Temperature control of vibrating heat-generating hardware using spray evaporative cooling in the nucleate boiling region

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TEMPERATURE CONTROL OF VIBRATING HEAT-GENERATING HARDWARE USING SPRAY EVAPORATIVE COOLING IN THE NUCLEATE BOILING REGION


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ABSTRACT

A temperature control approach using evaporative spray cooling of vibrating surfaces in the nucleate boiling region is proposed and verified experimentally. This is relevant to temperature control of heat-generating automotive vehicle components. By exploiting an experimentally calibrated dynamic correlation model to represent evaporative spray cooling of a flat test-piece, a PID controller has been adopted with emphasis focused on the choice of gain parameters to ensure both stability of temperature control, and favourable responses in terms of relevant performance measures. Optimum linearisation of the correlation model has been achieved by solving an appropriate Wiener-Hopf equation, mainly to undertake a practical stability assessment of the closed-loop temperature control system. To verify the predicted control system performance, experimental measurements have been obtained from an instrumented, and spray-evaporatively-cooled, flat test-piece exposed to displacement vibration from a shaker. Experimental testing, appropriate to automotive vehicle component applications, includes large-amplitude, low frequency vibration at 12 mm and 1.9 Hz, and low amplitude, high-frequency vibration at 0.02 mm and 400 Hz. To assess the effects of different PID controller gains on the thermal performance of the thermal management system, a coefficient of performance (COP) is used, defined as the ratio of heat power removal to the required pumping power. To achieve a reduction in the settling time, and an increase in the rise time of stable control, a PID controller with a negative proportional gain showed most promising results. A 10.5% increase in COP was achieved in comparison to a PID controller with positive gains. This information is useful for the design and optimization of thermal management systems using evaporative spray cooling.

1. Introduction

Thermal management plays a pivotal role in the electrification of both automotive vehicles [1] and aircraft [2, 3]. In these applications, high heat-flux levels > 1 MW/m² have increased the need for high energy conversion efficiency to meet stricter regulatory targets (such as those set by the EU for 2030 [4, 5]) to speed-up progress towards zero CO₂ emission transport systems [6]. This need has focused attention on electrification of vehicle powertrain and aircraft propulsion, with the aim of reducing their carbon footprint [7-9]. To meet the thermal management demands of high heat removal under controlled conditions, highly optimized cooling systems are needed. Addressing this need, Jafari et al. [10] reviewed state-of-the-art evaporative cooling systems for internal combustion engines, showing that there was considerable interest in evaporative spray cooling [11]).

There is a relatively large body of literature published on spray heat transfer characteristics. These include the review by Liang and Mudawar [12] on single-phase, nucleate boiling, critical heat flux [13];
and the review on film boiling and quench curves [14]. Droplet impact and heat transfer correlation models has also been reviewed by Breitenbach et al. [15]. In the past two decades, a number of studies have also revealed how key parameters affect the heat flux and surface temperature during cooling. These findings show how the regimes of heat transfer depend on spray characteristics, pressure, flow rate, and degree of subcooling. Pais et al. [16] for example, investigated the effect of surface roughness on air-atomized water spray cooling of copper surfaces polished with grit-sizes ranging from 0.3 to 22 μm. Smoother surfaces (<1 μm), where heat transmission is dominated by film conduction and evaporation, achieve greater cooling. Using anhydrous ammonia as the working fluid, Bostanci et al. [17] investigated vapor-assisted spray cooling of micro-structured surfaces with indentations and protrusions. For the micro-indentation and micro-protruding surfaces, heat transfer coefficients improved by 49 percent and 112 percent respectively, involving heat fluxes up to 500 W/cm². These improvements were attributed to an increase in the number of surface nucleation sites, and the length of the three-phase contact line. Thermal hysteresis was observed in cooling these surfaces, as shown by decreased surface superheat values for a given heat-flux during surface cooling-down as compared to when heating-up. Bostanci et al. [18] found that a multi-scale structured surface, which combines the benefits of both micro- and macro-structures, improves the heat transfer coefficient by 161 percent when compared to smooth surfaces. Elsewhere Bostanci et al. [19], observed that improved surfaces significantly boost CHF value because capillary forces inside the surface structures tend to better retain liquid, spreading the liquid layer more efficiently, thereby delaying the creation of dry patches.

The relationship between excess temperature \( \Delta T = T_{\text{wall}} - T_{\text{sat}} \) and heat flux, varies notably between the different regimes. And this reflects the flow physics encountered in those regimes. In the single-phase regime, a complex liquid film covers the surface. At a higher value of \( \Delta T \), nucleate boiling occurs where vapour bubbles are generated and promote the heat transfer rate. The heat flux increases rapidly with increasing excess temperature in this region, up to the critical (or peak heat flux). Beyond this, transition boiling occurs, where a vapour partially covers the surface and acts as an insulator; further increase in \( \Delta T \) leads to a decrease in \( q \). At even higher values of \( \Delta T \), atomization of the film occurs and this is followed by a complete vapour film. Surface roughness also affects the heat transfer coefficient. It is known that a high value of heat transfer coefficient occurs with very smooth and rough surfaces, but a lower value for a surface finish that is representative of a typical machining process [12].

Spray evaporative cooling offers a significant increase in heat flux compared with single-phase convection [20, 21]. Spray cooling, compared with flow boiling heat transfer, also offers significantly greater heat flux for a lower flow-rate [22, 23]. Lower mass-flow rates are very advantageous because they result in a significant reduction in the required pumping power and mass of associated hardware [24]. Effective flow-controlled spray cooling approaches have been proposed in [25] and [26]. To thermally manage power inverter modules used in automotive applications, a spray cooling system was developed and tested by Bostanci et al. [27]. The performance of this system demonstrated that spray cooling of power electronics is a viable solution for achieving large power densities while maintaining acceptable and uniform device temperatures. Spray cooling also enables a compact and efficient system design stemming from the use of a high temperature coolant at low flow rates.

Evaporative spray cooling meets all of the extra heat removal requirements for downsized internal combustion engines (ICEs) for conventional vehicles [11], and as ICE range extenders in hybrid electric
vehicles. Focusing on cooling of the electronic hardware in hybrid electric vehicles however, Mudawar et al. [28], reported that a feasible thermal management system should be able to dissipate heat in the flux range 1.5 - 2 MW/m² while maintaining temperatures below 150 °C (i.e. the silicon-based transistors limit). Evaporative spray cooling should meet this challenge, moreover its two-phase flow helps to eliminate hot spots, by providing a uniform temperature across all cooled surfaces [24]. Uniform temperature is vital in the thermal management of electronic hardware and Lithium-ion battery stacks [29], where there is risk of overheating [30, 31] and fires [32]. Further applications include indirect cooling, using pre-cooled air, which improves thermal performance of lithium battery packs [33]. Li-ion batteries have a narrow optimum working temperature range - exceeding the upper limit runs the risk of thermal runaway and fires; operating below the lower limit, reduces battery life. Operating at an optimal temperature maximizes power output, and also improves charging rate and longevity [34, 35].

Most potential applications of evaporative spray cooling in transport systems will be exposed to mechanical vibration, which can influence heat transfer [36][37]. This can be best seen in terms of two dimensionless parameters: Vibrational Reynolds Number \( Re_V = \rho_l a\omega d_i / \mu_l \), and Acceleration Number \( Ac = (\omega^2 a) / g \), where \( a \) is the vibration amplitude and \( \omega \) is frequency. The parameter \( Re_V \) is a measure of the increased turbulence that results from a vibrating surface [38] indicating how it can affect heat transfer. In fact, heat transfer changes resulting from vibration can be explained by turbulence changes to the boundary layer thickness, and generated nucleation sites [39], and by changes to the wetting-angle [40]. The parameter \( Ac \) can be used to explain the influence of experimentally-measured acceleration on spray cooling heat transfer, which has been investigated in various dynamic conditions [38, 42-44]. It is still difficult however to draw general conclusions from such studies owing to the complexity of spray heat transfer mechanisms and from differences in the experimental conditions.

The potential exists therefore, for evaporative spray cooling to achieve high heat transfer rates with low mass flow rates. There are however significant concerns remaining about the ability of the actual hardware to achieve stable control of an evaporative spray cooling system at the required heat flux and temperature. There is currently no published evidence of successful thermal management and control of hardware involving two-phase evaporative spray cooling of vibrating surfaces. In this paper, the design and performance of an experimental cooling control system is examined to confirm, in the presence of vibration, effective thermal management of heat generating hardware using spray evaporative cooling. The effect of choosing different gain values on the controller performance is examined in detail. The objective of the paper is to establish the efficacy of a particular control strategy for evaporative spray cooling of vibrating surfaces in terms of being able operate successfully at target temperatures needed for high-heat-flux cooling of a representative test piece.

2. Proposed thermal management system

The proposed thermal management system involves a ‘plant’ which here represents an evaporatively cooled test-piece (exposed to vibration) whose temperature is regulated by feedback
control. Figure 1 shows a high-level representation of the closed-loop feedback control system for temperature control of the hardware.

