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Transient thermal analysis of flash-boiling cooling in the presence of high-heat-flux loads

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ABSTRACT

A flash-boiling fluid rapidly cools and expands, responding to depressurization in as little as 10–100 ms. The presented dynamic cooling mechanism harnesses this phenomenon. Applications include pulsed, high-heat-flux ($\sim$100 W cm^{-2}) devices, particularly those requiring strict temperature stability ($\pm$5 °C) for a short duration ($\sim$0.1 to 10 s). A highly conductive graphitic foam is included as an extended surface, while enhancing phase-change phenomena. In addition to quantifying the rate of heat transfer as it varies spatially and temporally, the temperature stability of the surrogate heat source is evaluated. The experiments were designed using a statistical framework, allowing for the efficient generation of surrogate models. These surrogate models are used to explore the multi-parameter design space, identifying design criteria that optimize different performance objectives, such as temperature stability, efficiency, and cooling rate. An inverse-heat-transfer technique is applied to determine the dynamic rate of cooling during the event. Cooling rapidly peaks after 0.5–1 s, reaching approximately 30–50 W cm^{-2}, and steadily decays thereafter. The cooling device maintains stable system temperatures ($\pm$5 °C) during heat loads of up to 104 W cm^{-2}.

1. Introduction

Active cooling mechanisms are commonly implemented to constrain the temperature of high-power-density systems. The design of such cooling mechanisms for dynamically actuated high-heat-flux devices presents a significant challenge. Not only must the cooling device cycle rapidly, the system must also be designed for transient performance to prevent thermal overshoot related to time constants inherent in any thermal system. Even under idealized circumstances, thermal fluctuations cannot be entirely eliminated [1].

Cooling mechanisms are typically evaluated under steady-state operation, impeding the design of dynamic thermal systems. Often steady-state characterization is appropriate because the majority of applications are likewise rated for continuous operation. To suit highly transient thermal applications, many of these steady-state cooling techniques could be applied, albeit in a transient manner. Common convective cooling technologies used to mitigate high-heat-flux loads include pool and flow boiling [2–4], spray [5], jet-impingement [6], and microchannel cooling [7].

1.1. Flash boiling

In this work, flash boiling is utilized as a cooling technique for dynamically actuated, high-heat-flux devices. Flash boiling [8–10] from a liquid pool is unique in the respect that the phenomenon is inherently transient. Once a liquid body is rapidly depressurized to a condition below the vapor pressure, the substance enters a metastable state, defined by uniform superheat. In this state, the heat of vaporization required is provided by the specific internal energy of the liquid, resulting in rapid phase change. The vaporization of superheated liquids is often characterized by the Jakob number. In contrast, traditional nucleate boiling features phase change constrained by a thermal boundary layer close to the heated surface.

Superheat-enabled phase change rapidly peaks and subsequently decays with time as the excess specific internal energy is expended [11–13]. Despite the limited duration, the mechanism may be suitable for transient applications that require pulsed cooling for periods ranging from 100 ms to 10 s. In particular, the flash appears to result in an initial peak cooling that wanes to a quasi-steady state value [14]. The high initial rate of cooling could be engineered to counteract the latent heat of the device.

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1.2. Graphitic foams

In the present implementation, the flash-boiling cooling mechanism is paired with a super-macroporous graphitic foam, acting as an extended surface for enhanced heat transfer. Similar to porous metal foams, graphitic foams contain a large surface area exposed for convection; however, the effective thermal conductivity is typically many-fold higher at equivalent densities [15]. The enhanced thermal performance of graphitic foams is attributed to the high thermal conductivity (>1300 W m^{-1} K^{-1}) of the ligaments [15]. The flow resistance is higher for graphitic foams than their metallic counterparts [16,17].

The pore diameter of graphitic foams is typically on the order of 100 μm with 60–85% open porosity. Values for the internal-surface-area-to-volume ratio (i.e., internal area) are commonly given in the literature to be around 5000–50,000 m^{2}m^{-3} [18–21]. The internal area is calculated assuming smooth pore walls [20], and therefore represents the surface area available for convection and neglects the presence of localized surface features, such as microcracks [15]. The internal area differs from the specific surface area measured for adsorption applications, such as by BET measurements.

Convective heat transfer within the foam is characterized using the Nusselt number, Nu, at the pore wall. The following relationship applies to convective cooling with foam in a channel [16,22]:

$$Nu = h_{sf}\frac{D_e}{k_f} \tag{1}$$

where h_{sf} is the heat transfer coefficient at the pore wall, k_f is the thermal conductivity of the fluid, D_e is the equivalent solid-sphere particle diameter, calculated as [22,20]:

$$D_e = \frac{6(1-\varepsilon)}{A_{sf}} \tag{2}$$

where A_{sf} is the internal area and ε is the porosity.

