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# Transient thermohydraulic modeling of two-phase fluid systems

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**Abstract.** This paper presents a transient thermohydraulic modeling, initially developed for a capillary pumped loop in gravitational applications, but also possibly suitable for all kinds of two-phase fluid systems. Using finite volumes method, it is based on Navier-Stokes equations for transcribing fluid mechanical aspects. The main feature of this 1D-model is based on a network representation by analogy with electrical.

This paper also proposes a parametric study of a counterflow condenser following the sensitivity to inlet mass flow rate and cold source temperature. The comparison between modeling results and experimental data highlights a good numerical evaluation of temperatures. Furthermore, the model is able to represent a pretty good dynamic evolution of hydraulic variables.

## 1. Introduction

Because of increasing power electronics integration in terrestrial transportation (railway, automotive, aeronautic, ...), traditional cooling systems reach their maximal performances. Thanks to their heat transport capacity, two-phase fluid systems appear among identified alternative solutions. Among them are capillary two-phase fluid loops, appeared in the late 60's in space industry. Two technologies have been developed in parallel: the Capillary Pumped Loop (CPL) in the United States and the Loop Heat Pipe (LHP) in the USSR.

Before being used in ground transportation, some investigations are needed to estimate their performances and operation under gravity. In order to combine benefits of each technologies of loops (CPL and LHP), a hybrid loop has been developed taking advantages of gravity: the *Capillary Pumped Loop for Integrated Power* (CPLIP) [2, 3].

Alongside experimental studies [2–5], several modeling works have been conducted. If lots of steady state modeling are available in the literature [6, 7], only few transient models exist [8–10]. However, transient phases often induce major thermal constraints of electronic components and capillary pumped loops. Transient models are therefore essential.

This paper presents a transient thermohydraulic modeling system approach, initially developed for the CPLIP by Delalandre *et al.* [1], but also possibly suitable for all kinds of

two-phase fluid systems. After a description of the model, focus will be made on a classical counterflow condenser, which is an essential part of two-phase fluid systems. A sensitivity study will be exposed on this system to various parameters: inlet mass flow rate and cold source temperature.

## 2. Modeling description

In a system approach, the model exposed in this paper uses global variables to be able to predict quickly the general operation of the studied system. Contrary to Pouzet *et al.* [9] and Launay *et al.* models [10], both using submodels, it is based on a system's discretization by means of finite volumes method. Furthermore, like Kaya *et al.* [8], it uses mass enthalpy as energy variable rather than temperature for the phase change management. Mass flow rate and pressure are respectively mass balance variable and momentum balance variable.

### 2.1. Hypothesis

This modeling approach is based on the following hypothesis:

- Vapor is considered as an ideal gas;
- Two-phase flows are considered as homogeneous;
- The fluid is considered compressible whatever its physical state;
- Radial thermal resistance in metallic lines is neglected, as confirmed by local Biot number;
- Heat exchanges with the environment are taken into account.

### 2.2. Mass enthalpy approach and phase change management

Most commonly modeling approaches are based on temperature form of energy conservation equation. It is the case for the thermal management of solid parts. However, for a two-phase flow, temperature cannot define alone physical state because of phase change. Indeed, at fixed pressure, discontinuity of mass enthalpy  $h$  involves discontinuity of thermodynamic properties with temperature at saturation. Using rather mass enthalpy, continuity of these properties, as well as temperature, are contrary ensured.

Assuming homogeneous fluid hypothesis, thermodynamic properties are estimated as functions of vapor quality  $x$ , as exposed by Delalandre *et al.* [1]. This latter is evaluated by means of pressure and mass enthalpy  $h$ , as follows:

- If  $h < h_{l,sat}(P)$  then  $x = 0$ ;
- If  $h_{l,sat}(P) \leq h \leq h_{v,sat}(P)$  then  $x = \frac{h - h_{l,sat}(P)}{h_{v,sat}(P) - h_{l,sat}(P)}$ ;
- If  $h_{v,sat} < h$  then  $x = 1$ .

