Practical Considerations for Pulsating Heat Pipe and Embedded Heat Pipe Heat Spreaders

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Abstract

Increasing heat loads in power electronics and embedded computing applications require improved thermal management strategies. Embedded heat pipes and pulsating heat pipes (PHPs) are both passive two-phase technologies used for enhancing the thermal conductance of electronics heat spreaders. While conventional heat pipes are a mature technology, PHPs are an emerging technology that has demonstrated benefits in some thermal management applications. Here, operating principles, theoretical performance limits, and manufacturing considerations of both technologies are considered. Recent experimental results for a PHP with ammonia working fluid are presented. A 3U form factor conduction card with embedded PHP channels was tested over a range of sink temperatures (-10°C to 30°C) and fluid fill factors (50%, 64%, and 75%), at applied heat fluxes of up to 52 W/cm^2. At the optimum fill ratio of 64%, a 50% reduction in thermal resistance was observed.

1 Introduction

1.1 Power Electronics Thermal Management

As power electronics evolve, thermal management solutions using conduction only heat spreaders will not be sufficient for high heat flux requirements. Passive two-phase heat spreaders can be used to significantly enhance the conductance over baseline aluminum substrates. Embedded copper-water heat pipes are a proven technology for improving heat rejection in power electronics. Pulsating heat pipes (PHPs), also known as oscillating heat pipes, are an emerging two-phase thermal management technology that shows promise for certain electronics thermal management applications [1].

1.2 Heat Pipe Embedded Heat Spreaders

Heat pipes are passive heat transfer devices that transport heat by the two-phase flow of a working fluid [2]. They consist of a vacuum sealed metal envelope filled with a saturated working fluid and a porous wick structure. Heat applied in the evaporator region vaporizes working fluid from the wick. The vapor flows to the colder condenser region where it is condensed, with the resulting liquid returning to the evaporator by capillary action in the wick structure.

For terrestrial electronics cooling, copper-water heat pipes are the most common material/fluid combination. In many cases, copper-water heat pipes are embedded into aluminum heat spreaders, enabling the mechanical strength and mass of the original material, with significantly enhanced (up to 10×) thermal conductance.

1.3 Pulsating Heat Pipes

Conventional heat pipes utilize the latent heat of vaporization of a working fluid to transport heat, and rely on capillary action in a wick structure for liquid return. Pulsating heat pipes, however, operate via a different mechanism. A PHP consists of a continuous serpentine channel of capillary dimensions embedded in a substrate, as shown in Fig. 1. The serpentine is partially charged with a working fluid and hermetically sealed. When working fluid is introduced into the channels, it naturally distributes into liquid slugs and vapor plugs. Heat transfer between the heated section (evaporator) and the cooled section (condenser) is achieved through the pulsating action of the slugs and plugs generated by instabilities as vapor bubbles are generated and condensed in the evaporator and condenser, respectively.

While embedded heat pipes are a mature technology, with well understood operating principles, PHPs are an active area of research, with the operating mechanisms, heat transfer performance limits, and effective thermal conductances being active areas of study. In this paper, the characteristics of both embedded heat pipes and PHPs are discussed, with the goal of providing practical considerations for the selection and design of both technologies in different applications. This is supported by recent experimental PHP heat spreaders for modular power electronics.

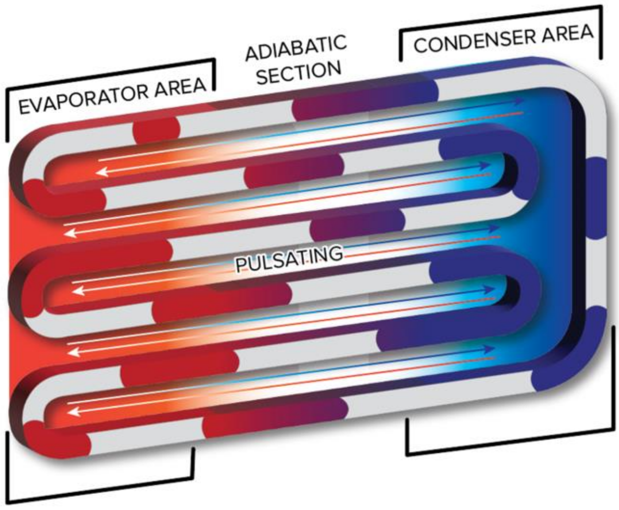

Fig. 1 Operating principles of a pulsating heat pipe (PHP)

