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EMBEDDED PULSATING HEAT PIPE RADIATORS

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ABSTRACT

With the aim of exploring potential applications of pulsating heat pipe (PHP) technology for space as well as terrestrial sectors, experimental study of embedded structures in an aluminum substrate subjected to conjugate heat transfer conditions, i.e. natural convection and radiation, has been carried out under different thermo-mechanical boundary conditions. To compliment the experimental study, system level 3D computational simulation of the complete experimental set-up has also been undertaken using commercial software. The effective thermal conductivity of the embedded structures has been estimated to be ~2500 W/mK by comparing experimental spatial temperature distribution on the plate with corresponding simulations. The study reveals that embedded PHP structures can be beneficial only under certain conditions. The effectiveness of such structures asymptotically levels off as its thermal conductivity increases beyond a particular limit. In addition, the degree of isothermalization of the radiator plate strongly depends on its thermal properties.

KEY WORDS: pulsating heat pipe, conjugate heat transfer, space radiators

1. INTRODUCTION

At present, there are many proven space radiator systems, such as [1]:

- Passive structure / honeycomb panels
- Conventional heat pipe embedded structures
- Metal Matrix Composites (AlSiC, Graphite/Al)
- Variable area actively controlled radiators
- Liquid pumped loop (NH_3)
- Loop Heat Pipe (LHP)/Capillary Pumped Loops

The concept of using an embedded pulsating heat pipe for space/terrestrial radiators is novel. To the best of knowledge of the authors, there is no information available in the open literature on this topic. In this background this work explores the applicability of an embedded Closed Loop Pulsating Heat Pipe (CLPHP) system so as to reduce the internal thermal resistance of 'radiator' plates. Such generic plates may find application in space environment as well as terrestrial applications. Embedded CLPHP aluminum structures are computationally simulated and experimentally tested under various boundary conditions including pure radiation and conjugate mixed mode conditions.

How effective will these systems be? To answer the questions we will first make an attempt to understand the situation from a 1-D point of view and then extrapolate the analogy to three dimensions. Considering a generalized case of 1-D heat transfer through a plate (Figure 1) along with the various resistances are given by [2]:

$$(T_1 - T_2)/\dot{q}'' = R_{th}\big|_{conduction} = \Delta x / k_{plate}$$

$$(T_2 - T_\infty)/\dot{q}'' = R_{th}\big|_{convection} = h^{-1} \qquad (1)$$

$$(T_2 - T_\infty)/\dot{q}'' = R_{th}\big|_{radiation} = (4 \cdot \sigma_r \cdot T_m^3)^{-1}$$

where

$$T_m = \left[\frac{(T_2^2 + T_\infty^2)(T_2 + T_\infty)}{4}\right]^{1/3}$$

The Biot Number compares the thermal resistance offered by the plate and the external convection:

$$Bi = \frac{conductive\ resistance}{convective\ resistance} = \frac{(\Delta x / k_{plate})}{(1/h)} \qquad (2)$$

A small Bi (<0.1) indicates that heat transfer is primarily limited by the external transport coefficient.

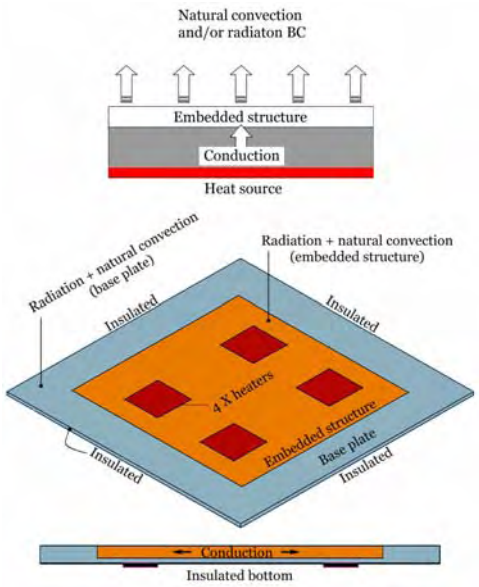

Figure 1: 1-D and 3-D schematic of the problem

Under pure radiative boundary, an analogous Conduction-Radiation Number Nr, is defined as [2]:

$$Nr = \frac{conductive\ resistance}{radiative\ resistance} = \frac{4 \cdot \sigma \cdot T_m^3 \cdot \Delta x}{k_{plate}} \quad (3)$$

