A theoretical investigation of a controlled hybrid mechanical/capillary pumped loop

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Abstract A series hybrid mechanical/CPL system is a way to improve the performance characteristics over a plain CPL system, as demonstrated by Schweickart et al\cite{1}. However, few hybrid loops actually operate because the control of the system remains difficult. An investigation of a controlled hybrid CPL is proposed based on a dynamic model, consisting of the coupling of a CPL model and a mechanical pump whose speed is controlled by a PID. We first theoretically determined the controller parameters versus the CPL characteristics in order to optimize the command for a given CPL. In a second part, some simulations of two different architectures were performed and analyzed. These results have confirmed that the hybrid system is very attractive to greatly improve the CPL performance.

1. Introduction

Two-phase capillary pumped loops such as CPL and LHP are passive and highly efficient systems for the cooling of a large range of electronic components. The maximum heat load operation is given by the maximum pressure drop across the wick (the capillary limit) or when vaporization deeply develops in the capillary structure (thermal limit). Since the thermal limit occurs at very high heat load, failures in operating conditions are largely due to the capillary limit. This becomes more and more critical for terrestrial transportation applications in which two-phase loops experience acceleration and dynamic changes of position in the gravity field and in which demanding heat load occur. As explain by Schweickart et al\cite{1}, a way of overcoming this limitation is to introduce a mechanical pump which would compensate the pressure drop of the loop. The resulting hybrid mechanical/capillary pumped loop (HCPL) gives thus a lot of new opportunities. For instance, the pump may be used to control the evaporator pressure drop so that gravity is compensated by the mechanical pump and does not affect the capillary pumping. In space industry, this could benefit to thermal characterization tests performed on ground in unfavorable gravity configuration, the maximum heat load being more representative of the one in 0g environment. The same advantages could be provided for the acceleration in terrestrial transportation applications. Moreover, the hybrid system allows to examine the use of some new working fluids with lower Global Warming Potential and toxicity even if the surface tension is low,

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which means a lower maximum capillary pressure. It can also permit to dimension the capillary evaporator for the loop nominal functioning point, and not for rare phenomenon.

Figure 1. Schweickart et al [1] hybrid loop schematic representation.

If the concept appears extremely fruitful, only a few studies in the literature are related to hybrid systems. The first one was done by Schweickart et al [1] at the NASA. They examined the coupling of a multi-evaporator CPL with a volumetric pump in series located at the inlet of the evaporators. In such configuration (Figure 1), the pump speed has to be controlled as the flow rate is still related to the heat load at the evaporator. They demonstrated the hybrid system operation based on a controller using a differential pressure sensor at the evaporator. A significant but still limited improvement of the performance (14 %) over a plain CPL was obtained. According their results, the fixed liquid/vapor interface disappear with this configuration, the loop becomes a mechanical two-phase loop with an improved evaporator.

More recently, Park et al. [2] and Crepinsek et al. [3] have built a hybrid two-phase loop with a volumetric pump for terrestrial military purpose and heat flux up to 100 W/cm² (Fig. 2). Their hybrid system consisted of a mechanical volumetric pump placed between the reservoir and the evaporator in a parallel configuration. The mechanical pump imposed a high and constant flow rate at the inlet of the evaporator in the first branch of the loop. The capillary wick of the evaporator collects a liquid flow rate accordingly to the heat flux imposed at the evaporator. The vapor then flows into the condenser placed on the second branch. The remaining liquid returns to the reservoir. The main advantage of this system is that it does not required pump control.

Figure 2. Park et al [2] hybrid loop schematic representation.
The objective of the Park et al hybrid loop was to increase the maximum heat load by establishing a continuous liquid flow under the wick. This system was also studied by Sarraf and Anderson [4] who demonstrated the system reliability on a LHP for high heat flux and high temperature (up to 200 °C) for electronic application. They proved the system operation up to a heat flux density of 67 W/cm².

Setyawan et al [5] investigated a LHP assisted with a mechanical pump (Figure 3). The aim of their work was to avoid dry-out during the starting of the loop. When the pressure at the evaporator, measured by a sensor, reached a set value, the volumetric pump started and flooded the evaporator. When the mechanical pump was switched off the nucleate boiling allowed the fixation of the liquid/vapor interface. This system has prevented the wick dry-out during starting and transient regime of the capillary loop. In all those studies, the used mechanical pump is volumetric. The coupling of a capillary evaporator with a non-volumetric pump was not yet investigated. We proposed in this work a theoretical and numerical investigation of the behavior and the control of a series HCPL close to the one presented by Schweickart et al [1], but with a non-volumetric pump. The objective is to extend the working range of the loop with a simple architecture while keeping the thermal performance of a capillary evaporator due to the fixed liquid vapor interface. First, the concept is briefly presented in the following section, then, a mathematical modeling is proposed. Numerical results are finally analyzed.

