An experimentally-verified temperature control simulation model for spray evaporative cooling of vibrating powertrain parts

Article (Accepted Version)

Thalackottore Jose, J, Sarmadian, A, Dunne, J F, Long, C A, Pirault, J-P and Rouaud, Cedric (2021) An experimentally-verified temperature control simulation model for spray evaporative cooling of vibrating powertrain parts. *International Journal of Heat and Mass Transfer*, 170. a121041 1-21. ISSN 0017-9310

This version is available from Sussex Research Online: <http://sro.sussex.ac.uk/id/eprint/96840/>

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.

AN EXPERIMENTALLY-VERIFIED TEMPERATURE CONTROL SIMULATION MODEL FOR SPRAY EVAPORATIVE COOLING OF VIBRATING POWERTRAIN PARTS

by

J. Thalackottore Jose^a, A Sarmadian^a, J.F. Dunne^{a*}, C.A. Long^a, J-P Pirault^a, and Cedric Rouaud^b

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

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

*^bRicardo plc
Shoreham-by-Sea, West Sussex BN43 5FG, UK*

ABSTRACT

A new simulation capability is presented to enable the performance of a hardware-based temperature control system to be assessed in thermally-managing heat-generating automotive vehicle powertrain parts. Temperature control is assumed to involve spray evaporative cooling of powertrain parts exposed to vibration. Two hypotheses are proposed to enable construction of a practical simulation that is both accurate and computationally efficient. The first is that a dynamic correlation model for single-nozzle spray evaporative cooling of a flat test-piece exposed to vibration, can be used as a reasonable model for multiple-nozzle spray evaporative cooling of component parts with curved cooling surfaces of non-horizontal orientation. The second is that the transient heat diffusion properties of a particular 3-dimensional component can be replaced by a 1-dimensional (1D) equivalence. To test this hypothesis, Finite Element models for two representative parts have been constructed and used to demonstrate the quality of the 1D heat diffusion equivalence, for which a fast Finite Difference solution can be exploited. To test the accuracy of test-piece surface temperature control simulation, an experimental test facility has been built in hardware, in which the temperature of two instrumented test-pieces exposed to vibration (from a shaker) are controlled by spray evaporative cooling. Each test piece is electrically-heated and the hardware control system is configured using PID control, for which appropriate gains are selected. Detailed comparisons of temperature control by hardware and simulation are given for the two test-pieces under static and dynamic conditions. Good agreement is generally obtained between simulated surface temperatures compared with measurements taken from both test-pieces. The paper shows that temperature control of a hardware-based control system using spray evaporative cooling of powertrain parts can be confidently simulated.

1 INTRODUCTION

Effective thermal management is important for electrification of both automotive vehicles [1], and aircraft [2, 3]. The motivation for better automotive thermal management stems from the CO₂ reduction benefits [4] made possible through the use of weight-reduced powertrain component parts, such as lighter power electronic hardware in battery electric vehicles, and lighter, more compact, combustion engines in hybrid electric vehicles [5][6]. The general need to reduce CO₂ emissions is driven by ever more stringent emissions regulations [7, 8]. Electrification is opening-up important new possibilities for both automotive powertrain and aircraft propulsion [9-11]. But to fully realise CO₂ emission reductions, highly optimized cooling systems are required. Conventional liquid-based cooling systems have largely reached their limitations of achievable heat flux levels. In exploring alternative cooling methods, Jafari et al. [12] reviewed evaporative cooling systems for internal combustion engines, to address the heat flux levels needed for highly-boosted engines [13]. Evaporative cooling is also seen as being suitable for thermal management of vehicle power electronics, batteries, and drives, in both electric and hybrid electric vehicles. In fact, it would appear that spray evaporative cooling could offer excellent thermal management potential for a range of temperature sensitive powertrain components because the heat flux is very much higher than single-phase convection [14-21]. And because mass-flow rates are significantly reduced, pumping power is correspondingly reduced.

Several key studies have examined the physics of spray cooling: e.g. a review by Liang and Mudawar on single-phase and nucleate boiling [22]; a review on critical heat flux associated with film boiling and quench curves in [23]; and a review on droplet impact and heat transfer correlation models by Breitenbach et al. [24]. Evaporation of spray droplets is a complex process with some similarities to the well-known pool boiling regimes, such as nucleate boiling, peak heat flux, and unstable transition to film boiling [25]. In the past two decades, efforts have been made to establish how key parameters affect heat flux and surface temperature during cooling. It has been consistently shown that key variables (such as the spray characteristics, pressure, flow-rate, and degree of subcooling) differently influence heat transfer modes. In fact, by varying surface heat load, the dynamic behaviour of even a single droplet impacting onto a hot surface [24], produces different regimes, i.e. from single-phase to nucleate boiling, through the transition to atomization, and finally to film boiling. The current knowledge of these different regimes is largely based upon empirical studies that have examined how dimensionless parameters respond to load changes. Studies of boiling and quench curves for example, show the importance of nucleate boiling in applications where large amounts of heat removal is needed [26], confirming that nucleate boiling has the highest average heat transfer rate.

Returning to the potential for spray evaporative cooling in vehicle electrification, the variation of heat flux when expressed in terms of excess temperature ($T_{\text{wall}} - T_{\text{sat}}$) typically shows several orders of magnitude increase over single-phase convection. Mudawar et al. [27] reported that a feasible thermal management system for hybrid electric vehicles should have the potential to

dissipate heat flux in the range 1.5 - 2 MW/m², while the temperature needs to be kept below the silicon transistor limit (~ 150 °C) and above a lower operating temperature limit. Spray evaporative cooling actually provides a uniform temperature across all cooled surfaces [28], eliminating hot spots. This is important for thermal management of Lithium-ion battery stacks [29] to protect against overheating [30, 31] and fires [32] because exceeding the upper limit risks thermal runaway whereas operating under the lower limit, reduces cell life. It could also be useful for indirect battery pack cooling [33], where pre-cooled air enables operation at an optimum temperature which improves power output, charging rate, and battery life [34, 35].

All applications of spray evaporative cooling involving transport vehicle components operate with some mechanically-induced vibration, known to influence heat transfer [36]. Several sources of vibration can be identified across a range of amplitudes and frequencies. Automotive vehicle bodies vibrate from rolling over rough terrain and from engine block vibration. Aircraft vibrate on the ground rolling over runway roughness, and in the air, from turbulence and engine imbalance. The effect of vibration on heat transfer is evident from two dimensionless groups: *Vibrational Reynolds*, and *Acceleration Number*. The Vibrational Reynolds Number, which is representative of the turbulence resulting from a vibrating surface [37], helps to understand how increased turbulence caused by vibration, can affect heat transfer. Changes in heat transfer resulting from vibration can also be explained by changes to the boundary layer thickness, nucleation sites [38], and wetting-angle [39]. The dimensionless Acceleration Number, which is helpful in assessing the impact of acceleration on spray cooling heat transfer, has been investigated in various ways i.e.: i) using a rotating arm test bench [40]; ii) by examining reduced gravity [41], and iii) by tests on aircraft undergoing parabolic flight [42, 43].

Although spray evaporative cooling offers very high heat transfer rates with small mass flows, there is considerable uncertainty in the ability to control cooling at the required heat flux. In fact, to establish conditions for stable control and assess robustness, and ultimately to tune the controller performance, it is necessary to be able to simulate a spray evaporative cooling control system prior to implementation in hardware. The difficulty in doing this however is that the physics of spray evaporative cooling is computationally too complex to be able to simulate real system behaviour within practical timescales using fundamental thermo-fluid-dynamic principles [13][44]. Therefore, use is generally made of empirical correlation models for the spray heat transfer. But to create a simulation capability for a cooling control system, involving vibrating components, it is not appropriate, or even possible, to construct dynamic correlation models for all possible component-part masses and geometries, and therefore simplification to the approach is needed. In addition, a good approximation is needed to the full 3-dimensional solid-body models used for the component-part transient heat diffusion process otherwise extremely lengthy simulation times result leading to a control simulation capability of no practical value.

This paper proposes just such a simulation approach. The objectives are to demonstrate model simplifications and to verify through experimental measurement the ability of the simulation to

rapidly produce results that emulate control of hardware. Two control simulation comparisons are used in the experimental verification: the first uses a dynamic correlation model for spray evaporative cooling that matches the hardware test-piece, and therefore only tests the approximation to the 3-dimensional transient heat transfer characteristic of the component-part test-piece body. The second comparison tests both the use of a simplified dynamic correlation model coupled with the approximation to the heat transfer characteristic of the component-part.

2. CONTROL SIMULATION MODEL SIMPLIFICATIONS

This section explains in general terms the component-part temperature control challenge, and the high-level requirements needed to create a control simulation. The objective of the control system is to maintain the coolant-side metal temperature at a target value but at the same time ensuring that all the heat removal occurs in the form of latent heat into the vapour. Figure 1 shows a general high-level representation of a powertrain component part, temperature-controlled by a closed-loop feedback control system.

Fig. 1. Temperature control diagram for closed-circuit spray evaporative cooling of a power-train part exposed to displacement vibration of the form $X = a \sin(\omega t)$.

Figure 1 actually shows two circuits. In one, a coolant pump supplies saturated liquid to a nozzle which delivers a full-cone spray of coolant droplets onto hot vibrating surfaces of the component part. Generated vapour is then condensed and returned to the suction side of the coolant pump via a condensate tank. The other circuit in Figure 1, shows the feedback signal for the (cool metal)

surface temperature compared with the target temperature to provide the required error control signal as input to the controller. At this stage, no particular type of control strategy or controller is specified since many different options can be implemented within the created simulation capability.

