Performance of heat pipes as capillary pumps: modelling and comparison with experimental results

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Abstract The objective of this paper is to present a theoretical investigation of the operational characteristics on a small-scale Capillary Pump Loop (CPL), focused on the CPL capacity to create flow in addition to its heat transfer capacity. A typical design of a CPL is composed of a capillary evaporator, a condenser, a two-phase reservoir, liquid and vapour lines. The capillary evaporator generates the required pressure pumping for moving the working fluid from the condenser to the evaporator section. The fundamental principles of the proposed modelling are: The overall pressure drop in the loop must be less than the maximum capillary pressure in order to ensure that the system will operate continuously. The major components of the CPL pressure drop are related to the flow in the wick structure, condenser, vapour and liquid lines. The wick structure present in the evaporator causes a flow restriction that affects the CPL performance, which is dependent on the wick permeability, a property of the porous material that describes its ability to transport the liquid under an applied pressure gradient. An experimental lab-scale installation is used for the validation of the theoretical analysis. The results showed that the proposed CPL modelling is able to describe the CPL performance very well.

Keywords Capillary pump; CPL; heat pipes; pressure head rise

Nomenclature

$A_w$ porous media surface [m²]
$C$ constant [-]
$d$ pipe diameter [m]
$f$ friction factor [-]
$g$ gravitational acceleration [m²/s]
$g_c$ gravitational conversion constant [-]
$G$ mass velocity [kg/s m²]
$h$ relative height [m]
$h_{e,v}$ enthalpy [kJ/kg]
$K_w$ porous media permeability [m²]
$L$ pipe length [m]
$m$ mass flow rate [kg/s]
$P_e$ applied heat load of evaporator [W]
$Q$ flow rate [m³/s]
**Reynolds number** [-]

**r_c** porous media radius [m]

**T** temperature [°C]

**u** velocity [m/s]

**μ** viscosity [kg/ms]

**ρ** density [kg/m³]

**σ_s** surface tension [N/m]

**Δp** pressure drop [Pa]

### Subscripts

- **b** buoyancy head
- **c** capillary force, condenser
- **e** evaporator
- **eff** effective
- **g** gravity head
- **h** hydraulic
- **L** liquid phase
- **max** maximum
- **o** reference point
- **v** vapour phase
- **w** wick

### Introduction

Capillary pumping two-phase loops have been continuously investigated for electronic cooling systems, satellite thermal control and other space applications. Most tests were performed in capillary evaporators using plastic or metallic porous wick as the capillary structure and anhydrous ammonia as the working fluid.

A Capillary Pump Loop (CPL) is designed for operation as a two-phase heat transfer device in passive mode, without the need of any mechanical pump for driving the working fluid from a low temperature sink to a high temperature source. Although earth-based applications of the CPL have been proposed, it is especially well suited for thermal management in spacecraft, where gravity and hence its potentially deleterious effects on the CPL’s operation are absent. The CPL uses capillary action for fluid transport and contains no moving parts. Multiple evaporators and condensers can be added at different locations in a CPL, allowing the use of a single loop to reject heat from multiple sources to multiple sinks, possibly at different temperatures. In contrast to the heat pipe, wicks are absent in most of the transport section of the CPL. Instead, liquid and vapour flow through smooth walled tubing, thereby reducing the frictional pressure losses and increasing the maximum potential fluid flow and heat transfer rates.

A typical design of a CPL (Figure 1) is composed of a capillary evaporator (responsible for generating the capillary forces that drive the working fluid), a condenser, a two-phase reservoir (to control the loop saturation pressure), liquid and vapour lines. The capillary forces are generated by the capillary evaporator, which
acquires heat and transfers it to the working fluid. Formation of vapour is responsible for the displacement of the liquid in the lines towards the condenser during the start up. The two-phase reservoir is used to set the operation temperature at which the entire loop will operate. The CPL works without moving parts and with very little power consumption. The fluid must be used in its pure state without contaminants, which enables its use in microgravity. It passively promotes the thermal control, allowing fine control on the operating temperature of components.

CPL development began in the 1960s but received special attention in the late 1970s. At this time, CPL began to be intensively investigated, proving to be operationally reliable for thermal control and able to transport heat over long distances with minor temperature difference. Excellent reviews of the development history and theory of operation are available in the literature [1, 2]. In recent years, numerous CPLs have been fabricated and ground-tested, several have been tested in flight experiments and designs have been selected for use on a few spacecraft missions [3]. Nevertheless, issues relating to the CPL’s reliability and robustness have limited its acceptance and implementation, and have been the primary focus in recent years [4–7].