2 Thermal Performance Limits

2.1 Heat Pipe Performance Limits

In conventional heat pipes, the working fluid is selected based on the operating temperature range and fluid properties, according to the heat pipe merit number, according to Eq. 1, where ρ_l is the liquid density, σ is the fluid surface tension, λ is the latent heat, and μ_l is the liquid viscosity [2].

$$M_l = \frac{\rho_l \sigma \lambda}{\mu_l} \quad (1)$$

The heat pipe merit number is derived from the heat pipe capillary limit, which is the primary performance limit for conventional heat pipes. The capillary limit is the maximum power the heat pipe can carry and still return the condensed liquid to the evaporator through the wick, and is calculated by balancing the capillary force against the liquid and vapor pressure drops and gravity head (Eq. 2).

$$\Delta P_c = \frac{2\sigma}{r_c} > \Delta P_l + \Delta P_v + \Delta P_g \quad (2)$$

Other heat pipe limits include the viscous limit, in which viscous forces hinder the flow of vapor, and the entrainment, or flooding, limit in which vapor shear forces hinder liquid return. Finally, there is the boiling, or dry-out limit, in which the heat flux is sufficiently high to dry out the evaporator. The heat flux limit for typical copper-water heat pipes is around 75 W/cm^2.

2.2 PHP Performance Limits

When selecting a PHP working fluid, several things must be considered. In general, fluids with steep saturation curves in the operating temperature range show higher performance. Considering other relevant fluid properties (liquid density ρ_l, liquid specific heat C_{pl}, compressibility Z, liquid viscosity μ_l, and latent heat h_{fg}) a PHP merit number can be assigned in Eq. 3 [3].

$$M_{PHP} = \frac{\rho_l C_{pl} \left(\frac{\partial P}{\partial T}\right)_{sat} ZRT_{sat}}{\mu_l h_{fg}} \quad (2)$$

The merit number aids in the selection of working fluids for PHPs; other considerations include the serpentine channel dimensions, fill ratio (typically between 20-80%), and superheat required for startup.

Some performance limits have been identified for PHPs [4]. The most important of these limits are the vapor inertial limit and swept-length limit. The vapor inertial limit occurs when the velocity of the vapor plugs is sufficiently large to overcome the meniscus of the liquid slugs, and the flow becomes annular. The swept length limit occurs when the evaporator region is sufficiently long that liquid slugs cannot pass through the full length. An additional operating limit for PHPs is the startup limit. There is a minimum amount of applied heat that is required to generate the chaotic fluid motion that provides the PHP with its heat transport capabilities. As a consequence, at low powers, the PHP will have higher thermal resistance until the pulsations are established.

Details of the calculations of the various PHP performance limit are found in the literature [4]. The calculation of these limits provides an operational envelope for a given PHP design. As an example, Fig. 2 plots the vapor inertial, swept length, and startup limit for a nominal PHP using ammonia as a working fluid. The PHP dimensions in this calculation include a channel hydraulic diameter of 1.6 mm, an evaporator length of 2.54 cm, a condenser length of 2.54 cm, and overall length of 15.24 cm. In this PHP, eight channel passes are filled with a 50% liquid fill ratio.

In this example, the vapor inertial limit is the primary limit at lower temperatures and is relatively constant at around 500 W. Above a temperature of around 60°C, the swept length limit becomes the dominant limit and reduces the maximum heat transport capability of the PHP. It is noted that in this configuration, the startup limit is predicted to be fairly low, while experiments suggest it may be somewhat higher than predicted. For PHPs, these limits are not as well established as those of conventional capillary-driven heat pipes, and additional experimental and theoretical work is needed to validate them. For the PHP designer, for a given application the PHP operating point should be well within the center of the operating envelope defined by the calculated limits.

