the chamber pressure by using a simple look-up-table, and then sets the coolant temperature to the pre-defined level of sub-cooling.

The mechanism for integrating the pressure and temperature control loops

Initial simulation tests on the pressure and temperature control loops showed that the control system worked well in the part-load condition. However it was not able to successfully deal with sudden changes of load of the sort typically expected in a road vehicle. In particular, the transient response from part- to full-load conditions was not acceptable. It became evident that changes in the pressure difference across the spray nozzle alone were not sufficient to provide the required degree of precision control. As a result of exploring various approaches to address this problem, it was found that the best results were obtained using a small additional coolant reservoir and a 3-way valve. The coolant in this additional reservoir is at lower temperature than the bulk of the coolant. A schematic diagram of the

main coolant supply and the injected lower-temperature coolant path is shown in figure 3. The governing equation for this 3-way valve is:

$$T_c = K T_r + (1 - K) T_m \quad (10)$$

where T_r is the temperature of the small reservoir of coolant, T_c is the (previously defined) temperature of the coolant sprayed into the chamber, and T_m is the temperature of the main coolant line. The temperature of the coolant sprayed into the jacket will be somewhere between the condensate temperature of the main mass of coolant and the temperature of the small reservoir. Parameter K in equation (10) is an additional control gain to regulate the mass fraction of total sprayed coolant taken from reservoir. This gain value is assigned using feedback from the engine load level, indicated as a solid line in figure 3. Different assumed expressions for K as a function of engine load and reservoir temperature were examined, the best being found to be a linear function, taking values between 0 and 1 corresponding to a range of loads with corresponding flux densities of 0.3 MW/m² to 1.5 MW/m², and with the reservoir temperature at a constant lower temperature of 40°C. For (low) engine loads that require cooling power densities less than 0.3 MW/m², the value of K is set to zero, which effectively switches-off the supply from the low temperature reservoir.

To summarise the control structure, measurements are assumed to be available of the coolant-side wall temperature, the spray chamber pressure, and the engine load level. The nozzle pressure and coolant temperature are then calculated (as described) to satisfy the required control modes for all heat flux ranges. A Simulink model of this strategy is shown in figure 4. Tuning of the 3 PID controller gains is achieved using a modified Ziegler–Nichols method [58]. The value of the mass fraction gain K to give the best evaporative cooling performance was found by trial and error. This was achieved by setting the degree of ‘sub-cooling’ in the range 5°C to 10°C for part load conditions.

4. TESTING THE SPRAY COOLING CONTROL MODEL USING SIMULATION

The control strategy described in Section 3 is now tested on three different scenarios: i) a compatibility test, ii) a robustness test, and iii) a varying engine load test. For each case, a predetermined variation of coolant-side wall temperature and load-dependent heat flux is specified as a function of time. The success of the control strategy is judged on its ability to track the prescribed coolant-side wall temperature within acceptable limits, in the presence of heat flux variation. Moreover, the boundary conditions for all simulations are defined as shown in Table 3 to be representative of the duty cycle of a highly boosted IC engine based on experimental results and benchmarking data [47-49, 52].

Table 3 – Defined Boundary Conditions for Control Strategies

	Parameter	Value
1	Coolant side metal temperature limit (degC)	160
2	Gas side metal temperature limit (degC)	225
3	Thermal management system response time to step change	Less than 4 sec
4	Main line Coolant temperature (degC)	95-128
5	Chamber pressure (bar)	1 - 2.4
6	Load level (MW/m ²) (part-load to full-load)	0.2-1.6

