Experimental investigation of a capillary-assisted high performance evaporator

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Abstract. A particular architecture for a miniaturized evaporator is proposed in which liquid is spread on the heat source using a thin capillary structure connected to two manifolds. The vapor produced by the heat source is extracted from the device using a separate outlet. A prototype has been designed, manufactured and integrated in a mechanically pumped loop in order to evaluate the evaporator efficiency in terms of heat transfer coefficient and induced pressure drop. In this paper, a capillary structure consisting in parallel rectangular micro-grooves is investigated with HFE7000 used as the working fluid. Results show that the evaporator is able to extract heat flux of the order of 10 W.cm\textsuperscript{-2} with heat transfer coefficients close to 104 W.m\textsuperscript{-2}.K\textsuperscript{-1} while inducing pressure losses below a few hundred Pascal. Direct visualizations of the flow in the capillary structure allowed a precise characterization of the evaporator functioning mode, in particular the existence of two distinct two-phase flow regimes depending on the flooding of the capillary structure: nucleate boiling and liquid films evaporation. In order to predict the occurrence of these operating modes, a hydraulic model of the evaporator assembly has been developed and validated thanks to dedicated experimental campaigns. In particular, this model is able to predict the separation between the nucleate boiling and evaporation mode and highlights the fact that the capillary limit is not the main restricting factor in terms of the device maximum manageable heat flux.

1. Introduction
In addition to the well-known current context of miniaturization of heat producing units which leads to rising heat flux to be handled, the increase of the total number of heat sources in a given system appears as an upcoming problematic to consider. In particular, the “more electrical aircraft” target investigated by aircraft manufacturers implies a significantly higher number of dissipating electronic components embedded in a given plane. In practice, this implies the necessity to develop lightweight and compact heat management devices which can easily be embedded in complete cooling systems such as two-phase loops. Then, numerous experimental and theoretical studies have been led on ow boiling in micro-channels evaporators [1,2] This approach, despite being promising in terms of heat transfer coefficient and maximum heat flux density, has two major drawbacks: first, for a given dissipated heat flux, the thermal performances of these devices highly depend on the liquid ow rate supplied to the device.

In particular, small outlet vapor quality has to be maintained in order to maximize the evaporator heat transfer coefficient [3,4]. Moreover, their simple architecture induces significant pressure drops as

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two-phase ow has to be forced into very small cross-sectional area by the pumping source of the loop. In the case of a secondary evaporator embedded in a capillary pumped loop, this might imply a reduction of the operating range of the complete system. For a mechanically pumped loop, the necessary pumping power may be significantly increased.

To address this issue, a concept of capillary-assisted evaporator has been developed by the ACT company [5]. Their prototype is designed to passively extract liquid from a primary ow using local capillary pumping. The collected ow rate is equal to the vaporization ow rate necessary to extract the applied heat ux. The vaporized ow and liquid overflow are then evacuated through two separated outlets. The use of local capillary pumping allows the stabilization of the liquid/vapor interface as well as self-adaptation of the device. Moreover, the pressure drop induced by the device can be negligible, as the ow through small cross-sections (in the capillary structure) is solely driven by capillary forces while the mechanically pumped ow can be maintained in arbitrary wide sections. Since this proof of concept, this design has been more thoroughly investigated.

On one hand, Park & Crepinsek [6] led an experimental study on a prototype similar to the one proposed by ACT. In their setup the vapor produced by the evaporator is directly sent to the loop reservoir in which it is condensed. By varying the liquid ow rate supplied to the evaporator, the authors showed the existence of an optimal range of this parameter. Indeed, the main liquid ow rate has to be high enough to prevent ow boiling in the liquid supply line but low enough to avoid the evaporator flooding. More recently, Jiang et al. [7] proposed a different loop architecture in which the vapor produced by the evaporator is condensed and then reinjected in the main liquid line. Likewise, a minimum liquid ow rate necessary to ensure proper functioning of the evaporator is highlighted. The start-up phase of the evaporator has also been investigated [8].