### 2.3. Discretization and network formalism

To solve balance equations, a finite volume method is used. The system is discretized using two staggered meshes, as shown on figure 1, for numerical convergence [11]: the red one (continuous lines, notation nodes "i") for mass and energy balances and the blue one (dotted lines, notation nodes "k") for momentum balance.

Written under dimensionless form for convergence, equations are transcribed under network formalism with electrical analogy for a better visualization and to use a specific network solver: ESACAP ©. This latter can effectively easily treat singularities problems and has proven its ability in others contexts by using GEAR algorithm and variable time step order. It gives so a very good compromise between precision (model size) and calculation time. Model resolution is done via four networks: three for the fluid (mass flow rate, pressure and mass enthalpy)

and one for solid parts (temperature). The latter is a conventional thermal network based on transcription of conductive and convective heat transfers and has fluid nodes temperatures as boundary conditions. As an example, the thermal network of a common line is given on figure 3d. These four networks are strongly coupled, as shown on figure 2.

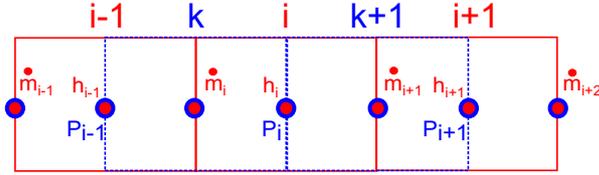


Figure 1: Meshes and variables locations

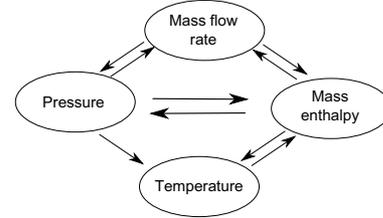


Figure 2: Networks coupling

#### 2.4. Balance equations and associated networks

2.4.1. *Mass balance:* The mass balance of the control volume “i” (cf. figure 1) is written:

$$\Omega_i \frac{d\rho_i}{dt} + \dot{m}_{ref}(\dot{m}_{i+1}^+ - \dot{m}_i^+) = 0 \quad (1)$$

where  $\Omega_i$ ,  $\rho_i$ ,  $\dot{m}_{ref}$ ,  $\dot{m}_i^+$  and  $\dot{m}_{i+1}^+$  are, respectively, volume, density and reference and dimensionless inlet and outlet mass flow rates of the volume “i”.

Under network formalism, this equation becomes:

$$Jdb_i + Gdb_i(\dot{m}_{i+1}^+ - \dot{m}_i^+) = 0 \quad (2)$$

where  $Jdb_i = \Omega_i \frac{d\rho_i}{dt}$  and  $Gdb_i = \dot{m}_{ref}$  are, respectively, the flux source and the conductance between the two nodes  $\dot{m}_i^+$  et  $\dot{m}_{i+1}^+$ .

The associated network is shown on figure 3a.

2.4.2. *Momentum balance:* According to [1], the momentum balance of the control volume “k+1”, between the nodes “i” and “i+1” (cf. figure 1) gives:

$$\begin{aligned} \frac{L_i + L_{i+1}}{2} \dot{m}_{ref} \frac{d\dot{m}_{i+1}^+}{dt} + \dot{m}_{ref}^2 \dot{m}_{i+1}^+ \frac{\dot{m}_{k+1}^+}{\rho_{k+1} S_{i+1}} - \dot{m}_{ref}^2 \dot{m}_i^+ \frac{\dot{m}_k^+}{\rho_k S_i} + P_{ref} S_{eq,i/i+1} (P_{i+1}^+ - P_i^+) \\ \pm \rho_{k+1} \frac{H_i S_i + H_{i+1} S_{i+1}}{2} g + |F_{visc,k+1}| = 0 \end{aligned} \quad (3)$$

with  $\rho_k = \frac{\rho_{i-1} \Omega_{i-1} + \rho_i \Omega_i}{\Omega_{i-1} + \Omega_i}$  and  $\dot{m}_k^+ = \frac{1}{2}(\dot{m}_i^+ + \dot{m}_{i+1}^+)$ .

