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RECENT ADVANCES ON ELECTRONICS AND SATELLITE THERMAL CONTROL USING CAPILLARY DRIVEN TWO-PHASE LOOPS

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Abstract. *With increasing processing speed and miniaturization of electronic devices, heat dissipation became a great issue on new designs. When dissipation of high heat fluxes is associated with microgravity environment and limited power, other technologies had to be developed in order to promote the required heat dissipation. For such conditions, capillary driven two-phase loops are used to dissipate heat and keep the electronics temperature within a narrow range. Because the driven force of such devices is the capillary force generated on an evaporation section, such devices do not require moving parts and energy consumption is dramatically minimized. The design of such systems still has issues to be solved, such as tilting limitations for heat pipes, startup procedures for capillary pumped loops and power management of several sources for loop heat pipes. Focusing on the development of such important technology, this paper presents a comprehensive study on capillary driven two-phase loops. Issues related on the design and tests of capillary driven two-phase loops, as well as working fluid selection based on the figure of merit, are presented and discussed. The results of this paper are focused on giving solid parameters for future development of capillary driven two-phase loops for ground and microgravity applications.*

Keywords*: capillary driven forces, heat transfer, thermal control and phase change.*

1. INTRODUCTION

With the development of electronics and telecommunication industries, the necessity of high processing speed and more compact equipments have grown the levels of heat to be dissipated from electronic devices. More compact electronic chips means the increase on the overall heat flux that need to be dissipated, which has become an issue on new designs. Such characteristic is also an issue in applications where the power is limited, thus the use of conventional thermal control devices is practically prohibited. In satellites applications, the thermal control of structures is important in order to allow the correct operation of its electronics. In order to help solving such issues, capillary driven two-phase loops have been used on the thermal control of electronics and structures during the last decade.

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Capillary driven two-phase loops, which comprehend heat pipes (HP), capillary pumped loops (CPL) and loop heat pipes (LHP) have been under development for several years. These devices present similar components: an evaporation section, liquid and vapor transport lines and a condensation section. Each one has its own characteristics regarding design and testing, but the operation principle is the same, which is the driven force generated on the evaporation section, where a porous wick is located. To promote heat transport from a source to a sink, a volatile working fluid is used, which is defined according to certain design aspects related to heat transport capabilities, wick structure and operation temperature.

After years of research and development, capillary driven two-phase loops have known to be reliable heat transport devices, which can be applied on a wide range of operation temperatures and heat fluxes levels. Their use had allowed the miniaturization of electronic chips and increased the processing speed and also the development of new satellite technology. Despite the great development achieved in the last years, several aspects regarding the design and tests still need to be investigated, such as tilting limitations of HP, start up issues of CPL and heat transport from several sources of LHP. To allow the comprehension of the development of such technology, a review on the research performed in the last years is presented. The objectives of this paper are to present the level of development on studies regarding capillary driven two-phase loops technology related to design and tests, as well as to give parameters on this device coupled with electro-hydrodynamically (EHD) technology.

2. HEAT PIPES - HP

Heat pipes were the first passive thermal management devices used in thermal control. It is mostly used on structures, batteries and electronics (avionics) thermal control. A heat pipe is a simple and reliable device, although presents some restrictions regarding the maximum capillary force developed in a porous structure. A heat pipe is composed of three distinct sections: the evaporator, the condenser and the adiabatic section. A representation of a heat pipe is presented by Fig. 1. Heat is absorbed from a heat source by the evaporator, transporting it through the adiabatic section and rejecting at the condensation section. To promote the heat transport, a volatile working fluid is used in the heat pipe. A porous structure, known as wick, is present on the entire heat pipe, which is responsible for generating the capillary forces in the evaporation section.

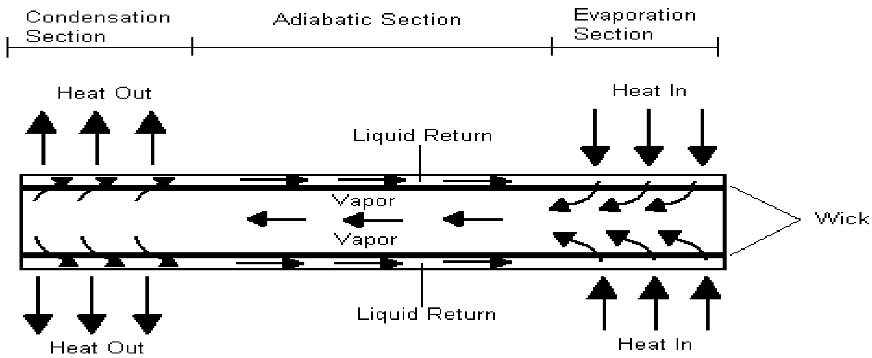


Figure 1 – Schematic of a heat pipe.

The generation of capillary forces in the evaporation section is related to the working fluid and the mean pore radius of the wick structure. According to Fig. 1, heat is transferred from the source to the working fluid through the evaporation section walls by conduction, which generates an evaporating meniscus in the wick structure. A thermodynamic equilibrium is verified on the pores where liquid changes to vapor. By temperature difference and Marangoni effects, known by the relation $d\sigma/dT$ where σ is the fluid's surface tension and T the temperature, vapor flows in the hollow through the adiabatic section toward to the condensation section. In this section, heat is rejected to the sink, condensing the vapor back to liquid. By capillary forces, liquid is pushed back to the evaporation section to complete the cycle.