Finally, Setyawan et al. [9] proposed a slightly different architecture. Their system, called a hybrid loop heat pipe with pump assistance, is based on the following principle: in usual condition, the loop operates as a regular LHP (in other words, the fluid motion is solely induced by capillary pumping at the evaporator). However if dry-out is detected, a mechanical pump placed upstream of the evaporator is turned on in order to push liquid back into the porous wick. The authors then demonstrate the ability of their system to increase the maximum heat flux that can be handled by the loop. All the hybrid evaporators architectures proposed above are based on the utilization of a ‘thick” porous media in which, similarly to a standard capillary evaporator, the capillary pumped liquid ow rate and the heat flux applied are in opposite direction. This implies the existence of a vapor layer between the liquid/vapor interface and the heat source and does not promote miniaturization.

In this context, an alternative capillary-assisted evaporator architecture has been developed and tested. In particular, the phase separation between the main liquid ow and the vaporized ow is ensured in a way allowing direct spreading of the liquid on the heated surface. The first objective of this paper is to detail the functioning mechanism of this evaporator as well as its performance in terms of heat transfer coefficient and induced pressure drop. In a second step, a hydraulic model aiming to predict the operating mode of the evaporator in function of two experimental parameters is presented and compared to the experimental data.

2. Experimental setup

2.1 Capillary-assisted evaporator

The evaporator is based on a planar capillary structure. In this study, the considered porous media is a silicon substrate in which 355 rectangular micro-grooves (40 µm wide and 75 µm deep) separated by 16 µm thick walls have been etched using a standard Deep Reactive Ion Etching (DRIE) process. The capillary structure covers a 2 x 2 cm² area corresponding to the heated section of the evaporator. Then, as represented on Figures 1 and 2, two liquid manifolds (called “distributors”) are placed directly onto the capillary structure. A 2 mm wide groove is etched on the bottom of the distributors to allow the liquid to ow into the capillary structure. The assembly is set into a hermetic container ensuring a pressurized contact between the capillary structure and the distributors as well as the connection of the distributors to the liquid inlet and outlet of the evaporator. The generated vapor is evacuated through a
dedicated outlet placed on one side of the evaporator in order to be reinjected in the liquid line downstream of the evaporator. Finally, a sapphire window placed on top of the evaporator allows direct observations of the internal ow distribution. The resulting prototype assembly is represented in Figure 4. Similar to the existing capillary-assisted evaporators mentioned in introduction, this architecture allows a passive regulation of the device, as the capillary structure only collects the fraction of the liquid ow needed on a main liquid line passing through the evaporator.

![Figure 1](image1.png)

**Figure 1** Top schematic view of the distributor/capillary structure assembly.

On the other hand, the major difference of this devices with the existing literature consists in the phase separation between liquid and vapor. Here, instead of a “thick” porous medium ensuring both phase separation and capillary pumping, the flow separation is ensured by the distributors, implying that the unique function of the capillary structure is to extract the needed liquid ow rate and to spread it on the heated surface.

### 2.2 Test bench

A test bench dedicated to this evaporator, schematically represented in Figure 4, has been designed and set up. It consists of a mechanically pumped two-phase loop in which the ow circulation is ensured by a volumetric pump (gear pump). A copper heating block placed directly in contact with the evaporator, is used as the heat source.

![Figure 2](image2.png)

**Figure 2** Schematics of the distributor/capillary structure assembly: (a) transversal cross section (left); (b) longitudinal cross section (right).
A control valve, placed between the evaporator liquid outlet and the liquid and vapor line connection allows the adjustment of the liquid pressure in the evaporator whatever the liquid flow rate of the loop. Indeed, as explained thereafter, this parameter has a strong influence on the functioning of the evaporator. The liquid/vapor mixture is then condensed in a counter-current heat exchanger connected to an external water circuit. After its condensation, the fluid passes through a mass-flowmeter. Finally, a constant pressure reference is set directly downstream of the pump by a two-phase reservoir linked to a thermostatic bath. The working fluid used is degassed HFE7000.