Scenario i): Compatibility Test

The first scenario examines the compatibility of the control system to manage the temperature at part- and full-load, and also an instantaneous transition between these two conditions. Figure 5 shows the prescribed variation of coolant-side wall temperature, and the heat flux over a duration of 210 seconds. As can be seen from figure 5, for $t \leq 30$ s, the heat flux through the wall is constant at 0.2 MW/m^2 ; it then increases linearly to the full-load conditions of 1.5 MW/m^2 at $t = 160$ s, and remains at this heat flux level for a further 30 s. The coolant-side wall temperature is varied in seven piece-wise linear steps each of 30 s duration as follows: i) 120°C for $0 \leq t \leq 30\text{s}$; ii) an increase to 125°C at $t = 30\text{s}$, then constant for 30s; iii) an increase to 135°C at $t = 60\text{s}$, then constant for 30s; iv) an increase

to 145°C at $t = 90\text{s}$, then constant for 30s; v) a decrease to 135°C at $t = 120\text{s}$, then constant for 30s; vi) a decrease to 130°C at $t = 150\text{s}$, then constant for 30s; and finally vii) a decrease to 120°C at $t = 180\text{s}$, then constant for 30s. The results of this compatibility simulation are shown in figures 6 and 7. Figure 6 (a) shows the variation of coolant delivery temperature and pressure difference across the spray nozzle; figure 6(b) shows the fractional breakdown of the coolant mass flow, and the total flow. Figure 7 shows both the coolant-side and gas-side wall temperatures.

Discussion of Scenario i) results

As can be seen from figure 6 the control inputs (i.e. the coolant temperature, coolant flow rates, and the pressure difference) are all used to achieve the required degree of control. As would be expected, to accommodate the gradual increase in heat flux, there is an overall reduction in coolant temperature. This is achieved by injecting water from the low-temperature reservoir, and from an overall increase in the total coolant mass flow - a consequence of the change in the pressure difference Δp . In addition to this overall reduction in coolant temperature, and the overall increase in mass flow, there are short-term variations in these parameters that affect control of the coolant-side wall temperature profile as shown in figure 5. Also shown in figure 5 is that the relatively large value of heat flux of 1.5 MW/m² is successfully dissipated by spray evaporative cooling using a coolant mass flow rate of 1.4 gm/s, which is very low compared with the use of single-phase liquid-coolant. The benefit of this very low flow requirement is that the pumping power is significantly lower than for a conventional engine cooling system. And because pumping power is a parasitic loss, its reduction translates into a corresponding reduction in fuel consumption and CO₂ emissions.

As shown in the insert in figure 7, the control strategy is successful in tracking the prescribed coolant-side wall temperature without steady-state error. But also within acceptable response time of less than 2 s. The value of the overshoot in the coolant-side

temperature is within acceptable engineering limits (i.e. $\leq 3^{\circ}\text{C}$) over the entire duration of 210 seconds. Finally, as shown in figure 7, the gas-side metal temperature remains below 200°C over the entire cycle, and this is well within the safe thermal limits for the aluminium alloys typically used for engine manufacture. This is important for structural integrity since the gas-side wall temperature must not be allowed to exceed predetermined limits.

Scenario II: Robustness test

The robustness of the proposed control strategy is now tested in its ability to track the prescribed coolant side wall temperature shown in figure 5 in the presence of a sinusoidal variation in the wall heat flux. This variation of heat flux is shown in figure 8, where there are 10 complete heat flux cycles over a duration of 210 second, each having a period of 21 second, with a minimum of 0.25 MW/m^2 and a maximum of 1.45 MW/m^2 . The pressure difference across the nozzle, and coolant delivery temperature, are treated as control inputs as shown in figure 9(a). The individual contributions, also the total mass flow rates, are shown in figure 9(b).

Discussion of Scenario ii) results

Figure 9(b) shows that a pump supplying 12 bar pressure with a flow rate less than 1.5 gm/s , will allow the controller to manage the thermal behaviour of the system even in the presence of significant periodic heat flux variations. Figure 9(b) also shows that although there are changes in the mass flow rate in the main coolant line, most of the control comes from modulation of the flow from the low-temperature reservoir. This is an important finding as it clearly demonstrates the need for a low-temperature reservoir and a 3-way valve. Moreover, the control command (i.e. the pump voltage), is smooth enough to implement in a practical context to provide the coolant mass flow rate shown in figure 9(b). Smoothness avoids erratic fluctuations and severe disturbances in the signal that could undermine the controller. As shown in figure 10, the results of the simulation for this scenario clearly confirm robustness of the proposed approach. In particular, the temperature tracking control is

achieved with high accuracy and reasonable response time. The fluctuation of temperature in this scenario is very low, falling within an acceptable operational range i.e. within $\pm 1.5^{\circ}\text{C}$. However between 65 seconds and 150 seconds, when the coolant-side wall temperature exceeds 135°C , the gas-side wall temperature does exceed 200°C .

