# Dynamic modelling of a Condenser/Water Heater with the Thermo-SysPro Library

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## Abstract

A new dynamic model of a water heater has been developed. The component model is meant to be used for power plant modeling and simulation with the ThermoSysPro library developed by EDF and released under open source license.
The model and the test conditions are fully described: modeling hypothesis, governing equations, parameter values and test transients.
To validate the model, three difficult transients were simulated: the islanding (sudden plant disconnection from the grid), flow reversal and zero-flow conditions inside the water heater.
Regarding the islanding scenario, the simulation results are very close to the experimental values measured on site. This transient demonstrates the physical validity of the model at it is fast and challenges the model equations in all operating conditions of the exchanger.

***Keywords:*** *Modelica; thermal-hydraulics ; heat exchanger ; water heater ; dynamic modeling; inverse problems*

## 1. Introduction

In the framework of the EUROSYSLIB project, a new library called ThermoSysPro has been developed.
The main objective of ThermoSysPro is to provide a generic library for the modeling and simulation of power plants and other kinds of energy systems. The meaning of the word 'generic' is here to be taken as the possibility to use the same library components to model different kinds of energy systems for different types of studies (sizing, control system verification, etc.).
The library is now routinely used for different purposes, see for instance [1 to 6]. An introduction to the library can be found in [5].
New developments are ongoing or planned to extend the scope of the library for uncertainties and state estimation.
The objective of this paper is to show how the library can be extended to include a new component to model a shell-and-tube heat exchanger, by using already existing components of ThermoSysPro.

## 2. Model of the condenser/water heater

### *2.1.* General presentation of the water heater

The water heater is a **two-phase** shell-and-tube heat exchanger (see Figure 1). The feedwater flows inside the tube bundle, while the steam and condensate flows outside these tubes (inside the cavity). In the water heater, there are three distinct areas: (1) the desuperheating zone, (2) the condensation zone, both located in the upper part of the component, and (3) the subcooled zone, located in the lower part of the component. In some water heaters, the condensate of the water heater located upstream from the current water heater is re-injected into the current water heater. During re-injection, part of the condensate may vaporize due to the pressure drop (this phenomenon is known as flash). The level of the condensate in the cavity is adjusted with a valve located at outlet of the water heater.

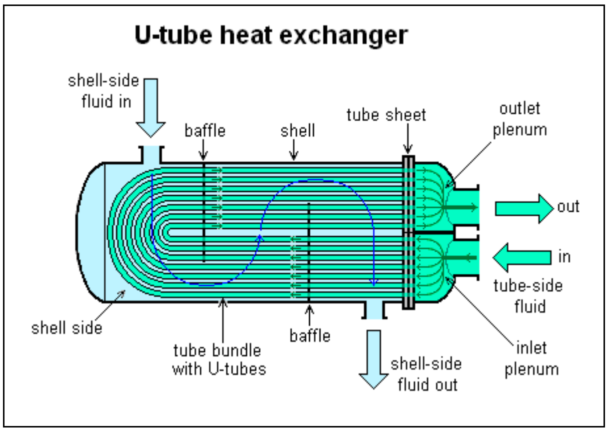


**Figure 1: Shell-and-tube heat exchanger**

## *2.2.* Description of the water heater model

The **DynamicWaterHeating** model represents the dynamics of the thermo-hydraulic phenomena of the hot fluid inside the cavity and of the cooling fluid which flows through the tube bundle. In particular, the model features the thermal exchanges between the fluid in the cavity and the cooling fluid flowing through the tube bundle.
The water heater is considered as a vertical or horizontal cylindrical cavity (as schematized in Figure 2), containing a U-bent tube bundle with the feedwater inlet and outlet located on the same side. The cavity is subdivided into the following zones:

A) **The desuperheating zone**, where the superheated steam, flowing into the heater, exchanges heat with the liquid flowing through the tube bundle, until it becomes saturated steam and enters the condensation zone. This zone is modelled by ‘Pipe 4’ in Figure 2.