$L_i$ ,  $L_{i+1}$ ,  $H_i$ ,  $H_{i+1}$  and  $S_{eq,i/i+1}$  are, respectively, the lengths and elevations of volumes “i” and “i+1”, and equivalent section of volume “k+1” as exposed in [1].  $P_{ref}$ ,  $P_i^+$ ,  $P_{i+1}^+$  are, respectively, reference, dimensionless inlet and outlet pressures of volume “k+1”;  $g$  is the gravity acceleration.  $F_{visc,k+1}$  represents viscous forces in volumes “k+1”.

This equation can be written much more simply under network formalism:

$$Jp_i + Gp_i(P_{i+1}^+ - P_i^+) = 0 \quad (4)$$

where  $Gp_i = P_{ref} S_{eq,i/i+1}$  and  $Jp_i$  (all other terms of equation (3)) are, respectively, the conductance and flux sources between the two nodes  $P_i^+$  and  $P_{i+1}^+$ .

Figure 3b shows the corresponding network.

2.4.3. *Energy balance:* The energy balance of the control volume “i” (cf. figure 1), based on mass enthalpy and using an upstream scheme, can be written as:

$$\rho_i \Omega_i h_{ref} \frac{dh_i^+}{dt} + \dot{m}_{ref} h_{ref} \dot{m}_i^+ (h_i^+ - h_{i-1}^+) - \dot{Q}_i - \dot{W}_{comp,i} = 0 \quad (5)$$

where  $h_{ref}$ ,  $h_{i-1}^+$  and  $h_i^+$  are, respectively, reference and dimensionless inlet and outlet mass enthalpies of the volume “i”.  $\dot{Q}_i$  is the heat flux received by this volume (calculated thanks to the thermal network) and  $\dot{W}_{comp,i}$  is the compression power (depending on pressures and mass flow rates). Under network formalism, this equation becomes:

$$DN_i \frac{dh_i^+}{dt} + GF_i (h_i^+ - h_{i-1}^+) - QS_i = 0 \quad (6)$$

where  $DN_i = \rho_i \Omega_i h_{ref}$  and  $QS_i = \dot{Q}_i + \dot{W}_{comp,i}$  are, respectively, the enthalpy capacitor and heat source applied to the node  $h_i^+$ .  $GF_i = \dot{m}_{ref} h_{ref} \dot{m}_i^+$  is the fluidic conductance (oriented according to the upstream scheme, figure 3c) between the nodes  $h_{i-1}^+$  and  $h_i^+$ .