Several studies have been performed in the past (Peterson, 1994; Chi, 1976) focusing on the determination of the factors that affect the heat pipe operation. It is well known from such investigations that the liquid phase plays an important role on the major pressure resistance in heat pipes, which must me carefully considered during the design. Such resistance was verified by Wang and Vafai (2000) during their experimental investigation, as the wick in the condensation section contributes a significant part of the total resistance. They also found that the operation temperature increases linearly with the applied power, which is also responsible for the maximum temperature difference within the heat pipe. Other considerations must be also performed, related to the capillary, boiling, sonic, viscous and tilting limitations. Such limitations are described in details by Chi (1976) and Peterson (1994). An extensive investigation on heat pipes was performed by Vasiliev (1998), where the state-of-the-art in heat pipe technology was presented and discussed, including design, construction, testing and analysis.

3. CAPILLARY PUMPED LOOPS - CPL

Capillary pumped loops (CPL) are considered efficient two-phase thermal management devices that have been used on thermal control of satellites, solar batteries and electronics. A CPL is able to transport heat from a heat source and dissipate it to a heat sink over long distances continuously, without the presence of moving parts. A CPL differs from a HP on the aspect that the wick structure is presented only at the evaporation section, characterized as the capillary evaporator (or pump). A CPL is able to transport heat over long distances with very little temperature differences and it is known to carry over 8000 W of power applied to the evaporation section. One of the greatest issues of CPL is the startup procedure, as the evaporator has a tendency of depriming due to a lack of liquid returning to its core causing the dry-out and consequently temperature overshoot.

The major components of a CPL are: a capillary evaporator, liquid and vapor lines, condensation and sub-cooling sections and a two-phase reservoir. A representation of a CPL is presented in Fig. 2. A CPL operates under variable conductance as the reservoir, used to control the loop operation temperature, holds both liquid and vapor in a thermodynamic equilibrium.

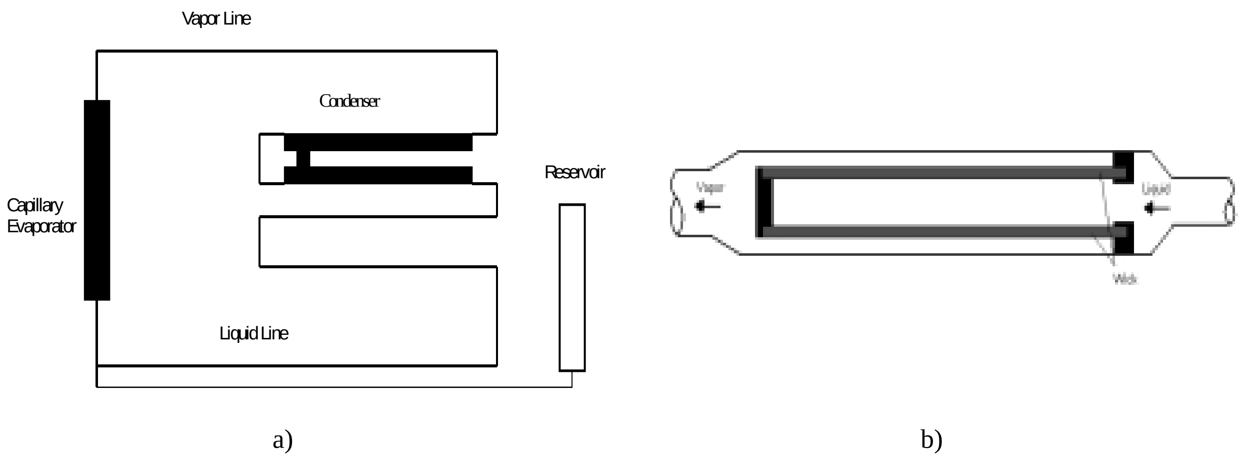


Figure 2 – a) schematic of a Capillary Pumped Loop; b) capillary evaporator assembling.

The CPL was first proposed by Stenger (1966) at the NASA Lewis Research Center. Basically the CPL uses capillary forces to pump a working fluid from a heat source to a heat sink. This system is able to transfer heat efficiently with a small temperature differential and no external power requirements (Faghri, 1995). Heat is acquired in a capillary evaporator and rejected on the condenser. The system is passively pumped by means of surface tension forces developed in a porous structure (called wick) located in the evaporator (Ku, 1993, 1994, 1997). Major advantages of a CPL include no moving parts and high heat transport capabilities. Disadvantages include that a CPL requires sub-cooling and the maximum pumping pressure is limited.

As the most important part of a CPL is the capillary evaporator, efforts have been applied on the correct design of such component, and different configurations have been suggested. Basically there are two different configurations for a capillary evaporator. The most common uses a direct connection from the liquid line at the evaporator's inlet. For this case, a result of an incomplete fluid's condensation or insufficient sub-cooling will result on the vapor transport direct to the evaporator's liquid core. As a consequence, the evaporator will deprime and its operation will be stopped because the liquid will be blocked by the vapor presented in the liquid core. A more efficient configuration, which avoids the accumulation of vapor in the liquid core, is the conception of an evaporator called "starter pump". This configuration is characterized by the presence of a bayonet coming from the reservoir, directly to the evaporator's liquid core. The use of such configuration avoids that the evaporator deprimes and stops its correct operation (Riehl, 2000). Although the capillary evaporator can still present failure, temperature overshoot very seldom takes place as the bayonet is constantly supplying liquid. When such behavior happens, it is normal do verify temperature swings from a lower to a higher level.

A P-T diagram, as shown by Fig. 3, represents the CPL operation. The reservoir temperature of a CPL has to be controlled, in order to set the saturation temperature at which the entire system will operate. The reservoir temperature is raised until the desired saturation temperature, in order to pressurize the entire system and flood the capillary evaporator with liquid prior to starting it.