This work focuses on reducing the $R_{conduction}$ by embedding a PHP in a radiator plate (see Figure 1). The pros and cons of such a design were analyzed by experimental and simulation studies. $R_{radiation}$ is generally reduced by increasing the surface emissivity of the radiator by suitable coatings. In terrestrial applications, forced convective conditions are suitably used to reduce $R_{convection}$.

2. EXPERIMENTAL DETAILS

The experimental setup was designed to generate new information on applicability / feasibility of embedded CLPHP concept as heat spreaders/ radiators under conjugate heat transfer conditions i.e. natural convection and radiation. The radiator base plate was made of aluminum with dimensions 350 X 350 X 5 mm^3, as shown in Figure 2. Grooves having a semicircular bottom were milled on this aluminum base plate inside which the CLPHP capillary tube (ID/OD: 2.0/3.0 mm, length 255 mm and inter-tube pitch 12.0 mm, 11 turns on each end, volume:17.5 cc) was fitted with application of heat sink compound. Four surface mountable flat mica heaters (50.8 X 50.8 X 1.0 mm^3) were mounted on the bottom side of

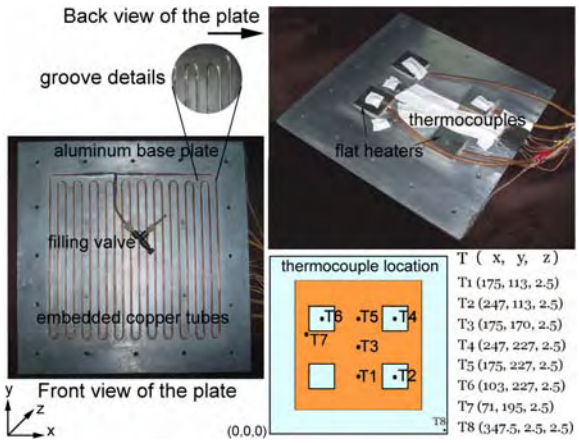

Figure 2: Details of the experimental setup

aluminum base plate. Eight K- thermocouples were used to measure temperature at important spatial locations. Data acquisition was done using 24-bit PC interface. For DC power input, a digitally controlled power source coupled with high precision (0.25% FS) digital multimeter was used. The entire base plate was insulated from the back side (where the heaters are located) and mounted on a tiltable frame. The setup was erected in a 'black' enclosure open from top and bottom for unhindered natural convective air currents. All experiments were performed with a FR = 50%. When no PHP action was desired, the CLPHP tube was dry and empty, under vacuum. The boundary condition on the top surface was conjugate i.e. combined natural convection + radiation ($\varepsilon \approx 0.95$). Global orientation of the plate was either vertical or horizontal. After proper filling and sealing of the CLPHP, desired heat input was given to the surface mounted heaters. Power input to individual heaters was increased from 20 W (1.2 W/cm^2) to 62.5 W (2.5 W/cm^2). Equal electrical power was dissipated by all four heaters. Temporal response of all thermocouples was recorded, from commencement of heating till quasi steady state was achieved.

3. MODELING AND SIMULATION

A 3-D tetrahedron computational grid domain (2.1e^5 grid points) was generated exactly representing the entire physical domain of the experimental set-up (with ICEMCFD/FLUENT®, see Figure 3 for details). Distinction is made between the grid volume corresponding to the radiator base plate and that corresponding to the embedded CLPHP. A constant heat flux boundary condition is assigned to the gird points corresponding to the surface heaters.