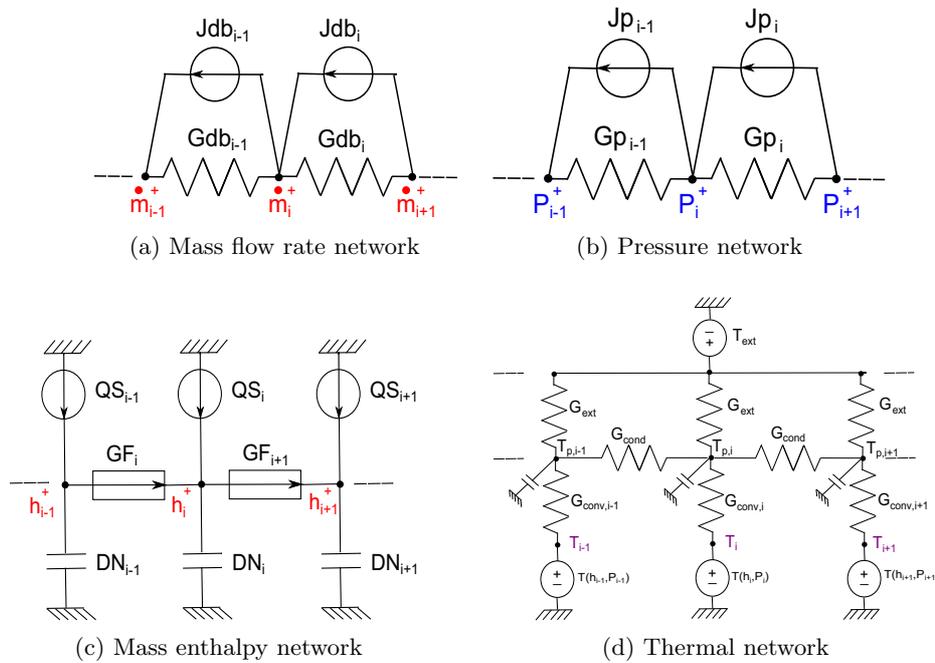


Figure 3: Schematic description of the four networks

### 3. Analysis and discussion

Previous works of Delalandre *et al.* [1], with this model, allowed to validate numerical results with experimental data for the CPLIP. These studies highlight good numerical results of temperature prediction (cf. figure 4) and the prediction of “overshoots” (with a slight overestimation) of hydraulic variables (cf. figure 5).

However, the model could still be improved by a more accurate management of the thermohydraulic dynamics of specific parts of the loop (condenser, tank, evaporator) [1]. In the context of this improvement, element by element, this paper aims at studying the only condenser followed by the liquid line. The validation of each part will be made in further work on global model of the CPLIP, knowing that experimental results of the only entire loop are available (the boundary conditions of each element being dependent on the others).

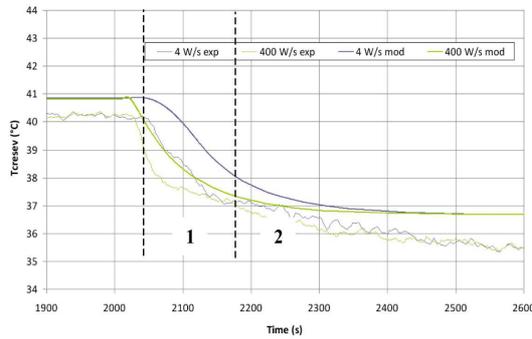


Figure 4: Evolution of temperature between tank and evaporator of the CPLIP during heat flux increases [1]

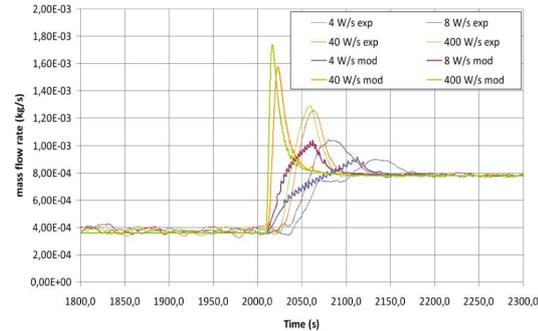


Figure 5: Outlet condenser mass flow rate responses to heat flux increases [1]

### 3.1. Presentation of the studied system

The condenser is a 12 mm inner diameter and 5 m long counterflow heat exchanger. In the external annular channel (14-18 mm), a water/ethylen glycol mixture flow cools the working fluid. The liquid line is 2 m long with 6 mm inner diameter [3].

The mass flow rate of cooling fluid  $\dot{m}_c$  is imposed constant and equal to  $0.5 \text{ kg.s}^{-1}$ , assuming the mixture incompressible. Its inlet temperature  $T_{c,in}$  is also imposed. For the working fluid, mass flow rate and mass enthalpy at the condenser inlet and pressure at the liquid line outlet are imposed as boundary conditions, as shown on figure 6a. Assuming the fluid is saturated vapor at entrance, the imposed inlet mass enthalpy corresponds to  $h_{in} = h_{v,sat}(P_{c1})$ ,  $P_{c1}$  being the pressure of the first node. This boundary condition is transcribed in the network by an upstream fictive node, characterized by  $P_I = P_{c1}$  and  $h_I = 2h_{v,sat}(P_{c1}) - h_{c1}$  (cf. figure 6b). Boundary conditions of mass flow rate and pressure are applied directly to, respectively, the entrance of the first node and the last node:  $\dot{m}_{c1} = \dot{m}_{in}$  and  $P_{L,NL} = P_{out}$ . Finally, heat exchanges with the environment are taken into account: an outside temperature  $T_{ext}$  and a heat transfer coefficient  $h_{ext}$  are imposed constant and respectively equal to  $23^\circ\text{C}$  and to  $15 \text{ W.m}^{-2}.\text{K}^{-1}$ .