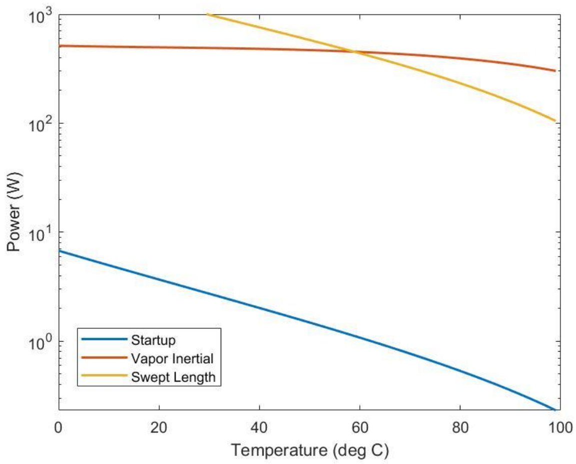

Fig. 2 Operating envelope of a nominal ammonia PHP

3 Practical Considerations

3.1 Geometric Considerations

When considering an embedded heat pipe or PHP solution for a given electronics thermal management application, there are a number of considerations to be made, in addition to fluid selection and evaluation of the various heat transfer performance limits. Certain geometric constraints may also influence the choice between the two solutions.

Both heat pipes and PHPs can operate in any orientation, within length limitations determined from the performance limits. The smallest thickness of an embedded heat pipe heat spreader is roughly 3.5 mm; below this thickness, the vapor space is not sufficient to carry large powers. However, PHPs can be made thinner than 3 mm, depending on the manufacturing process. On the other hand, there is an upper limit on the channel size for PHPs. Shown in Eq. 4, there is a critical diameter for PHPs channels based on the Bond number. Channels with dimensions above this are not able to sustain the liquid plugs that are integral to PHP operation.

$$D_c = 2\sqrt{\frac{\sigma}{g(\rho_l - \rho_v)}} \tag{4}$$

The size of the heat source is also a consideration. With embedded heat pipes, the size of the evaporator is taken into account in the design and layout of the heat pipes, but is generally not limited. For PHPs, as can be seen in the swept length limit, larger evaporators are detrimental to PHP performance. However, PHPs may be able to sustain higher heat fluxes (>75 W/cm^2) than conventional heat pipes if designed properly.

Heat pipe embedded heat spreaders are commonly planar, but heat pipes can be bent to conform to different geometries. However, there is a limit to bend radius of a heat pipe to prevent choking the vapor flow, and also limited by the stiffness of the wick structure. However, PHPs can be conformal to nearly any geometry, with a tight bend radius on the order of the channel diameter. This for example enables heat transfer around 90° bends, as shown later in this paper.

3.2 Manufacturing Considerations

Another consideration when selecting an embedded heat pipe heat spreader solution or PHP for a given application is the different manufacturing processes involved. Embedded heat spreaders are typically manufactured by machining grooves into the base aluminum and pressing the heat pipes into the grooves. The heat pipes may be bonded by solder or epoxy, and the surface then machined flat. This enables structures that have almost the same strength and mass as the base aluminum, but with significantly enhanced thermal conductance.

In the manufacturing of PHPs, a similar approach can be taken in which the channels consist of tubing which is embedded in a base material. Alternatively, the channels can be directly machined into a substrate, and a lid brazed or otherwise bonded to seal the channels. Additionally, PHPs are uniquely adapted to fabrication by additive manufacturing methods, where the channel structures can be directly printed. In a selective powder sintering additive manufacturing process, the challenge then becomes ensuring the channels are clear of residual powder.

4 Case Study – 3U Conduction Card Experimental Results

A 3U form factor conduction car, with dimensions of 160 mm × 100 mm x 3.38 mm, was selected as a common form factor for testing the thermal performance of embedded heat pipe and PHP heat spreaders. The heat spreader is designed to interface with a card retainer when mounted into a chassis. A CAD model of the PHP design, illustrating the layout of the capillary channels, is shown in Fig. 3. Note how the PHP channels conform to the 90° bend, enabling better transfer of heat to the card retainer and chassis. In this work, the PHP was fabricated by additive manufacturing of an aluminum alloy.