The CPL starts operating when heat is applied to fluid, through the capillary evaporator. Point (1) on Fig.3 represents the state of the vapor over the evaporating menisci in the wick. The vapor is then displaced in the vapor removal channels of the capillary evaporator with superheating (points 1-2). Then, the vapor flows adiabatically in the vapor lines (points 2-3) towards to the condensation section, being condensed (points 3-4). The condensed vapor flows to the sub-cooler where any remaining bubble is condensed and the liquid is then sub-cooled (point 4-5). The sub-cooled liquid flows adiabatically through the liquid line (point 5-6), returning to the capillary evaporator (point 6-7) to complete the loop, being absorbed by the porous wick again (point 7-8). The reservoir, as a two-phase accumulator, is used to control the pressure in the entire system (Faghri, 1995).

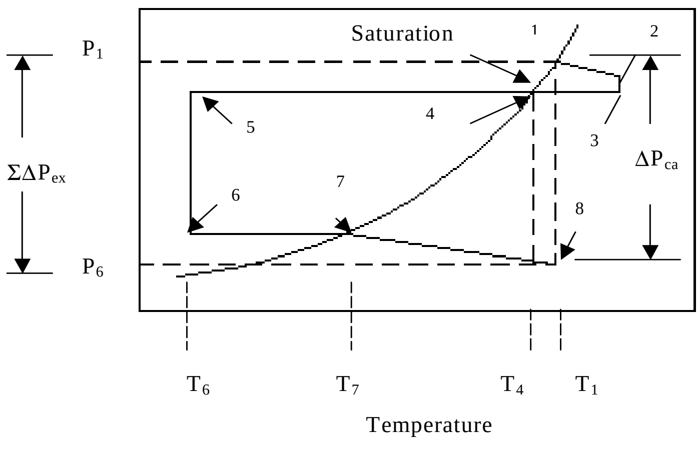


Figure 3 – Thermodynamic behavior of a CPL – P-T diagram (Faghri, 1995).

Typical design procedure for a CPL includes the determination of the maximum capillary pressure and the pressure drop over the loop, which is dependent on the heat load applied to the capillary evaporator. A CPL is limited by the maximum amount of heat that can be transported from one end to another. The equations to calculate the capillary limits presented here were proposed by Chi (1976), Peterson (1994) and Faghri (1995). The capillary limit is the primary limitation for how much heat may be transported by the CPL and it is represented by:

$$\left(\Delta P_{cap}\right)_{\max} \geq \Delta P_v + \Delta P_l + \Delta P_w + \Delta P_g \,, \tag{1}$$

where ΔP_{cap} (Pa) is the capillary pressure drop, ΔP_v (Pa) and ΔP_l (Pa) are the pressure drops in the vapor and liquid lines respectively, and ΔP_w (Pa) is the pressure drop in the porous material. The hydrostatic pressure drop, ΔP_g (Pa), is represented as:

$$\Delta P_g = \rho_l g h \,, \tag{2}$$

where ρ_l is the liquid density (kg/m^3), g is the gravity acceleration (m/s^2) and h is the height difference (m) between the evaporator and the condenser.

The maximum capillary pressure possible is a combination of factors, such as the fluid's surface tension, wick pore radius and the wetting angle between the fluid and the wick material. This angle is also known as the apparent contact angle. When a meniscus is formed at the liquid-vapor interface, the maximum capillary pressure can be calculated by the equation:

$$P_I - P_{II} = \sigma \left(r_1^{-1} + r_2^{-1}\right), \tag{3}$$

where σ is the surface tension (N/m), the subscripts I and II represent the phases and r_1 and r_2 are the radii of curvature. Equation (3) is referred as the Young-Laplace equation and defines the capillary pressure difference occurring across the meniscus separating the liquid and the vapor regions, as a function of the two principal radii of curvature (Carey, 1992).

This equation, in the case of $r_1 = r_2$ (i.e. a cylinder), can be simplified to become:

$$P_v - P_l = P_{cap} = \frac{2\sigma \cos\theta}{r_{pore}} \,. \tag{4}$$

where θ is the contact angle and r_{pore} is the mean pore radius (m). The maximum capillary pressure obtained by Eq. (4) must be greater than the summation of all other pressure drops that occur in a CPL. Such restrictions are pressure drop in the wick structure, pressure drop in the vapor and liquid lines, including the pressure drop across the condenser. The mean pore radius is critical to obtain high capillary pressures.

The wick limitation is related to a minimum pressure drop requirement between the evaporating and inlet surface of the wick. This pressure drop is required for displacing the liquid from the vapor line and filling the liquid line and the compensation cavity. This pressure drop can be achieved with a small radial temperature drop, ΔT_v (°C) in the wick according to the following equation, obtained by the evaluation of Fig. 3 (Faghri, 1995).

$$\left.\frac{dP}{dT}\right|_{\bar{T}} \left(\Delta T_v\right) \approx \Delta P_{cap} - \Delta P_w \approx \Delta P_{1-7} \,. \tag{5}$$