In order to evaluate this two-phase device in terms of cooling capabilities, the following instrumentation is set up. The liquid flow rate imposed by the pump is measured using the mass flowmeter downstream of the condenser (micro-motion CMFS010M, accuracy ±2.10^{-3} kg/h, range 0-8 kg/h). The absolute pressure at the entrance of the evaporator is measured by a Keller PA-33X pressure transducer (accuracy ±200 Pa, range 0-2 bar). Two differential pressure transducers Keller PD-33X (accuracy ±25 Pa, range 0-0.2 bar) are used. The first one measures the pressure drop between the liquid inlet (Figure 4, point A) and the liquid/vapor junction (Figure 4, point C) while the other measures the pressure drop between the liquid inlet and the vapor outlet (Figure 4, point B). Calibrated 0.5mm K-type thermocouples (accuracy ±0.2°C) are placed directly in the fluid flow at each inlet and outlet of the evaporator in order to measure the characteristic fluid temperatures. Finally, the evaporator wall temperature is measured with five 0.25 mm calibrated K-type thermocouples (accuracy ±0.2°C) bonded in dedicated grooves machined in the silicon substrate.
3. Evaporator operation and associated performance

3.1 Qualitative flow distribution

In this section, the control valve setting as well as the liquid flow rate supplied to the evaporator are set constant. Then, the applied heat flux is progressively increased from the onset of phase change in the evaporator until dry-out. During this sequence, the evaporator goes through 4 different operating modes:

- at low heat fluxes, the capillary structure appears overfed with liquid, in other words liquid freely flows from the distributor to the heated surface (Figure 5), thus the heat flux is extracted through nucleate boiling on the capillary structure. At the lowest heat fluxes, a part of the liquid flow directly exits the evaporator through the vapor outlet;
- at higher heat flux, the liquid is constrained within the capillary structure (Fig 6), allowing its capillary pumping from the distributor. As nucleate boiling is no longer observed, heat transfer then occurs through evaporation of the liquid films in the grooves, similarly to the evaporator area of a heat pipe;
- between these two functioning ranges (i.e. at intermediate heat fluxes), a transition mode is observed in which the nucleate boiling and evaporation modes are alternatively observed;
- finally, when a sufficiently high heat flux is applied, the free surface of the capillary structure is completely dried-out (Figure 7). The evaporator wall then achieves a significantly higher equilibrium temperature. As in these conditions the fluid is almost entirely confined under the distributors, direct visualizations of the phase structuration is impossible. Hence the flow repartition described in the right part of Figure 8 (flow boiling in the distributor groove) cannot be observed. However, the observation of vapor in the liquid outlets transparent tubes confirms this hypothesis.

3.2 Evaporator performance as a function of the applied heat flux

A situation similar to the one presented above is considered: the liquid flow rate supplying the evaporator is set at \( \dot{m}_{\text{tot}} = 3.0 \text{ kg.h}^{-1} \) and the control valve is set to induce a pressure drop \( \Delta P_{AC} \approx 100 \text{ Pa} \). The heat flux is then gradually increased from 3.4 to 9.8 W.cm\(^{-2}\). For each measurement point, acquisition of data starts once permanent regime has been reached. Considering the heated surface \( S \) of the evaporator, \( T_{\text{w}} \) the mean of the 5 wall temperatures measurements and \( \Phi_{\text{losses}} \) the evaporator heat loss to the ambient (measured by progressive heating of the vacuumed evaporator), the apparent heat transfer coefficient of the evaporator, is then defined as:

\[
h_{\text{app}} = \frac{\Phi - \Phi_{\text{losses}}}{S \times (T_{\text{w}} - T_{\text{sat}})}
\]