As can be seen, a test-piece, in contact with the heating source, is cooled using evaporative spray cooling. The objective of the controller is to keep the cooling process in the nucleate boiling regime to benefit from the heat removal in the form of latent heat of vaporization. In this regard, a PID controller has been designed to control the coolant-pump flow-rate to maintain the coolant-side temperature of the test piece (exposed to displacement vibration) at a target value. There are two circuits in Figure 1. The first circuit (in blue) shows the coolant flow from the coolant-pump to a full-cone nozzle on top of the test piece (where \( v \) is the volumetric flow rate). The saturated water, broken-up into droplets by the nozzle, impinges onto the test-piece surface, removing heat generated by the hardware. The vapour and accumulated droplets are drained into the vapour chamber, then into the condenser tank. Finally, the condensed water reaches the suction-side of the coolant pump to complete the circuit. The second circuit (in black) shows the temperature feedback signal from the cool-metal surface. This state is compared with the desired surface temperature (as a reference signal) to generate the error required to be used as an input signal for the PID controller. The control signal to the coolant pump completes this circuit.

2.1 Correlation model of the ‘plant’.

To undertake performance assessment and tuning of the thermal management system in Figure 1, a closed-loop feedback control system model is constructed to accurately model the entire physics during the cooling process (i.e. when the heat-generating hardware exposed to the vibration is cooled using evaporative spray cooling in the nucleate boiling region). To describe the physics of the evaporative spray cooling needed for the ‘plant’ model, the following empirical correlation model has
been constructed in [38] (using the Generalised Buckingham Pi method [45]) to represent evaporative spray heat transfer characteristics for the test piece under vibration in the nucleate boiling region:

\[
Bo = 1.2 \times 10^{-2}Ja^{1.392}\left(\frac{\rho^2}{\mu_l^3}\right)^{0.981}Re_v^{-0.987}\left(\frac{a}{H}\right)^{0.483}Ac^{0.499}
\]  

(1)

where \(Bo\) is Boiling number \((qH/\mu_l h_{fg})\) which includes heat flux \(q\), nozzle-to-surface distance \(H\), viscosity \(\mu_l\) and specific enthalpy \(h_{fg}\). Vibrational Reynolds Number \(Re_v = \rho_l a\omega d_H/\mu_l\) involving vibration amplitude \(a\), and angular frequency \((\omega = 2\pi f)\), Acceleration Number \(Ac = \omega^2 a/g\), \(Ja\) is the Jakob Number \((Ja = C_l\Delta T/h_{fg})\), which is the ratio of sensible to latent heat.

As was discussed in the development of dynamic correlations in the nucleate boiling regime in [38], functional forms can be derived based on modelling requirements and expectations. Dimensionless functional forms can be derived based on spray specifications and flow rate. Whether, for instance, it is for a design purpose that spray specifications are important (which is not the case in this study), or a special control approach that the pumping flow-rate (either mass flow rate, \(\dot{m}\), or volumetric flow rate, \(\nu\)) plays an important role on the wall temperature control, different dimensionless \(\Pi\)–terms can be obtained. In other words, these \(\Pi\)–terms can simplify measurement requirements, such as spray specifications which are usually costly and time consuming to obtain. They also make system identification easier.

In the thermal management design process of an evaporative cooling system, pumping flow-rate plays an important role on the wall temperature control. In addition, spray specifications should not be included in functional forms based on considerations for measurements (only a minimum number of sensors is desired to be embedded in an actual thermal management system).

It is well-known that not only for a nozzle with a different design but also for a single nozzle from one manufacturer, owing to the undetectable variability in machining, the flow field of the spray may differ, resulting in a sizeable error associated with using correlations. This may be the case even if the particular nozzle used, is consistent with information reported in the literature. This has been explained in [38], where the correlations are constructed and calibrated. There, based on the literature, it is explained that, there are concerns about using spray flow data from one nozzle to predict the heat transfer characteristics of a different nozzle, even if they have the same geometry. Overall, the ideal solution is to use the same nozzle [46] to obtain both the hydrodynamic and heat transfer characteristics, and then try to use a fitting function, or any other modelling approach, to make a unique correlation for every nozzle.

However, in a thermal management problem, a successfully tuned controller itself can compensate for the errors associated with using a correlation fitted to one nozzle, to act as a plant model for a different nozzle. This claim has been verified in [47], where the same correlation used in this study (i.e. equation (1), which is for a UNIJET nozzle) has been used as the plant model for two PJ nozzles.
To construct the plant model, a simple heat balance of the heat-generating hardware has been used to model the temperature control. A similar approach, adopted by Setlur et al. [48], Eberth et al. [49], Wagner et al. [50-52], and Henry et al. [53], results in a heat balance equation as follows:

\[ mC\dot{T} = \dot{Q}_{in} - \dot{Q}_{out} \]  

(2)

where \( m \) is the test-piece mass, \( C \) is the specific heat capacity of the cooled test-piece (in this case copper), \( \dot{T} \) is time-varying temperature of the test piece, \( \dot{Q}_{in} \) is the rate of heat release of the source, \( \dot{Q}_{out} \) is the rate of heat removal of the proposed cooling system. To obtain the coolant-side temperature, equation (1) is substituted into equation (2), and by setting \( \dot{Q}_{out} = qA \), gives a first-order (nonlinear) differential equation model for the test-piece temperature \( T_w \) which will shortly be linearized, a necessary step to ultimately undertake practical stability analysis. This correlation model can actually be linearised in a number of ways, and, although the experimental facilities will be described in Section-3, for the benefit of linearization, it is helpful to state the numerical values for a flat circular test-piece. The test-piece mass \( m = 404.4 \) gm, the diameter of \( d_H = 20 \) mm, and specific heat capacity (for copper) \( C = 385 \frac{J}{kgK} \). Parameters not assigned values, still appear as variables in the first-order (nonlinear) differential equation model, which becomes the ‘plant’ model, taking the form:

\[ mC\dot{T}_w = \dot{Q}_{in} - 1.2 \times 10^{-2} A \left( \mu_l h_{fg} \right) \left( \frac{C_l \Delta T}{h_{fg}} \right)^{1.392} \left( \frac{\sigma_l \sigma v}{\mu_l^3} \right)^{0.981} Re^{0.987} \left( \frac{a}{H} \right)^{0.483} Ac^{0.499} \]  

(3)

where \( \Delta T \) is the excess temperature in the term \(Ja = C_l \Delta T / h_{fg} \) (with \( T_{sat} = 99.6 \) °C, assumed constant at 1 bar), \( A \) is the area of the circular test-piece. Equation (3) represents the model for a single-input single-output (SISO) system, the input being \( v \), and the output being the time-varying wall temperature \( T_w \).

By considering equation (3) at a fixed dynamic operating condition, for example corresponding to nucleate boiling at 1 bar, nozzle-to-surface distance of \( H = 17 \) mm, vibration amplitude \( a = 12 \) mm, frequency = 1.9 Hz, and a pump flow-rate of 200 ml/min, various ‘constant’ terms result in equation (3), namely \( \dot{Q}_{in} = 1.2 \times A \times 10^6 W, \mu_l = 0.00028 \) Kg/m s, \( h_{fg} = 2257 \) kJ/kg, \( C_l = 4.219 \) kJ/kg K, \( \rho_l = 958 \) kg/m³, \( \sigma = 0.0589 \) N/m, and \( \frac{A}{mC} = 2.0177 \times 10^{-6} \) K/s² kg. Substitution into equation (3), gives a simpler form as follows:

\[ \dot{T}_w(t) = 2.421 - 7.536 \left( T_w - 99.6 \right)^{1.392} v^{0.981} \]  

(4)

It is recognized that some practical implementations will require a larger surface area than the test piece considered here (i.e. 20 mm diameter). This may also involve the use of multiple nozzles to provide the necessary coverage [54]. Nonetheless, if the heat flux for these applications can be correlated in terms of the main non-dimensional variables, as achieved here in equation (1), then a similar form to equation (4) should be obtained. The published literature on the effect of multiple-nozzle arrays on heat transfer in the nucleate boiling regime, supports the equivalence between single nozzles and arrays of multiple nozzles. Horacek et al. [55] for example, report that the average heat flux
for two nozzles, is about the same as for a single nozzle, assuming the same nozzle-to-surface distance. Lin and Ponnappan [56], report that the heat transfer trends for an array of 8 miniature nozzles were similar to those of a single nozzle. It is also suggested in [56] that the difference in heat flux, using an array of 48 nozzles (to cool a ‘large’ rectangular surface of dimensions 2.54 cm x 7.60 cm), was only around 30 per cent, compared with using a single-nozzle spray (to cool a smaller rectangular surface of dimensions 1 cm x 2 cm). Moreover, in a study published in [47], it is shown that construction of a practical simulation is possible. Namely, a dynamic correlation model for single-nozzle spray evaporative cooling of a flat test-piece exposed to vibration (equation (1)), can be used as a reasonable model for multiple-nozzle spray evaporative cooling of component parts with curved cooling surfaces in non-horizontal orientation. The simulation approach in [47] advancing what was achieved in [57].