2. Concept
Unlike volumetric pump, a non-volumetric pump such as centrifugal pump is not a flow rate generator. So the flowrate of the system is still fully imposed by the capillary evaporator. The non-volumetric pump could reduce the pressure drop of the loop by generating an opposite pressure difference.

In a first part, two configurations were examined (Figure 4): (i) the mechanical pump is located between the condenser and the reservoir (architecture 1) or (ii) at the inlet of the evaporator down-stream the reservoir (architecture 2). The approach is based on a modelling of the hybrid system consisting in coupling the dynamic model of CPL with the model of a controlled centrifugal pump. Some theoretical efforts were made to investigate the feedback control leading to relations between the controller parameters (PID) and the CPL characteristics (evaporator conductance, heat transfer coefficient at the condenser, pressure drop in line, etc.).

In a second part, some simulations of the controlled hybrid system, including a CPL model that was previously investigated and validated by Kaled et al [6], were performed.
3. Modelling

3.1 Model of the HCPL with a pump located at the inlet of the reservoir (architecture 1)

The HCPL model is based on the Dutour and Kaled [7] CPL modelling work. The loop dynamic is mainly controlled by the thermal inertia at the evaporator, the two-phase dynamic at the condenser, the hydraulic inertia in the liquid line and the reservoir dynamic. Considering the reservoir saturation temperature is very well controlled, the loop dynamic can be modelled with only three state equations (1) (2) and (3.1). Main parameters are shown on Figure 5 for architecture 1.

The wick and evaporator walls are represented with an equivalent capacitance $C$. $G$ is the total conductance between the global mass temperature $T_M$ and the vapor temperature, close to the vaporization interface, $T_v$. 
\[
\frac{dT_M}{dt} = \frac{Q(t)}{C} - \frac{G}{C} (T_M - T_v) \tag{1}
\]

In the condenser, the temperature in the two-phase zone is supposed uniform. The main hypothesis consists in assuming a constant liquid-vapor phase distribution in the condenser during transient phases. It leads to equation (2) with a constant void fraction \( \alpha \).

\[
\frac{dl_{2\varphi}}{dt} = \frac{m_v - m_l}{\alpha A_c (\rho_v - \rho_l)} \tag{2}
\]

The liquid inertia is represented in equation (3.1) with a momentum balance between the condenser outlet and the reservoir inlet. \( P_{R,low} \) is the pressure at the reservoir inlet and \( Z_R \) the height difference between the reservoir and the condenser:

\[
\frac{dm_1}{dt} = \frac{p_{sat}(T_{2\varphi}) - p_{R,low} - \Delta p_v - \rho Z_R}{L_c - l_{2\varphi}} \frac{h_{2\varphi}}{S_c} + \frac{\Delta p_{pump}}{L_c - l_{2\varphi}} \frac{h_{2\varphi}}{S_c} \tag{3.1}
\]

Some additional assumptions have been done: the flow inertia in the vapor line and in the reservoir are neglected compared to pressure drop. Heat is considered to be consumed in vaporization process, thus specific heat and heat losses are neglected. According to these assumptions, closure equations (4.1) to (4.4) and equation (4.6) are given by Dutour and Kaled [7]. The two-phase condensation coefficient (equation (4.5)) is given by the Wedekins et al [8] correlation for low liquid flow rate in an annular condenser. In this correlation a mean void fraction \( \alpha \) is considered.

\[
m_v = G \frac{(T_M - T_v)}{\Delta H_{lv}} \tag{4.1}
\]

\[
T_v = T_{2\varphi} + \frac{\Delta p_v + \Delta p_{2\varphi}}{a_{vap}} \tag{4.2}
\]

\[
h_{2\varphi} D_{ce} L_{2\varphi} (T_{2\varphi} - T_{sink}) = \Delta H_{lv} \left[ m_v - \rho_v \frac{m_l - m_v}{\rho_l - \rho_v} \right] \tag{4.3}
\]

\[
\frac{1}{h_{2\varphi \cdot D_{ce}}} = \frac{1}{h_{2\varphi \cdot D_{ci}}} + \frac{1}{h_{1\cdot D_{ce}}} + \frac{m(D_{ce})}{2 \Delta p_v} \tag{4.4}
\]

\[
h_{2\varphi \cdot l} = 0.7253 (\rho_v - \rho_l) \frac{\Delta H_{lv}}{\mu (m_v - m_l) / \rho_l} \tag{4.5}
\]

\[
p_{R,low} \approx p_{sat}(T_R) \tag{4.6}
\]