Scenario III: An engine test

The final test is to examine the proposed approach for engine cooling in an automotive vehicle application. This scenario corresponds to a critical situation where the level of heat flux suddenly jumps from a minimum (part-load) condition to maximum level (full-load). An example of this scenario is in rapid acceleration of a vehicle from rest. The heat flux profile, together with the prescribed variation in coolant-side wall temperature, is shown in figure 11. It would appear from this figure that changes in the heat flux and coolant-side wall temperature have an inverse relation to each other. Closer examination of the scale however shows that in relation to the 7-fold difference between the maximum and minimum heat flux, the coolant-side wall temperature is almost constant. An additional requirement is that the coolant-side wall temperature can, during the sudden increase, respond fully to the desired change in less than 3 seconds, and in less than 5 seconds, during a sudden decrease of heat flux.

Discussion of Scenario iii) results

The test results for Scenario iii) in figures 12 and 13, clearly demonstrate the ability of the proposed control approach to deal with realistic variations in engine load. Figure 12 shows that both the pressure difference across the nozzle, and the coolant flow rate, are within a feasible range for practical implementation. In this instance, the pump and nozzle combination are able to provide 11 bar differential pressure, and 1.6 gm/s of coolant. The temperatures for the engine test are shown in figure 13. The variation of gas-side wall temperature remains below 200°C , and is considered to be wholly acceptable. Tracking of

the coolant-side wall temperature is also satisfied without unacceptable over- and under-shoot (i.e. less than 8°C). The inserts in figure 11 show that the control, during a sudden transition of less 3 seconds duration, is able to track the wall temperature from part-load to full-load, and during a transition of less than 5 seconds, is successfully able to track temperature during a change from full-load to part-load.

The test results for these three scenarios confirm the potential of spray evaporative cooling together with the proposed control structure for an automotive engine application. The proposed cooling system is expected to provide improved engine efficiency. Successful management of heat flux from the engine block has also been demonstrated, with significantly lower coolant flow rates using a smaller coolant pump. The control system design issues to be examined in a realistic implementation are: i) selection of cost-effective sensors, ii) considerations of power and weight; iii) experimental verification, and iv) adoption of cooling system anti-erosion standards.

Finally, it is evident that since the simulations assume a rapid response of 4 seconds and a cool reservoir temperature of 50°C, rapid response at this reservoir temperature would not be suitable for operation in very warm climates where the response time would need to be lengthened (as appropriate for the IC engine in a Hybrid Electric Vehicle).

5. CONCLUSIONS

A new control structure is proposed for the thermal management of road vehicle internal combustion engines using spray evaporative cooling. The cooling methodology and control system are shown by simulation to have very good potential in the practical automotive propulsion application identified. The proposed controller is tested in three different scenarios to examine compatibility, robustness, and the thermal demands of rapid vehicle acceleration from rest. The physical model that underpins these simulations is based on a combination of lumped-parameter modelling, and published correlations associated with

spray evaporative heat transfer. The proposed control methodology and corresponding control system is shown to track the desired temperature profile with very acceptable precision and with wholly acceptable transient response performance. A vital design feature established for the proposed control system is the use of a three-way mixing valve, which regulates the injected flow of coolant from a separate low temperature reservoir into the main coolant line. Without this three-way valve and the low temperature reservoir, simulation results confirm that it would not be possible to achieve the degree of precision control needed for automotive engine cooling.

Acknowledgements

The authors wish to acknowledge funding support for this project from the EPSRC under Contract Number: EP/M005755/1. The technical support is also acknowledged of colleagues at Ford Dunton UK and Dearborn USA, Denso Italy, and the Ricardo Technical Centre Shoreham, UK.