At steady state, the heat transfer from the heated base of the extended surface is:

$$q_{conv} = A_{eff}h_{sf}(T_b - T_f) \tag{3}$$

where the effective area of the foam, A_{eff}, accounts for the fin efficiency, η_f, of the porous medium by treating the pore walls as microfins. The resulting effective area is: $A_{eff} = \varepsilon A_{base} + \eta_f A_{sf} V_{foam}$, where A_{base} is the area of the heated wall to which the foam is adhered and V_{foam} is the total volume of the porous medium.

The fin efficiency of a finite-length extended surface is [23]:

$$\eta_f = \frac{\tanh\left(\sqrt{2h_{sf}/k_s L_{fin}}L_s\right)}{\sqrt{2h_{sf}/k_s L_{fin}}L_s} \tag{4}$$

where k_s is the thermal conductivity of the foam ligaments, L_{fin} is the equivalent fin thickness, and L_s is the height of the foam structure. The equivalent fin thickness is determined using the Taylor model [22,24]:

$$L_{fin} = \frac{(1-\varepsilon)}{\varepsilon}\sqrt{\frac{12K}{\varepsilon}} \tag{5}$$

where K is the permeability of the porous medium.

Using the foregoing derivation, correlations for the Nusselt number have been reported in the literature for single-phase convective cooling. These correlations take the general form:

$$Nu = CRe^m Pr^n \tag{6}$$

where Pr is the Prandtl number and the Reynolds number, Re, is defined using the pore diameter and the filter velocity. For the variant of POCOFoam®used in the referenced work, which had a porosity of 82%, a pore diameter of 500 μm, and an effective thermal conductivity of 120 W m^{-1} K^{-1}, the following empirical constants were reported: $C = 0.018, m = 0.27$, and $n = 0.33$ [22].

The Bond (Bo), capillary (Ca), and Grashof (Gr) numbers are often used to characterize pool boiling in porous media [2,25–27]. In the referenced works, the cooling systems pair graphitic foams with various fluorocarbon working fluids. The viscous forces are considered to be negligible as indicated by the Grashof (~10^2 to 10^3) and capillary numbers (~10^{-6} to 10^{-5}). Meanwhile, the Bond number (~0.1) indicates that surface tension inhibits bubble departure; consequently, working fluids with lower surface tension are suspected to be preferable for cooling processes with liquid-vapor phase change. The predominance of surface tension is also believed to inhibit local convection near the pore wall [22,25,26]. These dimensionless numbers are examined later in Section 4.5 for the present flash-boiling experiments, which instead uses methanol as the working fluid.

2. Experimental approach

2.1. Experimental setup

The setup for the flash-boiling experiments has been described in a previous publication [28]. A description of the main components is included here for convenience. The flash-cooling device is depicted in Fig. 1. The upper portion of the device is a polycarbonate housing containing an internal and external chamber. Inserted into this housing is a heat-spreader subassembly, consisting of a heat spreader, electrical resistance heaters, and a graphitic foam. The graphitic foam, KFOAM®P1, is brazed to the topside of the heat spreader using TiCuSil®braze compound from Morgan Technical Ceramics. The thin-film resistance heaters (LR1, American Technical Ceramics) were fabricated on an aluminum nitride substrate and are adhered to the heat spreader with a standard lead–tin solder. Using an ohmmeter, two heaters with matching resistance were selected. These heaters also have aluminum-oxide covers that insulate the active element from the surroundings both electrically and thermally. The thermal mass of the heaters is negligible relative to the other components. The heat-spreader subassembly is mounted on a 4 mm polyetherimide (ULTEM™1000) baseplate. Details are provided in Table 1.

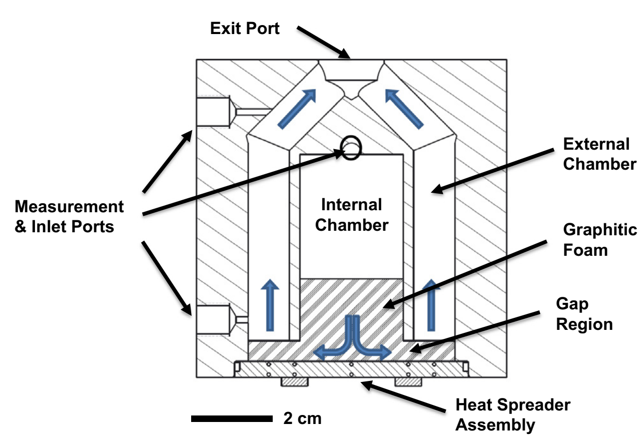

Fig. 1. The flash-cooling device consisting of a heat-spreader subassembly inserted into the polycarbonate housing. Relevant features of the device are annotated, and the direction of flow is indicated by arrows. The graphitic foam occupies the region with grouped hatches, extending from the internal chamber, through the gap region, and into the bottom of the external chamber. The locations of the thermocouples embedded in the heat spreader are indicated by circles. All features are to scale.