The pressure drops are considered to vary as the second order of the mass flow rate (equations (5.1) to (5.3)):

\[
\Delta p_l = f_{l,1} m_l + f_{l,2} m_l^2 \tag{5.1}
\]

\[
\Delta p_v = \Delta p_{groove} + \Delta p_v = f_{v,1} m_v + f_{v,2} m_v^2 \tag{5.2}
\]

\[
\Delta p_{2\varphi} = f_{2\varphi,1} \frac{L_{2\varphi}}{L_c} m_v + f_{2\varphi,2} \frac{L_{2\varphi}}{L_c} m_v^2 \tag{5.3}
\]

where \( f \) are the friction factors.
3.2 Model of the HCPL with a pump located at the outlet of the reservoir (architecture 2)
For architecture 2, equation (3.1) becomes equation (6). As the pump is no more situated on the liquid line.

\[
\frac{d\dot{m}_1}{dt} = \frac{p_{sat}(T_{2g}) - p_{R,low} - \Delta p_{v} - \rho_l g Z_R}{s_c} \frac{t_c - t_g}{s_g}
\]  

(6)

The other state equations and closure equations remain unchanged.

3.3 Mechanical pump and control model
The pressure generated by a centrifugal pump depends on the mass flow rate and of its rotational speed \(\omega_p(t)\) (equation (7)). To model a perfect mechanical pump, nominal values (which are intrinsic values of the chosen pump) are required. In a second step, equation (6) can be replaced by the pump characteristic given by the pump constructor.

\[
\Delta p_{pump} \approx \Delta p_{nom} \left( \frac{\omega_p(t)}{\omega_{p,nom}} \right)^2 - \Delta p_{nom} \left( \frac{\omega_p(t)}{\omega_{p,nom}} \right) \dot{m}_{pump}
\]  

(7)

Then, the time delay between the command set value and the pump set value is defined in equation (8):

\[
\frac{d\omega_p}{dt} = \frac{\omega_{p, set} - \omega_p}{\tau}
\]  

(8)

3.3.1 PID controller
To control the HCPL, the output which will be observed, must be defined. The chosen value is the pressure drop at the evaporator liquid-vapor interface \(\Delta p_{cap}\). The difference between \(\Delta p_{cap}\) and \(\Delta p_{cap, max}\) can then be determined. Moreover, a feedback loop can be achieved using \(\Delta p_{cap}\) in order to control the pump and keep the evaporator away from the capillary limit.

For architecture 1, the pump is between the condenser and the reservoir. Equation (9) is used to determine \(\Delta p_{cap}\) including the pressure drop in the pump and the inertia effect in the liquid line:

\[
\Delta p_{cap} = \Delta p_{grooves} + \Delta p_v + p_{sat}(T_{2g}) - p_{sat}(T_R) - \rho_l g (Z_R - Z_{ev}) + \Delta p_{l,2} + \Delta p_{wick}
\]  

(9)

With \(\Delta p_{l,2}\) the pressure drop between the reservoir and the evaporator inlet, \(Z_{ev}\) the height difference between the evaporator and the condenser and \(\Delta p_{wick}\) the pressure drop in the capillary evaporator wick.

For architecture 2, the mechanical pump is situated between the reservoir and the evaporator. Equation (10) is used to determine \(\Delta p_{cap}\):

\[
\Delta p_{cap} = \Delta p_{grooves} + \Delta p_v + p_{sat}(T_{2g}) - p_{sat}(T_R) - \rho_l g (Z_R - Z_{ev}) + \Delta p_{l,2} - \Delta p_{pump} + \Delta p_{wick}
\]  

(10)

A command law corresponding to a PID controller is then chosen to change the rotational speed of the mechanical pump (equation (11) and (12)) so that \(\Delta p_{cap}\) tends to its set value \(\Delta p_{cap, set}\):

\[
\omega_{p, set}(t) = K_p \cdot \varepsilon(t) + K_i \int_0^t \varepsilon(t) \cdot dt + K_d \frac{d\varepsilon(t)}{dt}
\]  

(11)

\[
\varepsilon(t) = \Delta p_{cap} - \Delta p_{cap, set}
\]  

(12)
The evaporator pressure difference is then controlled to not exceed \( \Delta p_{\text{cap,set}} \). When the pressure drop in the circuit is lower than this set value, the HCPL works like a CPL, i.e. in a pure capillary pumping mode. Then, when the capillary pressure difference reaches the set value, the mechanical pump is started. The rotational speed of the centrifugal pump is controlled with the PID to keep the evaporator pressure difference constant.