**Linearisation of the plant model.**

There is no closed-form solution to Equation (4) but it can be linearised in a number of ways and an exact solution is then trivial. The simplest way to linearize is to expand the nonlinearity in a Taylor series about an operating point then truncate, retaining only the linear term. For example, when the initial (temperature) condition $T_w = 110^\circ C$ is chosen as the expansion point, the linear model becomes.

$$\dot{T}_w(t) = -0.166 T_w(t) - 1419 \ v(t)$$

(5)

A comparison of the analytical solution of equation (5) with a numerical of equation (4) shows however a very significant difference, suggesting that linearization about the initial condition is not acceptable. An alternative linearization approach is to construct a least-square error solution [58] (over the entire time range). This is an iterative process involving numerical solution of equation (4) using a known initial condition, corresponding, for example, to the response to a step-input, or a free decay response. When an initial condition of $110^\circ C$ is used in the numerical solution (corresponding to a step input of $135^\circ C$), the least-square-error linear model obtained is:

$$\dot{T}_w(t) = -0.142 T_w(t) + 5755.6 \ v(t)$$

(6)

By contrast, when the initial condition of $140^\circ C$ is used (corresponding to a free-decay response), the least-square-error linear model obtained is

$$\dot{T}_w(t) = -0.147 T_w(t) + 5469.2 \ v(t)$$

(7)

Figure 2a and figure 2b show a comparison of the responses obtained by the numerical solution of equation (4) and the linearized model equation (6) for a step-input, and equation (7) for the free-decay response. It can be seen from figures 2a and 2b, that the least-square error linearized model responses are in excellent agreement when predicting the responses of the nonlinear model with the same initial conditions. However, when the equivalent linear models are switched, the predicted results are very poor (not shown). To attempt to overcome this deficiency, a single least-square-error linear model was constructed with the best overall performance for both the step-input and free-decay response. This model is:
\[ \dot{T}_w(t) = -0.134 T_w(t) + 5129 \nu(t) \]  

(8)

Analytical solutions to Equation (8), with initial conditions of respectively 110 and 140 °C (assuming constant \( \nu = 0.003 \text{ m}^3\text{s}^{-1} \), are:

\[ T_w(t) = 127.812 - 17.812e^{-0.134t} \]  

(9)

and

\[ T_w(t) = 12.188e^{-0.13363t} + 127.812 \]  

(10)

respectively, where the transfer function obtained for the linearised plant model equation (8) is:

\[ \frac{T_w(s)}{\nu(s)} = \frac{5129}{s+0.134} \]  

(11)

![Fig. 2. Responses obtained by numerical solution of equation (4) and the linearized model: a) equation (6) for the step-input, and b) equation (7) for the free-decay response; c) and d) numerical responses to equation (4) (with respective initial conditions of 110 and 140 °C), and the corresponding responses obtained from equations (9) and equation (10).](image-url)
Figure 2c and 2d show respective comparisons between the numerical responses to equation (4) (with initial conditions of 110 and 140°C), and the corresponding responses obtained from equations (9) and (10). It is evident from Figure 2c and 2d, that the level of agreement between the linearised model and the numerical solution of the full nonlinear model equation (4) is still poor. A more rigorous linearization process therefore needs to be followed such as the Wiener Hopf approach, as now explained.

**Obtaining the optimal fixed-order linear model using the Wiener Hopf approach**

To obtain the optimal model to replace equation (4), the Wiener-Hopf approach is used to obtain a fixed-order linear transfer function [59]. For the required input to a Finite Impulse Response (FIR) filter (i.e. the optimal filter weighting function), a random temperature history is generated by numerical simulation involving equation (4), with a white noise input. To generate the input temperature (as shown for example in Figure 3a), as supplied to the Matlab-Simulink model of the plant, the heat flux as the heat source in $\dot{Q}_{in} = qA$ and volumetric flow rate $v$, have respectively been randomly varied from 0 to $1.2 \times 10^6 \text{ W/m}^2$ and from 0 to 0.00333 $\text{ m}^3\text{s}^{-1}$. This satisfies the assumptions associated with the Wiener filter theory namely that both signal and noise are random processes with known spectral characteristics. Here, to ensure the simplest closed-loop stability analysis an optimal 1st-order linear model is chosen. The additive combination of randomly-generated input-temperature and white noise (shown in Figure 3b (in red)) has been correlated using the FIR filter. Figure 3b, shows that the FIR filter is successful in modeling the temperature with noise (where the output signal is in blue). The Mean Square Error (MSE) between two signals in Figure 3 is 0.77, and the transfer function of the optimal filter weighting function is:

$$H_{\text{FIR}}(s) = \frac{0.5s + 0.5}{s}$$

![Fig. 3. a) Numerically-generated temperature; b) the input signal to the Wiener-Hopf model; the temperature with additive white noise, and the FIR output signal.](image-url)
2.2 Stability analysis of the closed-loop transfer function

The design of a stable hardware controller, now proceeds using the best least-square solution to the nonlinear ‘plant’ model obtained by the Wiener-Hopf approach. Stability of the closed-loop control transfer function, is assessed iteratively using the root locus method (implemented using the Matlab Control System Toolbox). First, the root locus diagram is shown in Figure 4a for the (original) linearised open-loop correlation model of the plant (equation (11)).

![Root Locus Diagrams](image)

Fig. 4. *Root locus* diagrams: for (a) linearised open-loop correlation model of the plant, (b) linearised closed-loop feedback control with $P=0.02$, $I=0$ to 0.08 and $D=0$ to 0.004, (c) the designed controller with $P=2.1$, $I=0.08$ and $D=0.001$, and (d) closed-loop control system with negative proportional PID controllers; one with $P=-2.1$, $I=0.38$ and $D=0.001$ and another with $P=-2.1$, $I=0.7$ and $D=0.002$.

The stability of the closed-loop transfer function is shown in Figure 4b with proportional gains from 0 to 2 (in increments of 0.5); integral gains ranging from 0 to 0.08 (in increments of 0.02); and derivative gains from 0 to 0.004 (in increments of 0.001). As can be seen, the system is stable for all sets of PID gains. After finding the best integral gain to improve the steady-state error, as well as increasing the
stability using small values of derivative gains, the most promising PID gains are: P=2.1, I=0.08, D=0.001. Since the transfer function for a PID controller is:

\[ k_p + \frac{k_i}{s} + k_d s = \frac{k_ds^2 + k_ps + k_i}{s} \]  

the gain values thus found are chosen as starting gains for the hardware. Using the PID controller transfer function (equation (13)) with the starting gains, the linearised closed-loop feedback control system transfer function is given as follows:

\[ G(s) = \frac{5.129 s^2 + 10770 s + 410.3}{6.129 s^2 + 10770 s + 410.3} \]  

The root locus diagram for equation (14) shown in Figure 4c, gives stability information of a thermal management system using the PID controller. By a process of trial and error, both for the simulation and for the experimental hardware, it becomes apparent that another set of PID controller gains with a negative proportional gain of -2.1, namely P=-2.1, I=0.38, and D=0.001 can also make the system stable (as shown by the root locus diagram in Figure 4d). In fact, increasing the integral gain (to I=0.7) and doubling the derivative gain (to D=0.002) actually results in better stability and robustness of the system. The poles for these two sets of PID gains are shown in Figure 4d, along with robustness information in Table 1.). The linearized closed-loop control system transfer function for this PID controller (i.e. with P=-2, I=0.7, D=0.002) is:

\[ G(s) = \frac{10.26 s^2 + 10258 s + 3590}{9.258 s^2 + 10260 s + 3590} \]  

As expected, during the tuning process of the PID controllers in the actual control hardware, these two sets of PID controller gains (i.e. one with a positive and another with a negative proportional gain) were found to be promising. As can be seen from the root locus plot in Figure 5c and 5d, each set has its pros and cons. The pole for the positive proportional PID control system is closer to the imaginary axis. (It is -0.038 for the PID with P=2.1, I=0.08 and D=0.001, and -0.35 for the PID with P=-2, I=0.7 and D=0.002). Thus, the negative proportional PID controller can theoretically be more stable although it has a smaller range of stability for the PID controller with the larger integral and derivative gains (as shown in Figure 4d).

