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ANALYTICAL MODELING OF CAPILLARY BIPHASIC DEVICES FOR AERONAUTICAL APPLICATIONS

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Abstract

This paper highlights the analytical modeling made to show the viability of applying capillary biphasic devices inside an aircraft powerplant, focused on using this technology as an anti-icing system for an engine intake, with cooling oil as heat source. The model was developed replicating the intended thermal environment to explore the possibility of using them in aeronautical applications with variable operating conditions. First conclusion is that applying this technology in powerplant environment is viable. Also, a comparison between device architectures is made, concluding that CPL's architecture is more appropriate than LHP's due to possibility of independent regulation.

Nomenclature

CPL	Capillary Pumped Loop
HTC	Heat Transfer Coefficient
ISA	International Standard Atmosphere
LHP	Loop Heat Pipe
VFG	Variable Frequency Generator

A_{wick}	Transversal area of wick [m^2]	L_{ev}	Evaporator length [m]
$c_{s_{liq}}$	Specific heat coefficient for liquid [J/(kg/K)]	$L_{ev_{Transversal}}$	Evaporator Transversal length [m]
$c_{s_{vap}}$	Specific heat coefficient for vapour [J/(kg/K)]]	L_{vap_c}	Vapour length at condenser [m]
D_e	Exterior diameter [m]	L_{wick}	Wick length [m]
D_i	Interior diameter [m]	$\dot{m}$	Mass flow [kg/s]
ε_{wick}	Wick permeability [m^2]	μ_{liq_c}	Dynamic viscosity [Ns/m^2]
f_i	Friction factor in "i" [-]	Δp_{cap}	Capillary pumping [Pa]
G_{cc}/G_R	Conductance of compensation chamber/reservoir [W/K]	Δp_{wick}	Pressure loss at wick [Pa]
$G_{c_{liq}}$	Conductance of liquid part in condenser [W/K]	$\dot{Q}_{in}$	Heat load [W]
$G_{c_{vap}}$	Conductance of vapour part in condenser [W/K]	$\dot{Q}_{reg}$	Regulation heat [W]
G_{ev}	Conductance of evaporator [W/K]	r	Pore Radius [m]
$G_{ll_{amb}}$	Conductance of liquid line [W/K]	Re_i	Reynolds number at "i" [-]
h_c	Convection HTC in condensation [W/(m^{2}K)]	ρ_{liq}	Liquid density [kg/m^3]
h_{cn}	Natural Convection HTC [W/(m^{2}K)]	ρ_{vap}	Vapour density [kg/m^3]
$h_{c_{sink}}$	Convection HTC in external surface of condenser [W/(m^{2}K)]	σ	Surface tension [N/m]
h_{fg_c}	Latent heat of phase change [J/kg]	θ	Bubble's surface angle in wick pores [rad]
h_{lc}	Convection HTC in liquid side of condenser [W/(m^{2}K)]	T_c	Condenser Temperature [K]
H_{gro}	Height of grooves in wick [m]	T_{c0}	Subcooling Temperature [K]

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k_{eff}	Effective conductivity in wick [W/(mK)]	T_{cc}/T_R	Comp. chamber/Reservoir Temp. [K]
k_{liq}	Conductivity of fluid (liquid phase) [W/(mK)]	$T_{cc_{in}}$	Inlet of Comp. Chamber Temperature [K]
$k_{wall_{ev}}$	Conductivity at evaporator's wall [W/(mK)]	$T_{c_{wall_{int}}}$	Condenser's Interior wall Temperature [K]
$k_{wall_{cc}}$	Conductivity at Comp. Chamber's wall [W/(mK)]	T_{ev}	Evaporator Temperature [K]
k_{wick}	Conductivity of wick [W/(mK)]	$T_{ev_{wall}}$	Evaporator's wall Temperature [K]
L_c	Condenser length [m]	W_{ev}	Evaporator's width [m]
L_{cc}	Compensation chamber length [m]	w_{wick}	Wick's width [m]

1. Introduction

Heat transfer management is a key aspect within the nacelle of an aircraft powerplant. The wide range of conditions in which aircraft operate in terms of speed, pressure and temperature, implies expensive solutions based on dedicated systems to control pressure and temperature around the equipment to allow their proper performance in which the energy is not recovered. There is an opportunity to improve the management of heat loads within the nacelle, as nowadays there are systems able to provide cooling and heating using different power sources, when it is possible to transfer the heat from hot points to cold ones taking advantage of advanced heat transfer technologies and avoiding different systems and waste of energy.

There are many elements in the aircraft that require any kind of heat addition or extraction, such as the Ice Protection System for the air intake or wings leading edge, which need to be heated, or the engine oil system, that requires some cooling. For the moment, all these applications use dedicated systems: to avoid ice accretion, the first ones use electric surface heating or pneumatic inflating boots, and to provide cooling to engine oil, heat exchangers are used. However, a proper heat transfer between the hot and the cold side would allow to extract the heat from the hot points (engine oil), and use it for the cold ones (ice-protected areas).

Biphasic technology has been widely used in space application for years, mainly because of their efficiency, the small difference of temperature they manage and their passive performance. There has been diverse studies exploring aeronautics applications of these devices [1][2], but never implemented in a medium or large size aircraft.

The powerplant is the main heat source in an aircraft, and heat management within it is not optimized. Under CleanSky 2 activities, some applications have been studied to take advantage of biphasic technology as an efficient mean to transport heat between hot and cold points, avoiding with these ad hoc components to heat or cool down those elements which may need thermal conditioning.

Air intake requires a heating system to avoid ice accretion, which is currently solved by bleeding air from the engine. This solution has a negative impact on engine performances, and current tendency is reducing bleeds, maximizing efficiency of aircraft engines, and minimizing environmental footprint.

On the other hand, oil is currently used to refrigerate relevant equipment like the Variable Frequency Generator (VFG), and also for the engine itself. This oil has to be cool down using heat exchangers, heavy parts that require an air flow to extract the heat load from it. Considering this, not only weight should be taken into account, but also the associated drag to the intake because of that air flow. Other possibilities include the use of engine oil to remove heavy heat exchangers, and use nacelle cowls to dissipate the huge heat load it will involve.

To determine if a biphasic device is applicable for an aeronautical application is important to know, as a starting point, how these systems work and model it from an analytical point of view. The model implemented should be able to include the variable conditions of a flight mission that can affect the performance of the device, and these conditions are typically: speed, temperature, height and gravity load factor (accelerations). There are other important factors that should be considered, as the operating temperature range that will affect the selection of the working fluid, the varying heat load due to changes in the engine or generator regime, or the minimum temperature to have on the intake surface to ensure no ice accretion.

One of the objectives of the study was the selection of the best architecture for the biphasic device, as a Loop Heat Pipe or a Capillary Pumped Loop, checking the response of the system to the various conditions. Considering anti-ice system for the intake and oil cooling (electrical generator or engine) as selected application, different points of the flying envelope will be used to compare these two architectures.