B) **The condensation zone**, where the saturated steam condenses as a consequence of the thermal exchange with the tube bundle, turning into liquid water that enters the subcooled zone. This zone is modelled by ‘Pipe 2’ and ‘Pipe 3’ in Figure 2.

C) **The subcooled zone**, where the liquid inside the cavity continues to exchange heat with the liquid flowing through the tube bundle. This zone is modelled by ‘Pipe 1’ in Figure 2.

Four configurations of the model are possible (see Figure 2):

1. horizontal water heater, with desuperheating zone, condensation zone and subcooled zone,
2. horizontal water heater, with condensation zone only,
3. vertical water heater, with desuperheating zone, condensation zone and subcooled zone,
4. separate vertical water heater with desuperheating zone, condensation zone and subcooled zone.

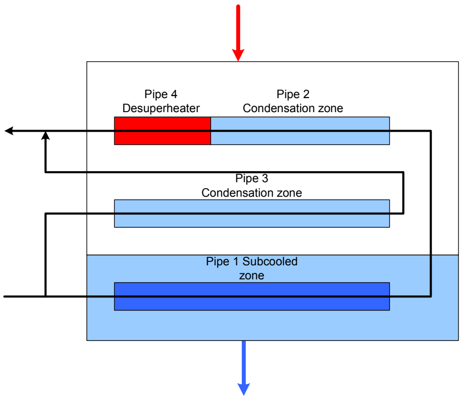


**Figure 2a: Horizontal water heater (1)**

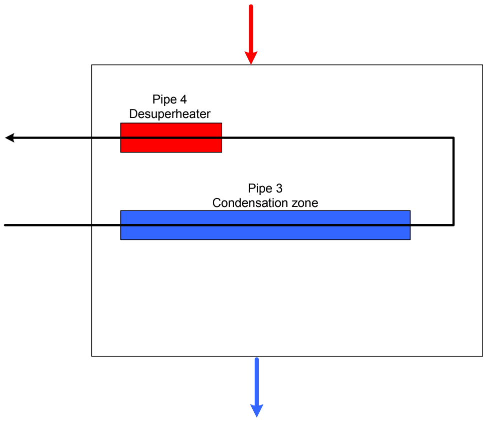


**Figure 2b: Horizontal water heater (2)**

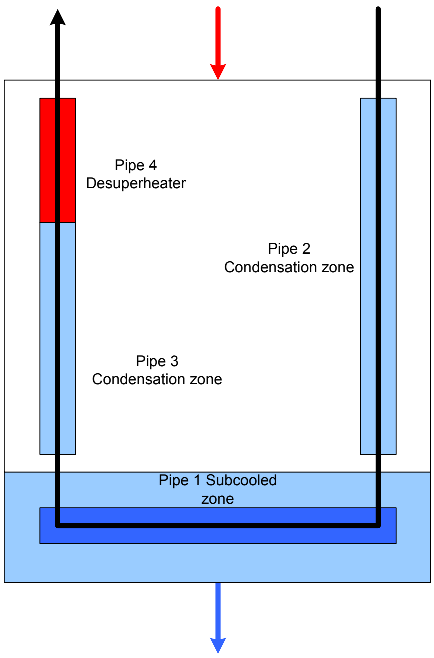


**Figure 2c: Vertical water heater (3)**


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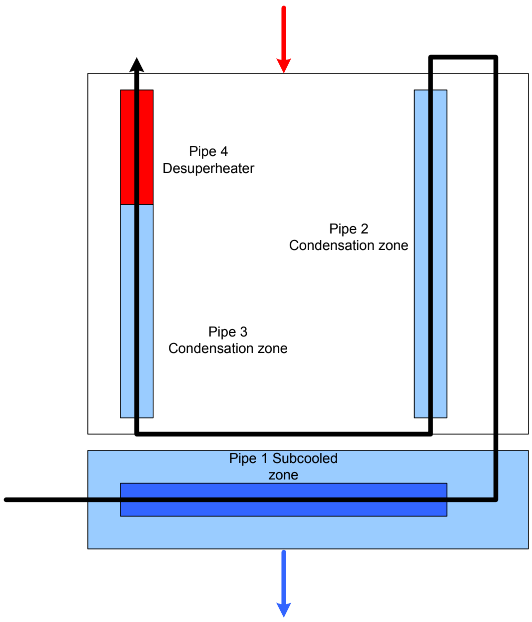


**Figure 2d: Vertical separate water heater (4)**

### *2.3.* Components of the water heater model

The **DynamicWaterHeating** can simulate all horizontal configurations as shown in Figure 2a and 2b.