The requirement of the proposed control simulation capability is to construct a closed-loop feedback control system model that includes a ‘plant’ model that accurately captures the entire process physics of a spray-evaporatively-cooled component part. A practical requirement of these simulation models is to enable computations to be undertaken in realistic timescales that are computationally efficient to enable the simulation to be used to assess performance and undertake tuning. Before explaining why and how spray evaporative cooling correlation models are used and needed, it is necessary to consider more closely the two required high-level simplifications mentioned to make it practically possible to simulate real system behaviour. In general, the first challenge is to make it possible to exploit a single experimentally calibrated correlation model for a single geometry, that can be used to approximate the characteristics of component parts with different surface geometry and orientation. The second challenge is to find a much-simplified equivalent model for the transient heat characteristics of the component part. To meet the control system simulation requirements, these simplifications represent two hypotheses which are best summarised as follows:

1/ It is hypothesised that a single dynamic correlation model calibrated using single-nozzle experimentally-measured data taken from a horizontal flat test-piece can be used to approximate the heat flux on a multiple-nozzle spray evaporatively cooled test-piece with a curved cooling surfaces in arbitrary orientation, and:

2/ It is hypothesised that to estimate particular metal temperatures of the real 3D (three-dimensional) solid-body test-piece under transient thermal conditions, an equivalent 1D cylinder, of appropriate size, can be found which gives very similar transient heat conduction characteristics, thus allowing 1D heat diffusion to be computed within the control simulations needed for tuning of the controller hardware.

A substitute dynamic correlation model

The first question is why a dynamic correlation model is needed. The reason as mentioned, is that the physics of spray evaporative cooling is computationally not yet of practical possibility even without boundary motion. With the additional complexity of boundary motion, the computation cost is several orders of magnitude higher again. Until the computation cost issue is addressed the practical solution is to exploit Dimensional Similarity. A dynamic correlation model has been constructed in [45] using the Generalised Buckingham Pi method [46]. This has been achieved by identifying the dependent variables assuming heat flux conforms to a function:

$$q = \varphi(\rho_l, \rho_v, h_{fg}, \mu_l, C_l, \sigma, T_c, T_w, P_n, P_{ch}, \Delta T_{sat}, \Delta T_{sub}, P_{at}, u_d \text{ or } u_m, \dot{m}, d_{32}, d_H, H, a, \omega, g) \quad (1)$$

where q is Heat Flux, ρ is Density, h_{fg} is Latent heat, μ is dynamic viscosity, C_l is specific heat, σ is surface tension, P is pressure, T is temperature, u is velocity, $\dot{m}$ is Mass flow rate, v is Volumetric flow rate, $\bar{v}$ is Average volumetric spray flux defined in [45] and α is the spray angle, d_{32} is the Sauter mean diameter, d_{05} is the mass or volume median diameter, d_H is Hydraulic diameter of the heating surface, H is the Height from nozzle to surface, a is the vibration Amplitude, ω is the Angular frequency, g is the gravitational acceleration, and where it is noted that variables $\dot{m}$ and d_{32} in equation (1) can respectively be replaced by v , or d_{05} . The inclusion of variables a and ω , distinguishes equation (1) from being a purely static correlation model. By use of the Generalised Buckingham Pi method [46], appropriate dimensionless groups give a compact relationship on which the heat flux depends.

In principle, the representation given by equation (1) can be fitted to measured experimental data taken from a test-piece with any particular geometry and orientation. To do this, corresponding sets of measured heat transfer data under dynamic conditions are needed. But a large number of test-piece geometries might be of interest which would require a large experimental test programme which would include the manufacture of a number of instrumented test-pieces. The cost alone of creating just one test-piece is considerable. Moreover, at the stage where performance information is needed, the component part cooling surface geometry might not be known in detail. And even if the required detail is known, and an instrumented test-piece created, the mass might turn out be too high to allow the desired range of dynamic amplitudes and frequencies to be achieved unless the shaker is sufficiently powerful. The hypothesis then, is that a flat test-piece, with achievably low mass, will provide a dynamic correlation model that can be adopted for the purposes of simulation involving a test-piece with arbitrary geometry and orientation.

A dynamic correlation model corresponding to a flat horizontal test-piece, cooled using spray evaporative cooling supplied by a UNIJET nozzle, has been obtained in [45] by fitting to measured experimental data takes the form:

$$Bo = 1.24 \times 10^{-2} Ja^{1.3920} \left(\frac{\rho_l^2 \sigma v}{\mu_l^3} \right)^{0.9809} Re_V^{-0.9869} \left(\frac{a}{H} \right)^{0.4825} Ac^{0.4998} \quad (2)$$

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 the convective and boiling heat transfer. This model will be used to test the dynamic correlation model hypothesis in Section 6.

Simplification of the test-piece solid body transient heat conduction characteristics

Turning attention to the heat characteristics of a general component part, in general to construct the transient information needed within a simulation, the 3D transient solution for a solid (with spatially-varying diffusivity) without internal heat generation, must be obtained from the heat diffusion equation:

$$\frac{\partial}{\partial x} \left(k \frac{\partial T}{\partial x} \right) + \frac{\partial}{\partial y} \left(k \frac{\partial T}{\partial y} \right) + \frac{\partial}{\partial z} \left(k \frac{\partial T}{\partial z} \right) = \rho C_p \frac{\partial T}{\partial t} \quad (3)$$

subject to appropriate boundary and initial conditions, where C_p is the specific heat capacity, k is the thermal conductivity, and ρ is the density of the solid. Although numerical solution of equation (3) is in itself not a major challenge, obtaining a transient solution for an arbitrary body shape within a practical timescale, is generally problematic. This is where the transient characteristic forms part of the plant model within a control simulation. The second hypothesis mentioned is that for control simulation purposes, an equivalent body can be found with regular geometry (i.e. in the y and z directions), that allows a very good approximation to transient heat conduction (in the x direction) be obtained by solving a 1D version of equation (3) i.e.:

$$\frac{\partial}{\partial x} \left(k \frac{\partial T}{\partial x} \right) = \rho C_p \frac{\partial T}{\partial t} \quad (4)$$

To test both hypotheses, a control simulation model (including an appropriate equivalence to equation (3) and subsequent solution to method applicable to equation (4)), will first be developed. The results from there will be discussed in Section 6 using experimentally-measured data obtained from instrumented component part test-pieces with complex 3D heat diffusion characteristics and complex cooling surfaces. The experimental facilities used to generate the test data are discussed in Sections 4 and 5. Before then, a section is now devoted to finding an equivalent 1D transient heat conduction characteristic for a 3d test-piece

3. AN EQUIVALENT 1D TRANSIENT HEAT CONDUCTION CHARACTERISTIC FOR TEST-PIECES

To enable the creation of a useful simulation capability involving the plant model in Figure 1, for an arbitrary three-dimensional test-piece, in particular, to simulate the heat conduction characteristics, a very computationally-efficient one-dimensional (1D) equivalence has to be found. Here a 3-dimensional Finite Element solution for heat conduction is used to demonstrate this using ANSYS. Once the 1D equivalence has been found, a fast numerical solution of the 1D heat diffusion equation (as explained in Section 4) can be embedded into the simulation. Although the approach is considered to be quite general, two test-pieces are used as vehicles to undertake the equivalence, namely to identify their test-surface temperature response with respect to thermal load changes, and to find a 1D metal thickness which gives a similar temperature response to the actual test piece. These vehicle test pieces (which are also used in the experimental tests in Section 6) are respectively referred to as the ‘flat’ (surface) test-piece and the ‘¼-cylinder’ test-piece (which represents part of an evaporatively-cooled boosted IC engine cylinder head).

A description of a Flat-surface test-piece and $\frac{1}{4}$ -Cylinder test piece

A flat test-piece chamber casing is shown in Figure 2a and the flat test-piece itself is shown in Figure 2b. This is internally-heated by six 250 Watt vertical electric cartridge heaters.

Fig. 2. Flat test-piece: (a) Chamber cross-section view, (b) From top to bottom, exploded view of shroud, bolts, flat test-piece, heater-block and shaker shaft (based on [48]).

Fig. 3. The $\frac{1}{4}$ -Cylinder test-piece: (a) photo of manufactured assembly; (b) Exploded view of assembly.

Figure 3 shows the '¼-Cylinder' test piece. The detailed design features of both of these two test-pieces will be explained, along with the experimental facilities, in Section 4.

Determination of equivalent 1D metal thickness

The approach to determine the equivalent 1D metal thickness required for simulating the temperature transient with respect to varying thermal load conditions similar to that in the test-pieces, uses a thermal finite element analysis (FEA) on the actual test-piece assembly to identify the temperature responses with varying load conditions. This was used to model the full 3D characteristics of the test-piece. FEA is undertaken with different spray cooling heat transfer coefficients but with the other boundary conditions kept fixed, the transient test surface temperature is computed corresponding to heater power changes for a range of spray cooling heat transfer coefficients that would typically be expected to occur on the actual test piece hardware. Mesh convergence studies were undertaken for both the ¼-cylinder and the flat test-piece models to ensure appropriate global element sizes were used. Transient temperatures on the cooled-surface were simulated for both models (with the same boundary conditions) corresponding to a number of decreasing mesh sizes. These were compared with the simulated temperature corresponding to a nominally-small mesh size. The converged mesh size was chosen to correspond to the largest mesh which gave a maximum absolute temperature-change of less than 0.1%. The flat test-piece element sizes were: 4 mm, 3 mm, and 2.5 mm, plus a small element size of 2 mm. The corresponding maximum absolute temperature differences from the small element temperature were: 0.2%, 0.096% and 0.035% respectively. Mesh convergence was therefore deemed to have occurred for an element size of 2.5 mm, since the difference change was also less than 0.1%. The ¼-cylinder test-piece element sizes chosen were: 10 mm, 5 mm, and 2 mm, plus a nominally-small element size of 1 mm. The corresponding maximum absolute temperature differences from the small element temperature were: 0.5%, 0.38% and 0.3% respectively. Mesh convergence was therefore deemed to have occurred for an element size of 2 mm, since the difference-change was less than 0.1%.

