A tree-on-a-chip: design and analysis of MEMS-based superheated loop heat pipes exploiting nanoporous silicon membranes

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Abstract. This paper reports the design, fabrication and analysis of a plant-inspired, MEMS-based superheated loop heat pipe (SHLHP) that would exploit nanoporous membranes to allow for operation with large capillary pressures and superheated liquid. The operating principles of SHLHPs differ from conventional designs in 1) the un-coupling of the working fluid from its saturation curve to eliminate limitations associated with temperature head and sub-cooling conditions and 2) the possibility of maintaining sub-saturation throughout the device to eliminate film condensation and improve the condenser thermal conductivity. Nanoporous silicon membranes integrated with DRIE channels are fabricated and characterized. The ability of the membrane to hold liquid under tension is tested by equilibrating water-filled device with various relative humidity and observing the cavitation events within individual voids underneath the membrane. Silicon membranes with desired functionality are further incorporated with patterned glass substrates to form prototype MEMS-based SHLHPs.

1. Introduction

Among many different technologies transferring heat, heat pipes (HPs) – closed-circuit, two-phase heat transfer devices in which a working fluid transfers its latent heat as it cycles between an evaporator and a condenser – have offered an attractive alternative in an array of applications¹, such as integrated circuits, systems for the storage of energy, vehicles and aircrafts, environmental controls in buildings, and energy efficiency of industrial processes. Unlike many other cooling technologies where external pumping power is required, HPs are passive systems that the central components required for HPs are wick structures that allow for the existence of a pressure difference, $\Delta P_{\text{cap}}$ [Pa] between the liquid and vapor phases of the working fluid based on capillarity, as described approximately by the Young-Laplace law:

$$\Delta P_{\text{cap}} = 2 \gamma \cos \theta_c / r_p$$

where $\gamma$ [Pa m] is the surface tension of the liquid, $\theta_c$ is the contact angle and $r_p$ [m] is the radius of the pore. This capillary pressure can drive the working fluid around its cycle if a difference of temperature exists between the evaporator and the condenser.

For applications regarding heat transfer over long distance or with liquid motions against gravity/acceleration field, conventional HPs with large pores of the wicks can no longer provide sufficient capillary pressure head to overcome the total pressure losses along the fluid cycle. Loop heat pipes
2. Operating principles and requirements

To gain a qualitative understanding of the differences between the conventional LHP and the SHLHP designs, one can compare their working cycles on P-T diagrams. Fig. 1 illustrates an idealized form of the working cycles of the conventional LHP and the sub-saturated SHLHP, with same pressure

Figure 1. Schematic cross-sectional views and working cycles on P-T diagram for (a) a conventional LHP and (b) a sub-saturated SHLHP. The numbered points in the cycles correspond to state of the working fluid at the labelled points. An acceleration, $g_e$, acts along the pipe axis. Heat enters at the evaporator with a rate $q$. $\Delta P_{\text{cap}}$ represents the capillary pressure required to overcome the total pressure drop in the entire loop. In P-T diagrams, The red lines indicate the working fluid is in vapor phase (Path 2-4); the blue lines (Path 5-1) indicate liquid phase.

(LHPs), Fig. 1a, have been developed to extend the limits of conventional HPs. The main design rules of LHPs are$^2$: 1) minimization of the distance that the liquid must move within the wick by placing a thin membrane in the evaporator, separating macroscopic conduits for the liquid and vapor, and 2) the introduction of a compensation chamber in the cycle to accommodate excess working fluid and allow the system to adapt to changing heat loads and temperatures. LHPs have been shown to provide robust operation and improved performance relative to conventional HPs, including higher heat load capacities and compatibility with a wider variety of architectures for adaptation to specific applications$^3$. However, in conventional LHPs, vapor-liquid coexistence in the compensation chamber forces the thermal cycle to remain close to the saturation and leads to two constraints – the temperature head and the sub-cooling conditions. Additionally, the condensate film in the condenser adds conductive resistance and may cause undesired oscillations in response to heat load steps$^4$.