The model is divided into sub-models of four different types which are connected together to make the full model (see Figure 3):

- 3 DynamicTwoPhaseFlowPipe models,
- 3 HeatExchangerWall models,
- 1 TwoPhaseCavity model,
- 3 Volume models.

By reassembling the sub-models, any other configuration of the water heater can be modelled.

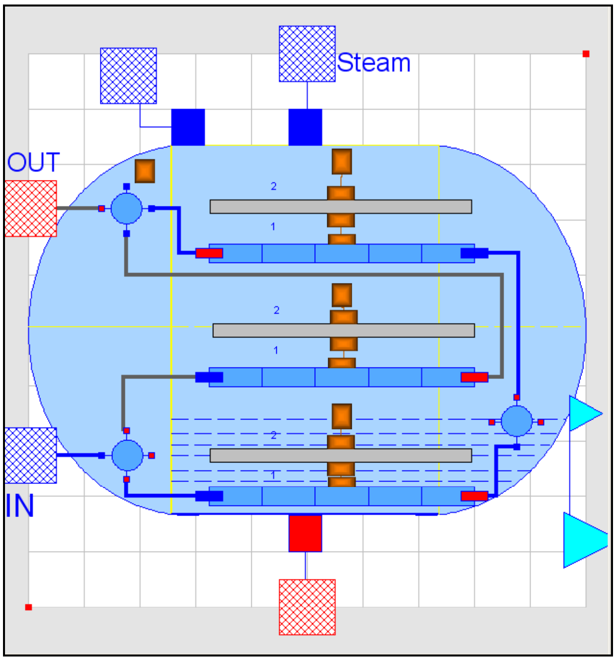


**Figure 3: Model of the water heater "DynamicWaterHeating "**

The description of each sub-model is given in the following section. Each sub-model in the model can be recognized by looking at its icon (see Figures 4, 5, 6 and 7).

## 3. Physics of the condenser/water heater

### *3.1.* DynamicTwoPhaseFlowPipe model

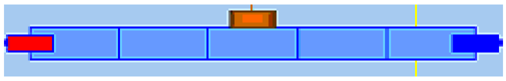

**Figure 4: Two-flow pipe model icon**

The model of the fluid flow in a cylindrical conduit is based on the dynamic mass, energy, and momentum balance equations, which are originally given as 1-D partial differential equations. The original distributed-parameter model is first discretised by using the finite-volume method. The model is formulated in order to correctly handle possible flow reversal conditions.

### Assumptions

- Homogeneous fluid in each mesh cell (same velocity for the liquid and steam phases);
- 1-D modelling (using the finite-volume method);
- The accumulation is considered in each mesh cell;
- The inertia of the fluid is taken into account;
- The phenomenon of longitudinal heat conduction in the metal wall and in the fluid is neglected;
- The thermo-physical properties are calculated on the basis of the average pressure and enthalpy in each mesh cell.