### 3.3.2 Analytical expressions of the controller coefficients

Dutour et al. [7] showed that an accurate solution of a nonlinear dynamic model of CPL is provided by a second-order linear system where the undamped natural frequency \( \omega_0 \) and system damping ratio \( \xi \) can be determined as functions of CPL parameters (evaporator equivalent conductance \( G \) and equivalent capacitance \( C \), global heat transfer coefficient at the condenser, friction coefficients and fluid properties). Following this approach, the transfer function of a CPL can be defined as:

\[
F(p) = \frac{K_D \omega_0^2}{p^2 + 2\xi \omega_0 p + \omega_0^2}
\]

The second order transfer function for a PID controller is simultaneously defined by:

\[
F(p) = \frac{K_P p^2 + K_D p + K_I}{p}
\]

The optimal control coefficients are then determined with parameters \( \omega_0 \) and \( \xi \) and by imposing the proportional coefficient \( K_P \):

\[
\begin{align*}
K_P &= 2 \cdot K_D \cdot \xi \cdot \omega_0 \\
K_I &= \frac{K_D}{4} \left( \frac{K_P}{K_D} - (2 \cdot \xi \cdot \omega_0)^2 + 4 \cdot \omega_0^2 \right)
\end{align*}
\]

### 4. Simulations

Some simulations were performed applying heat load steps at the evaporator.

#### 4.1 Performance simulations with the mechanical pump at the inlet of the reservoir (architecture 1)

For architecture 1 (Figure 4.1), the flow rate in the pump is the flow rate in the liquid line:

\[
\dot{m}_{\text{pump}} = \dot{m}_l
\]

Figure 6 shows the mass flow rate in the pump when heat load steps are applied to the evaporator. The liquid redistribution between the condenser and the reservoir leads to large overshoots of the mass flow rate. Thus, in this architecture, the control of the pump is highly influenced by the liquid flow dynamics and the applied power cycle at the evaporator. However as shown in Figure 7, it was possible for this power cycle to find a set of controller parameters leading to a robust control of the HCPL. As explain in paragraph 3.3.1, when the pressure drop in the circuit is lower than this set value, the HCPL works like a CPL, i.e. in a pure capillary pumping mode. Then, when the capillary pressure difference reaches the set value, the mechanical pump is started. The pump rotational speed depends of the heat load maintaining the evaporator pressure difference constant at the set difference pressure \( \Delta p_{\text{cap,set}} = 9 \text{ kPa} \). As a consequence, the HCPL was able to extend the CPL functioning range up to 2500 W for this simulation i.e. beyond the capillary limit of the wick (1200 W) and up to 40 kPa for the total pressure drop in the circuit. The HCPL maximum heat flux is defined by the maximum pressure difference the mechanical pump can generate.
A limitation was observable for this CPL configuration (unfavorable gravity position) when the heat load was decreasing. Actually, during this transient, the liquid flow rate is undershooting and a reversed flow rate could exist in the liquid line. As a consequence, the pressure drop in the pump drastically decreased and the control of the HCPL failed. Some additional considerations on this architecture including for instance a check valve have to be tested.

**Figure 6.** Small amplitude heat flux step at the evaporator from 100 W to 2500 W with architecture 1. a) Power cycle b) flow rate across the pump.

**Figure 7.** Small amplitude heat flux step at the evaporator from 100 W to 2500 W with architecture 1. a) Power cycle b) Pump differential pressure c) evaporator and circuit HCPL differential pressure.