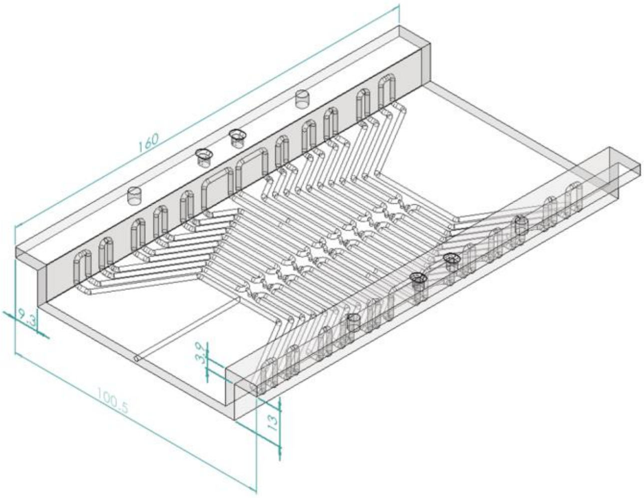

Fig. 3 3U PHP heat spreader design with fluid channel layout

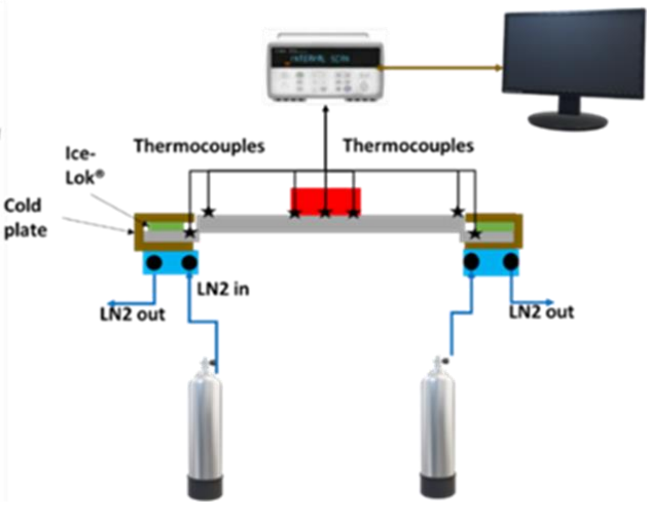

Fig. 4 Schematic of experimental setup for PHP thermal performance evaluation.

A schematic of the experimental setup used in the thermal performance evaluation is shown in Fig. 4. A 2.54 cm × 2.54 cm heat source is applied in the center of the conduction card. Each side of the conduction card is fixtured into a cold rail using a card retainer, which is temperature controlled and cooled by liquid nitrogen. Plunger-type thermocouples are used to measure the surface temperature of the PHP directly under the heat source. This temperature is used to calculate the experimentally determined thermal resistance, according to Eq. 5.

$$R_{th} = \frac{(T_{evap} - T_{sink})}{Q} \qquad (5)$$

In previous work, a heat pipe embedded conduction card and previous PHP design iterations were tested [5-7]. Here, additional test results for the PHP conduction card shown in Fig. 3 using ammonia as the working fluid at different liquid fill ratios are presented.

Three fluid fill ratios were evaluated: 50%, 64%, and 75%, defined as the volume fraction occupied by liquid at room temperature. For each fluid fill ratio, the thermal performance was evaluated at several different condenser temperatures (-10°C to 30°C). The experimentally measured thermal resistance for a 50% fill ratio is shown in Fig. 5. Each curve in Fig. 5 is obtained through an individual test. In each test, the cold rail temperature is maintained at the set value by modulating the flow of liquid nitrogen. Heater power is applied, and increased; at each step increase in power, the temperatures are allowed to reach a steady state. The measured temperatures are then used to calculate the thermal resistance per Eq. 5. Each test is concluded when the evaporator temperature reaches 80°C, representative of a maximum operating temperature for many electronics. Also shown in Fig. 5 is the measured thermal resistance of a plain aluminum conduction card of the same dimensions.

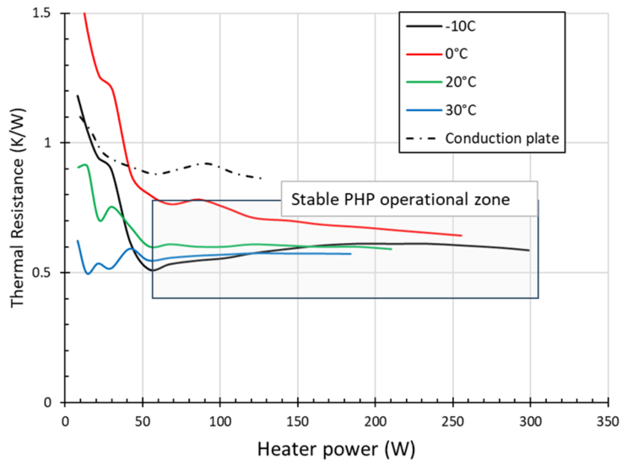

Fig. 5 Experimental PHP heat spreader thermal resistance, measured at different sink temperatures (50% fluid fill ratio)

From Fig. 5, it can be observed that the PHP design reduces the thermal resistance of the conduction card by up to 33% compared to the baseline conduction. At lower powers (< 50 W), the PHP has a higher thermal resistance, then converges to a lower value of around 0.6 K/W. Higher thermal resistance at lower heat flux is typical of PHPs.