The mean temperature $\bar{T}$ (°C) is taken as the average temperature between points 1 and 7, from Fig. 3. It should be noted that ΔP_{cap} - ΔP_w is the pressure drop in the loop from the evaporating to the inlet surface of the wick. The slope of the saturation pressure-temperature line at a given temperature can be calculated using the Clausius-Clapeyron relation as:

$$\left.\frac{dP}{dT}\right|_{\bar{T}} = \frac{i_{lv}}{\bar{T} v_{lv}} \,. \tag{6}$$

where i_{lv} is the latent heat of vaporization (J/kg) and v_{lv} is the liquid/vapor specific volume (m^3/kg). The pressure drop represented by $\Delta P_{1\text{-}7}$ is a function of dP/dT for a particular working fluid at a given mean temperature $\bar{T}$, and also depends on the geometric and thermal characteristics of the wick. The wick structure present in the evaporator and, for some applications, present also in the condenser, causes flow restriction that affects the CPL performance, which is dependent on the wick permeability. The wick structure permeability can be determined by expressions obtained in the past. The wick permeability (K, m^2) is a property of a porous material that describes the ability of the material to transport the liquid under an applied pressure gradient (Peterson, 1995). The wick thickness pressure drop has also to be calculated and the following expression can be used:

$$\Delta P_w = \frac{\mu \dot{Q}}{i_{lv} K \rho A_T} W_t \, , \tag{7}$$

where μ is the viscosity (Pa.s), $\dot{Q}$ is the applied heat load (W), A_T is the transversal wick area (m^2) and W_t is the wick thickness (m).Attention must be paid for an important factor of a CPL, called liquid line boiling limit, which is a result of the liquid sub-cooling after condensation to avoid boiling in the liquid line. The amount of required sub-cooling is related to the liquid pressure drop and should satisfy the following inequality:

$$\Delta T_{sc} \geq \left. \frac{dP}{dT} \right|_{\bar{T}} \left(\Delta P_l + \Delta P_g \right). \tag{8}$$

The liquid and vapor pressure drops are responsible for the highest capillary limitations in a CPL. The vapor pressure drop has a higher magnitude than the liquid pressure drop because the vapor velocity is much higher than the liquid velocity, causing greater pressure drop due friction with the channel walls. Liquid and vapor pressure drops can be derived from the Navier-Stokes equation, given the appropriate boundary conditions. Perhaps the most accepted one-dimensional pressure drop analysis is presented by Chi (1976). The derivations of such equations are presented by Chi (1976), Peterson (1994) and Faghri (1995). The final form of the pressure drop equations for both phases can be calculated by the equation:

$$\Delta P = f\left(\mathrm{Re}\right) \frac{\rho \, V^2 L}{2 D_h} , \tag{9}$$

where L is the length (m) of either liquid or vapor line, depending on the calculation procedure, V is the velocity (m/s), D_h is the hydraulic diameter (m) and f(Re) is the friction factor, dependent on the Reynolds number (Re) of each phase. The friction factor can be determined as follows (Chi, 1976):

$$f\left(\mathrm{Re}\right) = 64\,\mathrm{Re}^{-1} \qquad \text{if Re} < 2300, \tag{10}$$

$$f\left(\mathrm{Re}\right) = 0.316\,\mathrm{Re}^{-0.25} \qquad \text{if } 2300 \leq \mathrm{Re} \leq 20000 , \tag{11}$$

$$f\left(\mathrm{Re}\right) = 0.184\,\mathrm{Re}^{-0.2} \qquad \text{if Re}>20000. \tag{12}$$

Figure 4 shows the capillary limitations for a CPL in regard to its heat transport capability as a function of the saturation temperature. Figure 5 shows the operation behavior of a CPL, since its startup until the achievement of the steady state condition. One important point to be discussed is that, due to the intermittent vapor flow from the capillary evaporator, differential pressure readings over the evaporation section shows wide fluctuations. This is due to the acceleration of the vapor inside the vapor grooves and line, and deceleration at the condensation section. Figure 6 presents an example of differential pressure fluctuations oven the capillary evaporator of a CPL (Riehl, 2000).

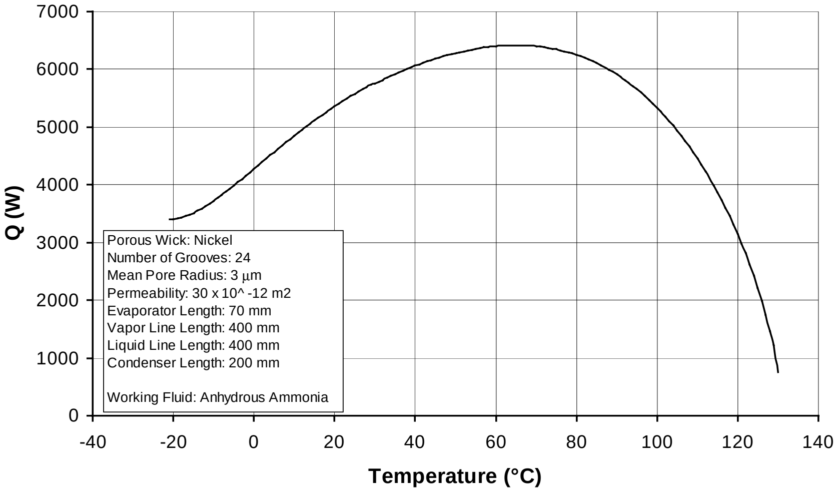


Figure 4 – Capillary limitation for a CPL

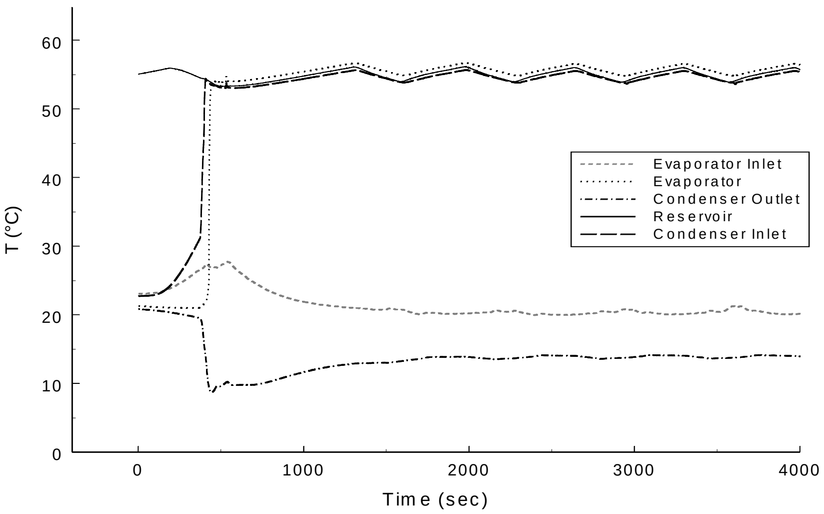


Figure 5 – CPL operation under constant temperature of the reservoir (Riehl, 2000).