4.2 Performance simulations with the mechanical pump at the outlet of the reservoir (architecture 2)
In this configuration (Fig. 4.ii) the liquid flow rate across the pump is the flow rate due to the vaporization (equation (17)). Actually, as the mechanical pump is a non-volumetric one, the flow rate in this branch is still imposed by the capillary evaporator since the evaporator pressure drop remains positive.

\[
\dot{m}_{pump} = \dot{m}_p = \frac{\dot{Q}_{ev}}{\Delta H_{lv}}
\]  

Equation (17)

Figure 8 shows the mass flow rate in the pump when heat load steps are applied to the evaporator. An important benefit compared to architecture 1 is that: (i) There is no overshoot consecutive to an increase of the heat load steps and (ii) the flow rate across the pump is not reversed when the heat load is decreased. This avoids the failure of the HCPL control as described in the previous subsection.

Figure 9 shows the simulation of the HCPL when the evaporator pressure difference is controlled to not exceed \(\Delta p_{cap,set} = 11\) kPa. As explain in paragraph 3.3.1, when the pressure drop in the circuit is lower than this set value, the HCPL works like a CPL, i.e. in a pure capillary pumping mode. Then, when the capillary pressure difference reaches the set value, the mechanical pump is started. The rotational speed of the centrifugal pump is controlled with the PID to keep the evaporator pressure difference constant. On Fig. 9.c) the well-controlled capillary pressure difference can be observed for both increasing and decreasing heat load steps during the hybrid pumping mode. During this mode, the total pressure drop in the circuit exceeded for 3 times the capillary limit. Returning to low heat loads, the pump is stopped when the pressure difference generated by the centrifugal pump become lower than a set value (here 100 Pa), the loop then works as CPL again.

![Figure 8](image_url)  

**Figure 8.** Small amplitude heat flux step at the evaporator from 100 W to 2500 W with architecture 2. a) Power cycle b) flow rate across the pump.

The simulation shows an important enhancement of the pumping performance: the HCPL extended the operating range from 1 200 W to 2 500 W with the chosen mechanical pump and a suitable condenser. As for architecture 1, the HCPL maximum heat flux is defined with the maximum pressure the mechanical pump can generate. This enhancement is also expected in the HCPL response to large amplitude power cycle. We simulated two different HCPL configurations: (i) when the pressure drop in the circuit is dominated by the pressure drop in the vapor line and (ii) when the pressure drops in both lines are comparable. As a consequence, the dynamics in the liquid line is negligible for the first configuration while it is not in the second one.
Figure 9. Small amplitude heat flux step at the evaporator from 100 W to 2500 W with architecture 2. a) power cycle b) Pump differential pressure c) evaporator and circuit HCPL differential pressure.

Figure 10. Architecture 2 with strong pressure drop in the vapor line. a) Power cycle b) Pump differential pressure c) evaporator and circuit HCPL differential pressure.
Figure 10 shows the simulation of the dynamics of the HCPL when large amplitude of steps (up to 2300 W) are applied considering the liquid line dynamics is negligible. The pump starts and stops efficiently for each step and the HCPL works well.

On Figure 11, the simulation was performed with comparable pressure drop in both lines. The consequence is that the pressure drop in the loop is affected by the liquid flow rate overshoot and could exceed the capillary limit during the transient as shown on Figure 11.c. However, the simulations demonstrated that the HCPL control is still robust with a pressure drop set at $\Delta p_{cap, set} = 9$ kPa.

![Figure 11. Architecture 2 when HCPL is affected by the liquid line dynamics. a) Power cycle b) Pump differential pressure c) evaporator and circuit HCPL differential pressure.](image)

5. Conclusions
The interest of a hybrid two-phase capillary loop (HCPL) with a non-volumetric pump was investigated in this work. The approach consisted in modelling the coupling between a CPL and a centrifugal pump. The control of the pump (the actuator) is based on PID controller based on a pressure drop sensor. Two architectures were simulated: (i) in architecture 1, the mechanical pump is placed at the inlet of the reservoir, (ii) in the second architecture, the mechanical pump is placed at the outlet of the reservoir. An expression of the PID parameters based on the CPL properties has been established. So that, based on the model, it is possible to find the suitable pump and command for a given CPL.

Analyses of the dynamics of the resulting system showed that this coupling could greatly extend the pumping performance of a CPL. For architecture 1, simulations also showed an improvement for increasing heat load steps. However, when the heat load is decreased, the control failed if a reverse flow occurred in the liquid line. This point has to be addressed and could probably be fixed using a suitable check valve. The best robustness was reached by architecture 2: the improvement of the functioning range of the loop and the control performance is demonstrated for increasing and decreasing heat load, for both large heat load steps and transient step with high flow rate peaks.
6. References

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