2. Mathematical Model

In this section, an analytical model will be developed for two different architectures of biphasic system: Loop Heat Pipe and Capillary Pumped Loop. The systems are very similar since both have an evaporator, a condenser and a compensation chamber to enable the circulation of a fluid along the loop, and taking advantage of the latent heat to

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being able of transferring higher amounts of heat power. To do so, a porous wick is integrated within the evaporator, in order to generate capillary pumping to make the vapour move from the evaporator to the condenser, where heat will be extracted and condensation will happen. The reservoir for the CPL (or compensation chamber for the LHP, depending on the architecture chosen) will keep the performance of the wick by keeping it wetted, and will help to stabilize the performance of the loop, as it will avoid that the subcooled liquid coming from the condenser arrives directly to the evaporator wick. The main difference between LHP and CPL is this element: in the case of LHP, the compensation chamber has a direct thermal link with the evaporator, which affects the performance of the system, while in CPL the reservoir is located apart from the evaporator, and usually equipped with a low power heat source that enables a regulation in operating temperature (see Figure 1). This characteristic has some benefits, as it allows the performance in optimum points, but it will affect starting and also performance stability.

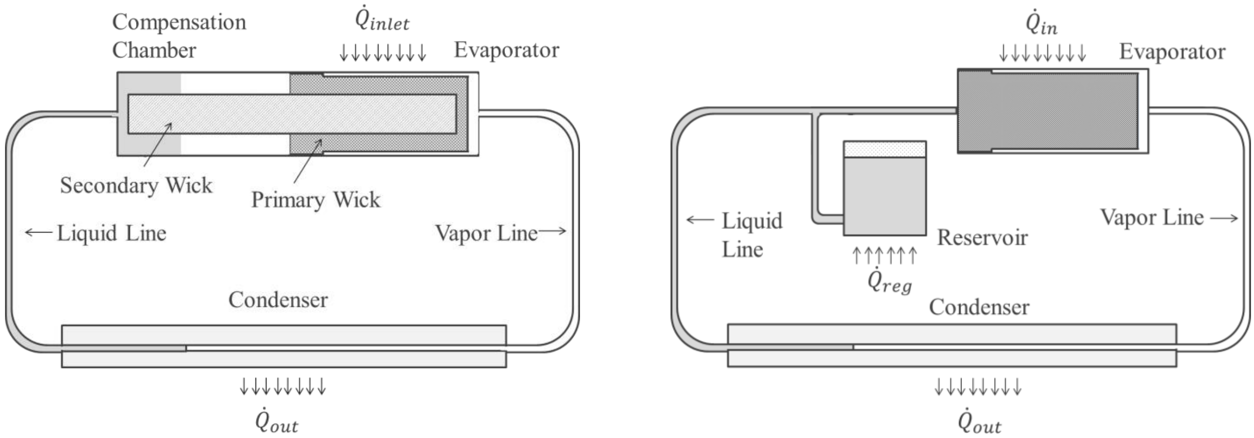

Figure 1. Schematic for a Loop Heat Pipe (LHP, left) and a Capillary Pumped Loop (CPL, right)

In order to develop an analytical model, some simplifications were made, always focusing on keeping track of the effects derived from the aeronautical environment. Therefore, the approximations may affect the accuracy of the results, but will keep the influence of those relevant variables for checking the viability of proposed aeronautical applications. Some particular characteristics of the biphasic system performance are not considered (as the wick performance, or the formation of non-condensable gas), but they should not affect the normal operation, at least from a comparative point of view. Also, it will be supposed that the devices will work on the fixed conductance mode, i.e., the subcooling length in the condenser will be limited and the operating temperature will be increased lineally according to the heat load. For lower heat loads, the operation mode is known as variable conductance, and the operating temperature diminish or keeps at a constant level until a minimum in which the length in the condenser is not enough for cooling the liquid.

The model will be valid in steady state, as transient performance is strongly related to biphasic heat transfer, something very complex to model analytically, and also highly affected by wick response, normally characterized by experimentation. It will also be considered that wick pores are totally wetted by liquid, avoiding any dry out phenomenon that may affect heat transfer effectivity in the wick.

On the other hand, the evolution of the fluid in the different sections of the loop will be studied, considering that phase changes will be at constant temperature and pressure (saturation condition). This way, in the Figure 2, real process is drawn in a continuous line, while the dotted line refers to the ideal process model in the present paper. A dash line is also set referring to the meta-stable condition in the wick, where the saturated vapour is formed from a subcooled liquid condition.

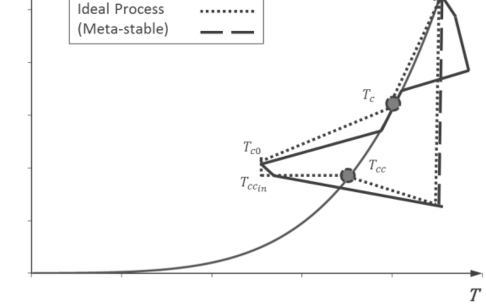

Figure 2. Evolution of the fluid in the different sections of the loop

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As inputs, the following variables will be considered apart from geometry (as general as possible, in order to allow a transferable model for different applications), materials and physical properties, and working fluid (it should be selected taking into account temperature limitations, surface tension and latent heat):

- Ambient temperature: it will depend on the flight phase and it will be model using International Standard Atmosphere (ISA), adding special conditions.
- External heat load: for the sake of simplicity, it is used as a direct input instead of characterizing the heat source.
- Cold sink Temperature: as heat is usually discharged directly to air flow, cold sink will be used as an important parameter to characterize the response of the system to variable atmospheric conditions.
- Load factor: which will include the effects of gravity for the proposed architecture (relative position between evaporator and condenser will be fixed by design as "geometry", but it can be easily changed).
- Total available length in condenser: it will affect operating temperature and subcooling.

On the other hand, the model will provide as outputs:

- Fluid temperature and pressure losses in each element.
- Heat fluxes between the different components.
- Working fluid mass flow.
- Vapour and liquid length in condenser.

The fluid will be characterized in accordance to [3], using reduced temperature for latent and specific heat, viscosity and thermal conductivity. These properties, as a function of the temperature in each point of the loop, will define Reynolds number, and with this, the fluid regime and the friction coefficients, very important to size pressure losses along the tubes [4]:

$$f_i = \begin{cases} \dfrac{0.125}{Re_i^{0.32}} + 0.0014 & Re > 200000 \\ \dfrac{0.184}{Re_i^{0.2}} & 10000 < Re \leq 200000 \\ \dfrac{0.316}{Re_i^{0.25}} & 2300 < Re \leq 10000 \\ \dfrac{64}{Re_i} & Re \leq 2300 \end{cases} \qquad (1)$$

Darcy's law for porous media will be used to size the losses along the wick, which is inversely proportional to wick permeability.

$$\Delta p_{Wick} = \frac{\dot{m}\, \mu_{wick} L_{wick}}{\varepsilon_{Wick}\, \rho_{liq}\, A_{wick}} \qquad (2)$$

Changes in the direction of the tubing will be inevitable in a nacelle, so a factor to include head losses shall be included. However, pressure losses during phase change in the condensation will not be included due to the low velocity of the flow inside the tubing.

The main contributor to pressure losses will be the gravity, in those cases in which the relative position between evaporator and condenser requires a minimum amount of capillary pumping. It should be noted that the aeronautical application that are proposed in the present paper will demand different values of acceleration, including some cases of negative "g-factor" that, even if the evaporator is placed in the lowest position, will require some amount of pumping to overcome that pressure loss, which in a normal situation with positive g will be helping to pump the fluid along the loop. In this sense, it will be interesting to minimize the vertical distance between evaporator and condenser while designing for an atmospheric application, because if different cases of acceleration are considered, they will have opposite effects and it will be necessary to size the system for the worst case anyway. The capillary pumping that should be able to overcome total pressure losses is given by Young-Laplace equation:

$$\Delta p_{cap} = \frac{2\sigma \cos\theta}{r} \qquad (3)$$

For thermal characterization, an electric analogy will be used [5], relating heat fluxes and temperatures, taking into account the architecture. One of the objectives of this paper is determining which is the best architecture or type of solution for an aeronautical application (Loop Heat Pipe or Capillary Pumped Loop), and thermal links will be relevant to characterize system performance.