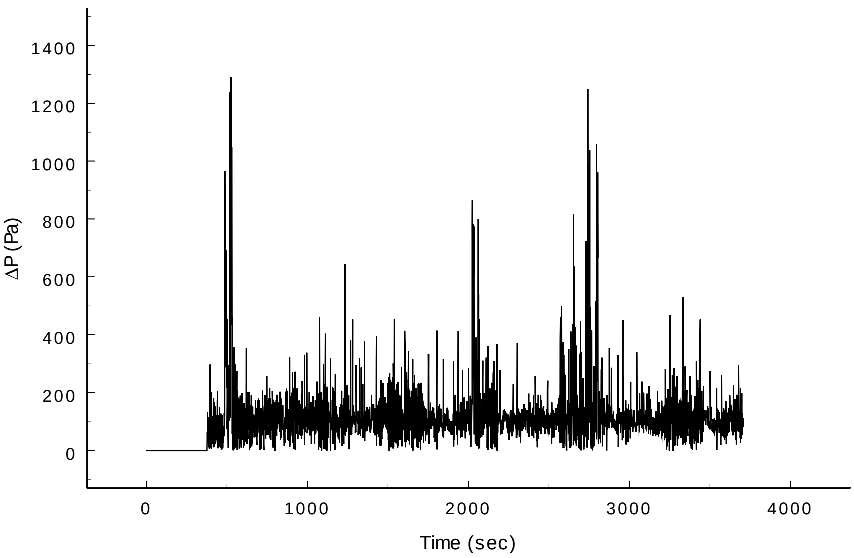


Figure 6 – Differential pressure readings over the evaporation section of a CPL (Riehl, 2000).

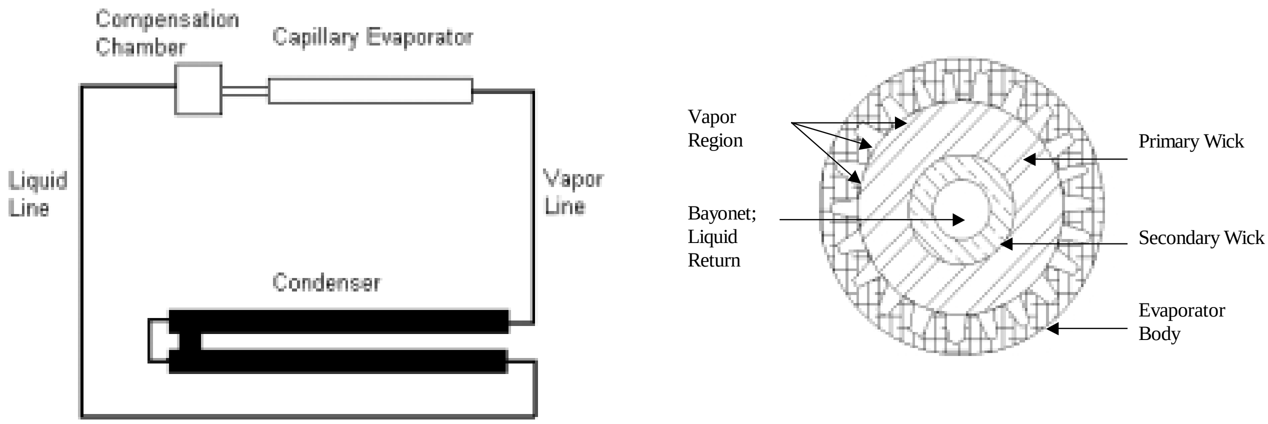
Extensive analytical and experimental investigations have been performed, focusing on the identification of the factors that affect the CPL performance. The main efforts are toward the capillary evaporator operation, which is responsible for generating capillary forces that drive the working fluid. Since the startup is an issue in capillary evaporators, LaClair and Mudawar (2000) investigated the transients prior to the initiation of boiling. They could conclude that the transient is linearly dependent upon the applied heat flux in the evaporator and the presence of a large thermal mass reduces the temperature differences during the preheat stage of a startup. Special attention must be paid in regard to the wick structure, since wicks with high thermal conductivity reduces the level of sub-cooling in the evaporator liquid core necessary to have a reliable startup. Such behavior was also investigated Yan (2000).

Thermal management systems using CPL technology have proved to handle more than one heat source, as investigated by Ku et al. (1998). On their experimental investigation, a CPL using anhydrous ammonia as working fluid with five parallel evaporators was tested, which was able to transport up to 3000 W or power. Tests showed that the CPL presented reliable operation when multiple evaporators and condensers were used, also presenting reliable startups. Ku and Hoang (1997) experimentally investigated a CPL using anhydrous ammonia as working fluid with four parallel starter pumps evaporators, showing that is able to transport up to 1600 W with little temperature difference over the loop. Experimental and analytical investigation of a small scale CPL has also been performed, focusing on its integration on a micro satellite platform to be tested under microgravity environment. Ground tests, using both acetone and anhydrous ammonia as working fluids, have shown that the proposed CPL configuration, using polyethylene wick, presents reliable startups and steady state behavior for heat transport up to 60 W (Reimbrecht et al., 2002; Riehl and Bazzo, 2002).