One problem which has plagued CPLs in both ground and flight tests is the difficulty in starting the evaporators. Typically, after a CPL successfully starts and enters into a steady operation mode it performs quite well and the operation is fairly predictable. However, before steady state operation can occur, a series of transient events, collectively referred to as the start-up, must take place where the thermophysical state of fluid through much of the loop changes quite dramatically. During the start-up, the locations occupied by liquid and vapour transiently shift to those corresponding to steady state operation. This repositioning of liquid and vapour occurs through the combined processes of vaporization, condensation, and multiphase fluid flow through various portions of the loop.

Figure 1. The schematic diagram of the capillary pumped loop.
In theory, vapour should form only in the vapour grooves of an evaporator. However, if sufficient superheat develops in the grooves before nucleation occurs, the wick and possibly the liquid core can also heat up to a temperature that is greater than the saturation temperature, and nucleation could potentially occur there as well. Additionally, vapour may be forced through the wick and injected into the liquid core during the initial pressure spike or pressure surge of the start-up. If the temperature of the liquid core is at a temperature near or at saturation, then vapour may remain there without condensing and in some cases may actually grow. Vapour presence in the liquid core has been found to fully or partially block liquid flow to the wick, which may lead eventually to deprime – the loss of capillary pumping action – of an evaporator [8, 9]. Also, vapour presence in the evaporator core has been to influence pressure oscillations in the CPL [10].

The two-phase reservoir has to be heated, prior to the start-up of a CPL, so that the operation temperature can be set. The entire loop will then operate at this temperature with slight variations owing to some superheat or subcooling. Upon setting an operating temperature in the reservoir, the internal pressure will raise which will fill the entire loop with liquid, causing the so-called pressure priming. When the loop is filled with liquid, the CPL is ready to start operating. Then, heat is applied to the capillary evaporator and as its temperature rises, only sensible heat is transferred to the working fluid. When the evaporator temperature reaches the same temperature as the reservoir, latent heat is transferred to the working fluid starting the evaporation process. A meniscus is formed at the liquid-vapour interface, which is responsible for developing the capillary pressure that will drive the working fluid. Vapour is displaced from the evaporator, which causes the displacement of the liquid in the channels allowing the vapour to reach the condenser. In the condenser, heat is removed and liquid will also present some subcooling. At the start-up, the excess of liquid present in the channels is displaced by the vapour back to the reservoir, which equalizes the right amount of working fluid for a given heat load. With such particularities, the CPL has been used to transfer heat over long distances with small pressure drops over the entire loop allowing its use in large systems. CPLs have been tested over different configurations, and it is known to transport up to 5000 W when capillary evaporators are used [11]. Without moving parts and because a CPL acts as a thermal diode, the working fluid cannot flow from the condenser to the evaporator by the vapour line.

Related, mainly to the porous structures (called wick) present in the capillary evaporator, great developments were achieved on CPLs. Different materials have been used as porous wick such as sintered nickel, stainless steel, titanium and ultra-high molecular weight polyethylene [12–14]. As the generation of capillary forces is dependent on the working fluid surface tension and wick pore size, CPLs have been investigated using methanol, acetone and anhydrous ammonia as working fluids. Several investigations have been performed towards the achievement of sintered nickel components with fine pore sizes [15, 16].

The working fluid must operate without impurities as an important condition to avoid the presence of non-condensable gases (NCG) in the loop. The presence of NCG in a CPL can cause a general failure of the capillary evaporator, but in general,
it is less likely to occur than in heat pipes. The presence of NCG can be minimized using compatible materials with the selected working fluid. Ensuring that a good vacuum condition is verified in the loop as well as using fluid with minimum contaminants also minimizes the presence of NCGs.

Nowadays, there are efforts to extend the application of capillary pump loops to commercial and industrial systems. The use of such devices offers many advantages regarding the flexibility in operation and application, as they are very efficient in transporting heat, even under a small temperature difference. Very few investigations have been conducted towards the use of small size capillary evaporators in order to manufacture a more compact CPL. Investigators [17] have reported that small evaporators have a tendency of depriming more easily, probably due to an insufficient subcooled liquid supply to the porous wick. More information regarding this matter is important especially when more compact CPLs need to be developed for specific applications.