In Figure 3 is displayed the thermal model using electric analogy for both types of device. On the left is displayed the schema for a LHP, and it is possible to see the thermal link between the compensation chamber and the evaporator, in this case, size with a conductance named G_{cc} to size the heat flux that goes from the evaporator's wall to the

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compensation chamber through the wall (conduction) [6]. In the case of a CPL this thermal link does not exist, as these two elements are physically apart. Nevertheless, some heat leak will be produced in any case through the wick in both cases, and it is described using G_w, which is defined for a cylindrical and a flat evaporator [7][8][9]. The other difference between the models will be the use of a $\dot{Q}_{reg}$ as a regulation heat for the reservoir In the CPL (right).

$$G_{cc} = \frac{\frac{D_{e_{cc}}^{\,2} - D_{i_{cc}}^{\,2}}{2} \pi \, k_{wall_{cc}}}{\frac{L_{cc} + L_{ev}}{2}} \tag{4}$$

$$G_w = \begin{cases} \dot{m}c_{s_{liq}} \left(\dfrac{1}{e^{\frac{\dot{m}c_{s_{liq}}}{2\pi k_{eff} L_{wick}} \ln\left(\frac{D_{e_{wick}}}{D_{i_{wick}}}\right)} - 1} \right) & (cylindrical) \\ \dot{m}c_{s_{liq}} \left(\dfrac{1}{e^{\frac{\dot{m}c_{s_{liq}} W_{wick}}{k_{eff} A_{wick}}} - 1} \right) & (flat) \end{cases} \tag{5}$$

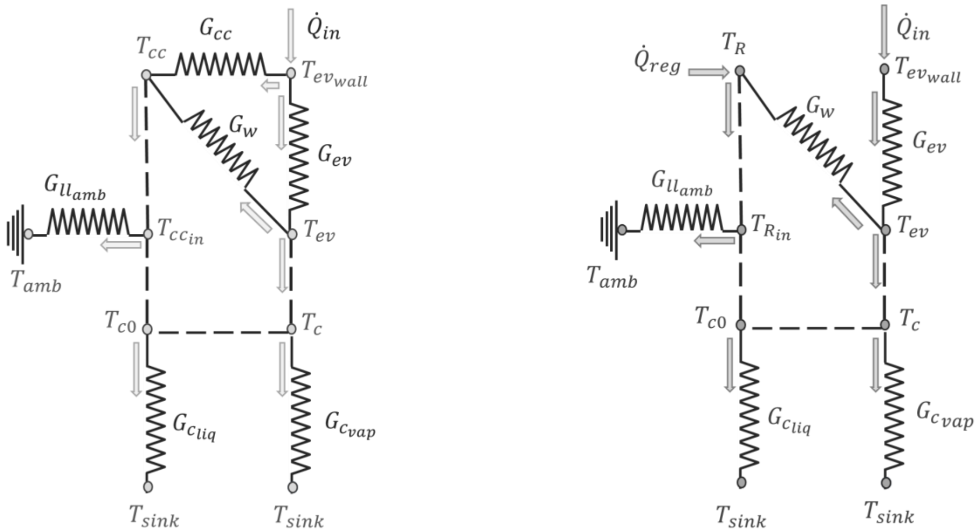

Figure 3. Thermal model for a Loop Heat Pipes (left) and a Capillary Pumped Loop (right)

The rest of the system is equivalent (just a little difference in notation to speak about the reservoir in the CPL instead of the compensation chamber for the LHP). It will be considered that the total heat load gets into the system for a single point in the evaporator ($\dot{Q}_{in}$), going through evaporator wall (G_{ev}). This last conductance is also provided for a cylindrical and a flat evaporator, to give the opportunity to implement different architectures which actually have an impact on thermal performance:

$$G_{ev} = \begin{cases} \dfrac{1}{\dfrac{\frac{1}{2\pi L_{ev} k_{wall_{ev}}}}{ln\left(\frac{D_{e_{ev}}}{D_{i_{ev}}}\right)} + \dfrac{1}{\frac{S_{contact_{ev-wall-wick}}}{\pi D_{i_{ev}} L_{ev}} \, \frac{2\pi L_{ev} k_{wick}}{ln\left(D_{i_{ev}}/(D_{i_{ev}} - H_{gro})\right)}}} & (cylindrical) \\ \dfrac{1}{\left(\frac{L_{ev} L_{ev_{Transversal}} k_{wall_{ev}}}{W_{ev}}\right)^{-1} + \left(\frac{S_{contact_{evwal-wickl}}}{L_{ev} \, L_{ev_{Transversal}}} \, \frac{L_{gro_{Longitudinal}} k_{wick}}{W_{ev} + H_{gro}}\right)^{-1}} & (flat) \end{cases} \tag{6}$$

This heat load in the system will be mostly evacuated through the condenser, but some will be also lost to the nacelle ambient through the liquid line (the loss in the vapour line was considered negligible). The system was designed to be conservative, and the heat dissipation through the cowls will be the limiting factor, so having it a bit overestimated is not a problem. To model the dissipation of heat through the condenser, two different conductances were proposed: one for the vapour phase in the first part of the condenser ($G_{c_{vap}}$), and a second one for the liquid part, after the condensation of the working fluid ($G_{c_{liq}}$) and therefore, causing the subcooling of the working fluid due to the fixed length of the condenser. It should be considered that a biphasic length will be inevitable and the heat dissipation through the condenser should be modelled taking this into account, however, this biphasic heat transfer requires a dedicated model that exceeds the scope of this study.

$$G_{c_{vap}} = \frac{\pi L_{vap_c}}{\dfrac{1}{h_c D_{i_c}} + \dfrac{1}{h_{c_{sink}} D_{e_c}} + \dfrac{\ln\left(\dfrac{D_{e_c}}{D_{i_c}}\right)}{D_{i_c} k_{wall_c}}} \tag{7}$$

$$G_{c_{liq}} = \frac{\pi\left(L_{condenser} - L_{vap_c}\right)}{\dfrac{1}{D_{i_c} h_{l_c}} + \dfrac{1}{h_{c_{sink}} D_{e_c}} + \dfrac{\ln\left(\dfrac{D_{e_c}}{D_{i_c}}\right)}{D_{i_c} k_{wall_c}}} \tag{8}$$

In these equations there are various heat transfer coefficients corresponding to different processes: convective condensation of the vapour (h_c), convective dissipation to the sink ($h_{c_{sink}}$) and conduction through condenser wall (k_{wall_c}), for the vapour side. On the other hand, for the liquid side of the condenser, there have been defined the convection within the liquid layer (h_{l_c}), the convective dissipation to the sink ($h_{c_{sink}}$) and again, the conduction through the condenser wall (k_{wall_c}). Hilpert correlation for cylinders in cross flow is used for $h_{c_{sink}}$ [10][11][12], and h_{nc} is obtained from an internal laminar flow with a Nusselt number of 4.36, assuming that the flow is completely developed. It should be noted that in the case of the liquid side of the condenser, this assumption may be not fully valid, as condensation will affect the evolution of the fluid.
Conversely, to consider laminar condensation it the next relation [10][13] was considered:

$$h_c = 0.555\left(\frac{\left(\rho_{liq} - \dfrac{\rho_{vap_c} + \rho_{vap_{ec}}}{2}\right) g \rho_{liq} k_{liq_c}^3}{\mu_{liq_c}\left(T_c - T_{c_{wall_{int}}}\right)}\left(h_{fg_c} + \frac{3}{8} c_{s\,liq}\left(T_c - T_{c_{wall_{int}}}\right)\right)\right)^{0.25} \tag{9}$$