### Mass balance equation

The mass balance equation in each cell is given by:

$$A \cdot \frac{d\rho_i}{dt} \cdot \Delta x = \dot{m}_{i-1:i} - \dot{m}_{i:i+1}$$

Taking the pressure and the specific enthalpy as state variables yields:

$$A \cdot \left[ \left( \frac{\partial \rho_i}{\partial P_i} \right)_h \cdot \frac{dP_i}{dt} + \left( \frac{\partial \rho_i}{\partial h_i} \right)_P \cdot \frac{dh_i}{dt} \right] \cdot \Delta x = \dot{m}_{i-1:i} - \dot{m}_{i:i+1}$$

### Energy balance equation

The energy balance equation in each cell is given by:

$$A \cdot \frac{d(\rho_i \cdot u_i)}{dt} \cdot \Delta x = \dot{m}_{i-1:i} \cdot h_{i-1:i} - \dot{m}_{i:i+1} \cdot h_{i:i+1} + \Delta W_i$$

with the specific internal energy given by:

$$u_i = h_i - \frac{P_i}{\rho_i}$$

Taking the pressure and the specific enthalpy as state variables yields:

$$A.\left( \left( h_i . \frac{\partial \rho_i}{\partial P_i} - 1 \right) . \frac{dP_i}{dt} + \left( h_i . \frac{\partial \rho_i}{\partial h_i} + \rho_i \right) . \frac{dh_i}{dt} \right) \cdot \Delta x = \dot{m}_{i-1:i} \cdot h_{i-1:i} - \dot{m}_{i:i+1} \cdot h_{i:i+1} + \Delta W_i$$

$h_{i:i+1}$ is the specific enthalpy of the mass flow $\dot{m}_{i:i+1}$ crossing the boundary between the cells $i$ and $i+1$. $h_{i:i+1}$ is related to the state variables $h_i$ and $h_{i+1}$ by:

$$h_{i:i+1} = \hat{s}(P_e) \cdot h_i + \hat{s}(-P_e) \cdot h_{i+1}$$

where $P_e$ is the Peclet number and

$$\hat{s}(x) = \frac{1}{1 + e^{-\frac{x}{2}}}$$ (see e.g. [8]).

When neglecting diffusion, the Peclet number is infinite, and

$$h_{i:i+1} = s(\dot{m}_{i:i+1}) \cdot h_i + s(-\dot{m}_{i:i+1}) \cdot h_{i+1}$$

where $s$ is the step function:

$$s(x) = \begin{cases} 1 & if \quad x > 0 \\ 0 & if \quad x < 0 \end{cases}$$

This simplification is known as the upwind scheme.

### Momentum balance equation

The momentum balance equation in each cell is given by:

$$1/A \cdot \frac{d\dot{m}_{i:i+1}}{dt} \cdot \Delta x = P_i - P_{i+1} - (\Delta P)^a_{i:i+1} - (\Delta P)^f_{i:i+1} - (\Delta P)^g_{i:i+1}$$

with respectively the acceleration, friction and gravity pressure losses given by:

$$(\Delta P)^a_{i:i+1} = \frac{1}{A^2} \cdot \dot{m}_{i:i+1} \cdot |\dot{m}_{i:i+1}| \cdot \left( \frac{1}{\rho_{i+1}} - \frac{1}{\rho_i} \right)$$

$$(\Delta P)^f_{i:i+1} = \zeta . \frac{\Lambda_i \cdot \Delta x_h}{2 \cdot D \cdot A^2 \cdot \rho_i} \cdot \dot{m}_{i:i+1} \cdot |\dot{m}_{i:i+1}|$$

$$(\Delta P)^g_{i:i+1} = \rho_{i:i+1} \cdot g \cdot (z_{i+1} - z_i)$$

By default, the flow is considered turbulent (Reynolds number Re > 2300).

The **Colebrook** correlation is used to compute $\Lambda_i$.

### Convective heat transfer within the U-tubes

The heat exchanged between the fluid and the wall is:

$$\Delta W(i) = h_c(i) \cdot \Delta S_2 \cdot (T_{w2}(i) - T(i))$$

### Convection heat transfer coefficient

The convection heat transfer coefficient $h_c$ between the fluid and the wall is computed using the **Dittus-Boelter** correlation.