The variation of this parameter with the applied heat flux is represented on Figure 8 with the corresponding functioning modes. Considering the points A, B and C represented on Figure 4 the measured pressure drops \( \Delta P_{AC} \) and \( \Delta P_{AB} \) as well as the deduced vapor pressure drop \( \Delta P_{BC} = \Delta P_{AC} - \Delta P_{AB} \) are represented on Figure 5. In terms of heat transfer coefficient (Figure 9), the nucleate boiling mode appears as mildly less effective than the evaporation mode. Indeed, given the manufacturing process of the capillary structure (plasma etching of a plain silicon surface), the heated surface is likely very smooth, inducing a small number of nucleation sites, hence an overall poor nucleate boiling performance. On the other hand, a sharp increase of \( h_{\text{app}} \) is observed during the transition to evaporation mode. Once stable evaporation is achieved, the heat transfer coefficient remains stable. On the contrary, once dry-out of the free surface occurs (partial at the penultimate point shown on and complete at the last one), the heat transfer coefficient is divided by approximately 5 which seems consistent with the significantly reduced (but not suppressed) wetted area in the assumed flow distribution.
Figure 5 Flow visualization and schematic representation of the phase distribution during nucleate boiling mode.

Figure 6 Flow visualization and schematic representation of the phase distribution during liquid films evaporation mode.

Figure 7 Flow visualization and schematic representation of the assumed phases distribution at dry-out.

Considering pressure drops (Figure 8 b), different tendencies can be observed. On one hand, apart from the dry-out points, the complete liquid line pressure drop ($\Delta P_{AC}$) only slightly decreases with the heat flux. Indeed, as a larger portion of the liquid flow rate is extracted from the main line by the evaporator (to absorb the increased applied heat flux) a lower flow rate (thus, a lower pressure drop) is expected between the liquid outlet and point C. Once dry-out occurs, this pressure drop increases significantly as a result of the transition to two-phase flow in the liquid lines.
Figure 8 Apparent heat transfer coefficient of the evaporator as a function of the applied heat flux (a), pressure drops induced by the different parts of the evaporator assembly as a function of the applied heat flux (b).

On the other hand, the vapor outlet pressure drop ($\Delta P_{AB}$) progressively decreases from the onset of boiling to the dry-out. Two successive phenomena can explain this variation: At first (in boiling mode), the diminution of the mean liquid level in the evaporator decreases the hydrostatic pressure difference between A and B. Then, during evaporation mode, as the capillary pumped flow rate in the micro-grooves rises, the associated menisci curvatures increases concurrently. As a consequence, a higher capillary pressure is induced leading to a decrease of the measured pressure difference. As capillary pumping is likely to be suppressed once dry-out occurs, the pressure drop then increases sharply.

Finally, the resulting vapor line pressure drop ($\Delta P_{BC}$), simply increases with the increase of the vaporized flow rate passing through, its reduction during dry-out indicates that the vapor flow rate in the vapor line is reduced as a significant part of the produced vapor is driven in the liquid line.

4. Analytical model of transition and dry-out

As seen in the previous section, the two-phase flow distribution is strongly influenced by the experimental parameters. The objective of the model developed thereafter is then to explain and predict the different operating modes observed using a simple hydrodynamic model.

4.1 Model presentation

As discussed earlier, starting from point A (Figure 4), two distinct paths can be taken by the liquid, depending whether it passes through the capillary structure (Figure 4, path ABC) or goes directly through the evaporator (Figure 4, path AC). As those two distinct lines have common starting and ending points, the pressure drops generated by those two parallel flows must be the same (in permanent regime). The pressure drop of the liquid line is set by the control valve setting and the liquid flow rate flowing through the AC path. Therefore, the flow in the line ABC has to adapt itself to this fixed pressure drop. The hypothesis developed here is that the different operating modes observed correspond to the only flow configuration allowing the ABC pressure drop to balance the fixed liquid line pressure drop.

4.2 Pressure drops evaluation

In practice, the pressure drop along the ABC path can be divided into four terms:

- the hydrostatic pressure $\Delta P_g = \rho_l g H$, H being the height difference (represented in Figures 5-7) between the liquid outlet and the liquid level in the evaporator, $\rho_l$ the liquid density and $g$ the gravity acceleration;
- the liquid pressure drop $\Delta P_{\text{grooves}}$ induced by the liquid flow in the capillary structure;
- the capillary pressure drop $\Delta P_{\text{cap}} = \gamma / R$, induced by the interface radius of curvature $R$ in the capillary structure (considering the significant extension of the liquid films, the longitudinal curvature radius is considered infinite). In the case of nucleate boiling, this term is considered equal to 0 as the capillary structure is flooded;
- the sum of regular and singular pressure drop $\Delta P_{\text{BC}}$ induced by the vapor flow from the vapor outlet (point B) to the liquid/vapor junction (point C).