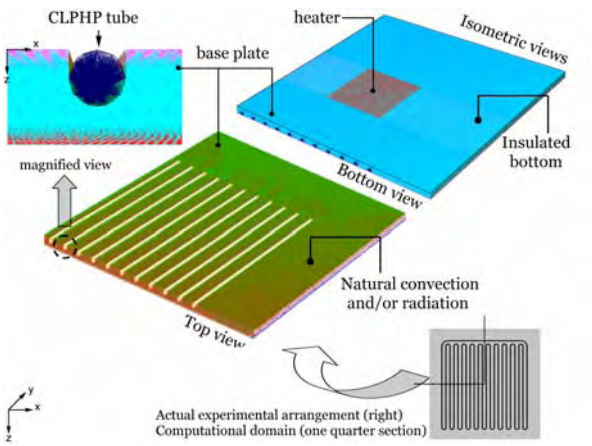

Figure 3: Details of the computational grid and the physical experimental domain

At the upper surface of the plate, conjugate boundary condition (natural convection + radiation) is applied. The lower and the side surfaces of the plate have insulated boundary condition[1]. All simulations were performed under transient conditions until a steady state is reached. Following assumptions are made: (i) All materials are isotropic and homogeneous, and (ii) perfect contact is maintained at all interfaces.

As there is no 'first principle' mathematical model yet available for the complex pulsating CLPHP phenomena [3, 4], here, it is modeled with an effective thermal conductivity approach (in conjunction with the experimental data), as described below:

1. For a given radiator plate orientation (horizontal or vertical), a given heat power input (all heaters were given the same power), the temporal variations of temperature at various locations (i.e. T1 to T8) were recorded till a steady state was achieved. This exercise was done under two conditions: (a) When the CLPHP tubes were filled with FR = 50% of deionized/ degassed water, rendering enhanced thermal conductivity to the plate. (b) When no working fluid was present in the CLPHP tubes, i.e. no enhancement in average thermal conductivity of the plate. The difference in respective steady state temperature at all eight locations on the plate in both the above situations (with and without CLPHP operation) was recorded.

[1]Since the entire experimental setup is symmetric about two perpendicular mid-planes, computations have been performed only on one quarter section with symmetric boundary conditions on the cutting planes. This situation is only true for horizontally oriented plate. For vertical orientation, thickening boundary layer leaves only one plane of symmetry.

2. Once the experimental data base as described above was generated, simulations were performed on the 3D-grid with exactly same boundary conditions as applied to experimental runs. The only unknown variable was the k_{PHP} of the embedded CLPHP tubes, which was iteratively changed in the simulations till a reasonably accurate matching of the steady state temperature distributions was obtained between the simulated results and the experimental values. Thus, an estimate of average effective thermal conductivity of the embedded CLPHP tubes could be ascertained.

4. RESULTS AND DISCUSSION

The 3D computational model was thoroughly validated with benchmark data generated with the plate without CLPHP action[2]. Once validation of the computational grid was established, simulations were first performed to predict the thermal performance of the plate under different boundary conditions. The only unknown parameter at this stage was the effective thermal conductivity, k_{PHP} of the CLPHP. For the purpose of simulation, k_{PHP} was varied from 200 W/mK to 5000 W/mK. That this range is representative and is applicable is suggested by literature [e.g.3-5]. Figure 4 shows typical simulation thermogram with aluminum and steel base plate respectively with 50 W/heater subjected to conjugate heat transfer on the top surface of the plate. The plate is horizontally oriented. Figure 5 compliments Figure 4 by depicting the variation of T_{max} and T_{min} with increase in effective k_{PHP}. From these simulations it is clear that:

- The advantage achieved by embedding a heat pipe structure, so as to decrease the conductive resistance of the plate, asymptotically approaches a finite value as thermal conductivity of the embedded structure increases. Therefore, it is not advisable to invest in a technology which increases the thermal conductivity of the radiator plate beyond a particular limit. In this context, CLPHPs are ideally suited. Their thermal performance may never be as good as conventional heat pipes but they are much simpler and cheaper to fabricate [6]. Also, they are not affected by conventional heat pipe limits (e.g. capillary limit). In parallel, if embedded CLPHPs can provide a composite thermal conductivity of the radiator plate which is just 'good enough',

[2] Validation results are not shown due to space constraints

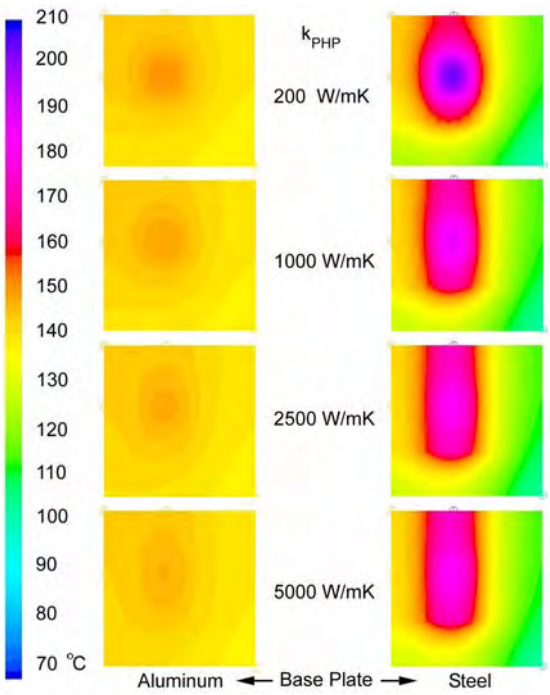

Figure 4: Simulation results for aluminum and steel base plate with varying thermal conductivity of CLPHP tubes with 200 W power input (50 W to each heater), horizontal orientation, under conjugate boundary conditions

then there is no real need to opt for a complicated and expensive technology which may be giving a better overall thermal conductivity. This scenario emerges because the heat transfer gets limited by external resistance (convection, radiation or conjugate) as compared to the internal conductive resistance (refer Eq. 1, 2). In the present simulations, relative advantage of the embedded structure decreases beyond ~ 3500 W/mK (see Figure 5).

- In case of aluminum base plate, the change in T_{max} and T_{min} on the plate, although clearly noticeable, is quite low even if the thermal conductivity of the CLPHP tubes is increased by an order of magnitude. (Refer Figures 4, 5). This scenario prevails because (a) the thermal conductivity of aluminum is itself quite high so enhancing it further by the embedded CLPHP does not help much, (b) the radiator base plate is quite thick, i.e. 5.0 mm which leads to a decrease in thermal resistance in the spanwise direction. As thermal conductivity of the base plate is reduced, the advantage of embedding CLPHPs for improving isothermalization is clearly manifested.

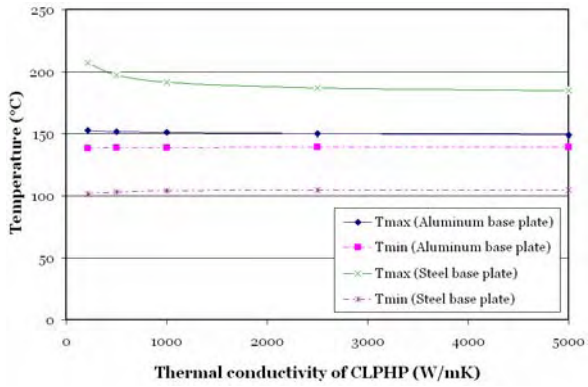

Figure 5: Variation of maximum and minimum plate temperature when k_{PHP} embedded CLPHP is increased (50 W supplied to each heater)

- For a given heat input, the maximum temperature will also depend on the base plate thermal conductivity. As the Bond Number criterion [6, 7] for critical diameter for CLPHP operation is temperature dependent, whether a CLPHP with a given working fluid will indeed work satisfactorily at the prevailing operating temperature needs to be addressed independently.