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### *3.2.* HeatExchangerWall model

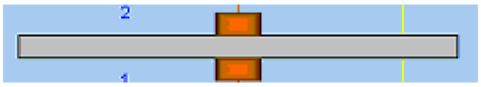

**Figure 5: Wall model icon**

The wall model describes the conductive heat flow through the wall of the tube bundle. The flow is positive when entering the tubes (going from side 2 to side 1 of the wall).

$$\Delta W_1(i) = \frac{2 \cdot \pi \cdot \lambda \cdot \Delta x(i) \cdot ntubes \cdot \left(T_w(i) - T_{w1}(i)\right)}{\ln\left((e + D)/D\right)}$$

$$\Delta W_2(i) = \frac{2 \cdot \pi \cdot \lambda \cdot \Delta x(i) \cdot ntubes \cdot \left(T_{w2}(i) - T_w(i)\right)}{\ln\left((2 \cdot e + D)/(e + D)\right)}$$

$$\Delta M_w \cdot c_{pw} \cdot \frac{dT_w}{dt} = \Delta W_2(i) - \Delta W_1(i)$$

### *3.3.* TwoPhaseCavity model

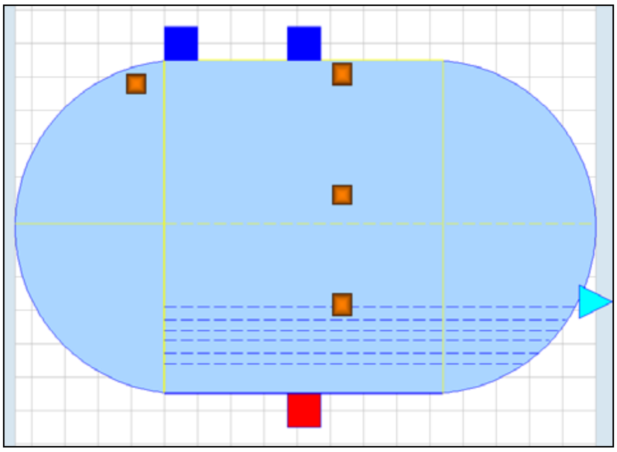

**Figure 6: Two-phase cavity model icon**

The cavity is modelled as a non-adiabatic two-phase volume, with vertical or horizontal cylindrical geometry. The physical model is based on a non-equilibrium, two-phase formulation of the fluid balance equations with a control volume approach. The two phases are supposed to be isobaric and will be referred to as liquid zone and steam zone, respectively.

The model features the condensation flow of the steam phase into the liquid phase, and reciprocally, the vaporization flow of the liquid phase into the steam phase.

The reasons for not assuming thermal equilibrium between the two phases are:

- The vapour may enter the cavity superheated (the vapour temperature is then higher than the saturation temperature).
- The liquid may be subcooled by the incoming drain and the wetted tube bundle (the liquid temperature is then lower than the saturation temperature).

***Assumptions***

- Accumulation of mass and energy is considered. Heat exchange between the liquid and steam phases is considered.
- Heat exchange between the liquid or steam phases and the wall is considered.
- Heat exchange between the water heater and the external medium (ambient) is considered.
- Pressure losses are not taken into account in the cavity.
- The liquid and steam phases are not necessarily in thermal equilibrium.
- The liquid and steam phases are assumed to be permanently in pressure equilibrium.

***State variables***

The state variables of the system are:

- the mean pressure in the cavity,
- the specific enthalpy of the liquid phase,
- the specific enthalpy of the steam phase,
- the temperature of the wall,
- the volume of the liquid phase.

The volume of the steam phase is bound to the volume of the liquid phase by the following equation:

$$V_l + V_v = V$$

***Mass balance equation in each phase***

$$\frac{d(\rho_l \cdot V_l)}{dt} = -\dot{m}_l^o + (1 - x_{mv}) \cdot \dot{m}_{drain}^e + \dot{m}_{cond} - \dot{m}_{evap}$$

$$\frac{d(\rho_v \cdot V_v)}{dt} = \dot{m}_v^e + x_{mv} \cdot \dot{m}_{drain}^e + \dot{m}_{evap} - \dot{m}_{cond}$$

where $\dot{m}_v^e$ is the mass flow of incoming vapor, $\dot{m}_{drain}^e$ is the mass flow of the incoming condensate of the water heater located upstream, $\dot{m}_l^o$ is the mass flow of outgoing condensate, $\dot{m}_{cond}$ is the condensation flow inside the cavity, and $\dot{m}_{evap}$ is the evaporation flow inside the cavity.