Then, the hydraulic equilibrium of the two lines is simply given by:

$$\Delta P_{AC} = \Delta P_{\text{grooves}} + \Delta P_{\text{BC}} - \rho g H - \Delta P_{\text{cap}}$$  \hspace{1cm} (2)

The first two terms (i.e. viscous pressure drops) can be estimated as a function of the vaporized mass flow rate $\Phi_{\text{vap}} / h_{lv}$. For the liquid flow along the grooves, two different areas represented on Figure 9 have to be considered:

- the closed zone (under the distributor), in which the flow section is a closed rectangle;
- the open zone (downstream of the closed zone), in which the liquid is vaporized along the open-top groove.

Given the very low Reynolds number in the micro-grooves (ranging between 1 and 10 for the experimental data presented in this paper), fully developed and laminar flow is assumed. Therefore, the pressure drop in the micro-grooves can be estimated by the following expressions [11]:

$$\Delta P_{\text{grooves}}^{\text{closed}} = 2P_0 \times \mu_l U_l \times \frac{L_{\text{closed}}}{D_h, \text{closed}}$$   \hspace{1cm} (3)

$$\Delta P_{\text{grooves}}^{\text{open}} = 2P_0 \times \mu_l U_l \times \frac{0.5 L_{\text{open}}}{D_h, \text{open}}$$   \hspace{1cm} (4)

In Eqs. 3 and 4, $P_0$ is the Poiseuille number of the flow through rectangular channels, estimated by Shah & London correlation. $D_h$ is the hydraulic diameter of the grooves, $\mu_l$ and $U_l$ are the liquid viscosity and mean velocity. $L_{\text{closed}}$ and $L_{\text{open}}$ represents respectively the length of the grooves under the distributors and in the vaporization area. The numerical values of these parameters are summarized in Table 1.

The liquid mean velocity is calculated assuming that the vaporized flow rate is equally distributed in each of the 4 flow directions (2 for each distributor) and in the N microgrooves of cross-sectional area $A_l$:

$$U_l = \frac{\Phi_{\text{vap}}}{4N_A h_{lv}}$$   \hspace{1cm} (5)

With $h_{lv}$ the fluid latent heat of vaporization. Finally, the 0.5 factor multiplying $L_{\text{open}}$ results from the inclusion of a linear diminution of mass flow rate along the groove due to its evaporation [11] (in the assumption of a homogeneous heat flux along the groove).
**Figure 9** Schematic view of the liquid flow considered along the micro-grooves in evaporation mode.

**Table 1** Numerical values of the geometric parameters in Eq. (3), (4) and (5)

|        | Open zone | Closed zone |
|--------|-----------|-------------|
| Po     | 18.0      | 15.3        |
| D_h [µm] | 63       | 52          |
| L [mm] | 3         | 1.5         |
| N      | 304       | 304         |

For the vapor flow, Idel’cik pressure drops memento [10] is used to estimate the pressure drop induced by each hydraulic element of the BC line. For brevity, this calculation is not detailed here, however the complete calculated pressure drop (including the vapor flow in the vapor lines and the liquid flow in the micro-grooves) can be estimated within a 6% error range by the following correlation (for a vaporization heat flux expressed in Watts and a pressure drop in Pascals):

\[
\Delta P_{BC} = 0.1802 \Phi_{vap}^2 + 11.62 \Phi_{vap} - 11.00
\]  

(6)

As mentioned above, the objective of this model is to predict the flow transitions experimentally observed, in particular the transitions from boiling to evaporation and from evaporation to dry-out. The definition of these two transitions is then the objective of the next section.