From the point of view of heat transfer, there will be some heat losses to the nacelle ambient in the liquid line taken into account the internal and external convection layers in the tube ($h_{l_{ll}}$ and h_{cn}respectively) and the conduction through the wall of it. In this model it is considered that the tube will have a circular shape, so the conductance for the heat transfer between compensation chamber inlet and the ambient would be:

$$G_{amb_{ll}} = \frac{\pi L_{ll}}{\dfrac{1}{D_{i_{ll}} h_{l_{ll}}} + \dfrac{1}{h_{nc} D_{e_{ll}}} + \dfrac{\ln\left(\dfrac{D_{e_{ll}}}{D_{i_{ll}}}\right)}{D_{i_{ll}} k_{wall_{ll}}}} \tag{10}$$

Vapour length is the region where condensation is produced and for the LHP was obtained on the basis that the heat leak between the compensation chamber and the evaporator is compensated by the subcooling, i.e., part of the heat load directly applied to the working fluid in the evaporator is dissipated in the condensation, and the additional heat that is evacuated because of the sub-cooling is the heat that is leaked through the wick and the thermal link between the compensation chamber and the evaporator. This hypothesis allows obtaining a value for the vapour length, which is obtained using a relation between conductances [5]:

$$L_{vap} = \frac{L_c}{1 + \dfrac{G_w}{G_{ev}} \dfrac{G_c S_{c_{liq}}}{G_{c_{liq}} S_c}} \tag{11}$$

To obtain the value for the different temperatures, heat balances were set in the nodes, using Clapeyron equation to relate the temperatures in the interfaces, as it has been considered that these points will be in saturated equilibrium state. In Figure 2 is shown the saturation curve where it is possible to see three points in saturated state: in the evaporator (T_{ev}), the condenser (T_c), and the compensation chamber (T_{cc}).The curve is approximated as a straight

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line corresponding to the local derivative and using the pressure loss for each interval to relate the different temperatures:

$$T_v - T_c = \frac{\partial T}{\partial p}\,\Delta p_v \xRightarrow{\text{for condensation process}} T_{ev} - T_c = \frac{\Delta p_{vap}}{C_{slope_{condes}}} \tag{12}$$

$$\xRightarrow{\text{for vaporization process}} T_c - T_{cc} = \frac{\Delta p_{liq}}{C_{slope_{vap}}} \tag{13}$$

The wall temperature in the evaporator was obtained considering that part of the heat entered through the evaporator is driven to the compensation chamber, so to obtained this temperature the heat balance in its node:

$$\dot{Q}_{in} = G_{cc}(T_{ev_{wall}} - T_{cc}) + G_{ev}(T_{ev_{wall}} - T_{ev}) \tag{14}$$

Furthermore, the heat for the working fluid in the evaporator will be the resulting of the subtraction of the heat leak to the compensation chamber through the walls, and not all of that heat will be used to change the phase of the fluid, as part of it is leaking through the wick and the fluid connection between the evaporator and the compensation chamber. With this:

$$G_{ev}\left(T_{ev_{wall}} - T_{ev_{if}}\right) = \dot{m}h_{fg} + \dot{m}c_{s_{liq}}\left(T_{ev_{if}} - T_{cc_{if}}\right) + G_{wick}\left(T_{ev_{if}} - T_{cc_{if}}\right) \tag{15}$$

There are two other temperatures that are not directly obtained with the heat balances, which are those corresponding to inefficiencies in the loop, the subcooling temperature (T_{c0}), and the inlet temperature to the compensation chamber ($T_{cc_{in}}$). For LHPs, there is an operating temperature for each set of external conditions, and for each of these operating temperatures will be a condenser length to avoid subcooling (optimum). On the other hand, for the inlet of the compensation chamber it should be noted that some heat will be dissipated through the liquid lines:

$$T_{c0} = T_{sink} + (T_c - T_{sink})e^{-G_{c_{liq}}/\dot{m}c_{s_{liq}}} \tag{16}$$

$$T_{cc_{in}} = T_{amb} + (T_{c0} - T_{amb})e^{-G_{amb_{ll}}/\dot{m}c_{s_{liq}}} \tag{17}$$

Finally, to obtain the mass flow through the loop it is possible to use an equation linking the variables as a closure:

$$\dot{m}h_{fg} = G_{c_{vap}}(T_c - T_{sink}) \tag{18}$$

In any case, the equations are strongly coupled and the analytical resolution is easier with an iterative process or using a simple program able to solve equations sets using the appropriate input data.

The previous mathematical model is not fully applicable to CPL, as the premise is that operating temperature is controlled externally by means of a regulating heat load, and therefore, it will not only be determined by the ambient temperature and the rest of external conditions. Also, a significant difference with the LHP model is the lack of the compensation chamber conductance, in this case called the reservoir (as it is shown in Figure 3, right). Therefore, all the heat introduced into the evaporator will be directly headed to the wick and the working fluid, so only a minimum part of it will be transferred to the reservoir.

$$\dot{Q}_{in} = \dot{Q}_{ev} = \dot{m}h_{fg} + \dot{m}c_{s_{liq}}(T_{ev} - T_R) + G_{wick}(T_{ev} - T_R) \tag{19}$$

Regulation heat will fix operating temperature at the desired level (usually, where the performance of the system is maximized with the minimum regulating heat load). For an operating temperature (a T_R given), the regulation heat will be:

$$\dot{Q}_{reg} = \dot{m}c_{s_l}\left(T_R - T_{R_{in}}\right) - \dot{m}c_{s_l}(T_{ev} - T_R) - G_{wick}(T_{ev} - T_R) \tag{20}$$

With this formulation, it will not be possible to assume that the resulting subcooling will be directly linked to the heat leak in the wick, and the vapour length will be dependent on the condensation temperature (T_c), that will be close to the operating temperature (Reservoir Temperature, T_R) but affected by the pressure losses. This could lead to non-steady cases in which the condenser will not be long enough to reach the operating temperature or too long (causing subcooling) because the operating temperature has not been selected appropriately.

From the design point of view, geometry will be vital to size a system able to manage the heat loads proposed, and therefore, for selecting the application. The shape and interfaces between elements is important for heat fluxes too, but also to limit pressure losses. These effects are easily seen with the model if a parametric study is carried out, but other aspects like stability or starting process may be also affected by these parameters and they are not contemplated in the model. Hence, the results obtained with the model, whose algorithm could be seen in Figure 4, should be taken as a first iteration in the design process, giving some insight about what is viable and what is not for aeronautical applications.