Figure 6 shows similar test results for a 64% fluid fill ratio. At low powers, the measured thermal resistance is high. It is not until a power of at least 50 W is applied that the PHP properly starts up, and the thermal resistance converges to a value of approximately 0.5 K/W, a 50% improvement over the baseline aluminum conduction card

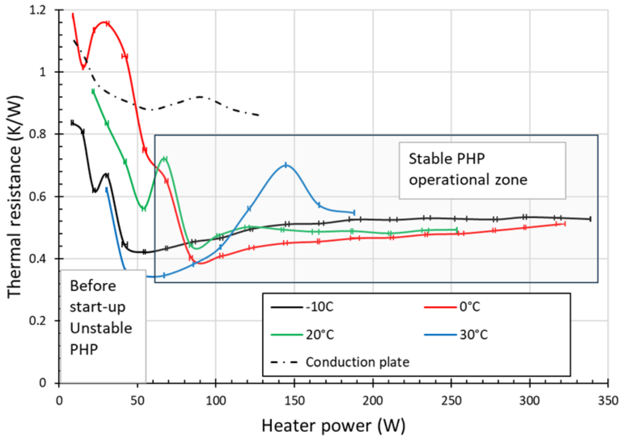

Fig. 6 Experimental PHP heat spreader thermal resistance, measured at different sink temperatures (64% fluid fill ratio)

Finally, the PHP conduction card was tested with a fill ratio of 75%. As shown in Fig. 7, at the lowest sink temperature of -10°C, the PHP conduction card operated quite well, with a thermal resistance of approximately 0.4 K/W. However, the thermal resistance increased with an increased sink temperature of 0°C, and for higher sink temperatures of 20°C and 30°C, the PHP would not start up at all.

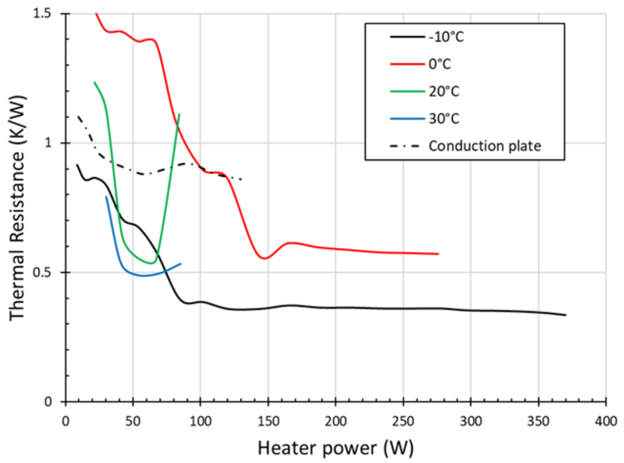

Fig. 7 Experimental PHP heat spreader thermal resistance, measured at different sink temperatures (75% fluid fill ratio)

From the above experimental results, for the particular PHP design configuration tested, a fluid fill ratio of 64% was determined to be optimal. Testing demonstrated the ability of the PHP to significantly reduce the overall thermal resistance of the conduction card (i.e., enhance the effective thermal conductance of the baseline material), enabling much higher applied powers (up to 340 W, versus 125 W for baseline conduction). For the footprint of the heater, this equates to a heat flux of up to 52 W/cm^2. The testing also demonstrated the challenges of startup at lower heat fluxes.

5 Conclusions

Both PHP and embedded heat pipe heat spreaders can provide significant improvement in the effective thermal conductance over that of pure conduction. Selection of a PHP or embedded heat pipe solution is dependent on several factors, including total power, heat flux, geometrical or mass constraints, and operating temperature range. Proper selection of the PHP working fluid and optimization of the filling ratio can significantly impact performance, and relevant performance limits must be calculated. Recent experimental results of an ammonia PHP conduction card at different sink temperatures and fluid fill ratios demonstrate the potential for significantly enhanced thermal conductance over the baseline aluminum. Future work will include further development of PHP heat transfer models for accurate predictions of performance and theoretical limits, as well as considerations of different PHP fabrication processes.

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