To further support this finding, the stability of the optimal model (obtained using the Wiener-Hopf approach) is also examined. The stability of both the open-loop and closed-loop transfer functions with the same PID gains have been examined also using a root locus diagram. As expected, the closest poles in both Figures 5a and 5b with the same ranges of P, I, and D gains are quite similar (0.040 for the best least-square solution to the nonlinear ODE of the ‘plant’ model, and 0.042 for the Wiener-Hopf derived model). The closest poles to the imaginary axis for PID gains of P=2.1, I=0.08 and D=0.001 on the root locus diagrams for both models are also very similar (-0.038 for the linearised model in Figure 4c, and -0.039 for Wiener-Hopf derived model in Figure 5c). In fact, examining the poles in Figures 4d and 5d for the negative proportional PID controllers, suggests the same trend. Moreover, the root locus diagrams in Figure 5c and 5d still support the previous finding that the pole for the positive proportional PID
control system is closer to the imaginary axis. Therefore, the closed-loop control system with positive gains is actually less stable in comparison to the system designed with a negative proportional PID gain. The only difference between the stability findings relates to the stability range of the PID controller with larger integral and derivative gains (P=-2.1, I=0.7 and D=0.002). In Figure 5d (corresponding to the Wiener-Hopf model) the s-domain for which the system is stable, has two complex conjugate poles of \(-0.37 \pm 36.7i\) but this does not indicate any reduction in stability.

Regarding the actual hardware, both the positive and negative proportional PID controllers have been implemented - their effects on the robustness are summarized in Table 1 (which are derived from results that will be discussed in Section 5). PID Controller Number 1 and Controller Number 2 are used in tuning the actual hardware without vibrational disturbance (i.e. under static conditions), whereas for the PID Controller Numbers 3, 4, and 5, the actual hardware was tested with vibration disturbances. Comparing the settling time and overshoot, the results for PID controller Numbers 1, 2, 3, 4, and 5, confirm better stability for the negative proportional gain controller. In fact, there are evident
reductions in settling time and overshoot for the PID controllers with negative proportional gains. Disturbance rejection for the responses in dynamic cases is good. Moreover, to check the level of uncertainty in the repeatability of the control, the same PID gains are attempted and the results are available in rows 4 and 5 of Table 1. The 3 s difference in settling time with a temperature difference of only 0.1 °C overshoot, offers an impressive level of repeatability which is further confirmation of the stability of the closed-loop control system.

Table 1. Summary of the robustness results for the hardware closed-loop control system.

<table>
<thead>
<tr>
<th>PID Controller No.</th>
<th>P</th>
<th>I</th>
<th>D</th>
<th>Disturbance</th>
<th>$q \text{ (} \frac{MW}{m^2}\text{)}$</th>
<th>$T_w\text{ (} ^\circ\text{C}\text{)}$</th>
<th>Settling time (s)</th>
<th>Overshoot (°C)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>2.1</td>
<td>0.08</td>
<td>0.001</td>
<td>No, Static</td>
<td>0.2-1.8</td>
<td>110-135</td>
<td>155</td>
<td>1.3</td>
</tr>
<tr>
<td>2</td>
<td>-2</td>
<td>0.7</td>
<td>0.002</td>
<td>No, Static</td>
<td>0.2-1.8</td>
<td>110-135</td>
<td>129</td>
<td>0</td>
</tr>
<tr>
<td>3</td>
<td>-2</td>
<td>0.7</td>
<td>0.002</td>
<td>Dynamic (a=12mm, f=1.9Hz)</td>
<td>0.2-1.8</td>
<td>110-135</td>
<td>61</td>
<td>0.5</td>
</tr>
<tr>
<td>4</td>
<td>-2</td>
<td>0.7</td>
<td>0.002</td>
<td>Dynamic (a=0.02mm, f=400Hz)</td>
<td>0.2-1.8</td>
<td>110-135</td>
<td>70</td>
<td>0.6</td>
</tr>
<tr>
<td>5</td>
<td>-2</td>
<td>0.7</td>
<td>0.002</td>
<td>Dynamic (a=0.02mm, f=400Hz)</td>
<td>0.2-1.8</td>
<td>110-135</td>
<td>67</td>
<td>0.7</td>
</tr>
</tbody>
</table>

Table 2 shows the proposed controller performance compared with published findings. This comparison includes both simulation and experimentally-measured results of temperature control and thermal management using spray cooling systems applied to electronic components and power plant (e.g. a superheater) - all without any vibrational disturbances. All of the cited literature noted the complexity of temperature control owing to nonlinearity, and the time-lag in the plant dynamics. Table 2 shows that all results have a higher settling time in comparison with the proposed control system while the overshoot is similar. The performance of the proposed closed-loop control system, resulting from use of negative proportional gains in the PID controller, has improved the settling time (by 37%) and the Coefficient of Performance (by 10.5%, COP). Overall, the proposed control system is stable and robust.

Table 2. Published temperature control and thermal management studies using spray cooling systems.

<table>
<thead>
<tr>
<th>Authors (Date)</th>
<th>Apply on</th>
<th>Coolant</th>
<th>Test Piece</th>
<th>Requirements</th>
<th>Control variable</th>
<th>Controller</th>
<th>Robustness</th>
<th>Settling /rise time (s)</th>
<th>Overshoot (°C)/%(%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Wang et al. [60] (2013), S¹</td>
<td>IGBT²</td>
<td>water</td>
<td>SPC³ wick</td>
<td>$q = 4 \text{ (} \frac{MW}{m^2}\text{)}$, $T = 153\text{ (} ^\circ\text{C}\text{)}$</td>
<td>Pump voltage</td>
<td>PID</td>
<td>868/-</td>
<td>2.1°/2.1 (%)</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>Fuzzy-PID</td>
<td>682/-</td>
<td>0.98°/0.98 (%)</td>
<td></td>
</tr>
<tr>
<td>Ding et al. [61] (2014), E⁴</td>
<td>IGBT</td>
<td>water</td>
<td>IGBT</td>
<td>$q = 0.64-0.68 \text{ (} \frac{MW}{m^2}\text{)}$, $T = 91\text{ (} ^\circ\text{C}\text{)}$</td>
<td>Pump voltage</td>
<td>PID</td>
<td>155/43</td>
<td>NA (small)</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>Fuzzy-PID</td>
<td>-</td>
<td>smaller</td>
<td></td>
</tr>
<tr>
<td>Sai and Reddy [62] (2016), S&amp;E</td>
<td>Power Plant</td>
<td>water</td>
<td>Superheater</td>
<td>$q = - \text{ (} \frac{MW}{m^2}\text{)}$, $T = 540\text{ (} ^\circ\text{C}\text{)}$</td>
<td>Coolant temp</td>
<td>PID</td>
<td>450/55</td>
<td>1.64/21.84</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>Fuzzy-PID</td>
<td>344/70</td>
<td>1.4/14.62</td>
<td></td>
</tr>
</tbody>
</table>

¹Simulation, ²Insulated gate bipolar transistor, ³Sintered porous copper wick, ⁴Experimental
3. Experimental facilities

To generate measured data for the thermal management system shown in Figure 1, an experimental test rig has been used. The test rig comprises a ‘plant’, involving an instrumented test-piece, a shaker, a controller, and a data acquisition system (DAQ). The pipework, the plant, test-piece, control unit, and DAQ system, are shown in Figure 6, and are explained in the following subsections.

3.1. Spray rig pipework

The cooling loop in Figure 6 contains the following components: a system of pipework (1/8 to 1/2 inch stainless steel pipes and compression fittings), a condenser (Denso RDP 583), a heater module (2 x 345 W band-heaters), a miniature heat exchanger, a pump (Micropumps MGD100P), K and T-Type thermocouples, two digital pressure transducers (Omega PXM309), a low-flow turbine meter (Omega FLR1009ST-D), and a condenser and separator tank. Starting from the pump (No. 1 in Figure 1), it circulates de-ionized water through the piping system. The flow meter (No. 16) reads the real-time flow rate measurements. The 345 W feed heater (No. 2) heats the water supply to the nozzle inlet (No. 4) just before the test unit. Silicone rubber flexible pipework carries water to and from the test-piece. The vent (No. 5) and a drain (No. 6) respectively transfer evaporated and accumulated droplets inside the test chamber (No. 13), into the condenser (No. 9) and the separator tank (No. 11). Then, water (from condenser and the tank), flows down into the condenser tank. A header tank (No. 8) is fitted at the highest point in the circuit to remove air and provide a positive feed. To complete the cooling cycle, the feed water cooler (No. 12) cools the condensate to a temperature below the operating maximum for the pump (100 °C).

Fig. 6. Experimental test rig diagram with flat-surface test-piece, DAQ, and control unit.
3.2. A flat-surface test piece.

The test piece is positioned on top of a mechanical shaker (Bruel & Kjaer V555) flat-surface. The purpose of the flat-surface test piece shown in Figure 7, is to make it possible to investigate the capability of surface temperature control, subject to heat flux requirements, first without, then with mechanical vibration.