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Condensation and evaporation mass flow rate inside the cavity

$$\dot{m}_{cond} = \begin{cases} x_v < X_{vo} \Rightarrow C_{cond} \cdot \rho_v \cdot V_v \cdot (X_{vo} - x_v) \\ x_v \geq X_{vo} \Rightarrow 0 \end{cases}$$

$$\dot{m}_{evap} = \begin{cases} x_l > X_{lo} \Rightarrow C_{evap} \cdot \rho_l \cdot V_l \cdot (x_l - X_{lo}) \\ x_l \geq X_{vo} \Rightarrow 0 \end{cases}$$

$C_{cond}$ and $C_{evap}$ being coefficients with inverse time dimensionality $\left[\frac{1}{t}\right]$, $X_{vo}$ and $X_{lo}$ denoting constants.

### *Energy balance equation in each phase*

The general form of the energy balance equation is given by:

$$\frac{d(\rho \cdot V \cdot u)}{dt} = \sum_e \dot{m}_e \cdot h_e + \sum_o \dot{m}_o . h_o + \sum W$$

Taking the pressure and the specific enthalpy as state variables yields:

$$V_l \cdot \left[ \left( \frac{P}{\rho_l} \cdot \left( \frac{\partial \rho_l}{\partial P} \right)_h - 1 \right) \cdot \frac{dP}{dt} + \left( \frac{P}{\rho_l} \cdot \left( \frac{\partial \rho_l}{\partial h_l} \right)_P + \rho_l \right) \cdot \frac{dh_l}{dt} \right] =$$
$$-\dot{m}_l^o \cdot \left( h_l^o - (h_l - \frac{P}{\rho_l}) \right) + \dot{m}_{cond} \cdot \left( h_l^{sat} - (h_l - \frac{P}{\rho_l}) \right)$$
$$-\dot{m}_{evap} \cdot \left( h_v^{sat} - (h_l - \frac{P}{\rho_l}) \right) + (1 - x_{mv}) \cdot \dot{m}_{drain}^e \cdot \left( h_{drainl}^e - (h_l - \frac{P}{\rho_l}) \right)$$
$$+W_{vl} - W_{lw} - W_{1t}$$

$$V_v \cdot \left[ \left( \frac{P}{\rho_v} \cdot \left( \frac{\partial \rho_v}{\partial P} \right)_h - 1 \right) \cdot \frac{dP}{dt} + \left( \frac{P}{\rho_v} \cdot \left( \frac{\partial \rho_v}{\partial h_v} \right)_P + \rho_v \right) \cdot \frac{dh_v}{dt} \right] =$$
$$\dot{m}_v^e \cdot \left( h_v^e - (h_v - \frac{P}{\rho_v}) \right) - \dot{m}_{cond} \cdot \left( h_l^{sat} - (h_v - \frac{P}{\rho_l}) \right)$$
$$+\dot{m}_{evap} \cdot \left( h_v^{sat} - (h_v - \frac{P}{\rho_v}) \right) + x_{mv} \cdot \dot{m}_{drain}^e \cdot \left( h_{drainv}^e - (h_v - \frac{P}{\rho_l}) \right)$$
$$-W_{vl} - W_{vw} - W_{2t} - W_{3t}$$

Energy accumulation at the wall

$$M_w \cdot c_{pw} \cdot \frac{dT_w}{dt} = W_{lw} + W_{vw} - W_{wa}$$

Heat exchange between the liquid and steam phases

$$W_{vl} = K_{vl} \cdot A_{vl} \cdot (T_v - T_l)$$

Heat exchange between the liquid or steam phases and the wall

$$W_{lw} = K_{lw} \cdot A_{lw} \cdot (T_l - T_w)$$
$$W_{vw} = K_{vw} \cdot A_{vw} \cdot (T_v - T_w)$$

Heat exchange between the water heater and the external medium

$$W_{wa} = K_{va} \cdot A_{va} \cdot (T_v - T_a)$$

In this equation, the vapor temperature is considered instead of the wall temperature to account for both the thermal resistance of the metallic wall and the thermal insulator of the cavity, in addition to the usual convective resistance. Consequently, $K_{va}$ is the global heat exchange coefficient between the vapor and the ambient. The liquid is neglected in this equation because the volume of liquid is small w.r.t. the volume of vapor.