References

- [1] Incropera, F. P. and DeWitt, D. P., 2002, *Fundamentals of heat and mass transfer*, John Wiley & Sons, Australia.
- [2] Gaugler, R.E., 1966, "An Experimental Study of Spray Cooling of High Temperature Surfaces," PhD thesis, Department of Mechanical Engineering, Carnegie Institute of Technology, Pittsburg, PV.
- [3] Nukiyama, S., 1966, "Maximum and minimum values of heat Q transmitted from metal to boiling water under atmospheric pressure," *International Journal of Heat and Mass Transfer*, 9, (12), pp. 1419-1433.
- [4] Wachters, L. H. J., and Westerling, N. A. J., 1966, "The Heat Transfer from a Hot Wall to Impinging Water Drops in the Spheroidal State," *Chem. Eng. Sci.*, 21, pp. 1047–1056.
- [5] Eugene, A., and Mizikar, A., 1970, "Spray cooling investigation for continuous casting of billets and blooms," *Iron Steel Eng.* 47 (6) pp. 53–60.
- [6] Pederson, C. O., 1970, "An Experimental Study of the Dynamic Behavior and Heat Transfer Characteristics of Water Drops Impinging upon a Heated Surface," *International Journal of Heat and Mass Transfer*, 13, pp. 369–381.
- [7] Hoogendorn, C. J., and den Hond, R., 1974, "Leidenfrost Temperature and Heat-Transfer Coefficients for Water Sprays Impinging on a Hot Surface," *Proc. 5th Int. Heat Transfer Conf.*, vol. 4, pp. 135–138.
- [8] Toda, S., 1972, "A Study of Mist Cooling (First Report: Investigation of Mist Cooling), *Heat Transfer: Japanese Research*," 1, pp. 39–50.
- [9] Toda, S., 1974, "A Study of Mist Cooling (Second Report: Theory of Mist Cooling and Its Fundamental Experiments), *Heat Transfer: Japanese Research*," 3, pp. 1–44,
- [10] Sasaki, K., Sugatani, Y., Kawasaki, M., 1979, "Heat transfer in spray cooling on hot surface," *Tetsu-to-Hagane (J. Iron Steel Inst. Jpn.)*, 65, pp. 90–96.
- [11] Monde, M., Kusuda, H., and Uehara, H., "Burnout heat flux in saturated forced convection boiling with two or more impinging jets," *Transactions of the JSME*, 46, pp. 1834–1843, 1980.
- [12] Bolle, L., and Moureau, J. C., 1982, "Spray Cooling of Hot Surfaces," *Multiphase Science and Technology*, Hemisphere, Washington, DC.
- [13] Shoji, M., Wakunaga, T., and Kodama, K., 1984, "Heat Transfer from a Heated Surface to an Impinging Subcooled Droplet (Heat Transfer Characteristics in the Non-wetting Regime)," *Trans Jpn. Soc. Mech. Eng.*, 50(451), pp. 716–722.
- [14] Yao, S. C., and Choi, K. J., 1987, "Heat Transfer Experiments of Mono-dispersed Vertically Impacting Sprays," *Int. J. Multiphase Flow*, 13(5), pp. 639–648.
- [15] Choi, K. J., and Yao, S. C., 1987, "Mechanisms of Film Boiling Heat Transfer of Normally Impacting Spray, *International Journal of Heat and Mass Transfer*," 30(2), pp. 311–318.
- [16] Katto, Y., and Yokoya, S., 1988, "Critical Heat Flux on a Disk Heater Cooled by a Circular Jet of Saturated Liquid Impinging at the Center", *International Journal of Heat and Mass Transfer*, 31(2), pp. 219– 227.
- [17] Deb, S., and Yao, S.C., 1989, Analysis on film boiling heat transfer of impacting sprays, *International Journal of Heat and Mass Transfer*, 32(11), pp. 2099–2112,.
- [18] Mudawar, I. and Valentine, W. S., 1989, Determination of the Local Quench Curve for Spray-Cooled Metallic Surface, *Journal of Heat Treatment*, 7, pp. 107–121.
- [19] Pais, M., Tilton, D. Chow, L., and Mahefky, E., 1989, "High heat flux, low superheat evaporative spray cooling," *Proceedings of the 27th AIAA Aerospace Sciences Meeting*, Reno, NV.
- [20] Tilton, D. E., 1989, "Spray cooling". Ph.D. Dissertation, University of Kentucky.
- [21] Sehmbey, M.S., Pais, M. R. and Chow, L.C., 1990, "Effect of surface material properties and surface characteristics in evaporative spray cooling,". *AIAA/ASME 5th Joint Thermophysics and Heat Transfer Conference*, AIAA 90, 1728, Seattle, WA.