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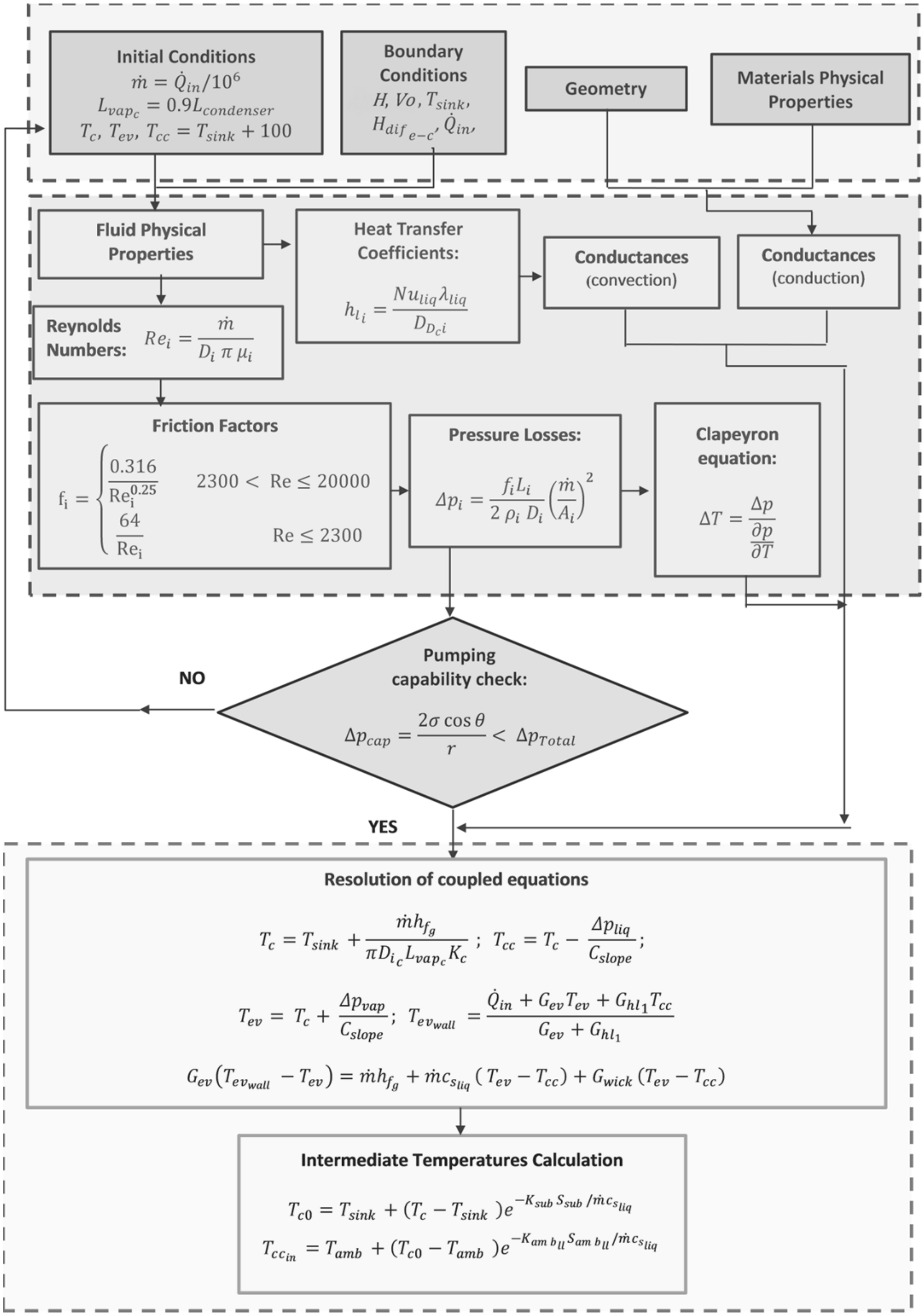

Figure 4. Model’s algorithm

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3. Results Analysis

The model created was implemented in a commercial software able to solve analytically all the equations, so it was easier to obtain fast results (because equations are coupled at different levels). Some parametric experiments were developed in order to study the effects of environmental and operational conditions. For LHP, the calculation will be as simple as shown in Figure 5: using heat load, geometry (including condenser length), operational conditions (gravity load factor, cold sink temperature, altitude, ISA) and materials and fluid properties, the model will provide temperature and pressure drop for the different components, giving information about vapour length in the condenser and the pressure margin. These two variables are useful to check if the system is working at its optimum point, as vapour length should be as close as condenser length as possible, ensuring that the whole length is being used to dissipate heat efficiently. On the other hand, pressure margin is indicating if the wick is able to overcome pressure losses, and possible limitations because of an excessive pore size. Lower pore size will lead to higher pumping capabilities in the wick, but it is more difficult to manufacture, and may have other problems related to non-steady performance. For CPL, additionally, it will be necessary to use an operating temperature as an input and it will be obtained the regulation heat load as an output, showing that even if the system is forced to work at its optimum level, it is possible that the regulation heat load is excessive or too costly to be acceptable.

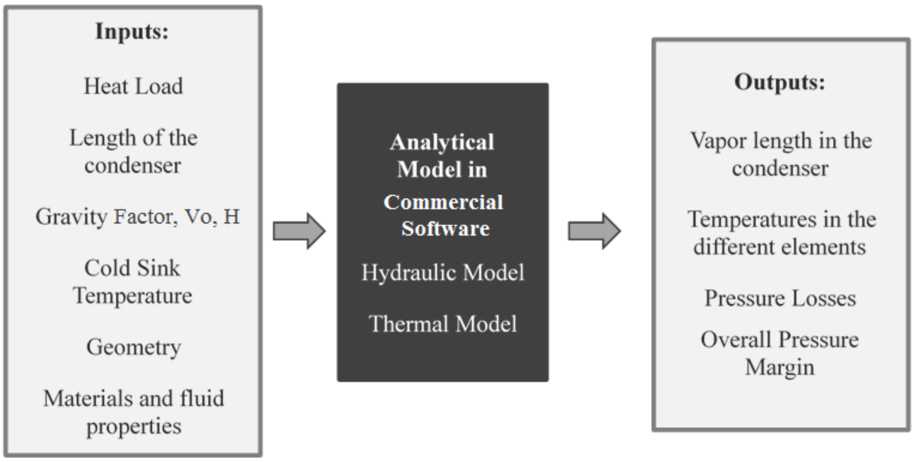

Figure 5. Calculation Algorithm for LHP

To determine the most appropriate working fluid, a figure of merit was used balancing heat transfer capacity and pumping capability (see equation (21)), concluding that, for the contemplated working fluids, methanol would be the best choice as it is able to operate in all the temperature range with the highest figure of merit in the interval (see Figure 6). Nevertheless, there are many other working fluids that may be suitable for other applications, but particularly in this case it is necessary also to consider limitations related to the restrictions to the substances that may be carried in an aircraft, and not only analyse their performance without considering the overall system in where these fluids are going to operate.

$$FM = \frac{\rho_l \sigma h_{lv}}{\mu_l} \quad (21)$$

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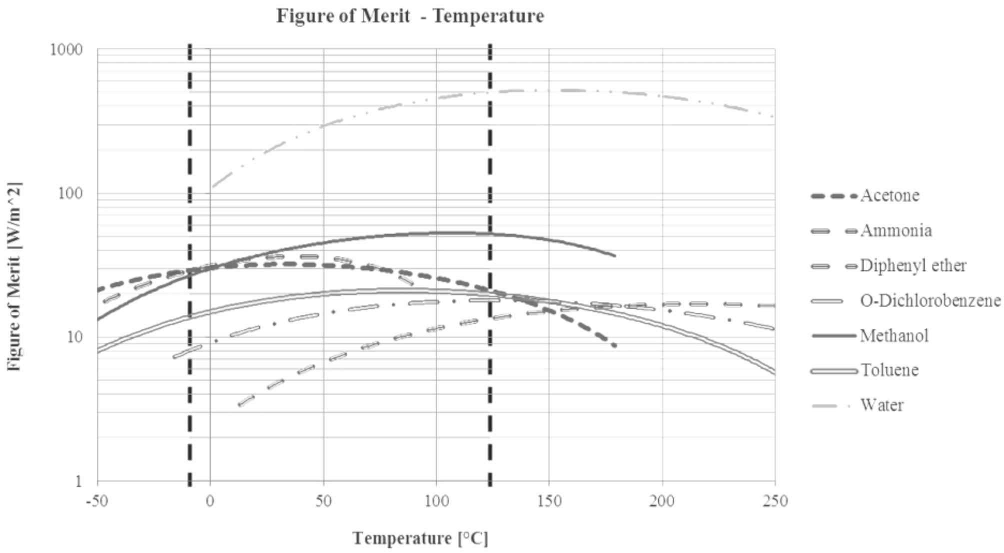