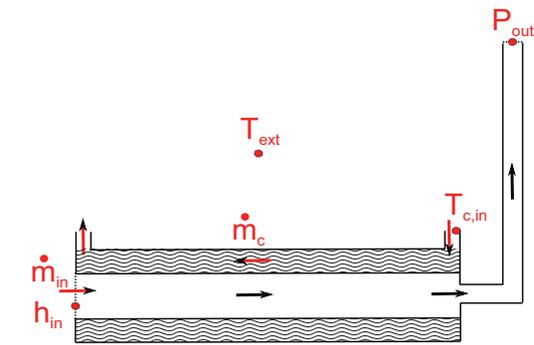
Discretization of the system and associated networks for the working fluid are respectively given on figures 6b and 7. Simulations are made with 600 nodes in condenser and 50 nodes in liquid line for the working fluid.

### 3.2. Sensivity analysis

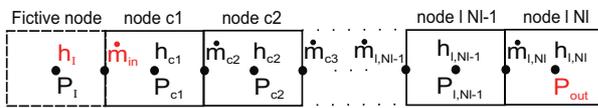
The study aims to analyse the sensitivity of the thermohydraulic operation of the system to some boundary conditions, during transient phases: inlet mass flow rate and cold source temperature.

**3.2.1. Sensitivity to inlet mass flow rate:** An increasing step of inlet mass flow rate is imposed. For the present study, four sollicitations are applied between steady-states corresponding to  $4.10^{-4} \text{ kg.s}^{-1}$  and  $8.10^{-4} \text{ kg.s}^{-1}$  inlet mass flow rates, as shown on figure 8. The difference between these sollicitations lies in the rate of this mass flow rate, imposed from  $0.2 \text{ kg.s}^{-2}$  to  $4 \text{ kg.s}^{-2}$ . Outlet pressure and inlet cold source temperature are imposed respectively to  $84300 \text{ Pa}$  and  $20^\circ\text{C}$ .

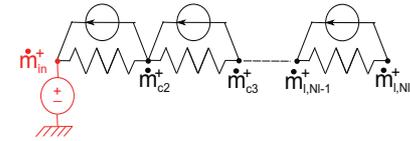
Figure 9 shows outlet condenser mass flow rate responses for these four sollicitations. Overshoots of this mass flow rate are highlighted during strong transient phases. Even if these overshoots are overestimated, especially for the higher increases, according to [1], these responses reveal the ability of the model to simulate the complex dynamics of the system (despite numerical oscillations resulting from discretization limitations).



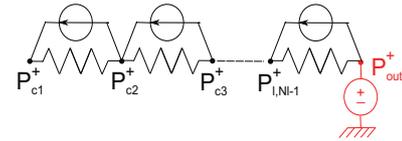
(a) Representation and boundary conditions of the studied system



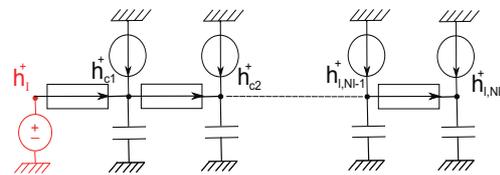
(b) Discretization of the system



(a) Mass flow rate network of the system



(b) Pressure network of the system



(c) Mass enthalpy network of the system

Figure 6: (a) Studied system and (b) discretization

Figure 7: Associated networks for the working fluid

Indeed, overshoots result of the competitiveness between condensation phenomena in the condenser and gravity forces in liquid line. The increase of the inlet mass flow rate implies initially an expansion of the two-phase area in condenser, which pushes liquid out of the condenser. Then an outlet mass flow rate damping happens thanks to gravity.