Motivated by vascular plants - the transpiration process of which is analogous to the operation of LHPs, we proposed, in the paper published previously$^5$, a new design of LHP - Superheated Loop Heat Pipe (SHLHP – Fig. 1b) with introductions of 1) nanoporous membranes allowing larger capillary pressures to be maintained between the liquid and the vapor, 2) removal of the compensation chamber such that the entire liquid path can become superheated and hence decoupled from the saturation curve, and 3) a regulator maintaining sub-saturated state throughout the loop, eliminating liquid from the vapor path.
differences across the vapor path ($\Delta P_{2-3}$) and the liquid path ($\Delta P_{6-7}$) and the same condenser temperature ($T_{\text{sink}}$).

As shown in the working cycle for the conventional case, in the evaporator wick surface (from which evaporation occurs), the liquid (represented as Point 1) is near thermodynamic equilibrium with the vapor in the evaporator (Point 2, slightly sub-saturated due to curved menisci). Path 2-3 represents the adiabatic motion of the vapor in the vapor path. As the vapor enters the condenser, the temperature drops until condensation occurs at a macroscopic, vapor-liquid interface on the saturation curve (Point 4 is the vapor; Point 5, liquid). Path 5-6 represent sub-cooling of the liquid before it leaves the condenser. The latent heat released upon condensation is evacuated to a heat sink. Path 6-7 represents the adiabatic motion of the liquid in the liquid path. As the liquid enters the evaporator (Point 8), it is heated by the conduction through the wick membrane. Due to the presence of vapor in the compensation chamber, Points 8 is brought to saturation. Path 8-1 corresponds to the liquid motion through the wick membrane to the evaporating meniscus; the liquid confined within the wick membrane becomes superheated.

The above working cycle descriptions for the conventional design illustrate three important conditions for the operation of a conventional LHP: 1) the capillary pressure ($\Delta P_{\text{cap}}$) developed by the porous wick must be able to overcome the total pressure drop in the entire loop:

$$\Delta P_{\text{cap, max}} \geq \Delta P_{\text{vap}} + \Delta P_{\text{liq}} + \Delta P_{\text{wick}}$$

(2)

2) The motive temperature head condition - $\Delta P_{2-8}$ is responsible for driving the fluid motion through all components except the wick membrane. Given that these two points are saturated (or nearly so for Point 2), the temperature difference $\Delta T_{2-8}$ is determined by the shape of the saturation curve. This temperature difference must be established across the wick to create the corresponding pressure difference that drives fluid around.

$$\Delta P_{2-8} \approx \left. \frac{dp_s}{dT} \right|_{T_{2-8}} \Delta T_{2-8}$$

(3)

where $dp_s/dT$ is the slope of the saturation curve at the evaporator temperature. 3) The sub-cooling condition - The heat leaked through the evaporator wick membrane into the compensation chamber, $q_{\text{leak}}$ [W], must be balanced by the sensible heat of returning cold working fluid:

$$q_{\text{leak}} = \frac{\Delta T_{2-8}}{R_{\text{wick,e}}} = QC_{p}^{\text{vap}} (\Delta T_{6-7})$$

(4)

This balance implies that the sub-cooling of the liquid before it enters evaporator, $\Delta T_{2-7}$, grows with the increasing of $\Delta T_{2-8}$ and the decreasing of the thermal resistance of the evaporator wick, $R_{\text{wick,e}}$ [K W$^{-1}$]. Combined with Eq. 3, the required sub-cooling is coupled to $\Delta P_{2-8}$ via $\Delta T_{2-8}$.

The working cycle for SHLHP illustrates the effect of eliminating the compensation chamber from the conventional design. Without vapor on the liquid side of the evaporator, Point 8 is no longer constrained to be on the saturation curve. As the demand for pressure drop grows with additional heat load, acceleration or viscous drag, the pressures in the liquid at Points 7, 8, and 1 drop deeper into the superheated region. This use of reduced pressure or even tension (i.e. negative pressure, as presented in Fig. 1b) lessens the demand for elevated pressure in the vapor at Point 2, and the temperature head condition (Eq. 3) of the conventional design does not apply. Further, $\Delta T_{8-7}$ is independent of the pressure load (via Eq. 4 and as the lack of dependence of $\Delta T_{2-8}$ on load). With a condenser membrane and a regulator that pins the fluid activity < 1, the entire working cycle moves below the saturation line. Replacing the saturated condensation process with condensation of sub-saturated vapor directly on the nanoporous membrane could substantially reduce the temperature drop in the condenser.