Figure 6. Figure of merit for different working fluids vs Operating Temperature

The main application studied was the use of these capillary biphasic devices for cooling down the oil used in the Variable Frequency Generator (VFG) to keep it within its temperature limits. This heat extracted from the oil will be used as an anti-ice system in the intake of the engine, so the overall heat load will be around 7000W in the worst case, with an external temperature of -40°C. This value is the maximum heat load the intake can dissipate, but it is not the maximum heat load to be dissipated by the VFG as it is estimated in 10kW. Considering this, it will be necessary to provide the system with a bypass to evacuate the exceeding heat.

To consider these capillary biphasic devices as suitable to be installed within an aircraft powerplant, it is necessary to study not only their capacity of transferring heat loads, but also their sensitivity to operational conditions (possibly the main reason why they have not been implemented before). In this theoretical study they have been limited to altitude, air mass flow, ambient temperature and gravity load factor.

The range where the system shall be able to perform has been established based on a flying envelope for a medium size transport aircraft, with an altitude starting from 0 to 10000 m, and ambient temperature corresponding to that altitude envelope considering ISA-40 to ISA+40. On the other hand, gravity load factor has been considered from 0.5 to 2g's, as the operation of the aircraft may involve manoeuvres that imply that range of accelerations. Finally, air mass flow is considering the effect of Calibrated Air Speed (CAS) that will vary with the different flight phase. This speed will have a strong impact on the heat transfer coefficient corresponding to intake's surface, and therefore will influence the heat dissipation capacity, increasing it with higher speeds, and hindering heat evacuation on ground and in hot days.

Other limitations for the system introduced in the analysis are the limit of 120 °C for the oil to keep its properties, and a lower limit of 40 °C, the minimum temperature on the surface of the intake to avoid ice accretion. This means that operating temperature shall be kept within these limitations.

When designing the best solution for the biphasic capillary device, it is important to size the different element for the whole range of operation. Big variations of heat load suggest the use of different condensers to avoid an excessive subcooling because of too long condensers, and the option of working with different operating temperatures. In Figure 7 is presented subcooling temperature versus heat load, for different condenser lengths (dashed line and dot-dashed line for longer condenser), and also for two different sink temperatures, for a selected operating temperature. It is easy to see that there is only one length for an operating temperature that avoids excessive subcooling for a wide range of heat loads, as the subcooling temperature early goes down when heat load is reduced. On the other hand, it is also visible that there is a direct relationship between condenser length (or heat evacuation area) and the maximum heat load that the system is able to evacuate for a fixed operating temperature. Additionally, the Figure 7 also includes the effect of ambient temperature. Simple lines represent a higher temperature, while double lines are describing the situation for -30 °C, where heat dissipation is very effective and, therefore, it is possible to evacuate even more than 7kW for an operating temperature of 83 °C, and proposed condenser lengths. Comparing to an operating temperature of -30°C, the differences are significant, and have a strong impact on the viability of the solution. As it was previously stated, surface temperature shall be, at least, 40°C to avoid ice accretion. With this graph it is possible to see that, for a fixed operating temperature, and using fixed evacuating areas, it is impossible to manage heat loads from 2 to 7kW to comply with the requirements of an anti-ice system. It is necessary to have a regulating system (i.e., move to a CPL type architecture or variant allowing different operating regimes), and adjust the area to dissipate the heat, using condensers in series or in parallel.

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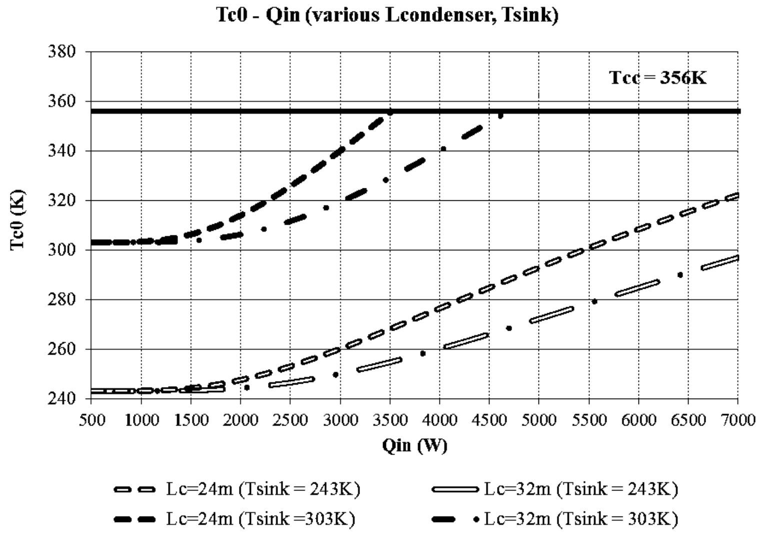

Figure 7. Subcooling temperature for different condenser lengths and external temperatures

Apart from checking the thermal performance of the system, it is important to pay attention to wick pumping capabilities too. The size of pores within the wick will have an important repercussion on the pressure margin, defined herein as the difference between the maximum capillary pumping and the actual pressure losses in the loop. For small pores, the pressure margin is wide, even if it falls for higher heat loads, but for bigger pores it ends up being negative even for the whole range of heat loads contemplated. Pumping capacity is inversely proportional to pore radius, so for higher pore sizes pressure margin will be lower. Nevertheless, it will not be a problem if pressure losses are stable for different heat loads, or not so high to overcome pumping capacity. In Figure 8 pressure margin is descending with heat load because it is for LHP architecture, where operating temperature increases for higher heat loads (with a fixed condenser length). For increasing operating temperatures, capillary pumping sinks, while pressure drops descends too, but slower.