Although CPL has proven to be reliable on the heat transport and transfer and considerable investigations have been performed over that last ten years, the CPL has fallen out of use due to the appearing of a more reliable technology developed by the Russian Space Program, as it will be discussed.

4. LOOP HEAT PIPES - LHP

As a response for the development of CPL technology, the Russian Space Program developed the so-called loop heat pipe (LHP) in the early 80’s. Such technology showed to be more reliable during the startup as the loop saturation temperature control is not required. The major difference between a CPL and a LHP is the configuration of the capillary evaporator, which presents an in-line two-phase reservoir and the presence of a secondary wick structure. Figure 7 presents a schematic of a LHP setup and the internal configuration of the capillary evaporator with two porous wicks.



a) b)

Figure 7 – a) Schematic of a LHP setup; b) cross section of the capillary evaporator.

Basically, the LHP has the same components of a CPL and its design can be done following the same procedure, only differing on the capillary evaporator configuration. The equations used to calculate the maximum capillary pressure developed in the evaporator and the pressure drop verified in each component can also be used on LHP.

The thermodynamic behavior of a LHP is presented on Fig. 8. The vapor generated at the porous wick external diameter (point 1) present in the evaporator is at the saturation condition. As it flows throughout the evaporator channels, it shows a slight superheating at the evaporator outlet (point 2) due to the heating process and the fluid absolute pressure decrease.

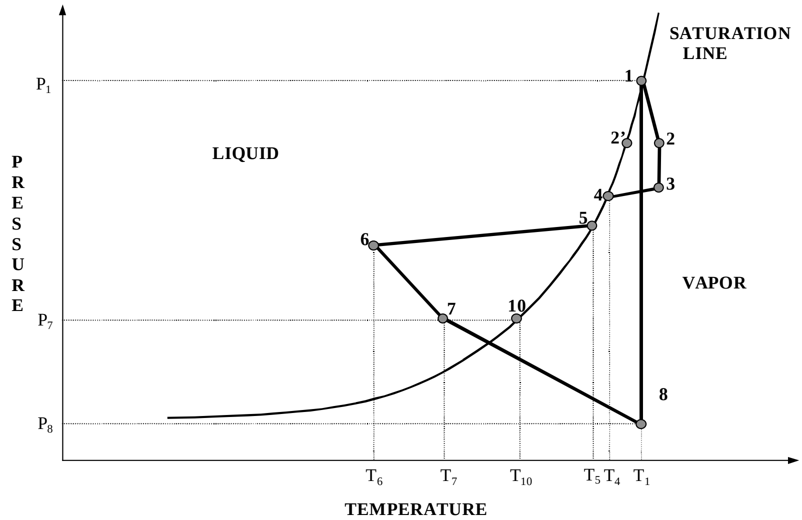


Figure 8 – P-T diagram for a LHP operation.

Considering that the vapor line is adiabatic, the vapor temperature remains constant. As the pressure loss continues throughout the line, the vapor becomes even more superheated in regard to the local saturation temperature, until it reaches the condenser inlet (point 3). Then, vapor begins to condensate inside the condenser (point 4), as the process takes place at the saturation line and both temperature and pressure decrease. When the vapor condensation is completed (point 5), the liquid is then sub-cooled until the condenser outlet (point 6). The sub-cooled liquid then flows throughout the liquid line, keeping its temperature due to the adiabatic flow. When liquid reaches the evaporator core (point 7), its pressure is P_7 and its temperature is T_7. As there is no flow between the compensation chamber and the evaporator during the steady state flow, the pressure P_{10} inside the compensation chamber must be equal to P_7 (Ku, 1999).

As shown by Fig. 7, the capillary evaporator presents two wick structures and the in-line compensation chamber. The primary wick structure with smaller pore radius is responsible for generating the capillary forces that will drive the working fluid from the heat source to the heat sink. The secondary wick structure presents a greater pore radius, which is responsible for draining excess liquid from the evaporator back to the compensation chamber or supplying more liquid when necessary. The correct sizing of the compensation chamber has been a challenge for designers, once there is not a maximum but a minimum volume. The compensation chamber must be able to handle the swings of the condensing flow and must be big enough to accommodate a liquid volume equal to the condenser volume (Ku, 1999). A LHP can operate at either variable of constant conductance depending only on the amount of liquid inside the compensation chamber. One of the major issues on the operation of LHP is the heat leak from the capillary evaporator to the compensation chamber, which may cause vaporization inside the compensation chamber. Experimental tests using different configurations of the compensation chamber were performed by North et al. (1997) and Gerhart and Gluck (1999), when high heat fluxes are applied to the evaporator in systems operating with two compensation chambers. It was observed that LHP could transport up to 0.7 kW/m^2 using ammonia as working fluid.

Several studies have been performed by Ku (1993, 1994, 1997, 1999) and Hoang and Ku (1995) toward the investigation of the operation characteristics of LHP. As in CPL, LHP presents great oscillations on the differential pressure during operation, which is caused by the liquid and vapor lines diameter, as well as the evaporation and condensation process since the vapor generated in the capillary evaporator accelerates while the condensing liquid decelerates causing pressure spikes due to the liquid incompressibility. O'Connel and Ku (1996) observed that upon decreasing the lines diameter, an amplification of the pressure oscillation takes place.