4.1 Effective thermal conductivity of the PHP

Figure 6 shows a time-temperature graph of the T_{max} (T6) and T_{min} (T8) on the horizontally oriented radiator plate with a power input of 200 W (50 W to each heater) subjected to conjugate boundary conditions on the top surface. For $t < t_0$ the CLPHP is operational while at $t = t_0$, the filling/charging valve of the CLPHP is opened allowing the working fluid inside to leak out, thus emptying the CLPHP. Therefore, for $t > t_0$, there is an increase of T_{max} and

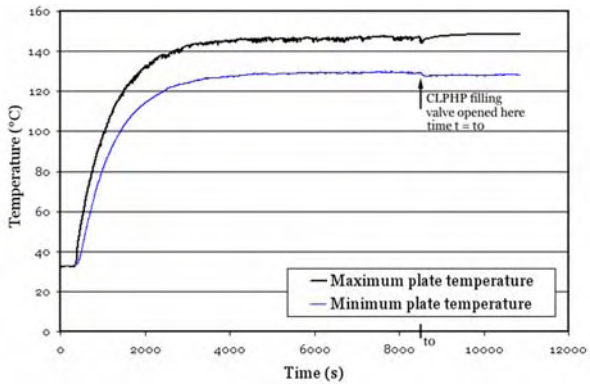

Figure 6: Time-temperature plot of T_{max} and T_{min} on the plate at 200 W input and vertical orientation.

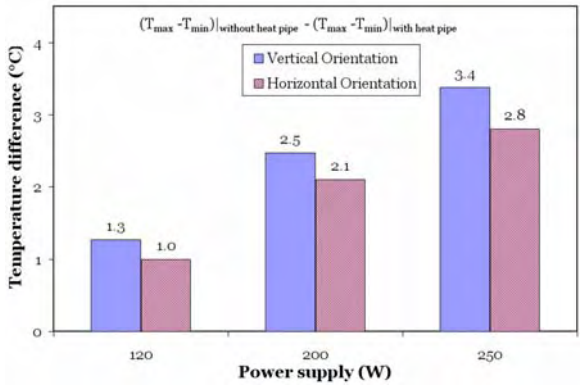

Figure 7: Experimental data for average change in ($T_{max} - T_{min}$) with and without the CLPHP at various heat input conditions

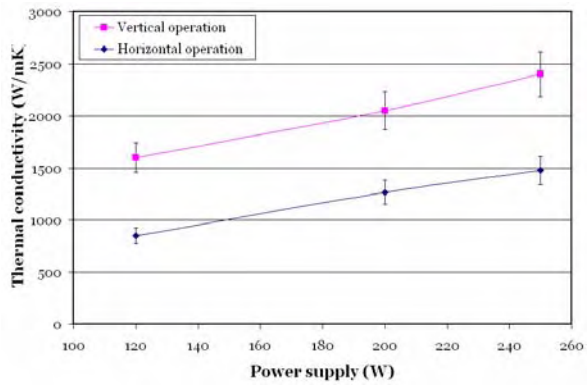

Figure 8: Estimated effective thermal conductivity of CLPHP under different operating conditions after comparison with respective simulations.

a decrease on T_{min}. As seen, the average value of (T_{max} - T_{min}) with working CLPHP is 16.8°C while the same increases to 19.3°C after the working fluid is removed from the device. Figure 7 summarizes the results for other similar experimental runs for different heat inputs. The data suggests that the CLPHP tubes were indeed rendering a higher k_{eff} to the radiator, although modest for aluminum base plate. In fact, this observation has already been made in the earlier simulation results (refer Figure 5).