Figure 7 shows the flat surface test-piece located in a cylindrical chamber, attached by a shaft, to the mechanical shaker. The test-piece and nozzle inside the chamber (fastened by two long bolts) can be shaken at different amplitudes and frequencies. The smooth copper test-piece, of 2 cm diameter, with thermal conductivity of 385 $Wm/K$, is heated electrically by six cylindrical Watlow 250W cartridge heaters vertically located in designated cylindrical holes inside the copper heater block. This is designed to provide strongly one-dimensional, heat conduction to the test-piece. Three T-Type thermocouples are used to measure temperature, the first of which is located 1 mm under the coolant surface of the test chamber, followed by two more thermocouples located evenly 5.5 mm below each other. These are to enable heat flux measurement. A shroud is also placed around the test-piece disc to guide the liquid film to the bottom of the chamber and out to drainage. A full-cone UNIJET® nozzle (TG tip Type) is located on top of the test chamber. All the joints and bolts are sealed with (LOCTITE® 5920™) copper paste which is able to intermittently stand a maximum temperature of 350°C.

![Diagram of test piece and nozzle](image_url)

Fig. 7. Flat surface test-piece: (a) Chamber cross-section view, (b) From top to bottom, exploded view of UniJet® TG Nozzle, flat surface test piece and heater block (based on [23]).
3.3. Data acquisition system and control unit

The main components of the National Instruments data acquisition system are listed in Table 3. As shown in Figure 6, the main processor, an NI cRIO-9035 (No. 17), is connected through its modules to the pump (A), the flow meter (B), an accelerometer (C), the pre-Heater (D), cartridge heaters (E), test-piece thermocouples (F), and a host computer via ethernet. In the host computer, FPGA and LabVIEW programming are configured for DAQ and control purposes, and a human-machine interface unit. Pump flow-rate (in the Micropumps MGD100P controller), cartridge heaters (Watlow 250W), and the feed heaters are operated using 0 to 5 V control signals. These control signals are fed from an NI9264 module in the cRIO-9035 NI hardware. The flow meter, and pressure transducers signals, are collected by an NI 9205 unit. Separate power supplies are used for the pressure transducers, flow meter, and pump. Two (EVR-25BF) power regulators are used to obtain the 0 to 5 V signals (from an NI 9264) and convert them to the required power for the feed and the cartridge heaters. Thus, the test-piece receives constant and variable heat loads through the cartridge heater for different control scenarios. The feed heater has a PID temperature controller to tackle the thermal inertia of the coolant feed. Therefore, the PID controller keeps the system stable, in terms of staying within the evaporative phase, even during rapid changes in pump speed (e.g. doubling the flow rate will only cause a temperature difference of ±5 °C).

<table>
<thead>
<tr>
<th>Module</th>
<th>Description</th>
<th>Absolute accuracy</th>
</tr>
</thead>
<tbody>
<tr>
<td>NI cRIO-9035</td>
<td>Main processor, Control algorithm</td>
<td>-</td>
</tr>
<tr>
<td>NI 9205</td>
<td>Analog inputs – 16 Bit – 32 Channels (Differential and Single-Ended)</td>
<td>4.9 mV</td>
</tr>
<tr>
<td>NI 9212</td>
<td>Thermocouple module – 24 Bit – 8 Channels</td>
<td>-</td>
</tr>
<tr>
<td>NI 9230</td>
<td>AI 24 Bit – 3 Channels</td>
<td>-</td>
</tr>
<tr>
<td>NI 9264</td>
<td>AO 16 Bit – 16 Channels</td>
<td>17 mV</td>
</tr>
<tr>
<td>NI 9472</td>
<td>DO 24 V – 8 Channels</td>
<td>-</td>
</tr>
</tbody>
</table>

The DAQ and control system unit are configured to provide flexibility and high precision - which is quantifiable by following NI procedures (for which the system accuracy relative to the input, is quantified to be 0.177%, with example results shown in Table 5). In addition, the control unit hosts the desired control algorithms. Temperature measurements from the thermocouples (located 1 mm under the coolant surfaces in the test-piece) are used as a state variable for the PID controller in the FPGA CRI0-9035 hardware to control the pump voltage. The amplitude and frequency of the shaker is controlled using a Feedback Instruments FG600 signal generator with a pure sine wave option. A Piezotronics PCB A 353B15 accelerometer (10.27 mV/g, 1 Hz - 10 kHz) is attached to the bottom of the drive-shaft, bolted to the shaker head. Corresponding acceleration signals were measured by the NI 9230 sound and vibration input module. This enables surface temperature control of the actual evaporative spray cooling system to be achieved during experimentation in the presence of vibration.

4. Experimental test procedure and data reduction

Table 4 gives the test and operating conditions for the flat test-piece. All the tests were undertaken at atmospheric pressure for both static conditions (without vibration) and dynamic conditions (with vibration). Two harmonic dynamic cases have been considered: a low amplitude, high-frequency test
at \(a=0.02\) mm and \(f=400\) Hz, and a high-amplitude, low frequency test at \(a=12\) mm and \(f=1.9\) Hz. These two cases are based on real vehicle test conditions [38], and had the highest heat flux deviations from the static case in steady-state conditions (a vibration amplitude of 12 mm is typical of vehicle-mounted hardware).

<table>
<thead>
<tr>
<th>(P_{\text{chamber}}) (bar)</th>
<th>(v) (ml/min)</th>
<th>(T_{\text{coolant}}) (°C)</th>
<th>(H) (mm)</th>
<th>Static</th>
<th>(f) (Hz)</th>
<th>(a) (mm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>60-225</td>
<td>80</td>
<td>17</td>
<td>✓</td>
<td>1.9</td>
<td>✓</td>
</tr>
</tbody>
</table>

### 4.1. Heat flux measurement

The heat flux was calculated using a numerical solution of the one-dimensional transient heat conduction equation. The Fourier one-dimensional transient conduction equation is as follows:

\[
\frac{\partial^2 T}{\partial x^2} = \frac{1}{\alpha} \frac{\partial T}{\partial t} \tag{16}
\]

where \(T\), \(x\), \(\alpha = \frac{k}{\rho C_p}\), \(t\) are respectively temperature, spatial position, thermal diffusivity, and time. For copper, density \(\rho\) and specific heat \(C_p\) are 8900 kg/m\(^3\) and 395 J/kg K. The finite difference method is used to approximate the temperature at different points across the thickness of the test piece by using discrete time steps. To achieve this, the material is discretized spatially into nodes. The spatial derivative on the LHS, and the temporal derivative on the RHS of equation (16) can be approximated by appropriate differences:

\[
\frac{T_{(i+1)}^{(n)} - 2T_{(i)}^{(n)} + T_{(i-1)}^{(n)}}{\delta x^2} = \frac{T_{(i)}^{(n+1)} - T_{(i)}^{(n)}}{\alpha \delta t} \tag{17}
\]

where subscript ‘\(i\)’ and superscript ‘\(n\)’ respectively refer to the current grid-point at time step. Grid points: ‘\(i-1\)’ and ‘\(i+1\)’ are those preceding and following grid-point ‘\(i\)’. The time index ‘\(n+1\)’ stands for the following time step. To obtain the temperature at the interior grid points, equation (17) is rearranged as follows:

\[
T_{(i)}^{(n+1)} = \delta Fo. T_{(i+1)}^{(n)} + (1 - 2\delta Fo)T_{(i)}^{(n)} + \delta Fo. T_{(i-1)}^{(n)} \tag{18}
\]

where \(\delta Fo = \frac{\alpha \delta t}{\delta x^2}\) is the Fourier number in which \(\delta t\) is the time step size, and \(\delta x\) is the distance between the grid points. Numerical stability criteria must be satisfied to achieve accurate results. The stability criteria are:

\[
\delta Fo(1 + \delta Bi) < 0.5 \tag{19}
\]
and
\[
\delta t < \frac{0.5 \delta x^2}{a(1 + \delta x h / k)}
\]

in which \( \delta Bi = \delta x h / k \) is the Biot number. To meet the stability criteria, the distance between the first and third thermocouple is split into 22 nodes. Thus, the distance \( \delta x \), between grid points, is \( 5.24 \times 10^{-4} \) m. The time-step \( \delta t \) is set as 0.0004 s. The time-step is much smaller than the sampling rate, so the boundary temperatures (i.e. T1 and T3) must be interpolated between the measured temperatures. To interpolate the measured temperatures, piecewise polynomials are used.