The LHP operation does not present one of the greatest problem of CPL, which is the startup procedure as liquid is always present in the evaporator liquid core. Cheung et al. (1998) experimentally investigated the startup of a LHP. As the LHP presents a direct liquid return to the capillary evaporator, dryout issues are relatively solved. Up to 50 levels of startup were tested and it was verified that the dryout phenomena simply did not occur. It was observed that the capillary evaporator could operate with a heat transfer rate of as little as 2 W. A phenomenon that must be carefully considered is the hysteresis on LHP configurations, since the operation temperature could not be achieved for certain power levels (Kaya and Ku, 1999). Microgravity tests of LHP have been performed (Ottenstein and Niemberg, 1998; Baker et al., 1998; Kurwitz and Best, 1997; Kozmine et al., 1996), showing that such device presents high efficiency on the heat transport and transfer. Loop heat pipes showed, in microgravity, that the hydrostatic effect of the liquid column does not exist, which allows that the meniscus interface in the capillary evaporator presents a negligible curvature, which contributes for an instantaneous startup.

Watson et al. (2000) investigated the steady state operation of a LHP, built to transport up to 800 W using ammonia as working fluid, using sintered nickel as wick structure. It was verified that this LHP presented good operationability at different sink temperatures, showing its reliability on the heat transport process. Ku and Birur (2001a, 2001b, 2001c), Ku et al. (2001) and Bienert and Nikitkin (2001) present an extensive experimental investigation of LHP with several heat sources. It was verified that a LHP with several heat sources are feasible, but the issue lies on the correct sizing of the compensation chamber. On multiple evaporators and compensation chambers LHP, only one compensation chamber will present both vapor and liquid phase while the others will be flooded with liquid only. The compensation chamber that presents vapor and liquid will control the LHP operation temperature. The LHP presented reliable startup procedure when only one or both evaporators were in use, which shows that a LHP can be used to transport heat from multiple heat sources. Even with extensive experimental and reported analytical studies on LHP, several issues still need to be investigated in order to improve the knowledge of such thermal management system.

The choice of a working fluid is important to have a better or worse operationability of a CPL or LHP. The working fluid selection is based on the fluid's Figure of Merit. For the case of HP, the Figure of Merit is calculated in regard to the liquid phase, which causes the major pressure loss. As for CPL and LHP the major concern is in regard to the vapor flow throughout the vapor line, as the liquid pressure drop (ΔP_{liq}) is negligible when compared to the vapor pressure drop (ΔP_{vapor}), the Figure of Merit for CPL and LHP systems is based only on the transport properties in regard to the vapor phase. Focusing on such aspect, a Figure of Merit (N) calculation procedure for CPL and LHP was presented by Dunbar and Cadell (1998) as

$$N = \frac{\sigma}{L^{-1.75}\mu_v^{0.25}\rho_v^{-1}}, \tag{15}$$

where L is the vapor line length (m). Figure 9 presents the Figure of Merit of various working fluids, using Eq. (15). From Fig. 9, it can be noticed that for a wide range of operation temperature, ammonia is still the best working fluid for CPL and LHP. When a CPL or LHP must operate at low temperatures (below -70 °C), especially when freezing conditions for ammonia are a concern, propylene has been used. The choice for the proper working fluid is still related to the operating temperature range, chemical interaction with the system parts, maximum heat load, working fluid wetability at the porous wick, which are all design parameters for CPL and LHP.

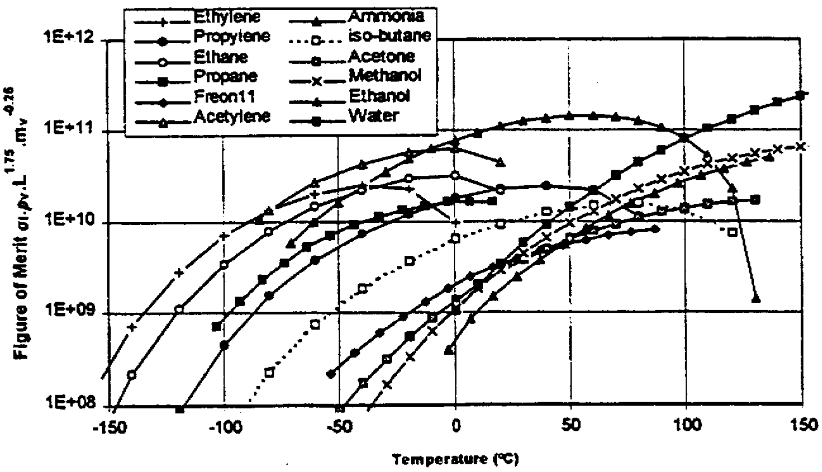


Figure 9 – Figure of Merit for working fluid selection (Dunbar and Cadell, 1998).

5. ELECTROHYDRODYNAMIC TECHNOLOGY - EHD

The electrohydronamically technology (EHD) has recently been used for thermal control and heat transport. An application of an electric field of up to 50 kV and very little current (in the range of microampere) has been used. Such power supplies are found in household appliances, such as color TVs. The use of this technology is important and has shown to improve the performance of capillary evaporators in CPL and HP (Bryan and Seyed-Yagoobi, 1997). The EHD phenomenon is an active enhancement method, used to improve the convective boiling heat transfer in many applications, but it can be offset by potentially high-pressure drops. Ground applications of EHD technology are extensive and include commercial heat exchangers equipment for refrigeration and air conditioning, electronic cooling, cryogenic and process industry and laser medical/industrial cooling applications. A breakthrough of a recent application of EHD technology, assembled in a capillary evaporator, is its use to promote the thermal control of plasma engines of spacecrafts and satellites.

The EHD technology involves the interaction of electric and flow fields in a dielectric fluid medium. Such interaction can promote an electrically induced fluid motion and interfacial instabilities that are caused by an electric body force. Bryan and Seyed-Yagoobi (1997, 2000) present a fundamental investigation on EHD phenomena, where they could conclude that EHD forces can generate significant enhancements in the convective boiling heat transfer coefficient, but it can create high pressure drops.