3. MEMS-based SHLHP design

The global design of a sub-saturated SHLHP is similar to that of reversible loop heat pipes (RLHPs) in which the vapor and liquid paths are separate and there are wick membranes in both the evaporator
Two of the major challenges for the successful realization of SHLHPs are the potential for increased hydraulic resistance relative to conventional wicks and the increased proneness to dry-out due to boiling along the superheated liquid.

A desired membrane should meet the following criteria: 1) uniform, sub-micrometer to nanometer pores, 2) large permeability to avoid introducing excessive liquid tension, and 3) ability to sustain high pressure difference without fracture or collapse. We proposed a layer of nanoporous membrane be supported by microchannels (Fig. 2) that connect to the liquid paths. The nanoporous layer allows for the generation of large $\Delta P_{\text{cap}}$ between the liquid and vapor phases; the supporting microchannel layer presents a high permeability to flow and serves as a structural support of the thin nanoporous layer. Further, to ensure the continuing operation of a SHLHP after dry-out or boiling events, we propose hydraulically independent, redundant liquid paths (Fig. 3) to provide passive robustness to cavitation.

### 3.1. Materials and methods

For the prototype SHLHP, we proposed a MEMS-based, microscale SHLHP design, fabricated completely on silicon and glass wafers (Fig. 3). Nanoporous silicon membranes were prepared by electrochemical anodic etching of highly doped, p-type, <100> oriented silicon wafers with a resistivity of 0.003 $\Omega$-cm. One silicon wafer was mounted between a Teflon cell and an Aluminium bottom electrode with one side of the silicon wafer in contact with the electrolyte, a 50:50 (v/v) solution of 49% hydrofluoric acid and 95% ethanol in the Teflon cell. Electrochemical etching was done under constant current density of 30 mA cm$^{-2}$ for 20 minutes, resulting in a porous silicon layer of approximately 20 $\mu$m in thickness with a pore diameter of 5-10 nm (Fig. 2).

### 4. Results and discussion

#### 4.1. Model analysis for steady-state operation

We presented in our previous paper\textsuperscript{5} a unified model for steady state operation of both the conventional and superheated designs. By numerically solving the coupled heat and momentum balances and local thermodynamic equilibria (equations not shown), we can model the steady-state evaporator temperature at fixed heat flows and $T_{\text{sink}}$. Fig. 4 shows the operating curves for both conventional and SHLHP designs with geometry parameters identical to our MEMS-based device; evaporators are above the condensers (i.e. $g_e = 1g$). The evaporator temperatures of the SHLHP are more than 100ºC lower than those of the conventional case.
4.2. Stability limit of the nanoporous silicon membrane

The insert in Fig. 4 shows the predicted liquid pressure profile in the saturated MEMS SHLHP as a function of heat load; the tension introduced into the SHLHP can be more than -3 MPa. We estimated the ability of the silicon membrane to hold liquid under tension by equilibrating water-filled device with various relative humidity and observing the cavitation events within individual glass cavities underneath the membrane (Fig. 5). Our membrane sustains a cavitation pressure (probability reaches 1/2) of ~-27 MPa at 15ºC; 100% of the voids survived tension up to -15 MPa, much larger compared to the tensions required for the MEMS SHLHP operation.

5. Conclusions

This paper reports the design, fabrication, model analysis and membrane characterization of a MEMS-based SHLHP that exploits nanoporous silicon membranes to allow for operation with large tension in the liquid to provide better performance compared to conventional LHP. This MEMS-based SHLHP allows us to demonstrate for the first time the use of liquid at negative pressure in technologies. After scale-up, SHLHPs could be particularly valuable in applications such as cooling of avionics and energy management in buildings in which heat must be transferred over large distances and against gravity or acceleration.

References

[1] Faghri A 1995 Heat Pipe Science and Technology (New York: Taylor and Francis)
[2] Ku J 1999 Proc. Intl. Conf. on Environmental System (Denver, Colorado: Society of Automotive Engineers, Inc)
[3] Maydanik Y F 2005 Appl. Therm. Eng. 25 635-657
[4] Pouzet E, Joly J L, Platel V, Grandpeix J Y and Butto C 2004 Int. J. Heat Mass Transfer 47(10-11) 2293-2316
[5] Chen I T, Amit P and Abraham D S 2014 AIChE J. 60.2 762-777
[6] Kim B H and Peterson G P 2005 J. Thermophys. Heat Transfer 19(4) 519-526