### 4.3 Transition and dry-out definition

First, the transition point between boiling and evaporation is defined as the vaporization heat flux for which the grooves are no longer flooded and, in the same time, the meniscus curvature at the entry of the “open zone” is still infinite. In other words, the transition point can be defined as the vaporization heat flux \( \Phi_{vap}^{tr} \) which satisfies the following equation (derived from Eq. (2)):

\[
\Delta P_{AC} + \rho g H_0 = \Delta P_{\text{grooves}}(\Phi_{vap}^{tr}) + \Delta P_{BC}(\Phi_{vap}^{tr})
\]  

(7)
Indeed, in the case where $\Phi_{\text{vap}} < \Phi_{\text{vap}}^{\text{tr}}$, the pressure drop generated by the vaporized flow is too small to balance the left hand side of Eq. (7). The only solution is then to reduce the left-hand side by increasing the liquid level in the evaporator (inducing $H < H_0$), in which case nucleate boiling is inevitable. On the contrary, if $\Phi_{\text{vap}} > \Phi_{\text{vap}}^{\text{tr}}$, the left hand side is too small, meaning that a non-zero meniscus curvature at the entry of the open zone is necessary to maintain equilibrium, which implies evaporation along the groove.

Then, the dry-out point is supposed to be imposed by the capillary limit of the evaporator. In other words, it is the vaporization heat flux inducing the maximum meniscus curvature at the end of the “open zone”. It can thus be defined as the vaporization heat flux solution of the following equation:

$$\Delta P_{AC} + \rho g H_0 + \Delta P_{\text{vap}}^{\text{max}} = \Delta P_{\text{closed}}^{\text{grooves}}(\Phi_{\text{vap}}^{\text{max}}) + \Delta P_{\text{open}}^{\text{grooves}}(\Phi_{\text{vap}}^{\text{max}}) + \Delta P_{BC}(\Phi_{\text{vap}}^{\text{max}})$$

(8)

4.4 Experimental validation

For a given evaporator geometry and capillary structure, the model developed above allows the determination of the vaporized heat flux as a function of the total pressure drop of the liquid line $\Delta P_{AC}$. However, if the value of $\Delta P_{AC}$ is directly measured on the test bench, the exact value of the vaporized heat flux is yet unknown as only a part of the heat flux provided by the heating block is absorbed by the vaporization of the fluid. Indeed, an energy balance applied to the evaporator gives the following expression for the heat flux $\Phi_{\text{tot}}$ dissipated by the heating block:

$$\Phi_{\text{tot}} = \Phi_{\text{vap}} + \Phi_{\text{AC}}^{\text{sensible}} + \Phi_{\text{AB}}^{\text{sensible}} + \Phi_{\text{v}}^{\text{sensible}} + \Phi_{p}$$

(9)

With $\Phi_{\text{AC}}^{\text{sensible}}$ and $\Phi_{\text{AB}}^{\text{sensible}}$ the sensible heat fluxes respectively absorbed by the non-vaporized and vaporized flow rates:

$$\Phi_{\text{AC}}^{\text{sensible}} = (\dot{m}_{\text{tot}} - \frac{\Phi_{\text{vap}}}{l_{\text{v}}} \times C_{p_l}(T_{\text{in}}^{\text{out}} - T_{\text{in}}^{\text{in}}))$$

(10)

$$\Phi_{\text{AB}}^{\text{sensible}} = \frac{\Phi_{\text{vap}}}{l_{\text{v}}} \times C_{p_l}(T_{\text{sat}} - T_{\text{in}}^{\text{in}})$$

(11)

$\Phi_{\text{v}}^{\text{sensible}}$ is the heat flux absorbed by the vapor superheating:

$$\Phi_{\text{v}}^{\text{sensible}} = \frac{\Phi_{\text{vap}}}{l_{\text{v}}} \times C_{p_v}(T_{\text{out}}^{\text{out}} - T_{\text{sat}})$$

(12)

$\Phi_p$ represents the heat loss to the ambient mentioned in section 3.2. In these expressions, $C_{p_l}$ and $C_{p_v}$ are respectively the specific heat of the liquid and vapor phases, $T_{\text{in}}^{\text{in}}$ and $T_{\text{in}}^{\text{out}}$ are the inlet and outlet measured liquid temperatures and $T_{\text{out}}^{\text{out}}$ is the measured outlet vapor temperature (at point B). $T_{\text{sat}}$ is the saturation temperature of the fluid, deduced from the liquid pressure measured at the evaporator inlet.