A corresponding transient temperature solution for the equivalent (single metal) 1D model is obtained with the same boundary conditions (using a finite difference scheme explained in Section 4). By examining a range of 1D metal thicknesses, the best equivalence is the thickness that produces the set of transient temperatures (corresponding to the range of spray cooling heat transfer coefficients examined), with minimum rms difference between the FE and 1D solutions. From the FE model simulation, the surface heat flux was identified corresponding to the heater power and the applied heat transfer coefficient. This heat flux was then applied as the heater power in the 1D simulations.

Fig. 4. a) Flat test-piece assembly temperature via FEA, b) Test surface at which the spray cooling BC is applied, c) Surfaces at which the surrounding steam cooling BC is applied.

For the (copper) flat test-piece, FEA was undertaken for the heater-block and test-piece assembly for three different spray cooling heat transfer coefficients but with the same heater power input and surrounding heat loss boundary conditions. Figure 4 shows the FE model for the flat test-piece and the prediction of the instantaneous temperature at a particular instance in time.

By following the procedure described, the equivalent 1D (copper) metal thickness was found to be 80 mm which gives transient surface temperature response very similar to the actual flat test-piece. Figure 5a shows, for three different spray cooling heat transfer coefficients, the simulated surface temperature against time for both the flat test-piece FEA and the 1D simulation (for an 80 mm thick 1D copper layer) for varying heater power (shown in Figure 5b).

For the ¼-Cylinder assembly, the FEA was more complicated because several metals are involved, which required thermal contact conductance boundary conditions be applied at contact surfaces between different parts. The conductance values for the metal combinations used for surface contacts are as follows: for Steel-Steel: 1900 W/m²K; for Steel-Aluminium: 6000 W/m²K; for Aluminium-Copper: 1700 W/m²K; for Aluminium-Aluminium 11400 W/m²K; and for Copper-Steel: 1700 W/m² K.

Fig. 5. Flat test-piece FEA and 1D simulation results. a) test-surface temperature b) heat flux load.

Figure 6 shows the FE model for the $\frac{1}{4}$ -Cylinder test-piece and the prediction of the instantaneous temperature at a particular instance in time. The boundary conditions correspond to: i) heater power of 800 W, ii) spray cooling convection with a heat transfer coefficient of 10^4 W/m² K, and a liquid coolant temperature of 85 °C, iii) a surrounding surface steam cooling convection with a heat transfer coefficient of 10^2 W/m²K, and a vapour temperature of 100 °C.

Fig. 6. a) 1/4-cylinder test-piece assembly temperature via FEA, b) Test surface at which the spray cooling BC is applied, c) Surfaces at which the chamber steam cooling BC is applied d) Surfaces at which the surrounding air-cooling BC is applied.

Figure 7 shows for two spray cooling heat transfer coefficients, the transient temperature for the 1/4-Cylinder test-piece assembly for boundary conditions of: i) heater power of 800 W ii) spray cooling convection with a heat transfer coefficient of $10^4 \text{ W/m}^2\text{K}$, and a coolant temperature of 85°C , iii) chamber steam cooling convection with a heat transfer coefficient of $10^3 \text{ W/m}^2 \text{ K}$ and a vapour temperature of 100°C and iv) surrounding surface air cooling convection with a heat transfer coefficient of $30 \text{ W/m}^2 \text{ K}$, and an ambient air temperature of 25°C . By following the procedure described, the equivalent 1D (aluminium) metal thickness was found to be 160 mm which gives transient surface temperature response very similar to the actual flat test-piece.

Fig. 7. ¼-cylinder assembly FEA and 1D simulation results. a) test-surface temperature b) heat flux load.

It can be seen from Figures 5 and 7 for both the flat-test piece and the ¼-Cylinder assembly, that there is a very good agreement between the transient temperatures obtained from the FE models and the 1D simulation for the entire range of spray cooling heat transfer BCs. The identified 1D metal thicknesses of 80 mm and 160 mm now form the basis of plant model control simulations to replicate the experiments as well as optimum case scenarios, which is described in the following sections.

4. ASSEMBLY OF THE CONTROL SIMULATION MODEL

With the two model simplifications of Section 2 (particularly the component part transient heat diffusion equivalence in Section 3) this section gives the numerical details of how the cooling

system shown in Figure 1 is simulated. First regarding the computation of the heat flux for the test piece heat removed by spray evaporative cooling, the dynamic correlation Equation (2) is used. As hypothesised (and indeed justified) in Section 2, this same flat test piece dynamic correlation is adapted for use in simulating the cooling control system of Figure 1 with any type of test piece and any type of controller using UNIJET nozzle data. With the nozzle to test-surface distance and test-surface diameter set, the boiling number given by Equation (2), plus the liquid coolant water properties and the temperature, can be used to obtain the heat flux, which is applied as the coolant-side boundary condition. This boundary condition is input information to the 1D numerical solution of component part transient heat diffusion equivalence discussed (in context) in Section 3. The details of this solution are now given.

Finite Difference Solution of the 1D equivalent model

The full 3D time-dependent thermal FEA simulation requires around 12 hours of CPU time to simulate 2000 s of physical time (using an Intel i7 workstation). It is therefore impractical for use in real-time control. An efficient alternative is to use a 1-D equivalent model, based on a finite-difference approximation [16], which can run in under 5 minutes. The numerical model is spatially discretised, using grid points in the spatial direction as shown in figure 8.

Fig. 8. 1D Numerical model grid structure.

Equations for each grid point have been derived using Taylor Series approximation for the 1-D Fourier diffusion equation (4), written without loss of generality in the spatially independent diffusivity form as:

$$\frac{\partial^2 T}{\partial x^2} = \frac{1}{\alpha} \frac{\partial T}{\partial t} \quad (5)$$

where T is the temperature, x is the position, α is the thermal diffusivity, and t is the time. Using a central finite difference, the spatial derivative is approximated as

$$\frac{\partial^2 T}{\partial x^2} \approx \frac{T_{(i+1)} - 2T_i + T_{(i-1)}}{\delta x^2} \quad (6)$$

and the temporal derivative is approximated as:

$$\frac{1}{\alpha} \frac{\partial T}{\partial t} \approx \frac{T_i^{(n+1)} - T_i^{(n)}}{\alpha \delta t} \quad (7)$$

where subscript ' i ' stands for the grid point; ' $i-1$ ' and ' $i+1$ ' are the grid points to the left and right of the grid point i respectively. Superscripts ' n ' and ' $n+1$ ' denote the current, and following time-steps respectively. Substituting equations (6) and (7) into (5), and defining the Fourier number, $\delta Fo = \alpha \delta t / \delta x^2$, gives an equation for temperature at grid points as follows:

$$T_{(i)}^{(n+1)} = \delta Fo \cdot T_{(i+1)}^{(n)} + (1 - 2\delta Fo)T_{(i)}^{(n)} + \delta Fo \cdot T_{(i-1)}^{(n)} \quad (8)$$

where δt is the time step size, and δx is the distance between the grid-points. An imaginary grid point is used to calculate the temperature at the boundary grid-point where there is actually no grid point to the right or left of the grid point of interest. The equation for the temperature of the imaginary grid point for a heat flux boundary condition is given as:

$$T_{imag} = T_{adj} + \frac{2\delta x q}{k} \quad (9)$$

where q is the heat flux, and where subscripts ' $imag$ ' and ' adj ' represent the imaginary outside and inside grid points respectively. The boundary conditions for heating and cooling are applied at the end grid points, as heat fluxes. The instantaneous heat fluxes applied, are then solved for the imaginary temperatures i.e. equation (9). To achieve numerical stability for a given spatial distance between grid points, the temporal discretisation must meet the stability criterion:

$$\delta Fo(1 + \delta Bi) < 0.5 \quad (10)$$

where $\delta Bi = \delta x h / k$ is the Biot number, which in turn determines time step size as follows:

$$\delta t < \frac{0.5 \delta x^2}{\alpha(1 + \delta x h / k)} \quad (11)$$

This 1D solution methodology was already applied in Section 3 to the equivalent flat-test piece and the equivalent $\frac{1}{4}$ -Cylinder models. More details can now be given. Assuming for the 1D solution, a copper Flat test-piece and an aluminium $\frac{1}{4}$ -Cylinder test-piece, the respective material properties are: Thermal Conductivity k , 385 and 150 W/mK; Density ρ , 8960 and 2700 kg/m³; and Specific Heat Capacity C_p , 390 and 910 J/kgK. A default fixed time-step size of 1 ms was used (many times smaller than required to meet the stability criterion for a model with a minimum metal thickness of 7 mm – the typical thickness of a downsized 100 kW/L engine cylinder head combustion chamber section). The maximum time-step size needed to meet this stability criterion for a metal thickness of 7 mm at the highest spray evaporative cooling heat transfer coefficient of 50,000 W/m² K is 14 milliseconds.