Having generated the experimental data as given above for various power levels and operating orientations (i.e. T1 to T8, (T_{max} - T_{min}) with and without heat pipe), simulations were again performed to estimate the effective thermal conductivity of the CLPHP tubes. The value of k_{PHP} of CLPHP tubes was iteratively improved in the 3D model till a reasonably good fit to the experimental data (steady state spatial temperature) was obtained. Figure 8 gives the results by following this procedure. The following conclusions can be drawn:

- Effective k_{PHP} increases with increasing heat flux. This is inline with the earlier observations [e.g. 8-10]. This is a direct consequence of the fact `that the thermal power input is the driving force for the self sustained oscillations inside the device, leading to higher internal heat transfer coefficients.
- The CLPHP operates better in vertical orientation than in the horizontal orientation. This observation also supports earlier studies. It should be noted that the applied heat flux in the present experimental runs varies from 1.2 W/cm^2 to 2.5 W/cm^2. It has been pointed out in the literature that a certain minimum heat flux > ~ 5 W/cm^2 is needed to make the thermal performance independent of operating orientation [6]. In the present experiments this condition was not met and therefore the effective conductivity was different for vertical and horizontal operation.
- It is seen from the results, simulations as well as from experimental data that the temperature differentials with and without the CLPHP are not very high because of the aluminum base plate. If the base plate was made of a material having lower thermal conductivity then the estimation of effective k_{PHP} would have been more accurate. Nevertheless, the exercise has indicated effective k_{PHP} values which are inline with the trends being projected in the literature.

5. CONCLUSIONS

PHPs have emerged as promising candidates which can cater to high waste heat and flux density in modern electronic equipment, including space and terrestrial sectors. With this background, a CLPHP structure embedded in an aluminum substrate has been studied under conjugate boundary conditions. The following are the major conclusions:

- In those situations where external heat transfer coefficient limits the heat throughput (i.e. cases like natural convection or pure radiation, where the external 'h' is low), the thermal conductivity of the base plate on which a heat pipe (conventional or PHP) is embedded does not play a 'major' role in improving the heat transfer scenario, after a particular minimum value of its effective thermal conductivity is attained. In the present case, the base plate is made of 5 mm thick aluminum and so putting an embedded PHP is only marginally improving the performance. The bare aluminum

base plate is itself doing a good enough job. If the plate thermal conductivity is low, the relative advantage achieved by embedded PHP systems is appreciably higher. In space applications, such a material may be a composite which gives a weight/strength advantage as compared to aluminum, but is thermally not that attractive. The spanwise thermal conductivity of such a composite may be tailored by suitably embedding a CLPHP structure.

- Under the boundary conditions studied, the typical values of thermal conductivity achieved by PHPs (900 W/mK – 2500 W/mK) are quite attractive for its use as embedded structure in radiator plates.
- Simulations indicate that relative advantage of any enhanced thermal conductivity device embedded in a radiator plate (such as heat pipes, PHPs etc.) will decrease as the absolute value of thermal conductivity of the enhancement device/structure increases beyond a particular value.

NOMENCLATURE

Bi : Biot Number = hL_c / k_s
Bo : Bond Number = $\sqrt{Eö}$
D : diameter (m)
Eö : $(D \cdot g \cdot \rho_{liq}) / (\sigma / D)$
FR : volumetric filling ratio (V_{liq}/V_{total})
g : acceleration due to gravity (m/s^2)
h : heat transfer coefficient (W/m^2K)
k : thermal conductivity (W/mK)
L : Length (m)
Nr : Conduction-Radiation Number
$\dot{q}''$: heat flux (W/m^2)
R_{th} : thermal resistance (Km^2/W)
T : temperature (°C or K)
x, y, z : length or axes

Greek symbols

ε : surface emissivity
σ : surface tension (N/m)
σ_r : Stephan-Boltzman const. = 5.67e-8W/m^2K^4
ρ : density (kg/m^3)

Subscripts

∞ : ambient
c : characteristic
liq : liquid
m : mean
s : solid

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