The objective of this paper is to present a theoretical investigation of the operational characteristics on a small-scale CPL, focused on the CPL capacity to create flow in addition to its heat transfer capacity. An experimental lab-scale installation, described analytically in [18] and the modified configuration in [19, 20], is used for the validation of the theoretical analysis. The results showed that the proposed CPL modelling is able to describe the CPL performance very well.

**Modelling of a CPL**

The CPL design is related to the maximum capillary pressure that can be developed by the wick structure and working fluid. The overall pressure drop in the loop must be less than the maximum capillary pressure in order to ensure that the system will operate continuously. Thus the driving pressure difference $\Delta p_{\text{eff}}$ is given by:

$$\Delta p_{\text{eff}} = \Delta p_{c,\text{max}} + \Delta p_b - \Delta p_{\text{total}}$$

(1)

where $\Delta p_{c,\text{max}}$ is the pressure drop due to maximum capillary force, $\Delta p_b$ is the pressure drop due to buoyancy force effect, and $\Delta p_{\text{total}}$ is the total pressure drop in the overall loop.

The pressure drop due to maximum capillary force is given by:

$$\Delta p_{c,\text{max}} = \frac{2\sigma_L}{r_c}$$

(2)

with $\sigma_L$ the surface tension and $r_c$ the radius of the porous media. The pressure drop due to buoyancy force effect is given by:

$$\Delta p_b = (\rho_l - \rho_v)gh$$

(3)

with $\rho$ the density of liquid and vapour respectively and $h$ the relative height.

The major components of the CPL pressure drop are related to the flow in the evaporator (wick structure), condenser, vapour and the liquid lines. Thus the total pressure drop in the overall loop is given by:
The pressure drop in the evaporator due to liquid flowing through the porous media (based on the Darcy’s law) may be written as:

$$\Delta p_w = \frac{\mu m L_w}{\rho_w K_w A_w}$$  \hspace{1cm} (5)

where $L_w$ the length, $A_w$ the surface, and $K_w$ the permeability of the porous media (wick) and $m$ the mass flow rate.

The pressure drop in the condenser is related to the mass velocity $G$ of the liquid, the velocity of the liquid $u_l$ and the velocity of the vapour $u_v$ through the equation:

$$\Delta p_c = \frac{G}{g_c} (u_v - u_l)$$  \hspace{1cm} (6)

where $g_c$ is the gravitational conversion constant.

The mass velocity $G$ and the liquid velocity $u_l$ are given by the equations:

$$G = \frac{4 \rho_l Q_l}{\pi d_{l,h}^2} \quad \text{and} \quad u_l = \frac{4 Q_l}{\pi d_{l,h}^2}$$ \hspace{1cm} (7)

and the flow rate of the liquid $Q_l$ and the vapour $Q_v$ are related with the equation:

$$Q_v = \frac{\rho_v}{\rho_l} Q_l$$ \hspace{1cm} (8)

where $\rho_l$ is the density of the liquid and $\rho_v$ is the density of the vapour.

Substituting equations (7) and (8) into equation (6) it is obtained that:

$$\Delta p_c = \frac{16}{g_c \pi \rho_l \rho_v \rho_v \pi^2 d_{l,h}^4} \left( \frac{\rho_l}{\rho_v} - 1 \right) \rho_v \rho_l^2$$ \hspace{1cm} (9)

The pressure drop due to liquid flowing through the liquid head line is given by:

$$\Delta p_l = f \frac{L_l}{d_{l,h}} \frac{\rho_l u_l^2}{2}$$ \hspace{1cm} (10)

where $L_l$ the length and $d_{l,h}$ the hydraulic diameter of the liquid head line. The pressure drop due to vapour flowing through the vapour head line is given by:

$$\Delta p_v = f \frac{L_v}{d_{v,h}} \frac{\rho_v u_v^2}{2}$$ \hspace{1cm} (11)

where $L_v$ the length and $d_{v,h}$ the hydraulic diameter of the vapour head line. And finally the pressure drop due to gravitational head effect is given simply by the equation:

$$\Delta p_g = \rho g h$$ \hspace{1cm} (12)
In the above equations \( f \) is the friction factor dependent on the Reynolds number of each phase, \( f(Re) \). The friction factor can be determined as follows:

\[
f(Re) = \begin{cases} 
\frac{64}{Re} & \text{for } Re < 2300 \\
\frac{0.316}{Re^{0.25}} & \text{for } 2300 \leq Re \leq 20000 \\
\frac{0.184}{Re^{0.2}} & \text{for } Re > 20000
\end{cases}
\] 