The 1D solution forms an essential part of the control simulation. Figure 9 shows a system model for a control simulation for a version of the system shown in Figure 1 (in which here, a PID controller has been adopted to control the coolant-side metal temperature). The nodes of the 1D heat diffusion solution are explicitly shown. The controller output is representative of the cooling pump control voltage used with a lookup table containing flow rate information as a function of pump voltage. For the $\frac{1}{4}$ -Cylinder test-piece (which has two nozzles) the flow rate was appropriately split, whereas the boiling number computations involve one nozzle, where only one cooling heat flux is applied as the boundary condition in the 1D equivalent model.

Fig. 9. Control model plant diagram showing nodes to obtain 1D heat diffusion solution.

5. EXPERIMENTAL FACILITIES

The experimental rig is now described which is capable of controlling in hardware, the temperature of various dynamically-excited test-pieces using spray evaporative cooling. This rig uses de-ionised water coolant. The temperature-controlled test-piece is excited dynamically by a shaker, itself controlled by National Instruments hardware. The test-piece is also instrumented with sensors to measure temperature at various locations and accelerometers to measure motion kinematics, with data captured from a data acquisition system and control unit system using NI hardware and LabView software. Figure 10 shows the entire cooling control system hardware which comprises the following components: a condenser (Denso RDP 583), a heater module (Two 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 condenser and separator tanks. The pump (No. 1 in Figure 10) circulates de-Ionized water measured by the flow meter (No. 16). The Feed

Heater (No. 2) supplies heated water to the nozzle (No. 4) and test-piece via silicone rubber flexible piping. Vent (No. 5) transfers vapour to the condenser (No. 9); Drain (No. 6) takes remnant droplets from the test chamber (No. 13) to the separator tank (No. 11). Water flows from the condenser to the condensate tank. A header tank (No. 8) removes any air from the system to provide a small positive feed pressure to the inlet of the pump. A feed water cooler (No. 12) drops the condensate temperature to below the maximum operating limit for the pump (100 °C).

Fig. 10. Experimental cooling control system hardware.

Heating and Instrumentation details of the test pieces.

During experimental testing, the test-piece is positioned on top of a (Bruel & Kjaer V555) mechanical shaker. The details of two particular test pieces will be discussed: the flat test-piece and the 1/4- cylinder test-piece described in Section 3. The 2-cm diameter flat test-piece (shown in Figure 2b) with its nozzle fitted inside the chamber can be shaken at different amplitudes and frequencies at the same time as being heated by six vertical 250W cylindrical Watlow cartridge heaters located inside a copper heater block. This provides a strongly one-dimensional heat conduction to the test piece. There are three T-Type thermocouples fitted, one located 1

millimetre under the coolant surface of the test chamber, the other two located 5.5 millimetres below each other. This is to facilitate heat flux measurement. A shroud is also placed around the test-piece disc to guide any accumulated liquid film to the bottom of the chamber for drainage. A full-cone UNIJET® (TG tip Type) nozzle is located on top of the test chamber. All joints and bolts are able to withstand a maximum temperature of 350°C.

The ¼-cylinder test piece (Figure 3b) is designed to emulate cooling of a three-dimensional surface to approximate a hybrid electric vehicle range extender engine cylinder head. The ¼-cylinder test-piece has four times the cooling surface area of the flat test-piece and comprises: a test chamber, a heater block, and an insulated base. There are eight Watlow 250W cartridge heaters: six are located in the copper heater block underneath the test piece; two are located at the top of the test chamber and inside the castings (that emulate fuel injector and valve guide bosses). A single Type K thermocouple is located 1 mm under the coolant surface of the test chamber. A dovetail joint between heater block and test-piece allows for differential thermal expansion between the copper heater block and aluminium. Two Bete PJ8 atomising misting nozzles are located on the top of the test chamber.

Data acquisition system and control unit

The main components of the National Instruments data acquisition system are listed in Table 1.

Table 1. Data Acquisition System.

Module	Description
NI cRIO-9035	Main processor, Control algorithm
NI 9205	Analog inputs – 16 Bit – 32 Channels (Differential and Single-Ended)
NI 9212	Thermocouple module – 24 Bit – 8 Channels
NI 9230	AI 24 Bit – 3 Channels
NI 9264	AO 16 Bit – 16 Channels
NI 9472	DO 24 V – 8 Channels

As shown in Figure 10, NI cRIO-9035 main processor (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, a human-machine interface unit. Pump flow rate (in Micropumps MGD100P controller), cartridge heaters (Watlow 250W), and the band heaters are operated using 0 to 5 V control signals. These control signals come from an NI9264 module in the CRIO-9035 National Instrument. Flow meter and pressure transducers signals are collected by the NI 9205. Figure 11 shows an implementation of the test-piece spray evaporative cooling temperature control system.

Fig. 11. Temperature control system using spray evaporative cooling.

The pre-heater has a PID temperature controller to tackle the thermal inertia of the coolant feed. The system remains stable, staying within the evaporative phase even during rapid changes in pump-speed by the PID controller (e. g. doubling the flow rate will only cause a temperature difference of $\pm 5^\circ\text{C}$). Two (EVR-25BF) power regulators are used to obtain the 0 to 5 V signals (from NI 9264) and convert them to the required power for the band and the cartridge heaters. Thus, the test-pieces receive constant and variable heat loads through the cartridge heater for different control scenarios. Measurements from thermocouples (located 1 millimetre under the coolant surfaces in either test piece) are used as a state variable for the PID controller in the FPGA CRIO-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 for the flat-test piece, and on top of its test chamber, for the $\frac{1}{4}$ -cylinder test piece. Acceleration signals were measured by the NI 9230 sound and vibration input module.

6. EXPERIMENTAL TEST PROCEDURE AND DATA REDUCTION

Table 2 shows the testing plan and operating conditions for the flat and ¼-cylinder test pieces. All the test runs have been undertaken with the cooling system at atmospheric pressure.

Table 2. Test plan and operating parameters showing shaker amplitudes and frequencies .

Test case	$P_{chamber}$ (bar)	v (ml/min)	$T_{coolant}$ (°C)	H (mm)	Static	f (Hz) / a (mm)	1.9	6	100	200	400
¼-cylinder	1	60-130	80	NA	✓	0.01	-	-	-	-	✓
						0.08	-	-	-	✓	-
						0.09	-	-	✓	-	-
						1	-	✓	-	-	-
						3	-	✓	-	-	-
Flat	1	120-225	80	17	✓	0.02	-	-	-	-	✓
						12	✓	-	-	-	-

For flat test-piece temperature control, data was examined for two dynamic cases: a high-frequency, low-amplitude vibration at frequency $f=400$ Hz and amplitude $a=0.02$ mm, and a low-frequency, high-amplitude vibration at $f=1.9$ Hz and $a=12$ mm. This latter case emulates maximum vibration amplitudes of around 12 mm, typically used to test vehicle-mounted hardware. These two vibration cases use appropriate forms of the dynamic correlation Equation (2), obtained in [45]. Collecting data for the ¼-cylinder test-piece however proved more difficult, because its higher mass only allows for a maximum vibration amplitude of 3 mm. This limitation is one of the reasons for proposing in Section 2, the use of the flat test-piece dynamic correlations for all test-pieces.

The heat flux through the flat test-piece test area can be calculated during experimental testing using the numerical solution to the 1D transient heat conduction equation discussed in Section 4 (in particular using equations (8) and (9)). To meet the numerical stability criteria given by equations (10) and (11), the distance between the first and third thermocouple is split into 22 nodes. Thus, the distance between the grid points δx is 5.24×10^{-4} m. The time step δt 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) had to be interpolated between the measured temperatures. To interpolate the measured temperatures, Piecewise Polynomials are used. The surface temperature gradient $\nabla T = \partial T / \partial x$ needed for the transient heat flux calculation, can be obtained rather than using equation (7) by using a second-order backward finite difference i.e.:

$$\frac{\partial T}{\partial x} \approx \frac{3T_i - 4T_{i-1} + T_{i-2}}{2\delta x} \quad (12)$$

Setting the temperature controller operating conditions and selecting

To ensure the safety, the test-piece temperature control system is put through two phases of trials. Phase-1 is a safety trial. Using previously fitted dynamic correlation equation (2), various

operating parameters, setpoints, and thermal requirements must be evaluated. Operation of the temperature control system follows a step-by-step plan shown in Figure 12.

Fig. 12. Procedure for setting the controller hardware operating conditions.

First, the initial operating conditions fixed during the experiments are set by the operator in a human-machine interface (HMI). A graphical user interface (GUI) is programmed in LabVIEW. The iterative part using the dynamic correlation examines the boundaries, taking into consideration the applicability of the spray system to maintain the regime within nucleate boiling. During all the operating conditions, the nucleate boiling mode must be capable of providing enough heat removal required for the particular application. Choosing suitable setpoints for the spray cooling system is as important and challenging as the control task [47]. The desired parameters (such as surface temperature and flow rate) and requirements (e.g. heat flux) are completely dependent on each other. To confirm that measurement errors do not produce unacceptable scatter in the experimental test results, a thorough uncertainty analysis was undertaken by quantifying

parameter uncertainties and measurement errors. All the parameter uncertainties and measurement errors are given in Table 3.

Table 3. Uncertainties in measured and calculated parameters.