Heat exchange between the liquid and the tube bundle 'Pipe 1'

$$\Delta W_1(i) = h_{conv1}(i) \cdot \Delta S_{ext1} \cdot (T_l - T_{w1}(i))$$
$$W_{1t} = \sum \Delta W_1(i)$$

Heat exchange between the steam and the tube bundle 'Pipe 2'

$$\Delta W_2(i) = h_{cond2}(i) \cdot \Delta S_{ext2} \cdot (T_v - T_{w2}(i))$$
$$W_{2t} = \sum \Delta W_2(i)$$

Heat exchange between the steam and the tube bundle 'Pipe 3'

$$\Delta W_3(i) = h_{cond3}(i) \cdot \Delta S_{ext3} \cdot (T_v - T_{w3}(i))$$
$$W_{3t} = \sum \Delta W_3(i)$$

Heat exchange between the steam and the tube bundle 'Pipe 4'

$$W_{4t} = \dot{m}_v^e \cdot (h_v^e - h_v^{sat})$$

Heat transfer convection coefficients

The heat transfer convection coefficient $h_{conv}$ between the water and the outside wall of the tube bundle is computed using the **Kern** correlation [7] .

The **Nusselt** correlation is used to calculate the heat transfer coefficients $h_{cond}$ between the steam and the outside wall of the tube bundle, in the condensation zone.


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### *3.4.* Mixture homogeneous Volume model

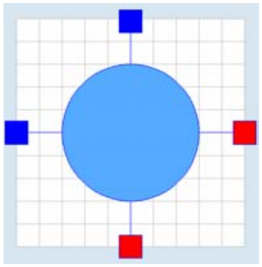

**Figure 7: Mixing volume model icon**

This sub-model describes the mixing of one-phase flow fluid.

***Mass balance equation***

$$\frac{d(\rho \cdot V)}{dt} = \sum_e \dot{m}_e + \sum_o \dot{m}_o$$

***Energy balance equation***

$$\frac{d(\rho \cdot V \cdot u)}{dt} = \sum_e \dot{m}_e \cdot h_e + \sum_o \dot{m}_o . h_o + \sum W$$

## 4. Validation of the condenser/water heater

### *4.1.* Modelica model of the condenser/water heater

To simulate the complex dynamic physical behaviour in normal and accidental conditions of the condenser/water heater model, a test model called "TestDynamicWaterHeating" has been developed by assembling the necessary components from the **ThermoSysPro** library (cf. Figure 8). The test model includes the level control system.

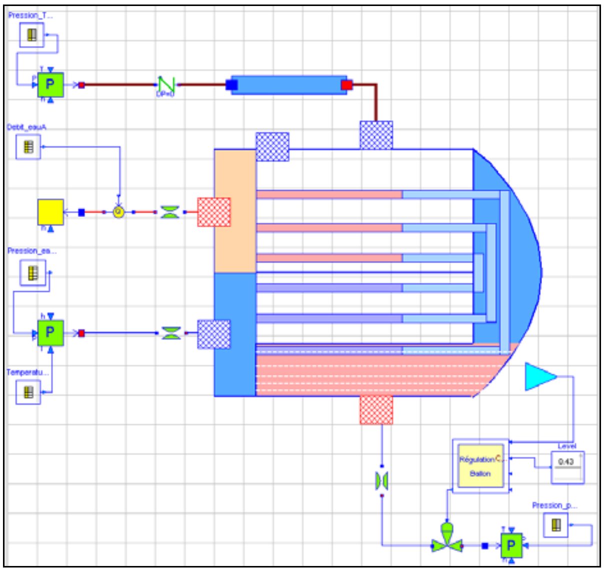

**Figure 8: Model of the water heater "TestDynamicWaterHeating "**

### *4.2.* Data implemented in the model

All geometrical data were provided to the model (tubes and exchangers lengths, diameters, volumes, corrective terms for the heat exchange coefficients, corrective terms for the pressure losses, etc.). The plant characteristics are given in Figure 11 (cf. Appendix).