Finally, Eq. 9 to 12 have been used to obtain the following expression of the vaporized heat flux:

$$\Phi_{\text{vap}} = \frac{\Phi_{\text{tot}} - \Phi_p - \dot{m}_{\text{tot}} C_{p_l}(T_{\text{in}}^{\text{out}} - T_{\text{in}}^{\text{in}})}{l_{\text{v}} C_{p_v}(T_{\text{out}}^{\text{out}} - T_{\text{sat}})}$$

(13)

Then, an experimental campaign dedicated to this model validation has been carried out: several measurement series similar to the one described in section 3 have been conducted for various values of
In practice, for each series, the valve setting and liquid flow rate were fixed as a function of the desired pressure drop. In any cases, the liquid flow rate in the loop was set between 1.8 and 3 kg.h⁻¹.

The set of experimental points obtained are then reported on Figure 10 representing $\Phi_{vap}$ versus $\Delta P_{AC}$. The two theoretical curves obtained with Eqs. (7) and (8) are also reported on this figure.

![Graph showing $\Phi_{vap}$ versus $\Delta P_{AC}$ with experimental points and theoretical curves.](image)

**Figure 10** Comparison of the theoretical transition and maximum heat flux (plain lines) with the experimental points (symbols).

On one hand, the model seems to accurately predict the transition between the boiling and evaporation mode. The transition experimental points (star symbols) are found on both sides of the transition curve. As this transition range is not considered by the model (which consider a single transition point), this tendency seems rational. Nevertheless, future work will be necessary to clarify the mechanisms at the origin of this relatively large transition zone (in particular transient phenomena).

On the other hand, the capillary limit predicted by the model seems to be significantly overestimated, in particular for the highest values of $\Delta P_{AC}$. In fact, the maximum heat flux appears weakly dependent of the liquid pressure drop. In practice, this can be explained by the fact that as the evaporation apparent heat transfer coefficient does not vary significantly with the applied heat flux or the liquid pressure drop, the wall superheating linearly increases with the heat flux applied. Hence, the approximatively constant maximum heat flux observed could correspond to the maximum superheating of the wall before the occurrence of nucleation below the distributors discussed previously (Fig.8). As this phenomenon seems to significantly disrupt the capillary pumping of liquid through the micro-grooves, this might lead to a dry-out of the capillary structure before the capillary limit is reached.

5. **Conclusions**

The design of a prototype and the corresponding experimental data obtained allowed to prove the viability of a particular concept of hybrid evaporator. The qualitative operation of this two-phase device has been highlighted, in particular the existence of two main operating modes imputed to hydraulic equilibrium of the two outlet lines. A specific model has then been developed to predict the two-phase flow repartition in the evaporator as a function of two experimental parameters (pressure drop of the evaporator assembly and vaporized heat flux). This model was able to accurately predict the frontier between evaporation and boiling in function of these two parameters. Nevertheless, several interrogations subsist:
the physical origin of the transition mode remains to be understood and included in the evaporator functioning model;

- given that the occurrence of boiling during the evaporation mode seems to be at the origin of the maximum heat flux manageable by the evaporator, the resulting dry-out mechanism needs to be clarified in order to accurately predict the dry-out heat flux.

Finally, the transition and dry-out models applied to this evaporator for the same liquid pressure drop range with methanol as a working fluid predict a maximum heat flux two to three times superior to HFE7000 whereas the transition heat flux stays at the same order of magnitude. Thus, in order to investigate the potential of this prototype and to confirm the reliability of the analytical models, further tests with methanol as a working fluid will be performed.

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