Parameter	Uncertainty (%)	Units
Volumetric flow rate	± 0.6 (of full scale)	ml/min
Thermocouples	± 0.4	$^{\circ}\text{C}$
Accelerometer frequency	± 5	Hz
Pressure transducers	± 0.25	bar
Diameter	± 1	mm
Length	± 1	mm
Heat flux	± 2.6	MW/m^2

With the exception of heat flux, the uncertainties given in Table 1 are taken from the manufacturer's calibration information for each measuring instrument used. The uncertainty in the calculation of heat flux, is obtained using the method by Moffat [49], which, at the maximum experimentally-measured data point of 1.96 MW/m^2 , gives an uncertainty of 2.6% (which is deemed wholly acceptable).

Tuning the PID gains under static and dynamic conditions

The control hardware shown in Figure 11 is tuned and used to control the metal temperature of the two test-pieces described. The presence of vibration acts as disturbance to the temperature controller. Before the application of vibration, the PID gains in the controller in Figure 11 are tuned in a largely trial and error process because the control system plant model is so strongly nonlinear, the prospect of using theoretical PID tuning methods such as Ziegler-Nichols is limited. It was initially considered more likely that static gains would provide a useful starting point to assigning gains for dynamic cases.

After many trials, the most acceptable static gains which minimized both overshoot and undershoot, and gave a reasonably fast settling time, were found to be $P=2.1$, $I=0.08$, and $D=0.001$. Settling time however plays a vital role in the thermal management and energy efficiency of powertrain parts where typically, the response should be less than around 20 seconds in real time. The combined test-piece, test rig, and pipework can however result in a relatively high thermal inertia, which in some cases can produce a settling time as high as 155 seconds. Simulated control systems by contrast, are capable of producing much more realistic (shorter) time constants (i.e. less than 20 seconds). The static gains which produced the best settling time however were found to involve a negative proportional gain (e.g. $P=-2.1$), plus an increased integral gain (keeping the same derivative gain). Higher proportional gain would initially increase the error (during the rise time), but later, during the settling time, integral

control would dominate and correct the system. This reduced the settling time from 155 s to 98 s (a 37% improvement) with only a 1 second increase in settling time for cooling setpoints from 135 °C to 110 °C. A PID controller with negative proportional gain is unusual as it can suffer a diminished phase margin, and more severe overshoot but it is known to work in certain circumstances. One reason why this type of PID controller is workable for the cooling system in Figure 11 is that the pump works at a low speed and provides a flow rate of around 110 ml/min during the rise time. As soon as the error is positive, the flow-rate surges to the highest possible value (around 230 ml/min) trying to compensate for the overshoot. Conversely, the flow-rate for negative proportional gain, peaks at around 220 ml/min during the rise time, then gradually drops to around 160 ml/min to reach the setpoint (135 °C). There is a reverse trend from a flow-rate threshold when the covering diameter of the spray cone exceeds the area of the surface target. From that specific flow-rate (threshold), the volumetric flux (a measure for effective volumetric flow rate) reduces, and as a consequence, the heat transfer rate reduces accordingly. It is in fact stated on the UNIJET® nozzle manufacturer data sheet, that the spray angle varies from 50° to 61°. This explanation for the reduction in heat transfer rate is supported by Schwarzkopf et al. [50]. It means from the flow-rate threshold, the trend for the heat transfer is counter intuitive: the higher the flow rate, the lower the heat transfer rate (which for our configuration is estimated to be above 200 ml/min). The same reverse trend has been observed by Zhang et al [40]. Thus, for negative proportional PID controller, it was concluded that the shift between these two trends of heat transfer could have resulted in a further increase in the test piece surface temperature and therefore a reduced settling time. Moreover, a negative proportional gain which resulted in having a higher volumetric flow rate during the rise time, would suppress the effect of vibration frequency due to a higher droplet velocity, which is consistent with the experimental results in [51]. This means that a negative PID proportional gain controller could be more reliable and adaptive to dynamic conditions including with high-frequency vibration. To better understand the influence of different gains on thermal performance and energy efficiency, a coefficient of performance (COP) has been defined as the ratio of the heat (power) removal to the required pumping power i.e.:

$$COP = \frac{Q=q\pi d^2/4}{VI} \quad (13)$$

where, Q, V, and I are respectively the heat removal provided by the spray cooling, the pump voltage and current. To smooth out trends in the COP, by filtering-out noise from random short-term fluctuations, a 10-point moving average is used. Since experimental evidence shows that vibration reduces heat transfer [45], it is expected that *dynamic* disturbances reduce COP. because the controller orders the pump to compensate for a reduction in heat transfer, thereby consuming more power. (Note that use of negative proportional gain, can increase heat transfer

rate. In one example, an average COP value of 4.95 was obtained - an increase of 21% compared with use of a PID controller with positive gains).

Tuning the PID controller gains for the dynamic case was made possible using the static gains as the starting point. In fact, through experience, a broadly accurate guide to the PID controller gain tuning process is to increase the derivative gain to reduce overshoot. The difference observed between overshoot and undershoot with and without the derivative gain clearly supports this guiding principle. Furthermore, a subtle change in the computed COP value confirms that the pump is better managed by the increased derivative gain. Despite attempts to find more appropriate PID gains through trial and error tuning, ultimately the static case gains were used for the dynamic case for both high frequency at low amplitude and low frequency at high amplitude, namely: $P=-2$, $I=0.7$, and $D=0.002$.

7. A COMPARISON OF TEMPERATURE CONTROL BY HARDWARE AND SIMULATION

In this section, the test-piece temperature control simulation capability developed in Sections 3 and 4 is tested to assess its accuracy in predicting the behaviour of the temperature control hardware described in Section 5. First, for temperature control of the flat test-piece (described in Section 3), a comparison is made under static conditions, between the use of the numerical simulation capability (developed in Sections 3 and 4) as compared with the corresponding behaviour of the hardware temperature control system shown by Figure 11. A comparison is then made under dynamic conditions, between flat test-piece simulation and the behaviour of the hardware control system. Simulation is then used under both static and dynamics conditions, to show 'fast' control of the flat test-piece temperature.

Attention then switches to the $\frac{1}{4}$ -Cylinder test-piece and tuning the hardware to achieve appropriate temperature control. Comparisons are then made, first under static, then dynamic conditions, to show $\frac{1}{4}$ -Cylinder test-piece temperature control by simulation and the corresponding behaviour of the hardware control system.

The flat test-piece temperature control comparison: hardware versus simulation

Simulations were first undertaken for the flat test-piece to emulate the static behaviour of the experimental hardware with the same PID control gains and the same flow rate limits. Figure 13 shows the comparison between the simulated and measured surface temperatures and flow rates. The PID control gains were: $P=2$, $I=0.08$, and $D=0.001$, and the maximum and minimum flow rate limits were set to 110 ml/min and 220 ml/min respectively.

Figure 14 shows a comparison of surface temperatures and flow rates by simulation and experiment measurement with the flat test-piece being exposed to vibration at a frequency of 400 Hz and an amplitude of 0.02 mm. The PID control gains were: $P=-2$, $I=0.7$ and $D=0$, and the maximum and minimum flow rate limits were 110 ml/min and 220 ml/min respectively.

Figure 15 shows, for 'faster' temperature control, surface temperatures and flow rates obtained by simulation only. The flat test-piece is again exposed to vibration at a frequency of

400 Hz and an amplitude of 0.02 mm but with the integral gain reduced such that the PID control gains were: $P=-2$, $I=0.4$ and $D=0$. These values were arbitrarily chosen to provide the faster response with minimum overshoot at the peak setpoint temperature. Here dynamic and static results are shown for comparison. Nominally ‘faster’ temperature control, was achieved by reducing the lower flow rate limit to zero. When the controller outputs a signal requiring a flow rate below 110 ml/min, the flow in the simulation was switched-off. This was to simulate the scenario where the pump could not operate satisfactorily at a pressure low enough to produce a flow rate below 110 ml/min. The maximum flow rate limit was kept at 220 ml/min, which was at the maximum pressure the coolant pump in the hardware system could operate at.

Fig. 13. Flat test-piece static simulation of surface temperature control and flow rate compared with experimental measurement, also showing the 1D equivalent thermal load heat-flux supplied by the heater (top).

Fig. 14. Flat test-piece dynamic ($f=400$ Hz, $a=0.02$ mm) simulated surface temperature control and flow rate compared with the experimental measurement, also showing the 1D equivalent thermal load heat-flux supplied by the heater (bottom).

Fig. 15. Flat test-piece surface temperature control and flow rate under static and dynamic conditions ($f=400$ Hz, $a=0.02$ mm) obtained for 'fast' response control simulations, also showing the 1D equivalent thermal load heat-flux supplied by the heater (top).

Discussion of the flat test-piece simulation and hardware control comparisons

Good agreement can be seen in Figure 13 between the simulated surface temperatures and that measured from hardware. The surface temperature in the simulation generally follows the same trend throughout the period of temperature rise, although during the last 20% of the rise, the measurement from hardware shows a temperature increase that has been drastically reduced. The reason for this behaviour is speculated to occur from the change in boiling regime which has significantly increased the rate of heat transfer. The difference in the trend associated with the temperature rise can also be seen in the flow-rate. Thus, during this rise, the simulated surface temperature reaches the setpoint faster, and the flow-rate starts to increase until the temperature reaches a genuine steady-state. Whereas, on the hardware, this effect is significantly held-back owing to the delay in reaching the setpoint. The dynamic simulation and hardware measurement comparison in Figure 14, follow a similar trend during the rise and fall of the surface temperature with respect to the increase and decrease of the load. The simulation could not quite maintain the surface temperature at the setpoint and is seen to be oscillating for much of the time. This probably occurred from use of a negative proportional gain, which in the case of measurement, was used to avoid temperature overshoot. It is seen to work well on the hardware whereas in the simulation, the flow needed to keep the temperature at the setpoint was unstable owing to a lack of system 'damping'. The controlled surface temperature rise-time simulated in Figure 15, was reduced to approximately 7 s to reach the setpoint, which was similar in both static and dynamic cases. During the period of temperature fall, with the reduced thermal load, the flow rates were at maximum to reduce the temperature faster, but response times were similar to the other experiments because the maximum flow rates were similar.