- [21] Klinzing, W.P., Rozzi, J.C., and Mudawar, I., 1992, "Film and transition boiling correlations for quenching of hot surfaces with water sprays," *J. Heat Treat*, 9(2), pp. 91–103.
- [22] Tilton, D. E., Tilton, C. L., Moore, C. J., and Ackerman, R. E., 1994, "Spray cooling for the 3-D cube computer. In: *Intersociety Conference on Thermal Phenomena in Electronic Systems*, Washington, DC, USA.
- [23] Yao, S.C., and Cox, T.L., 2002, "A general heat transfer correlation for impacting water sprays on high-temperature surfaces," *Exp. Heat Transfer* 15(4), pp. 207–219.
- [24] Nasr, G.G., Sharief, R., James, D.D., and Jeong, J.R., 1999, "Studies of high pressure water sprays from full cone atomizers," *ILASS-Europe'99*, Toulouse, pp. 1–6.
- [25] Yang, J., Chow, L.C., and Pais, M.R., 1996, "Nucleate boiling heat transfer in spray cooling," *Journal of Heat Transfer*, 118, pp. 668–671.
- [26] Bernardin, J. D., and Mudawar, I., 1997, "Film Boiling Heat Transfer of Droplet Streams and Sprays, *Int. J. Heat Mass Transfer*," 40(11), pp. 2579–2593.
- [27] Hsieh, S-S., Fan, T-C, and Tsai, H-H., 2004, "Spray cooling characteristics of water and R-134a. Part-I: nucleate boiling," *International Journal of Heat and Mass Transfer*; 47, pp. 5703–5712.
- [28] Hsieh, S-S, Fan, T-C., and Tsai, H-H, 2004, Spray cooling characteristics of water and R-134a. Part-II: transient cooling. *International Journal of Heat and Mass Transfer*; 47, pp. 5713–5724.
- [29] Estes, K. A., and Mudawar, I., 1995, "Correlation of Sauter mean diameter and critical heat flux for spray cooling of small surfaces," *International Journal of Heat and Mass Transfer*, 38, pp. 2985–2996.
- [30] Mudawar, I., and Estes, K.A., 1996, "Optimizing and predicting CHF in spray cooling of a square surface," *Journal of Heat Transfer*, 118, pp. 672–679.
- [31] Aguilar, G., Majaron, B., Verkruysse, W., Zhou, Y., Nelson J.S., and Lavernia, E.J., 2001, Theoretical and experimental analysis of droplet diameter, temperature, and evaporation rate evolution in cryogenic sprays, *International Journal of Heat and Mass Transfer* 44(17), pp. 3201–3211.
- [32] Rini, D.P., Chen, R-H., and Chow, L. C., 2002, "Bubble behavior and nucleate boiling heat transfer in saturated FC-72 spray cooling," *J Heat Transf*, 124, pp. 63–72.
- [33] Di Marzo, M., Tartarini, P., Liao, Y., Evans, D., and Baum, H., 1993, Evaporative cooling due to a gently deposited droplet," *International Journal of Heat and Mass Transfer*, 36, pp. 4133–4139.
- [34] Inada, S., and Yang, W-J., 1994, "Film Boiling Heat Transfer for Saturated Drops Impinging on a Heated Surface," *International Journal of Heat and Mass Transfer*, 37(16), pp. 2588–2591.
- [35] Milke, J. A., Tinker, S. C., and di Marzo, M., 1997, "Effect of dissolved gases on spray evaporative cooling with water," *Fire Technology*, 2nd Quarter, 33(2), pp. 99–114.
- [36] Rybicki, J. R., and Mudawar, I., 2006, "Single-phase and two-phase cooling characteristics of upward-facing and downward-facing sprays," *International Journal of Heat and Mass Transfer*, 49(1-2), pp. 5–16.
- [37] Selvam, R. P. , Lin, L., and Ponnappan, R., 2006, "Direct Simulation of Spray Cooling: Effect of Vapour Bubble Growth and Liquid Droplet Impact on Heat Transfer," *International Journal of Heat and Mass Transfer*, 49(23–24) pp. 4265–4278.
- [38] Visaria, M., and Mudawar, I., 2008, "Effects of high subcooling on two-phase spray cooling and critical heat flux," *IntJ Heat Mass Transf*, 51, pp. 5269–5278.
- [39] Visaria, M., and Mudawar, I., 2008, "Theoretical and experimental study of the effects of spray inclination on two-phase spray cooling and critical heat flux," *International Journal of Heat and Mass Transfer*, 51, pp. 2398–2410.
- [40] Visaria, M., and Mudawar, I., 2009, "Application of two-phase spray cooling for thermal management of electronic devices," *IEEE Trans Compon Packag Technol*, 32, pp. 784–793.