Finally, the surface temperature gradient \( \nabla T = \frac{\partial T}{\partial x} \), needed for the transient heat flux calculation, is obtained by a second-order backward difference scheme as follows:
\[
\frac{\partial T}{\partial x} \approx \frac{3T_i - 4T_{i-1} + T_{i-2}}{2 \delta x}
\]

Table 5 gives the uncertainties in measured and derived quantities. To determine the uncertainty in the calculation of heat flux, a method by Moffat [63] has been used. The highest expected uncertainty in the calculation of heat flux, at maximum experimentally-measured data point of 1.96 MW/m\(^2\) is found to be 2.6%.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Uncertainty (%)</th>
<th>Units</th>
</tr>
</thead>
<tbody>
<tr>
<td>Thermocouples</td>
<td>±0.4</td>
<td>°C</td>
</tr>
<tr>
<td>Volumetric flow rate</td>
<td>±0.6 (of full scale)</td>
<td>ml/min</td>
</tr>
<tr>
<td>Accelerometer frequency</td>
<td>±5</td>
<td>Hz</td>
</tr>
<tr>
<td>Pressure transducers</td>
<td>±0.25</td>
<td>bar</td>
</tr>
<tr>
<td>Diameter</td>
<td>±1</td>
<td>mm</td>
</tr>
<tr>
<td>Length</td>
<td>±1</td>
<td>mm</td>
</tr>
<tr>
<td>Heat flux</td>
<td>±2.6</td>
<td>MW/m(^2)</td>
</tr>
</tbody>
</table>

### 4.2. Setting operating conditions and control architecture

Safety and efficacy are of prime importance in thermal management. The question is whether the operating parameters and control algorithm is safe to implement in the hardware? Some minor deviations from desired setpoints can be acceptable, but the control scheme must not harm the hardware. Efficacy measures how effective the thermal management is in fulfilling the setpoints for variable loads.

To secure the safety and test the efficacy, the thermal management system is put through two phases of trials. ‘Phase one’ is the safety trial. Using previously fitted correlations, operating parameters, setpoints, and thermal requirements must be established. Moreover, since disturbances to the hardware take the form of vibrations of different amplitudes and frequencies (with resulting differences in acceleration), a dynamic correlation containing vibrational parameters and acceleration,
must be considered in the thermal management system. In this regard, the following correlation models fitted to the UNIJET® nozzle data for static and dynamic cases (constructed in [38] using the Generalised Buckingham Pi method [45]) are used in the thermal management system i.e.:

\[ Bo = 3.9 \times 10^{-2} Ja^{1.558} \left( \frac{\rho_l^2 \sigma v}{\mu_l^3} \right)^{0.501} \]  \hspace{1cm} \text{Static} \hspace{1cm} (22)

and

\[ Bo = 1.2 \times 10^{-2} Ja^{1.392} \left( \frac{\rho_l^2 \sigma v}{\mu_l^3} \right)^{0.981} Re_v^{-0.987} \left( \frac{a}{H} \right)^{0.483} Ac^{0.499} \]  \hspace{1cm} \text{Dynamic} \hspace{1cm} (23)

where \( Bo \) is Boiling number (\( qH/\mu_l h_{fg} \)). \( Re_v = \rho_l a \omega d_H/\mu_l \) is the Vibrational Reynolds Number. \( Ac = \omega^2 a/g \) is the Dimensionless Acceleration Number. \( Ja \) is Jacob number (\( Ja = C_l \Delta T/h_{fg} \)), which considers both convective and boiling heat transfer.

The correlation models have been constructed using Generalized Buckingham \( \pi \)-theorem [45]. To develop a functional form, independent key parameters (including surface roughness, subcooling degree, nozzle-to-surface distance etc.) identified for the static conditions in the literature have been considered. However, the next step in generalized \( \Pi \)-Theorem is to define invariant parameters according to the test plan. In this study the surface roughness, chamber pressure, the degree of subcooling and nozzle-to-surface distance are constant and therefore are invariant parameters. The functional forms used for this study (equations (22) and (23)) therefore do not include \( \Pi \)-terms associated with the invariant parameters which is a result of Generalized Buckingham \( \pi \)-theorem. The range of applicability of the correlation is available in Table 6 as follows:

<table>
<thead>
<tr>
<th>( P_{chamber} ) (bar)</th>
<th>( v ) (ml/min)</th>
<th>( T_{coolant} ) (°C)</th>
<th>( H ) (mm)</th>
<th>Static (without vibration)</th>
<th>Dynamic (with vibration)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>60-225</td>
<td>80</td>
<td>17-21</td>
<td>✓</td>
<td>0.02-12</td>
</tr>
</tbody>
</table>

The thermal management system consults a step-by-step algorithm shown in Figure 8.

Because surface conditions affect the critical heat flux and heat transfer coefficient, a different control strategy would be needed to control heat transfer and temperature for different surface conditions. The experimental measurements reported here have only used one test-piece with a machined-surface (which is observed as being ‘smooth’, although the surface-roughness was not actually measured). To identify the control parameters needed to take account of varying surface conditions, a new set of experimental measurements would be needed. These measurements would need to be taken from several different test-pieces with the same geometry and nozzle parameters, but with differing surface conditions. Without such measurements, understanding the way in which PID controller gains are affected by spray mass flow-rate for example, can only be speculated. There are in fact contradictory findings published in respect of surface roughness and spray parameters. The review by Liang and Mudawar [12], notes a contradiction stemming from the absence of any form of examination involving the relevant parameters affecting complex spray mechanisms during droplet impact. The review recommends careful and systematic experimental measurements be taken to examine the influence of key parameters on heat transfer.'
No hysteretic heat flux $q$ versus excess temperature DT measurements have been observed in the measurements reported. This probably stems from the low mass flux of the spray as a function of heat input. But as reported in the work by Bostanci et al. [17], hysteresis has been widely observed in the nucleate boiling literature involving very wetting liquids like R-113 and FC-72. However, the water used in the current research has a rather high cohesive forces owing to hydrogen bonding, and is therefore not as good as R-113 and FC-72 at wetting surfaces. But if hysteresis is observed, the best control approach might be to advance or delay the controller output during the rise and fall time. Evidence of hysteresis in measured heat flux during surface cooling and heating can be found in the publication by Jose et al. [47] for a test piece with a three-dimensional surface (i.e. figures 16-19 in [47]). But using the same correlations as currently reported, a tuned-controller with different gains, is able to meet the temperature control requirements in the presence of hysteresis.

**Fig. 8. Procedure for setting the operating conditions (Phase one: Safety test).**

First, the initial operating conditions to be applied during each experiment, are set by the operator in a human-machine interface (HMI). A graphical user interface (GUI) is configured in LabVIEW. The iterative
part using the correlations, examines the boundaries, taking into consideration the applicability of the spray system in maintaining nucleate boiling.

Choosing suitable ‘setpoints’ for the spray cooling system is as important and challenging as the control task [64]. The desired parameters (such as surface temperature and flow rate) and the requirements (e.g. heat flux), dependent on each other in a complex way. In addition, several dynamic cases produce substantially different results from the static results (heat flux and excess temperature). Therefore, the flow rates and spray patterns which result in effective cooling at the same setpoints are significantly different. The trade-off between setpoints, boundaries, spray specifications, and the specified requirements, have largely been determined by trial and error and prior experience [38].

Once the desired operating parameters and suitable boundaries are successfully fixed, the thermal management system ‘moves to ‘Phase Two’. Phase two is the efficacy trial which demonstrates how well the control scheme works in tracking the trajectories induced by variable heat loads. This is achieved by an automatic control loop, and by a variable heat load generator. Figure 9 shows the sensor-based control diagram containing two closed-loop PID controllers, a setpoint generator, the HMI, the test section, and the CompactRIO (NI cRIO-9035) with the FPGA containing I/O modules.

Fig. 9. Sensor-based control system diagram.
As can be seen in Figure 9, the flat test piece is exposed to displacement vibration of the form \( X = \text{asin}(\omega t) \) generated by the shaker. The PID controller tuning process is a manual activity which will be further explained in the results and discussion section. With a real-time processor, the FPGA in the CompactRIO sends the control signals through a PCI bus, and receives sensor data. This data requires high-speed logic and precise timing (e.g. the control signals to the power regulators for the cartridge heaters, band heaters, and the pump). Simultaneously, the human-machine interface (with a GUI) runs on the host computer to allow monitoring of the system state, and setting of the operating parameters.

5. Controller tuning and thermal management effectiveness in static and dynamic cases

The results for the flat test piece are given in the following three subsections. First, the manual tuning process of the surface temperature controller under static conditions, and the efficacy and robustness of the tuned controller are assessed. Second, the impact of different controller gains on the control and thermal performance are discussed. In addition, the effectiveness of the thermal management system in saving pump energy is assessed in terms of a coefficient of performance (COP). Finally, the statically-tuned controller is used for dynamic conditions. The thermal management system has been tested in dynamic cases with the highest departure from static results in order to check the efficacy and suitability of the proposed thermal management system.

5.1. Control system tuning and thermal management

Tuning of the surface temperature is a manual process. This has been achieved by trial-and-error using the closed-loop (feedback) control system plant hardware (explained in section 3.2). Figure 10 shows the temperature tracking and pump response to the applied setpoints, plus the induced heat flux. The temperature setpoints are typical requirements, and the chosen operating ranges are based on the thermal management needed for range-extended hybrid electric vehicle [28], or an all-electric vehicle cooling systems and electronic components (i.e. between 110°C to 135°C with the heat flux varying between 0.2 to 1.96 MW/m^2). After several trial-and-error attempts, with a negative feedback PID controller, the most acceptable gains which sufficiently minimise the overshoot and undershoot, and gave rise to a reasonably quick settling time during the same trajectories, were found to be P=2.1, I=0.08, and D=0.001 respectively. The settling time plays a vital role in thermal performance and energy efficiency of an internal combustion engine (ICE) [22] in range-extended hybrid electric vehicles. The thermal management systems required for most of the electronic components needs to have a high response (less than 100 seconds). The test rig and pipework used however, suffered from large thermal inertia. For this reason, the settling time for a change in temperature from the two setpoints of 110°C to 135°C in Figure 10a was 155 seconds. (Test rigs of miniature scale are suitable for the investigation of electronic component cooling).