Tuning the hardware control system for use with the 1/4-Cylinder test-piece

Before attempting to show a comparison between temperature control simulations and experimental measurement of the ¼-cylinder test-piece, some preliminary explanation is needed of the measures taken to get the hardware control system shown in Figure 11 to function as intended. To put this into an application context, Mudawar et al. [25] has reported that feasible thermal management applications for hybrid electric vehicles should have the potential to dissipate heat within the range of flux 1.5 - 2 MW/m². There is no published evidence yet, of successful surface temperature control involving spray evaporative cooling in the presence of vibration. To address this need, the ¼-Cylinder test-piece has been used to establish if the temperature control system in Figure 11 has adequate precision, flexibility, and viability to control surface temperature by mainly using coolant flow-rate control (using varying pump speed). A total of three PID controllers (all implemented from NI LabVIEW) have been adapted to achieve coolant-side metal temperature control and verification: one PID controller acts as the temperature controller, another controls the coolant pump speed, and the third controls feed-water temperature. The temperature PID controller is actually PI (since the PID gains are: P=2, I=0.6, D=0), and is used to follow the desired cooled-metal surface temperature during variable

thermal load-steps, with temperature measured by a single embedded thermocouple 1 mm beneath the surface inside the test chamber. The PID controller, used for flow-rate control, has gains $P=0.003$, $I=0.008$, and $D=0.05$. This achieved flow stability, important to avoid on-off switching of the pump should the surface temperature PID controller be suddenly switched-off. The third PID controller (with gains $P=3$, $I=0.78$, $D=0.09$) maintains the feed-water temperature at the desired set-point to maintain a constant degree of sub-cooling at the inlet. This is achieved with a 0 to 5 V controllable power regulator (United Automation, EVR-25BF, AC Burst Fire Controller).

Figure 16 shows the controlled surface temperatures during different set points and thermal loads both in a static condition and with vibration at a frequency of 6 Hz, and an amplitude of 3 mm. The right-hand-side axis on Figure 16 shows the pump flow rate. The static results were used to obtain a good control performance by trial and error tuning the PID controllers (i.e. low steady-state error and settling time). The dynamic condition was to establish that the controller can adapt to the dynamic environment created by the shaker, and also to show the effect vibration has on the control of temperature. The arrows on Figure 16, show variable thermal load fluctuations of between $\pm 35 - 40$ W, which are superposed at the end of each of the step changes when a steady-state temperature has been reached and the steady-state control error is zero.

Fig. 16. Cooled metal surface temperature hardware control as a function of time for the $\frac{1}{4}$ -cylinder test piece under static and dynamic conditions.

The results in Figure 16 clearly show that the controller is successfully adapting to the dynamic condition with a steady-state error of around 1 °C at set-points of 120 °C and 125 °C during the heating mode. During the cooling mode (involving steps from the higher to the lower thermal loads, i.e. after 3400 s) where there is one step from 130 °C to 125 °C, the steady state error is also about 1 °C. This steady-state error is present in both static and dynamic cases. The reason for this is evident from the flow-rate where the pump-speed has reached maximum before the end of the cooling step and not able to provide higher flow-rate of above 130ml/min needed for PJ8 nozzles. To confirm this, and to show that when maximum speed is not reached, temperature control is robust, two consecutive thermal load changes were taken, from the minimum of 280 W to a maximum of 800 W, and back to 280 W, with the desired surface temperatures ranging between 110 °C and 125 °C. Figure 17 shows the corresponding results, where it can be seen that the flow-rate range is now sufficient and that control is achieved without any steady-state error.

Fig. 17. Cooled metal surface temperature control as a function of time for the ¼-cylinder test piece under static and dynamic conditions involving high-to-low, and low-to-high thermal load steps.

It is also evident in Figure 17 that single to two-phase transitions occur and these play a prominent role in the control of spray evaporative cooling. For spray cooling during heating and cooling of the ¼-cylinder test-piece (i.e. above 114 °C for static and 116 °C for dynamic cases), vibration appears to reduce the two-phase spray cooling-rate. This demonstrates that the controller works better in the static case. By contrast, during the single-phase regime (i.e. below 114 °C for static and 116 °C for dynamic) vibration has an enhancing effect on the surface

temperature and therefore the controller uses lower flow-rate variations in comparison to the static case. The flow rate however, is not an appropriate measure in the comparison because the nozzle spray-angle varies with supply pressure. For single-phase, the results are consistent with the enhancing effect of vibration to the heat transfer as mentioned in [10].

The ability to control the temperature of the $\frac{1}{4}$ -cylinder test-piece is now useful to provide a comparison with the predictions using the control simulation (developed in Sections 3 and 4) in the same way as for the flat test-piece. However, before showing the comparison it is evident that the current $\frac{1}{4}$ -cylinder test-piece is actually flawed as a means of testing the practical time scales of powertrain components and gives further justification for the use of simulation. Apart from the flow-rate issues associated with maximum pump speed being reached, for the nozzles used, the more serious concern is the thermal inertia of the heater block, which does not allow temperature changes on the coolant-side to occur on time-scales that are close to the behaviour of an actual engine. The current design generates time spans of 116 minutes 40 seconds (i.e. 7000 s). It is for this reason that the following simulation results use the dynamic correlation model for a UNIJET® nozzle giving response time scales of less than 20 seconds.

The $\frac{1}{4}$ -cylinder test-piece temperature control comparison: hardware versus simulation

Simulations were first undertaken for the $\frac{1}{4}$ -cylinder test-piece to emulate the static behaviour of the experimental hardware with the same PID control gains and the same flow rate limits. Figure 18 shows the comparison of surface temperatures and flow rates for PID control gains were: $P=2$, $I=0.6$ and $D=0$, and the maximum and minimum flow rate limits set to 65 ml/min and 120 ml/min respectively.

Figure 19 shows a $\frac{1}{4}$ -cylinder test-piece surface temperature and flow rate comparison between simulation and experimental measurement. The test-piece is exposed to vibration at a (low) frequency of 6 Hz and an amplitude of 3mm. The PID control gains were: $P=2$, $I=0.6$ and $D=0$, and the maximum and minimum flow rate limits were set to 65 ml/min and 120 ml/min respectively. Figure 20 shows the $\frac{1}{4}$ -cylinder test piece surface temperatures and flow rate comparison with faster surface temperature control. Both dynamic and static results are shown for comparison. The test-piece is exposed to vibration at both high and low frequencies: 400 Hz, and amplitude of 0.02 mm, and at 6 Hz with an amplitude of 3mm. The same PID control gains were used i.e.: $P=2$, $I=0.6$ and $D=0$. Faster surface temperature control was achieved by reducing the minimum limit of flow rate to zero and increasing the maximum flow rate limit to 135 ml/min. When the controller outputs a signal to demand a flow rate below 65 ml/min, the flow was switched-off. This was to simulate the scenario where the pump could not operate satisfactorily at a pressure low enough to produce a flow rate below 65 ml/min. The maximum flow rate limit was increased to 135 ml/min because when vibration is present, the simulated maximum flow rate of 120 ml/min was insufficient to keep the surface temperature at the maximum setpoint value.

Fig. 18. Controlled surface temperature and flow-rate for the $\frac{1}{4}$ -cylinder test-piece in static conditions: simulation and experimental measurements, also showing the 1D equivalent thermal load heat-flux supplied by the heater (bottom).

Fig. 19. Controlled surface temperature and flow-rate for the $\frac{1}{4}$ -cylinder test-piece for dynamic conditions: simulation and experimental measurements, also showing the 1D equivalent thermal load heat-flux supplied by the heater (bottom).

Fig. 20. Controlled surface temperature and flow-rate for the 1/4-cylinder test-piece for both static and dynamic conditions: experimental measurements only, also showing the 1D equivalent thermal load heat-flux supplied by the heater (bottom).

Discussion of the 1/4-Cylinder test-piece simulation and hardware control comparisons

The simulation results in Figure 18 were close to the experimental hardware results with a very similar trend in the temperature rise and fall. The only difference being several degrees of temperature offset during the temperature rise and fall. It is evident that the simulated surface temperature was lower than experimental measurement throughout the rise and fall periods. This occurs because higher heat flux was calculated by the correlation model for the same minimum and maximum flow rate as in the experiment. The surface temperature comparison in Figure 19, between experiment and simulation, was not in particularly good agreement during the temperature rise. The simulated surface temperature reaches the first setpoint faster than in the hardware. This was believed to be caused by the lower heat flux being calculated by the correlation model for the same flow-rates as for the hardware. But it should be noted that the calculated heat flux, was insufficient to keep the temperature at the maximum setpoint value for the maximum flow-rate (which was achieved by the hardware). The opposite trend in the simulated temperature can be observed during the period of temperature fall. The surface temperature appears to cool down faster than the hardware. The simulations in Figure 20, with reduced minimum flow rate and increased maximum flow rate, show a faster response in the rise and fall time for all the static and dynamic cases simulated. The temperature rise time, from 110 °C to 125 °C, was reduced to around 20 s whereas the fall time was reduced to around 400 s. This shows that faster response times can be achieved in a physical system having a broader range of

flow rates, and especially when the flow can be switched-off during the rise time (which in the case of an electric vehicle range extender engine, could be operated at the peak temperature most of the time where the heat losses are minimum).