- [41] Mudawar, I., Bharathan, D., Kelly, K., and Narumanchi, S., 2009, "Two-Phase Spray Cooling of Hybrid Vehicle Electronics," *IEEE Transactions on Components and Packaging Technologies*, 32(2).
- [42] Tilton, D. E., and Tilton, C. L., 2010, "Thermal Management System for Evaporative Spray Cooling," US Patent US 7,836,706 B2.
- [43] Cheng, Wen-Long., Zhang, Wei-Wei., Chen, Hua., and Hu, Lei., 2016, "Spray cooling and flash evaporation cooling: The current development and application," *Renewable and Sustainable Energy Reviews*, 55, pp. 614–628.
- [44] Cader, T., Westra, L.J., and Eden, R.C., 2004, "Spray cooling thermal management for increased device reliability," *IEEE Trans Device Mater Reliab*, 4, pp. 605–613.
- [45] Silk, E.A., Golliher, E.L. and Paneer Selvam, R., 2008, Spray cooling heat transfer: Technology overview and assessment of future challenges for micro-gravity application, *Energy Convers Manag*, 49, pp. 453–468.
- [46] Anglart, H., Alavyoon, F., and Novarini, R., 2010, Study of spray cooling of a pressure vessel head of a boiling water reactor, *Nuclear Eng Des*, 240, pp. 252–257.
- [47] Wagner, J., Marotta, E., and Paradis, I., 2001, "Thermal Modeling of Engine Components for Temperature Prediction and Fluid Flow Regulation," SAE Tech. Paper 2001-01-1014.
- [48] Wagner, J., Ghone, M., Dawson, D., and Marotta, E., 2002, "Coolant Flow Control Strategies for Automotive Thermal Management Systems," SAE Tech. Paper 2002-01-0713.
- [49] Wagner, J., Srinivasan, V., and Dawson, D., 2003, "Smart Thermostat and Coolant Pump Control for Engine Thermal Management Systems," SAE Tech. Paper 2003-01-0272.
- [50] Eberth, J., Wagner, J., Afshar, B., and Foster, R., 2004, "Modeling and Validation of Automotive 'Smart' Thermal Management System Architecture," SAE Tech. Paper 2004-01-0048.
- [51] Henry, R., Koo, J., and Richter, C., 2001, "Model Development, Simulation and Validation, of Power Train Cooling System for a Truck Application," SAE Paper 2001-01-1731,.
- [52] Setlur, P., Wagner, J., Dawson D., and Marotta, E., 2005, "An advanced engine thermal management system: Nonlinear control and test," *IEEE/ASME Trans. Mechatron.*, 10(2), pp. 210–220.
- [53] Page, R., Hnatzuk, W., and Kozierowski, J., 2005, "Thermal Management for the 21st Century—Improved Thermal Control & Fuel Economy in an Army Medium Tactical Vehicle," SAE Paper 2005-01-2068.
- [54] Kim, J., 2007, "Spray Cooling Heat Transfer: The State of the Art", *International Journal of Heat and Fluid Flow*, 28(4), pp. 753-767.
- [55] Ghodbane, M., and Holman, J.P., 1991, "Experimental study of spray cooling with freon-113," *Int. J. Heat Mass Transfer* 34, pp. 1163–1174.
- [56] http://www.bete.com/pdfs/BETE_PJ-metric.pdf (Date accessed: 21st March 2017)
- [57] Jafari, S., Dunne, J.F., Langari, M., Yang, Z., Pirault, J-P, Long, C. A., and Thalackottore-Jose, J., 2017, "A review of evaporative cooling system concepts for engine thermal management in motor vehicles," *Proceedings of the Institution of Mechanical Engineers, Part D: Journal of Automobile Engineering*, 231 (8). pp. 1126-46.
- [58] Tajjudin, M., Rahiman, M. H. F., Arshad, N. M., and Adnan, R., 2013, "Robust Fractional-Order PI Controller with Ziegler-Nichols Rules," *World Academy of Science, Engineering and Technology International Journal of Electrical and Computer Engineering*, 7(7), pp. 1034-1041.