(13a) (13b) (13c)

Usually the flow is in the laminar region, \( Re < 2300 \), and by substitution of equation (13a) into equations (10) and (11) we get:

\[
\Delta p_f = \frac{128 \mu_L}{\pi d_{f,h}^4} Q_f \quad \text{and} \quad \Delta p_v = \frac{128 \mu_v L_v \rho_v}{\pi d_{v,h}^4} Q_v
\]

(14)

By combining equations (5), (9), (12) and (14) it is obtained:

\[
\Delta p_{\text{total}} = \frac{\mu_L}{\rho K_w A_w} \frac{m}{g_c \pi^2 d_{c,h}^4} \left( \frac{\rho_v}{\rho_f} - 1 \right) \rho_f Q_f^2 + \frac{128 \mu_L L_f}{\pi d_{f,h}^4} Q_f + \frac{128 \mu_v L_v \rho_v}{\pi d_{v,h}^4} Q_v + \rho_v g h
\]

(15)

Equation (15) gives the total pressure drop \( \Delta p_{\text{total}} \) in the loop from parameters which are either directly measurable (pipe length and diameter, flow rate) or material properties which can be found in the bibliography (viscosity of the fluids).

**Coupling fluid dynamics with heat transfer**

By means of the relation between fluid velocity, mass flow rate, applied heat load of the evaporator, \( P_e \), and enthalpy, \( h_{f,v} \), it is possible to couple fluid dynamics with thermodynamics magnitudes. The fluid velocity may be expressed as:

\[
u = \frac{4 m}{\rho \pi d_{f,h}^2} = \frac{4 P_e}{\rho \pi d_{f,h}^2 h_{f,v}}
\]

(16)

and by substituting into the above derived relations for pressure drops, we get:

\[
\Delta p_w = \frac{\mu_L}{\rho K_w A_w} \frac{m}{h_{f,v}} = \frac{\mu_L}{\rho K_w A_w} \frac{P_e L_w}{h_{f,v}}
\]

(17)

\[
\Delta p_e = \frac{16}{g_c \pi^2 d_{c,h}^4} \left( \frac{\rho_v}{\rho_f} - 1 \right) \rho_f Q_f^2 = \frac{16 Q_e P_e}{g_c \pi^2 d_{c,h}^4 h_{f,v}} \left( \frac{\rho_v}{\rho_f} - 1 \right)
\]

(18)

\[
\Delta p_f = f \frac{L_f}{d_{f,h}} \frac{\rho_f u_f^2}{2} = C \frac{\mu_L}{\rho_f d_{f,h}^4 h_{f,v}}
\]

(19)
The pressure drop due to buoyancy force effect may be expressed as:

$$\Delta p_b = (\rho_f - \rho_v)g h$$  \hspace{1cm} (21)

Under natural circulation conditions, when the relative vertical distance between the evaporator and condenser is large enough, the loop operates without a pump, and the flow is driven dominantly by the buoyancy-generated pressure head.

The maximum power that can be removed by the natural convection for a given loop configuration is given by substituting the equations (2), (12), (17)–(21) into equation (1). By rearranging terms, the following equation can be derived:

$$P_e = \frac{2\sigma L_l}{r_c} - \rho_f g h + (\rho_f - \rho_v)g h$$  \hspace{1cm} (22)

From the above equation the maximum removal power of the capillary loop can be obtained.

**Results and discussion**

The experimental lab-scale installation, presented in [19], is used for the validation of the theoretical analysis. The wick thickness is $20 \times 10^{-3}$ m, the transversal area is $3 \times 1.131 \times 10^{-4}$ m$^2$, the pore radius is $3.17 \times 10^{-5}$ m and its permeability is $1.43 \times 10^{-11}$ m$^2$, as given by the manufacturer. The length of the liquid line is 3.67 m, the length of the vapour line is 3.05 m and the diameter of the loop is inches ($0.0127$ m). The properties of the liquid are those for 20°C (density $\rho_l = 998.2$ kg/m$^3$, viscosity $\mu_l = 100.2 \times 10^{-5}$ kg/ms, and surface tension $\sigma = 71.97 \times 10^{-3}$ N/m) and of the vapour are those for 100°C (density $\rho_v = 0.595$ kg/m$^3$ and viscosity $\mu_v = 1.206 \times 10^{-5}$ kg/ms).