Although the current focus is about testing the safety and efficacy of a spray-based thermal management system, several practical modifications that could be useful in reducing the settling time were also examined in a further trial-and-error examination. It was found that the best option is to use a negative proportional gain (P=−2.1 in Figure 10b). It also found that a larger integral gain (comparing with Figure 10a) is beneficial but keeping the same derivative gain. The negative proportional control gain would normally act to make the error worse (during the rise time). However, after some time (during the settling time), the integral action would dominate and correct the system. This effectively
works to reduce the settling time from 155 s to 98 s (a very significant 37% improvement with a negligible 1 s increase in settling time in cooling, during setpoints from 135°C to 110°C). Having a larger integral gain (0.38 compared to 0.08 in Figure 10a) finally adjusts the controller to maintain a negative feedback loop. Although a negative proportional controller is in general unusual, since it can suffer from diminished phase-margin, and more severe overshoot, it can work well in certain circumstances. Based on the desirable cost benefit approach in developing the thermal management system proposed here, it has been demonstrated to be superior to a controller with a set of positive PID gains.

Fig. 10. Static case temperature tracking and the pump response to the applied setpoints: for (a) a positive proportional gain of $P=2.1$ with $I=0.08$ and $D=0.001$; (b) a negative proportional $P=-2.1$ with a large integral gain of $I=0.38$ and $D=0.001$; (c) and (d) Associated heat fluxes and COP measurements.

An explanation of why and how this type of controller is workable is evident from the pump response shown in Figure 10. For the regular PID controller (Figure 10a), the pump works at low speed and provides a flow rate of around 110 ml/min during the rise time. As soon as the error becomes positive,
the flow-rate surges to its highest possible value (around 230 ml/min) to address the overshoot. Conversely, for negative proportional PID gain (Figure 10b), the flow-rate peaks at around 220 ml/min during the rise time, then gradually drops to around 160 ml/min to reach the setpoint (135 °C). Based on previous experience in developing the correlations [38], it was noticed that there is a reverse trend when the splash diameter of the spray cone exceeds the area of the target surface. From that specific flow-rate (which can be deemed to be a ‘threshold’), the volumetric flux reduces. Therefore, the heat transfer rate reduces accordingly - a hypothesis supported by Schwarzkopf et al. [65]. Note: it is actually stated on the UNIJET™ nozzle manufacturer’s data sheet, that the spray angle varies from 50° to 61°, which means from the threshold, the heat transfer trend is the opposite: the higher the flow-rate, the lower the heat transfer rate. The threshold flow-rate for the current experimental facility is around 200 ml/min. The same reverse trend has been observed by Zhang et al [42]. Thus, for the negative proportional PID controller, it was concluded that the shift between these two identified trends in heat transfer during the rise-time between 90 and 110 seconds (and 20 seconds of reverse trend) could have resulted in a further increase in the surface temperature and therefore a reduced settling time. This reverse trend above 200 ml/min is supported by Figure 10b when a steady-state error of 1 °C has occurred after passing the 200 ml/min. It should be noted that the differences between heat transfer rates before and after this threshold are subtle. However, the difference is of practical significance to be used for the purpose of adjusting the settling time to approach a more realistic time constant.

The proposed controller initially raised several important questions. The first question concerns why a lower flow-rate has not been considered by setting a minimum margin in order to have a smaller heat transfer rate to do the same task (i.e. a reduced settling time). The answer is: because the UNIJET nozzle is not able to produce a full cone spray at low mass flow rates owing to the lower nozzle pressure differences. But imperfections in the spray flow field could change the expected trends in the cooling regimes. For example, a very low flow rate may not be able to maintain nucleate boiling for the required high heat fluxes and would therefore raise safety issues. For this reason, it has not been tested. In addition, a low flow-rate (e.g. around 50 ml/min, shown in figure 10c around 65 seconds) cannot achieve the required high heat fluxes. Moreover, a negative proportional PID gain which resulted in higher (rather than lower) volumetric flow rate during the rise time, would suppress the effects of vibration frequency owing to higher droplet velocity, which is consistent with the experimental results in [61]. This means during the rise time, that a negative-proportional gain PID controller could in theory be more reliable and adaptive in dynamic conditions including high-frequency cases. The dynamic results in section 4.3 will support the hypothesis.

The second question concerns other possible control solutions such as controlling the nozzle-to-surface distance, changing the degrees of subcooling [67], introducing a pulsating spray flow field [68, 69], to control either volumetric flux or flow-rate. Control of the nozzle-to-surface distance is not fast enough owing to the need for a mechanical mechanism in the form of a micro positioning slide [66]. The inertia of the feed heaters does not allow for rapid change in the inlet temperature required to control the degrees of subcooling. Also, the calculation of the volumetric flux to reflect a proper trend of the effective spray flow-rate impinging on the target surface, is not feasible in a design of a cooling system in which a minimum number of sensors is required. Even using existing empirical correlations for volumetric flux [70] such as given by:
\[ \bar{v} = \frac{v}{\pi \left( H \tan\left( \frac{\theta}{2} \right) \right)^2} \]  

(24)

needs a real time spray angle (\(\theta\)) measurement which adds uncertainty to the plant. These issues leave room for improvement in the design of the test facility. For example, a high response speed solenoid valve would make possible a pulsating spray cooling module. The empirical correlations could be updated using Strouhal number (\(St = \frac{f_d}{v}\)), to include the effect of a pulsed-spray flow field which would also benefit from a variable pump speed module. Overall, using a PID controller with a negative-proportional gain, although unusual, is still workable and fulfills the aims of this study. The efficacy and robustness of a PID controller with negative-proportional gain, both without and with vibration is now examined.

5.2. The effect of PID controller gain changes without test-piece vibration: static cases

The effect of proportion gain changes

To better understand the effects of PID gain changes on the thermal performance and energy efficiency, a coefficient of performance (COP) has been included in figure 10c and 10d for the heat fluxes. The COP is the ratio of the heat removed to the pumping power given as:

\[ COP = \frac{Q = Q_d a^2 / 4}{VI} \]  

(25)

where, Q, V, and I are respectively: the heat removal by spray cooling, the voltage and the current supplied to the pump. To smooth out noisy fluctuations in the COP, a 10-sample moving average is used. As can be seen in Figure 10d, a negative proportional gain not only achieves a higher heat transfer rate, but also gives rise to an average COP=22.56 which is a 10.5% increase in the performance compared with a regular PID controller (Figure 10c).

Figure 11 shows results for two more sets of data obtained during the tuning process. First, up to 450 s, with a slightly larger integral gain (I=0.48), and after 450 s, with a smaller proportional gain (P=-2). A further decrease in the settling time was observed with the higher integral gain of I=0.48. This was caused by the integral gain counteracting the negative proportional gain after a longer time. Consequently, the spray flow-rate has been maintained in the reverse trend for a longer time. As was explained in Section 5.1, this is a reason for the faster settling time.

The lower heat transfer rate could be supported by the 1 °C steady-state error (at around 130 s) while the pump was operating at around 215 ml/min. A disadvantage of this, is that after the integral gain was compensated for, the effect of a negative proportional gain, the thermal management system was unstable and fluctuated between two trends (at 170 s in Figure 11a). This resulted in a ‘bouncing’ response, although a derivative gain twice the previous magnitude (i.e. D=0.002), was able to stabilize these fluctuations. The result for a smaller proportional gain (P=-2) is consistent. Since the integral gain is now dominant, the fluctuations occur earlier producing a longer settling time (but still faster than a positive proportional gain). But these PID settings are not recommended because the COP values of 22.25 and 22.52 in Figure 11b are not promising.
Fig. 11. Static case temperature response to the applied setpoints (between 110 and 135 °C) and load steps between 0.2 and 1.95 MW/m²: (a) Temperature tracking and the pump response for proportional gains of -2.1 and -2 with I=0.48 and D=0.002, (b) Associated heat flux and COP measurements.
Fig. 12. Static case temperature response to the applied setpoints (between 110 and 135 °C) and load steps between 0.2 and 1.8 MW/m²: (a) Temperature tracking and the pump response for a larger integral gain of 0.7; (b) Associated heat flux and COP measurements.
**The effect of Integral gain changes**

Figure 12 shows the effect of changing the integral gain in another trial and error experiment. The PID controller has the larger integral gain value of 0.7 but the same proportional and derivative gains as Figure 11a. This was to reduce the fluctuations in the pump response to the temperature trajectory. In comparison to Figure 11a, the oscillations in the flow rate are reduced. The settling time increased since on average during the rise time, the flow rate was in a reverse trend (i.e. above 200 ml/min) for a shorter time. The overshoot was slightly better, but the undershoot was worse (2.4°C). A disadvantage of using a larger integral gain is the average COP, as is shown in Figure 12b, was less than those with a smaller integral gain of 0.48. As soon as the influence of the PID gains was discovered through the tuning process, the thermal management system was challenged in *dynamic* cases. The next section examines these results.