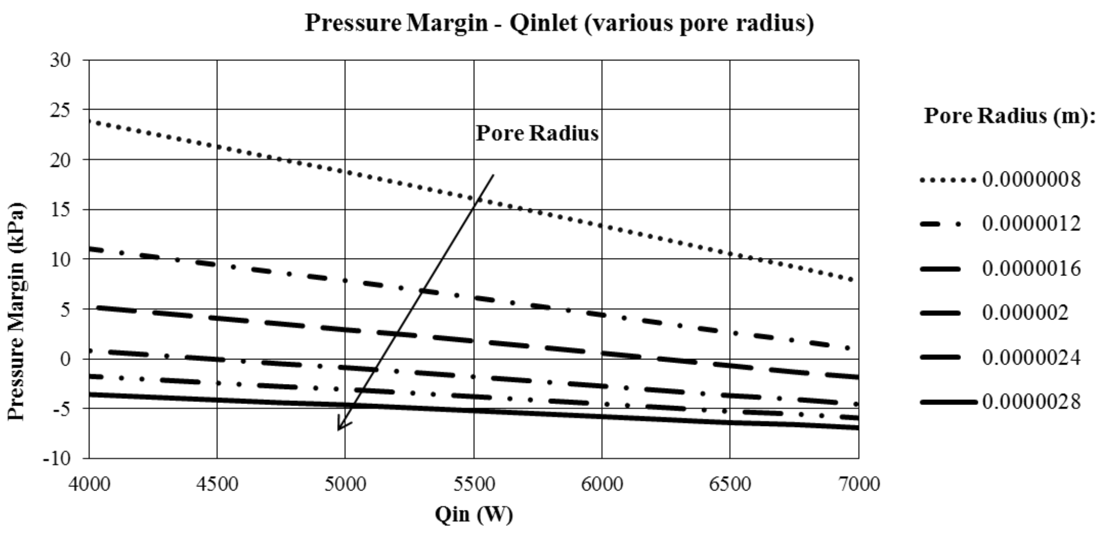

Figure 8. Pressure Margin versus Heat Load for different pore sizes in LHP

System architecture have other effects in the response to operational conditions. It was mentioned in the first paragraphs of this analysis that the effect of different flying conditions had been studied, not only to prove viability, but also to decide which of the initially considered configurations was more suitable. Analyzing thermal performance it has been easily seen that regulation will be needed for an aeronautical application, considering the wide flying envelope, but pumping capacity should also be taken into account. Pressure margin is strongly related to operating temperature, therefore, to ensure a enough pumping for CPLs it will be as easy as selecting an operating temperature that allows it for the desired range of heat load. For LHPs it will not be so obvious because operating temperature will be varying with heat load. Furthermore, external conditions will have an impact too:

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Increasing altitude is usually associated to lower external temperatures and lower densities. The first factor will allow higher heat transfer coefficients, but will force to have higher heat load extractions from the oil. On the other hand, lower densities are accompanied by lower mass flow rates, what will affect heat dissipation in the opposite way of the previous one, as it decreases with altitude (see Figure 9). Nonetheless, density related effect is not as significant as temperature's, and pressure margin will be reduced with growing altitudes.

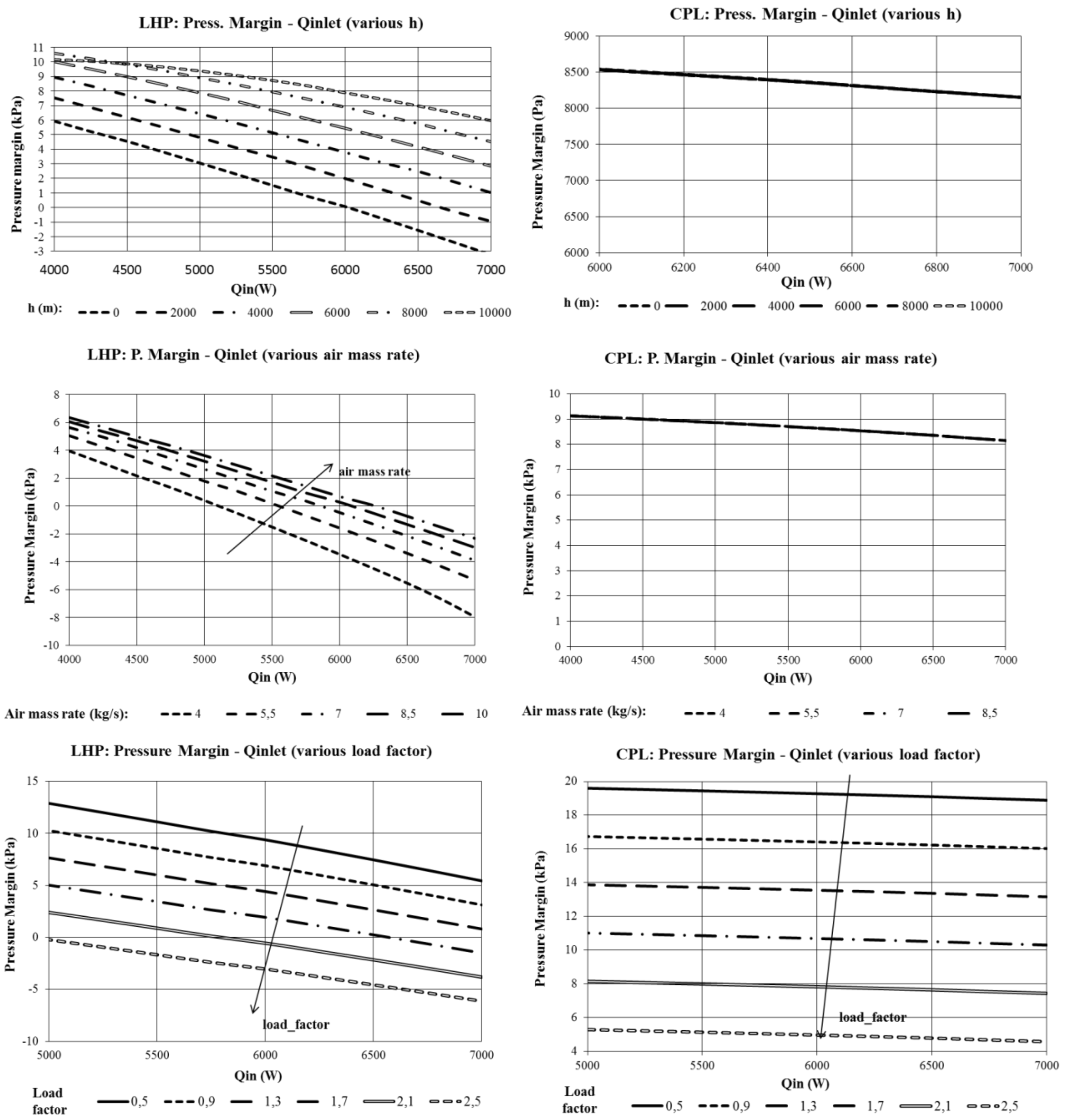

Figure 9. Pressure margin versus heat load for different altitude in LHP device (up, left) and in CPL (up, right); air mass flow in LHP (center left) and CPL (center right), and gravity load factor for LHP (bottom left) and CPL (bottom right)

The same effect is observed for air mass flow through the intake, which in this case is considering the effect of Calibrated Air Speed on pressure margin. For CPL, it will not affect capillary pumping, and only have a slight effect on pressure losses, so the lines are virtually in the same place. For LHP, conversely, speeding up the aircraft increases air mass flow, enabling an easier heat transfer in the condenser, diminishing operating temperature and increasing pressure margin. Lines will move up for higher mass flow rates, so operating range could be extended with higher mass flow rates, even if they diminish with heat load (see Figure 9).

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The last factor considered to define flying envelope is gravity load factor. This feature is especially important if evaporator is in lower position, as gravity factors greater than one will increment pressure losses in the same factor, limiting operating ranges due to insufficient pumping capacity. Therefore, this effect will not be different for various architectures (as it can be seen in Figure 9), but will be more constraining in the case of LHP as heat load have a direct effect on pressure margin and will reduce the range of operating ranges.