8. CONCLUSIONS

A simulation capability has been created to assess the temperature control performance of a hardware-based control system involving spray evaporative cooling, for thermally-managing component parts exposed to vibration, in an automotive vehicle powertrain. Two hypotheses, proposed to construct the simulation capability, have been tested and shown to be effective. The first hypothesis, involving the use of a flat test-piece dynamic correlation model for spray evaporative cooling of parts exposed to vibration, has been shown to be acceptable for spray evaporative cooling of component parts with curved cooling surfaces in non-horizontal orientation. And in respect of the second hypothesis, very successful finite-element-based testing confirms that the transient heat diffusion properties of particularly complex 3-dimensional component part body can be replaced by a 1-dimensional equivalence for which a fast finite difference solution is available.

An experimental test facility has been built in hardware, to examine the temperature control performance of two particular instrumented and electrically-heated test-pieces exposed to vibration from a shaker. These component parts represent a flat test-piece and a $\frac{1}{4}$ -cylinder test-piece. Comparisons of temperature control by hardware and simulation for these two test-pieces under static and dynamic conditions, show that temperature control of the hardware-based system can be confidently simulated. The design of the $\frac{1}{4}$ -cylinder test-piece, did not prevent a comparison with simulation, but for two reasons, it was found to be inappropriate for realistic hardware testing. This was because the thermal inertia of its heater-block made the thermal response time-scales unrealistically long by nearly two orders of magnitude. The second reason was that the total mass did not allow adequately high vibration amplitudes to be reached by the shaker i.e. no greater than 3 mm. Simulated modifications to the $\frac{1}{4}$ -cylinder test-piece show that temperature control is indeed possible on realistic time scales. This type of design modification actually shows that the creation of a spray evaporative cooling simulation capability is justified. The rms temperature differences between static and dynamic control model simulations, and the corresponding experimental measurements, were 1.93°C and 8.48°C respectively for the flat test-piece; and 1.74°C and 2.46°C respectively for the $\frac{1}{4}$ -cylinder test-piece. The flat test-piece being exposed to significantly higher vibration amplitudes than the $\frac{1}{4}$ -cylinder test-piece.

Acknowledgments

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.

The authors also wish to acknowledge additional financial and technical support of Ricardo Shoreham Technical Centre, UK.

References

- [1] Kim J, Oh J, Lee H. Review on battery thermal management system for electric vehicles. *Applied Thermal Engineering*. 2019;149:192-212.
- [2] Jafari S, Nikolaidis T, Thermal Management Systems for Civil Aircraft Engines: Review, Challenges and Exploring the Future. *Applied Sciences*. 2018; 8(11):2044.
- [3] Doty J, Yerkes K, Byrd L, Murthy J, Alleyne A, Wolff M, Heister S, Fisher T S. Dynamic thermal management for aerospace technology: review and outlook. 2017;31:86-98.
- [4] Tsiakmakis S, Fontaras G, Dornoff J, Valverde V, Komnos D, Ciuffo B, Mock P, Samaras Z. From lab-to-road & vice-versa: Using a simulation-based approach for predicting real-world CO2 emissions. *Energy*. 2019;169:1153-65.
- [5] Karagöz Y, Balci Ö, Gezer O, Kale S, Köten H. "Performance analysis of a hybrid lightweight vehicle with downsized engine." *Energy Sources, Part A: Recovery, Utilization, and Environmental Effects* 42, no. 12 (2020): 1513-1525.
- [6] Kotten H, Gul Z, Yilmaz M, Savci IH, 2018, "Advanced Numerical and Experimental Studies on CI Engine Emissions ", *Journal of Thermal Engineering*, Volume:4, Issue:4, Pages:2234-2247, doi: 10.18186/journal-of-thermal-engineering.434044.
- [7] Mera Z, Fonseca N, López J-M, Casanova J. Analysis of the high instantaneous NOx emissions from Euro 6 diesel passenger cars under real driving conditions. *Applied energy*. 2019;242:1074-89.
- [8] Benajes J, García A, Monsalve-Serrano J, Martínez-Boggio S. Emissions reduction from passenger cars with RCCI plug-in hybrid electric vehicle technology. *Applied Thermal Engineering*. 2020;164:114430.
- [9] Ranasinghe K, Guan K, Gardi A, Sabatini RJE. Review of advanced low-emission technologies for sustainable aviation. *Energy*, 2019;188:115945.
- [10] Deisenroth DC, Ohadi MJE. Thermal Management of High-Power Density Electric Motors for Electrification of Aviation and Beyond. *Energies*. 2019;12:3594.
- [11] Brelje BJ, Martins JRRA. Electric, hybrid, and turboelectric fixed-wing aircraft: A review of concepts, models, and design approaches. *Progress in Aerospace Sciences*. 2019;104:1-19.
- [12] Jafari S, Dunne JF, Langari M, Yang Z, Pirault J-P, Long CA. 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*. 2017;231:1126-43.
- [13] Jose JT, Dunne JF, Pirault J-P, Long CA. Spray Evaporative Cooling System Design for Automotive Internal Combustion Engines. *ASME 2018 Internal Combustion Engine Division Fall Technical Conference: American Society of Mechanical Engineers*; 2018. p. V002T07A8-VT07A8.
- [14] Cai C, Liu H, Jia M, Yin H, Xie R, Yan P. Numerical investigation on heat transfer of water spray cooling from single-phase to nucleate boiling region. *International Journal of Thermal Sciences*, 2020, 151:106258.
- [15] Langari M, Yang Z, Dunne JF, Jafari S, Pirault J-P, Long CA. Multiphase computational fluid dynamics–conjugate heat transfer for spray cooling in the non-boiling regime. *The Journal of Computational Multiphase Flows*, 2018;10:33-42.
- [16] Long CA. *Essential heat transfer*: Longman; 1999.
- [17] French CJST. Taking the heat off the highly boosted diesel. *SAE Technical Paper 690463*, 1969.
- [18] Finlay I, Boyle R, Pirault J, Biddulph T. Nucleate and film boiling of engine coolants flowing in a uniformly heated duct of small cross section. *SAE Technical Paper*; 1987.

- [19] Jafari S, Dunne JF, Langari M, Yang Z, Pirault J-P, Long CA. Control of Spray Evaporative Cooling in Automotive Internal Combustion Engines. *ASME Journal of Thermal Science and Engineering Applications*, 2018;10:041011.
- [20] Panão MR, Moreira AL. Intermittent spray cooling: a new technology for controlling surface temperature. *International Journal of Heat and Fluid Flow*. 2009;30:117-30.
- [21] Bostanci H, Altalidi S, Nasrazadani S. Two-phase spray cooling with HFC-134a and HFO-1234yf on practical enhanced surfaces. *Applied Thermal Engineering*. 2018;131:150-8.
- [22] Liang G, Mudawar I, Review of spray cooling—Part 1: Single-phase and nucleate boiling regimes, and critical heat flux. *International Journal of Heat and Mass Transfer*, 2017;115:1174-205.
- [23] Liang G, Mudawar I. Review of spray cooling—Part 2: High temperature boiling regimes and quenching applications. *International Journal of Heat and Mass Transfer*. 2017;115:1206-22.
- [24] Breitenbach J, Roisman IV, Tropea C. From drop impact physics to spray cooling models: a critical review. *Experiments in Fluids*, 2018;59:55.
- [25] Cheng W-L, Zhang W-W, Chen H, Hu L. Spray cooling and flash evaporation cooling: the current development and application. *Renewable and Sustainable Energy Reviews*. 2016;55:614-28.
- [26] Lin Y-K, Zhou Z-F, Fang Y, Tang H-L, Chen B. Heat transfer performance and optimization of a close-loop R410A flash evaporation spray cooling. *Applied Thermal Engineering*, 2019;159:113966.
- [27] Mudawar I, Bharathan D, Kelly K, Narumanchi S. Two-phase spray cooling of hybrid vehicle electronics. *IEEE Transactions on Components and Packaging Technologies*. 2009;32:501-12.
- [28] Zhao X, Yin Z, Zhang B, Yang Z. Experimental investigation of surface temperature non-uniformity in spray cooling. 2020;146:118819.
- [29] Lei S, Shi Y, Chen GJATE. A lithium-ion battery-thermal-management design based on phase-change-material thermal storage and spray cooling. 2020;168:114792.
- [30] Feng X, Zheng S, Ren D, He X, Wang L, Liu X. Key characteristics for thermal runaway of li-ion batteries. *Energy Procedia*. 2019;158:4684-9.
- [31] Xie Y, He X-j, Hu X-s, Li W, Zhang Y-j, Liu B. An improved resistance-based thermal model for a pouch lithium-ion battery considering heat generation of posts. *Applied Thermal Engineering*. 2020;164:114455.
- [32] Du J, Liu Y, Mo X, Li Y, Li J, Wu X, et al. Impact of high-power charging on the durability and safety of lithium batteries used in long-range battery electric vehicles. *Applied Energy*. 2019;255:113793.
- [33] Yang Y, Yang L, Du X, Yang Y. Pre-cooling of air by water spray evaporation to improve thermal performance of lithium battery pack. *Applied Thermal Engineering*. 2019;163:114401.
- [34] Ding Y, Cano ZP, Yu A, Lu J, Chen Z. Automotive Li-ion batteries: current status and future perspectives. *Electrochemical Energy Reviews*. 2019;2:1-28.
- [35] Varini M, Campana PE, Lindbergh G. A semi-empirical, electrochemistry-based model for Li-ion battery performance prediction over lifetime. *Journal of Energy Storage*. 2019;25:100819.
- [36] Mishra SK, Arora A, Chandra H. Application of Vibration on Heat Transfer-A Review. *i-Manager's Journal on Future Engineering and Technology*. 2019;15:72.
- [37] Sathyabhama A, Prashanth S. Enhancement of boiling heat transfer using surface vibration. *Heat Transfer—Asian Research*. 2017;46:49-60.
- [38] Atashi H, Alaei A, Kafshgari M, Aeinehvand R, Rahimi S. New Pool Boiling Heat Transfer in the Presence of Low-Frequency Vibrations Into a Vertical Cylindrical Heat Source. *Experimental Heat Transfer*. 2014;27:428-37.