Upon applying the EHD technology on capillary evaporators, Mo et al. (1999) observed that a polarized EHD force could also assist or substitute capillary forces to collect, guide and pump liquid condensate in regions of high electric field intensity. Basically, a spring type electrode was inserted in the liquid core of a capillary evaporator, which was responsible for promoting the EHD forces, as presented in Fig. 10.

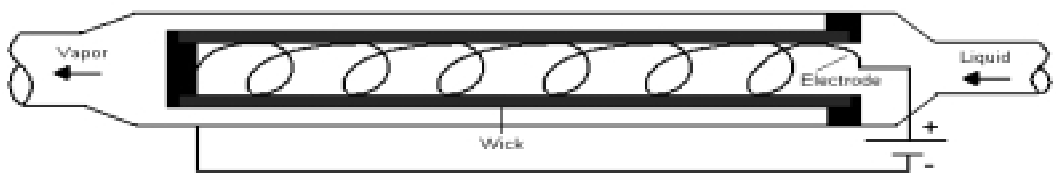
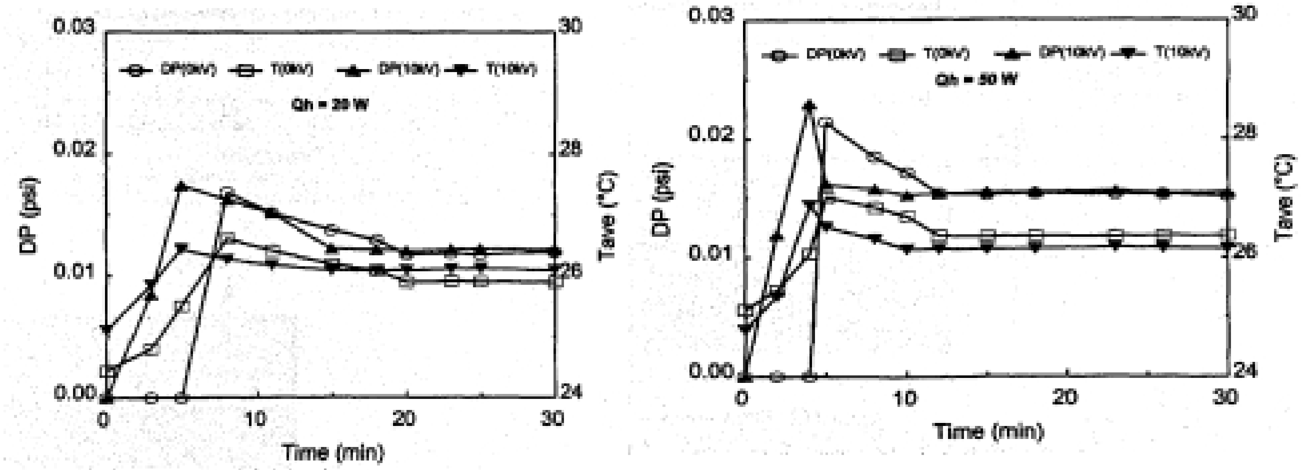


Figure 10 – Schematic of an EHD-assisted capillary evaporator.

The application of such EHD coupling would promote the flow of liquid to a more intense electrical field and vapor to the less intense electrical field. Thus, vapor would migrate from the liquid core to the outer wick diameter of the capillary evaporator, which would lead to a faster and more reliable startup.

During experimental tests, Mo et al. (1999) observed that when using of EHD-assisted capillary evaporators, the electric field was responsible for a startup time considerably reduced by as much as 50%, as presented on Fig. 11. Although the use of EHD-assisted capillary evaporators can decrease the startup times, it was observed that this time reduction is less significant at higher power levels. This is because the menisci in the capillary evaporator are formed more easily at higher heat load levels and produce higher capillary forces.



a) b)

Figure 11 – Startup test of an EHD-assisted capillary evaporator: a) time reduction on a 20 W EHD-assisted CPL; b) time reduction on a 50 W EHD-assisted CPL (Mo et al., 1999).

Experimental results on EHD-assisted capillary evaporators have presented a solution for startup issues of CPL. Further investigations on EHD-assisted capillary evaporators used in LHP still need to be performed, since this is a growing and more reliable technology when compared to CPL. Certain aspects of using EHD technology on LHP capillary evaporators have to be taken in account, since its configuration is slightly different than those used in CPL, although its investigation must be performed.

6. CONCLUSION

A survey on capillary driven two-phase loops, used for thermal management of electronics and satellites has been presented. From what could be verified, heat pipes still will be used when very limited areas need to have a precise thermal control along low heat dissipation requirements. Heat pipes can also be used in local thermal management of structures of aircrafts and spacecrafts, where localized heat dissipation is a concern. Capillary pumped loops, although known as reliable two-phase thermal management systems, have fallen out of use due to the appearing of the loop heat pipe, which has shown to be a more reliable and effective on the thermal control. Capillary pumped loops still have applications, especially when a fine temperature control is required, which is the case of electronic chips and other components. Loop heat pipes are currently in use for thermal control of electronics, avionics, and space engines, aircrafts and spacecrafts structures and have shown to be very reliable on the thermal management. Capillary driven two-phase loops with EHD-assisted technology has been presented as a solution for startup issues at low power on CPL and even LHP.

Although extensive investigations have been performed in the past, further investigations are still necessary in order to better understand and control the issues on thermal management using capillary driven two-phase loops. Certain aspects related to heat pipes, such as tilting and gas loaded heat pipes, still need to be investigated. In the case of CPL and LHP, the correct design can lead to a better thermal management related to startup issues of the capillary evaporator, as well as evaporator depriming and re-priming, which could lead such equipments to several applications.

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