After all these comparisons, it seems that choosing CPL is a better option than LHP for this kind of operation. Nevertheless, it should be taken into account that this ability of the system of being immune to changing conditions is not free, and keeping operating temperature at the desired point may not be interesting from the cost point of view. Regulation heat has also been sized to see the real cost of imposing an operational temperature, as shown in Figure 10. It can be seen that regulation heat increases for growing operating temperatures, logically, as it is necessary more energy to heat the fluid over the minimum temperature, and it is particularly onerous for lower heat loads as it can represent almost 50% of the heat load (anyhow, it should be noted that all these cases have been studied for a fixed condenser length, which may not be practical at all). For a particular operating temperature, regulation heat increases for growing heat load until reaching a maximum value where it goes down again. This maximum is displaced to greater heat loads for higher operating temperatures. Regulation heat is necessary to overcome the temperature drop due to subcooling, and therefore, it is directly proportional to the evolution of subcooling temperature for different operating temperatures: for growing operating temperatures the step will be greater and it will be needed more heat to control the performance of the device. On the other and, for a given operating temperature, temperature step (i.e., the difference between condensation temperature and subcooling temperature) will be diminishing until reaching a certain heat load, where the decrease in temperature step is speeded up because condensation length grows fast, and therefore, heat extraction, until the optimum point where subcooling is minimum and also regulation heat. However, this regulation heat is also proportional to fluid mass flow, which is lineally growing with heat load according to the model, and in combination with the subcooling temperature evolution give rise to the mentioned maximum.

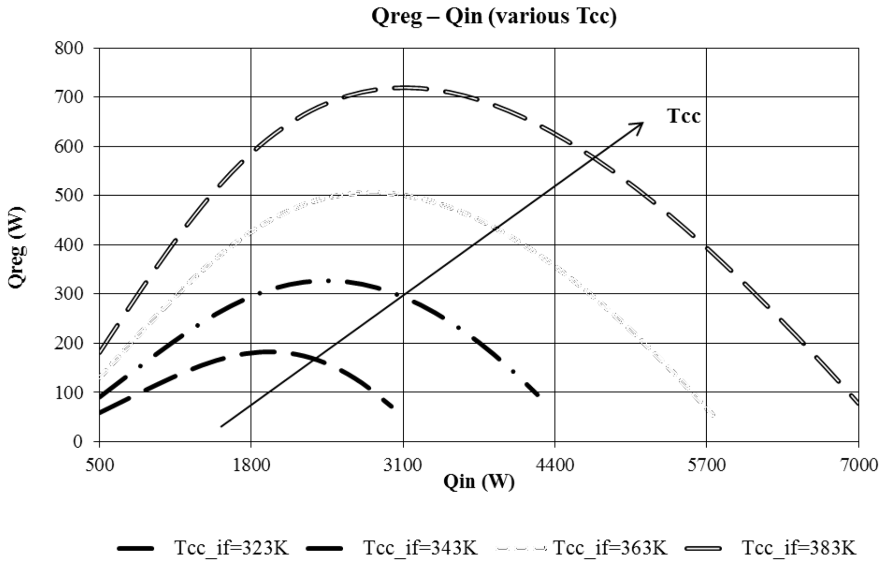

Figure 10. Regulation heat versus Heat Load, for different operating temperatures

Finally, as the objective of this study was to prove the viability of using biphasic technology, it has not been detailed the matching of the heat loads between the source and the sink. Once in the real platform, it will be necessary to install some bypass systems to evacuate to nacelle cowls in case that heat load coming from the VFG overcomes the intake dissipation capability, because the external temperature is too high or the surface temperature is close to its limit. On the other hand, there may be cases where the heat coming from the VFG is not enough to avoid ice accretion on the intake's surface, so alternate sources shall be considered. However, this represents an opportunity to study alternate architectures using different sources and sinks, using more evaporators and condensers placed in series or in parallel to maximize performance and match the needed conditions, once the preliminary viability of using this technology has been checked.

5. Conclusions

Current trends in aircraft powerplant heat transfer management lead to solutions based on dedicated systems making it poor in terms of efficiency. This gives an opportunity to implement biphasic technology to transfer heat from a cooling application to other part of the powerplant where that heat may be used for something else. In this paper it

has been proposed the use cooling oil of the VFG to obtain the necessary heat to provide engine's intake of an anti-ice system to avoid ice accretion. Also, to obtain higher heat loads, it could be used engine oil, removing also the heat exchanger needed to cool it down, and use instead nacelle panels to dissipate the heat.
Biphasic technology is promising for this type of applications, as they have been widely used and highly developed for years in space industry. Current researches are focused on extending their use to other industries, as the present article intended. However, it is still necessary further advances in wick manufacturing and materials compatibility for higher ranges of temperature to enable the use of hotter heat sources and greater heat loads.
With this study, it has been concluded that it is possible to use biphasic capillary devices to transfer a heat load of around 7kW maximum, even if it is necessary to further look into the most suitable architecture regarding the number of elements and their relative positions, which would increase the heat load to the necessary level. Along this paper it has been described some problems related to the wide range of heat loads intended to manage, and what is more, having the system working in different operative conditions for its installation in an aircraft.
The model proposed describe performance of biphasic devices from two perspectives: thermal performance and hydraulic limits, and allows exploring viability in a simple way, providing some information about the limitations. It is subjected to some geometrical information, which has been simplified to be applicable to more design cases. With it, is possible to select applications from the point of view of performance, giving an idea of the modifications that will be needed (more evaporators, a greater dissipation area, lower pore size). Nevertheless, it is important to note that this model is not considering transient phenomena that have an important impact on these devices' performance.
After the comparison explained in the previous section, CPL devices are more appropriate for the aeronautical application due to their capacity of regulating temperature and thus, control their response to varying conditions of different flying conditions. However they have more problems in starting process than LHP, and may have some problems of drying out in the wick, something that may be unacceptable from a certification point of view, considering the safety requirements imposed to aeronautical systems. There are other possible architectures that include more robust wicks with an integrated with the compensation chamber/reservoir that could overcome those transient regime problems, and be more competitive than a CPL system, if those other systems include bypass and regulation systems that enable their use in the whole flying envelope.
As expected, the main problem of these capillary biphasic devices is their unknown response to ambient temperatures when installed in an aircraft, something that this model intended to cover. Considering the numerous simplifications made, trends and orders of magnitude should be considered as fully valid when using it, in any case, it has been observed that:

- High sink temperatures make difficult to manage great heat loads, like those present in the case of a take-off in a hot day, where the anti-ice system would not be needed, but cooling down the oil would be vital. To manage high heat loads will be necessary to increase operating temperatures, investing power in regulating the device, and use great dissipation areas, that may lead to wider subcoolings in other conditions (something far from the optimum condition).
- Altitude will have a less significant effect than temperature, but will have a positive impact at high altitudes from the point of view of dissipating heat. On the other hand, it will demand more heat from the oil, which may not be available. In CPL system will not have impact in hydraulic performance, although it will make easier for LHP systems to perform in higher altitudes due to the decrease in operating temperature.
- Flight speed, and with it, intake air mass flow, will have an analogous impact as the previous variable, but it should be pointed out that it will be necessary to have some air mass flow to allow enough heat dissipation (natural convection is significantly less effective).
- Gravity load factor will be more important if the vertical position of evaporator and condenser are very different, and also if the evaporator is in lower position as normal attitude of the aircraft will force to have a positive capillary pumping and higher than pressure losses. Considering this, evaporator and condenser shall be located horizontally whenever possible to limit the effects of gravity.

The next steps to furtherly study the use of biphasic technology in aeronautical applications should be experimentation to check the device's performance with transient effects, but also to study wick response. Also, other materials could be proposed for the manufacturing of the system (it has been proposed aluminium and steel), taking into account compatibility with working fluids. The model has also being used to compare the performance with Orto-dichlorobenzene and toluene, but, as predicted by the figure of merit predicted, their hydraulic performance was poorer.

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