**5.3. The effect of PID controller gain changes with test-piece vibration: dynamic cases**

Experimental evidence shows that vibration generally leads to an attenuation of convective heat transfer rate. As a consequence, when the test-piece is made to vibrate, the controller instructs the pump to compensate for the deterioration in the rate of heat transfer. This consumes more power and thus reduces the COP for the *dynamic* cases below those of *static* conditions. This section investigates the performance of the controller with vibration, in particular, the effect that different gains have on heat flux.

Figure 13 shows a data set during the tuning process for large amplitude vibration a=12 mm at a frequency $f=1.9$ Hz. On the left-hand side of Figure 13 up to 840 seconds, a trial involving an unsuccessful set of PID controller gains with is compared with a PID controller with larger gains. From 840 seconds onwards, a much smaller set of gains was also considered to allow comparison. It can be seen that there are oscillations in the temperature, also the undulating pump response for the higher gains clearly indicates the significant influence of different gains. Having the smaller PID controller gain values of $P=1$, $I=0.4$, and $D=0.001$, smooths both the pump response and the undulating temperature. Furthermore, the settling time is also improved. The problem with small gains is the 1.8°C steady-state error produced, also the longer return time.

Taking into consideration the detrimental gain effect for dynamic cases, plus the need to test the adaptive capability of a PID controller tuned for static cases, a PID controller with gain values: $P=-2$, $I=0.7$, and $D=0.002$ was chosen for the results in Figure 14. The gains in Figure 14 were the same for the static case in Figure 12. As expected, the rise time is improved compared with the static case owing to the impeding effect of vibration. Moreover, a lower average COP value of 18.06 in Figure 14b (compared to 19.80 for the static case) indicates increased power consumption for the same heat flux (up to 1.8 MW/m²). Beyond t=400 s, a PI controller with gains: $P=-2$, $I=0.7$, $D=0$ (rather than $D=0.002$) is tested to establish whether the derivative gain is genuinely having any influence. Broadly speaking, derivative gain in a PID controller has the effect of reducing overshoot. It is evident that the differences between overshoot and undershoot, with and without the derivative gain, actually justifies derivative control. Furthermore, the subtle difference in COP, being smaller for the PID, suggests the pump is managed by the derivative gain.
Fig. 13. Temperature response induced by applying temperature setpoints and load steps for large-amplitude vibration of 12 mm: (a) Temperature tracking and pump response for a PID with large gains following by a smaller gain PID, (b) Heat flux disturbances and real-time COP measurements.
Fig. 14. Temperature response induced by applying setpoints (between 110 and 135°C) and load steps (up to 1.8 MW/m²) for large-amplitude vibration of 12 mm: (a) Temperature tracking and pump response for a PID and a PI controller, (b) Heat flux disturbances and real-time COP measurements.
Fig. 15. Temperature response induced by applying setpoints (between 110 and 135 °C) and load steps for high-frequency vibration of 400 Hz: (a) Temperature tracking and pump response for PI and PID controllers, (b) Heat flux disturbances and real-time COP measurements.
Figure 15 shows a dynamic case at high frequency. The test-piece was shaken at an amplitude of \(a=0.02\) mm and frequency \(f=400\) Hz. The same PI and PID controller gains for the large amplitude case appear to successfully manage the pump speed to enable it to achieve surface temperature tracking. The results in Figure 15 show a reasonable rise time, overshoot, and undershoot, and that successful thermal management for loads between 0.2 MW/m\(^2\) to 1.8 MW/m\(^2\) was achieved. Also, the effect of derivative gain was consistent with the previous explanation (i.e. error reduction and less settling time for a PID controller). Similarly, there is a corresponding increase in overshoot, undershoot, and settling time of 0.3 - 0.4 °C, 0.4 - 0.5 °C and 16 - 20s, for the PI controller. As expected, the COP for the PID controller, shown in Figure 15b, was lower than for the PI controller. Finally, repeatability of the results was examined by introducing an extra step load for the PID controller. The resulting temperature differences were 0.1 °C for overshoot and undershoot, and a 3s difference in the rise and settling times. The coolant flow rate during the repeatability test was similar to the previous load step, suggesting that overall, the experimental measurements were repeatable.

6. Conclusions
A thermal management system, using evaporative spray cooling, has been designed, built, and experimentally-tested to assess its ability to control the temperature of heat-generating hardware subject to heat flux requirements. To model the system, an experimentally calibrated dynamic correlation model has been used to represent the physics of evaporative spray cooling of a flat test-piece exposed to vibration. To enable stability analysis of the thermal management system using a PID controller, the nonlinear correlation model has been replaced by an optimum 1st-order linear model obtained by solving an appropriate Wiener-Hopf equation. This allows practical stability assessment of closed-loop temperature control for particular choices of PID controller gains. To verify the predicted control system performance via simulation, experimental measurements appropriate to automotive vehicle component applications, includes large-amplitude, low frequency vibration, at 12 mm and 1.9 Hz, and at low amplitude, high-frequency vibration, at 0.02 mm and 400 Hz. The effects of different PID controller gains on the thermal management system performance under static and dynamic conditions, has been assessed using a coefficient of performance (COP), defined as the ratio of heat power removal to the required pumping power. The main conclusions of the study are:

1. Trial-and-error tuning of the hardware suggests that using the best PID gains for a closed-loop control system can give very reasonable thermal management performance in terms of temperature control, percentage overshoot, settling time, and coefficient of performance (COP).
2. A negative proportional-gain PID controller, with positive integral and derivative gains, was found to be stable, and proved to be superior in terms of performance, to a PID controller with all positive gains. In fact, the use of this negative proportional-gain PID controller gave the largest coefficient of performance (COP) at 22.56, and a reduction of 37% in the settling time compared to a PID controller with positive gains.
3. The best negative proportional-gain PID controller for the static case also gave the best performance in dynamic cases confirming the robustness of the controller.
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References


Nomenclature

\( \bar{v} \)  average volumetric spray flux, \( \bar{v} = \frac{v}{\pi (H \tan \frac{\alpha}{2})^2} \) (m\(^3\)s\(^{-1}\)/m\(^2\))

A  surface area (m\(^2\))

a  Amplitude of vibration (m)

\( Ac \)  dimensionless acceleration, \( Ac = \omega^2 a / g \) (-)

Bo  Boiling number, \( Bo = \frac{qH}{\mu_l h_{fg}} \) (-)

C  specific heat (kJ/kg k)

d  diameter (mm)

D  derivative gain

e  error

f  Frequency (Hz)

h  specific enthalpy (kJ/kg)

H  height (m)

I  Integral gain

Ja  Jakob number, \( Ja = C_l \Delta T / h_{fg} \) (-)

k  thermal conductivity (W/m K)

P  pressure (kPa)

P  proportional gain

q  heat flux (kW/m\(^2\))

Re  Reynolds number, \( Re_v = \frac{\rho_l a \omega d_H}{\mu_l} \) (-)

St  Strouhal number, \( St = \frac{fd}{v} \) (-)

T  temperature (K)

t  Time (s)

V  voltage
Weber number, $We = \rho_i (u_m^2 \text{ or } \bar{v}^2) d_32 / \sigma$ (-)

$I$ current

$Q$ heat (kW)

$x$ position (m)

$\delta Bi$ Biot number, $Bi = x h / k$ (-)

Greek symbols

$\Delta T$ temperature difference (K)

$v$ volumetric flow rate ($m^3/s$)

$\alpha$ thermal diffusivity $\alpha = \frac{k}{\rho c_p} (m^2/s)$

$\theta$ temperature ratio

$\mu$ dynamic viscosity ($Kg/m \cdot s$)

$\rho$ density ($kg/m^3$)

$\sigma$ surface tension (N/m)

$\omega$ angular velocity (Hz)

$\vartheta$ spray cone angle

Subscripts/superscripts

$at$ atmospheric

$c$ coolant

$ch$ chamber

$f$ liquid phase

$g$ vapour phase and gravitational force

$H$ heating surface

$i$ grid point

$l$ liquid

$n$ time step

$ref$ reference temperature

$s$ surface

$sat$ saturation

$sub$ subcooling

$V$ vibration

$w$ Wall

Acronyms

$CHF$ critical heat flux

$COP$ coefficient of performance

$FIR$ finite impulse response

$GUI$ graphical user interface

$HMI$ human-machine interface

$ICE$ internal combustion engine

$MSE$ mean square error

$PID$ proportional integral derivative
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