- [39] Wang Z, Xing Y, Liu X, Zhao L, Ji Y. Computer modeling of droplets impact on heat transfer during spray cooling under vibration environment. *Applied Thermal Engineering*. 2016;107:453-62.
- [40] Zhang Y, Pang L, Liu M, Xie Y. Investigation of spray cooling: Effect of different heater surfaces under acceleration. *International Communications in Heat and Mass Transfer*. 2016;75:223-31.
- [41] Yoshida K-i, Abe Y, Oka T, Mori YH, Nagashima A. Spray cooling under reduced gravity condition. *Journal of heat transfer*. 2001;123:309-18.
- [42] Baysinger K, Yerkes K, Harris R, Michalak T, McQuillen J. Design of a microgravity spray cooling experiment. 42nd AIAA Aerospace Sciences Meeting and Exhibit 2004. p. 966.
- [43] Sone K, Yoshida K, Oka T, Abe Y, Miri Y, Nagashima A. Spray cooling characteristics of water and FC-72 under reduced and elevated gravity for space application. *IECEC 96 Proceedings of the 31st Intersociety Energy Conversion Engineering Conference: IEEE*; 1996. p. 1500-5.
- [44] Chen J, Zhang Z, Xu R, Ouyang X, Jiang P. "Numerical Investigation of the Flow Dynamics and Evaporative Cooling of Water Droplets Impinging onto Heated Surfaces: An Effective Approach To Identify Spray Cooling Mechanisms," *Langmuir*, vol. 32, no. 36, pp. 9135–9155, Sep. 2016, doi: 10.1021/acs.langmuir.6b02205.
- [45] Sarmadian A, Dunne J.F., Long C.A., Thalackottore Jose J., Pirault J-P, and Rouaud C. Heat flux correlation models for spray evaporative cooling of vibrating surfaces in the nucleate boiling region, *International Journal of Heat and Mass Transfer*, 160, 2020, 120159.
- [46] Sonin, A.A., 2004. A generalization of the Π -theorem and dimensional analysis. *Proceedings of the National Academy of Sciences*, 101(23), pp.8525-8526.
- [47] Petrus B, Zheng K, Zhou X, Thomas BG, Bentsman J. Real-time, model-based spray-cooling control system for steel continuous casting. *Metallurgical and materials transactions* 2011;42:87-103.
- [48] Jose JT. Spray Evaporative Cooling of Vibrating Surfaces with Application to Automotive Combustion Engines: PhD Thesis. University of Sussex; 2020.
- [49] Moffat RJ. Describing the uncertainties in experimental results. *Experimental thermal and fluid science*. 1988;1:3-17.
- [50] Schwarzkopf J, Cader T, Okamoto K, Li B, Ramaprian B. Effect of spray angle in spray cooling thermal management of electronics. *ASME 2004 Heat Transfer/Fluids Engineering Summer Conference: American Society of Mechanical Engineers Digital Collection*; 2004. p. 423-31.
- [51] Zhao X, Zhang B, Xi X, Yin Z. Analysis and prediction of single-phase and two-phase cooling characteristics of intermittent sprays. *International Journal of Heat Mass Transfer*. 2019;133:619-30.

Nomenclature

$\bar{v}$	average volumetric spray flux ($\text{m}^3\text{s}^{-1}/\text{m}^2$)
A	surface area (m^2)
a	Amplitude of vibration (m)
Ac	dimensionless acceleration (units of g)
Bo	Boiling number (-)
C_p	specific heat (J/kg K)
d	diameter(mm)
D	derivative gain
E	Error
δFo	Fourier Number
f	Frequency (Hz)
g	Gravitational acceleration
h	specific enthalpy (J/kg)
H	height (m)
I	Integral gain
Ja	Jacob number
k	thermal conductivity (W/m K)
$\dot{m}$	Mass flowrate (kg/s)
P	pressure (Pa)
P	proportional gain
q	heat flux (W/m^2)
Re	Reynolds number
St	Strouhal number ($\frac{fd}{v}$)
T	temperature (K)
ΔT	Temperature Difference ($^{\circ}\text{C}$)
t	Time (s)
δt	Time step-size
u	Velocity (m/s)
V	voltage
We	Weber number
I	current
Q	heat (kW)
x	position (m)
δx	Grid length
δBi	Biot number

Greek symbols

v	volumetric flow rate (m^3/s)
α	thermal diffusivity (m^2/s)
θ	temperature ratio
μ	dynamic viscosity (kg/m s)
ρ	density (kg/m^3)
σ	surface tension (N/m)
ω	angular velocity (Hz)
ϑ	spray cone angle
φ	function

Subscripts/superscripts

<i>adj</i>	adjacent
<i>at</i>	atmospheric
<i>c</i>	coolant
<i>d</i>	droplet
<i>ch</i>	chamber
<i>f</i>	liquid phase
<i>g</i>	vapour phase and gravitational force
<i>H</i>	heater surface
<i>i</i>	grid point
<i>imag</i>	Imaginary
<i>l</i>	liquid
<i>n</i>	time step
<i>m</i>	modified
<i>ref</i>	reference temperature
<i>s</i>	surface
<i>sat</i>	saturation
<i>sub</i>	subcooling
<i>V</i>	vibration
<i>v</i>	vapour
<i>w</i>	wall
32	Sauter Mean
05	median

Abbreviations

<i>HTC</i>	Heat Transfer Coefficient
<i>BC</i>	Boundary Condition

List of Figures

- Fig. 1. Temperature control diagram for closed-circuit spray evaporative cooling of a power-train part exposed to displacement vibration of the form $X = a\sin(\omega t)$.
- Fig. 2. Flat test-piece: (a) Chamber cross-section view, (b) From top to bottom, exploded view of shroud, bolts, flat test-piece, heater-block and shaker shaft (based on [48])
- Fig. 3. The $\frac{1}{4}$ -Cylinder test-piece: (a) photo of manufactured assembly; (b) Exploded view of assembly.
- Fig. 4. a) Flat test-piece assembly temperature via FEA, b) Test surface at which the spray cooling BC is applied, c) Surfaces at which the surrounding steam cooling BC is applied.
- Fig. 5. Flat test-piece FEA and 1D simulation results. a) test-surface temperature b) heat flux load.
- Fig. 6. a) $\frac{1}{4}$ -cylinder test-piece assembly temperature via FEA, b) Test surface at which the spray cooling BC is applied, c) Surfaces at which the chamber steam cooling BC is applied d) Surfaces at which the surrounding air-cooling BC is applied.
- Fig. 7. $\frac{1}{4}$ -cylinder assembly FEA and 1D simulation results. a) test-surface temperature b) heat flux load.
- Fig. 8. 1D Numerical model grid structure.
- Fig. 9. Control model plant diagram showing nodes to obtain 1D heat diffusion solution.
- Fig. 10. Experimental cooling control system hardware
- Fig. 11. Temperature control system using spray evaporative cooling.
- Fig. 12. Procedure for setting the controller hardware operating conditions
- Fig. 13. Flat test-piece static simulation of surface temperature control and flow rate compared with experimental measurement, also showing the 1D equivalent thermal load heat-flux supplied by the heater (top).
- Fig. 14. Flat test-piece dynamic ($f=400$ Hz, $a=0.02$ mm) simulated surface temperature control and flow rate compared with the experimental measurement, also showing the 1D equivalent thermal load heat-flux supplied by the heater (bottom).
- Fig. 15. Flat test-piece surface temperature control and flow rate under static and dynamic conditions ($f=400$ Hz, $a=0.02$ mm) obtained for 'fast' response control simulations, also showing the 1D equivalent thermal load heat-flux supplied by the heater (top).
- Fig. 16. Cooled metal surface temperature hardware control as a function of time for the $\frac{1}{4}$ -cylinder test piece under static and dynamic conditions.
- Fig. 17. Cooled metal surface temperature control as a function of time for the $\frac{1}{4}$ -cylinder test piece under static and dynamic conditions involving high-to-low, and low-to-high thermal load steps.
- Fig. 18. Controlled surface temperature and flow-rate for the $\frac{1}{4}$ -cylinder test-piece in static conditions: simulation and experimental measurements, also showing the 1D equivalent thermal load heat-flux supplied by the heater (bottom).
- Fig. 19. Controlled surface temperature and flow-rate for the $\frac{1}{4}$ -cylinder test-piece for dynamic conditions: simulation and experimental measurements, also showing the 1D equivalent thermal load heat-flux supplied by the heater (bottom).
- Fig. 20. Controlled surface temperature and flow-rate for the $\frac{1}{4}$ -cylinder test-piece for both static and dynamic conditions: experimental measurements only, also showing the 1D equivalent thermal load heat-flux supplied by the heater (bottom).