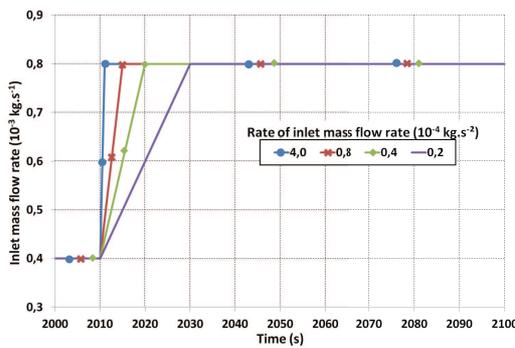


Figure 8: Inlet mass flow rate step solicitations

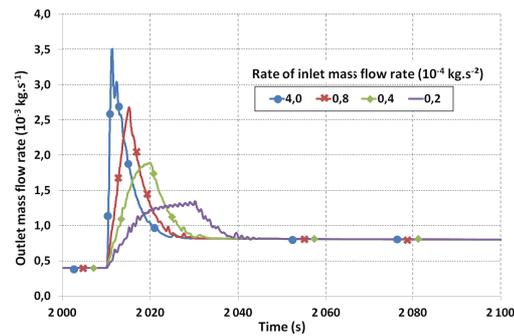


Figure 9: Outlet mass flow rate responses to inlet mass flow rate increases

To illustrate this dynamics, the evolution of the two-phase area length in condenser is shown in figure 10. We can observe that the faster the inlet mass flow rate increase is, the faster the new two-phase area length is reached. The multiple steps which can be seen on the rising part of the curves are directly linked to the spatial discretization of the condenser.

Finally, the evolution of fluid temperatures in condenser can be observed in figure 11, during a transient phase corresponding to a rate of inlet mass flow rate of  $0.4 \text{ kg.s}^{-2}$ . This one also highlights the increase of the two-phase area with inlet mass flow rate, corresponding to a constant saturation temperature of  $75^\circ\text{C}$ , since the beginning of the condenser. Furthermore,

we can note that final subcooling of working fluid to cold source temperature of  $20^{\circ}\text{C}$  (due to oversizing of the condenser) takes place further in the subcooling area, always as a result of the increase of two-phase area length.

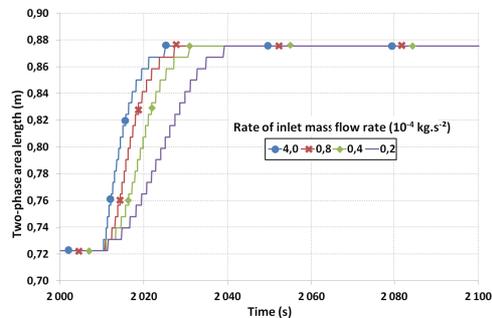


Figure 10: Two-phase area length calculation in condenser

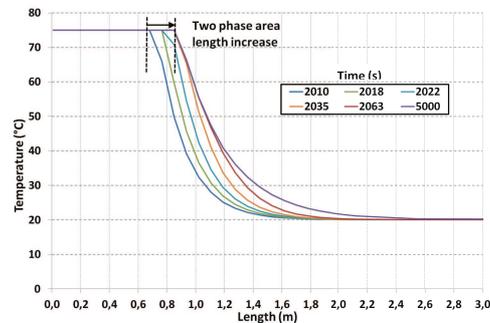


Figure 11: Iso-times temperatures in condenser during an inlet mass flow rate solicitation ( $0.4 \text{ kg.s}^{-2}$ )

*3.2.2. Sensitivity to cold source temperature:* An increasing step of inlet cold source temperature is imposed from  $20^{\circ}\text{C}$  to  $25^{\circ}\text{C}$  with rates from  $0.25^{\circ}\text{C.s}^{-1}$  to  $5^{\circ}\text{C.s}^{-1}$ . Inlet mass flow rate and outlet pressure are imposed respectively to  $0.4 \text{ kg.s}^{-1}$  and  $84300 \text{ Pa}$  for this sensitivity analysis.