FIGURE CAPTIONS

Figure 1. Values of heat transfer coefficient in different cooling techniques, taken from [1].

Figure 2. Schematic diagram of a spray cooling system and relevant parameters.

Figure 3. Schematic diagram of the spray cooling control structure.

Figure 4. Simulink model for the spray cooling system with associated controller.

Figure 5. Prescribed variations of: (a) coolant-side wall temperature and (b) heat flux in the compatibility scenario.

Figure 6. Variation of: (a) coolant delivery temperature and pressure difference across the injector nozzle and (b) coolant mass flow rates in the compatibility scenario.

Figure 7. Variations with time of the coolant-side and gas-side wall temperatures in the compatibility scenario.

Figure 8. Prescribed variation of heat flux with time in the robustness scenario.

Figure 9. Variation of: (a) coolant delivery temperature and pressure difference across the injector nozzle and (b) coolant mass flow rates in robustness scenario.

Figure 10. Variations with time of the coolant-side and gas-side wall temperatures in the robustness scenario.

Figure 11. Prescribed variation with time of the coolant-side wall temperature and heat flux in the engine test scenario.

Figure 12. Variations with time of (a) coolant delivery temperature and pressure difference across the injector nozzle and (b) coolant mass flow rates in the engine test scenario.

Figure 13. Variations with time of the coolant-side and gas-side wall temperatures in the engine test scenario.

LIST OF FIGURES

Figure 1. Values of heat transfer coefficient in different cooling techniques, taken from [1].

Figure 2. Schematic diagram of a spray cooling system and relevant parameters.

Figure 5. Prescribed variations of: (a) coolant side wall temperature and (b) heat flux in the compatibility scenario.

Figure 6. Variation of: (a) coolant delivery temperature and pressure difference across the injector nozzle and (b) coolant mass flow rates in the compatibility scenario.

Figure 7. Variations with time of the coolant-side and gas-side wall temperatures in the compatibility scenario.

Figure 8. Prescribed variation of heat flux with time in the robustness scenario.

Figure 9. Variation of: (a) coolant delivery temperature and pressure difference across the injector nozzle and (b) coolant mass flow rates in robustness scenario.

Figure 10. Variations with time of the coolant-side and gas-side wall temperatures in the robustness scenario.

Figure 11. Prescribed variation with time of the coolant-side wall temperature and heat flux in the engine test scenario.

Figure 12. Variations with time of (a) coolant delivery temperature and pressure difference across the injector nozzle and (b) coolant mass flow rates in the engine test scenario.

Figure 13. Variations with time of the coolant-side and gas-side wall temperatures in the engine test scenario.