The results show that the above proposed CPL modelling is able to describe well the CPL performance. A sample of the comparison diagram between theoretical and experimental results for the pressure head rise vs. flow rate is given in Fig. 2. It is obvious that for higher flow rates there is a very good agreement between theoretical and experimental results. For flow rates lower than 1.5 lt/h there is a disagreement due to the unsteady character of the flow at the start-up period. As the time passes the temperatures are stabilized and the capillary pump loop operates steadily.

Figure 3 shows the maximum power that can be removed from the capillary pump loop as a function of the liquid flow rate. For the experimental installation and the conditions mentioned above, derives from Fig. 3 that the maximum power for liquid flow rate 1 lt/h is equal to 2598 W, for 2 lt/h is 2589 W, and for 3 lt/h is 2562 W.

The pressure head rise versus input heat flux of the evaporator is shown in Figure 4 for different flow rates. Input heat flux of the evaporator affects very much the
Figure 2. Comparison between theoretical and experimental results for the pressure head rise vs. flow rate.

Figure 3. The maximum removal power from the capillary pump loop as a function of the liquid flow rate.
pressure head rise (or effective pressure drop). The function is linear and the liquid flow rate does not play a significant role. Another result from this figure is that the maximum input heat flux of the evaporator or the maximum power is between 2560 and 2590 W which is the same result derived from Figure 3. The conclusion is that the liquid flow rate does not affect the maximum power that can be removed from the capillary pump loop very much and as a result we may use a round value of 2600 W independent of the liquid flow rate. Applying the developed modelling to any installation with known geometry and flow conditions, it is possible to estimate the maximum power that can be removed by the capillary pump loop.

Pressure head rise as a function of the porous media surface can be calculated using equation (15). These results are given in Figure 5 for three different liquid flow rates. The main result is that for any liquid flow rate there is a minimum value of the porous media surface necessary to start the flow. As it is clear, the higher the flow rates, the larger the surface necessary for the loop operation. Another result is that as the porous media (or wick) surface increases, for a specific flow rate, the pressure head rise increases firstly quite steeply and then tends asymptotically to a constant value. The same behaviour may be seen for the pressure head rise as a function of the permeability of the wick, which means that these parameters affect very much the operation of a capillary pump loop.

Conclusions

A mathematical model was developed for the theoretical investigation of the operational characteristics on a small-scale Capillary Pump Loop (CPL). An
experimental lab-scale installation was used for the validation of the theoretical analysis.

Usually, in the CPL research area information such as temperature is given to show the performance of a CPL. Such information is given in the first part of the present research work [20] concerning the experimental investigation of a CPL installation. On the contrary, there is a lack of information in the literature concerning pressure drop and especially flow rates of a CPL. Since the target of the present research work is to use a CPL, instead of a centrifugal pump, in installations where flow rates are critical, the presented theoretical model is focused on the prediction of the pressure head rise and the flow rates of a CPL.

From the comparison diagram between theoretical and experimental results for the pressure head rise vs. flow rate, it is obvious that for higher flow rates there is a very good agreement between theoretical and experimental results. For lower flow rates there is a disagreement due to the unsteady character of the flow at the start-up period. During the start-up period, the temperature of the liquid inlet increases up to a maximum value and then drops down to a steady state temperature leading to a steady state operation of the capillary pump loop.

The liquid flow rate does not affect the maximum power that can be removed from the capillary pump loop very much and thus, for any installation with known geometry and flow conditions, it is possible to estimate a round value of the maximum power independent of the liquid flow rate.

The major components of the CPL pressure drop are related to the flow in the wick structure, vapour and the liquid lines. The wick structure present in the evaporator causes flow restriction that affects the CPL performance, which is dependent
on the wick transversal area and permeability, a property of the porous material that describes its ability to transport the liquid under an applied pressure gradient.

The results showed that the proposed CPL modelling is able to describe very well the CPL performance.

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