Figure 12 shows outlet mass flow rate responses to the four inlet cold source temperature increases. Overshoots of this mass flow rate are also observed during these solicitations. Here again the amplitude of the overshoot increases with the rate of inlet cold source temperature increase. Thus inlet mass flow rate and inlet cold source temperature increases have the same consequences on the hydraulical dynamics of the condenser. Like the previous study, an increase of two-phase area length is also highlighted. Indeed, the thermal flux exchanged between working fluid and cold source fluid remaining the same in steady state, this length must grow to compensate the decrease of temperature difference between the two fluids. Numerical oscillations lead to think that spatial discretization is not sufficient. However, it should be noted that the number of nodes is a choice of a system approach for modeling a complete system. Remember that 650 nodes are used for the working fluid, what is already considerable for such approach.

Evolution of fluid temperatures in condenser during a transient phase corresponding to a rate of cold source temperature increase of  $0.5^{\circ}\text{C.s}^{-1}$  is shown in figure 13. Furthermore, the final subcooling temperature of working fluid, corresponding to cold source temperature thanks to the oversized condenser, is reached at approximately the same location during the transient phase. Thus, the increase of the two-phase area length is compensated by the decrease of the difference between saturated working fluid and cold source temperatures, due to the increase of cold source temperature.

#### 4. Conclusion

The transient thermohydraulic modeling of two-phase fluid systems, presented in this paper, has been developed in a system approach. This latter involves the use of stronger hypotheses than for an accurate approach of phenomena. Thus, for example, the phase change is managed assuming a homogeneous fluid. Based on the finite volumes method, this model further has the particularity to use a network representation of mass, momentum and energy conservation equations. The writing of these equations allows the creation of four strongly coupled networks having as variables: temperature for solid parts and mass flow rate, pressure and mass enthalpy for the fluid. This latter was preferred to the temperature for a better phase change management.

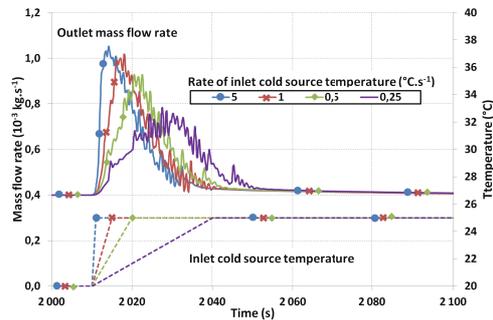


Figure 12: Outlet mass flow rate responses to inlet cold source temperature increases

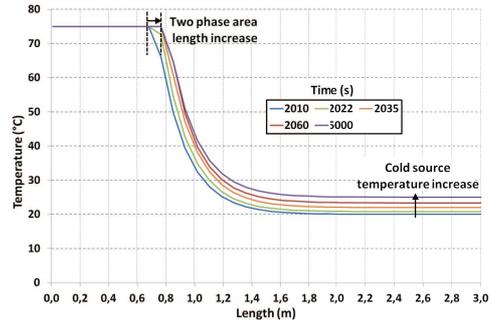


Figure 13: Iso-times temperatures in condenser during an inlet cold source temperature increase ( $0.5^{\circ}\text{C}\cdot\text{s}^{-1}$ )

Despite the disadvantages of the system approach, this model has been validated by works of Delalandre *et al.* [1] on a capillary pumped loop. Thus, it was highlighted a good prediction of temperatures and a pretty good hydraulic dynamics in transient phases. In the context of improving the loop model, element by element, a sensitivity analysis has been presented on a classical component of two-phase fluid systems: a counterflow condenser. This analysis was able to highlight the strong influence of severe increases of inlet mass flow rate and cold source temperature on the hydraulic dynamics, during the transient phases. In particular, high “overshoots” of the outlet mass flow rate have been shown. The competitiveness between increase of two-phase area length in condenser and the gravity forces in the following liquid line has been emphasized.

Future work will be to improve the transient thermohydraulic modeling of Delalandre *et al.* [1] of the capillary pumped loop. Indeed, a better modeling of the tank and the evaporator seems necessary to transcribe more precisely the thermohydraulic behavior of these components, the purpose being always to stay in a system approach.

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