### *4.3.* Calibration of the model

The calibration phase consists in setting (blocking) the maximum number of thermodynamic variables to known measurement values (enthalpy, pressure) taken from on-site sensors for 100% load. This method ensures that all needed performance parameters, size characteristics and output data can be computed.

The main computed performance parameters are:

- the correction coefficient of the heat transfer coefficient inside the condensation zone,
- the correction coefficient of the pressure loss coefficients inside the tube bundle (pipes),
- the pressure loss coefficients of the pipeline between the steam turbine and the water heater,
- the maximum Cv values of the extraction valve and the valves positions.

### *4.4.* Simulation scenario: islanding

In order to challenge the dynamic simulation capabilities of the model, a high amplitude transient, called islanding, that occurs when the plant is suddenly disconnected from the normal energy discharge network, is simulated. This transient is used to check and validate the physics taken into account in the model and the numerical robustness of the model as it runs the water heater model into very different operating regimes. This allows to test the validity and applicability range of the model equations, and the numerical robustness of the Modelica implementation when using Dymola.

### *4.5.* Boundary conditions of the model

The boundary conditions of the model (scenario profiles) are presented in Figure 9.

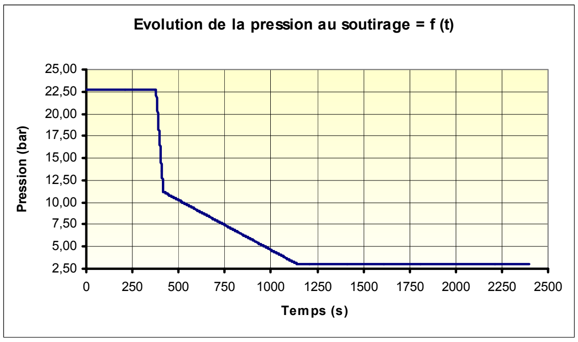


**Figure 9a: Outlet pressure of the steam turbine**

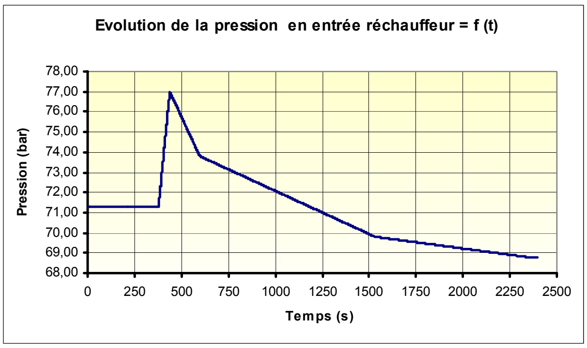


**Figure 9b: Inlet pressure of the feed water**

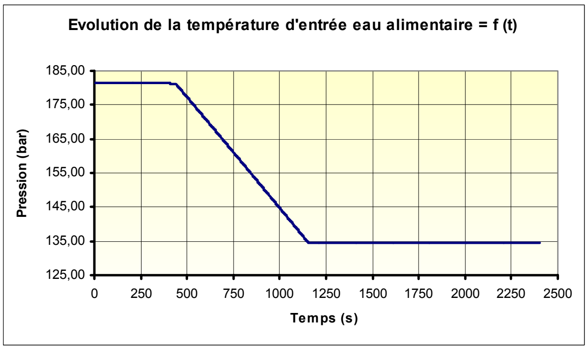


**Figure 9c: Inlet temperature of the feed water**

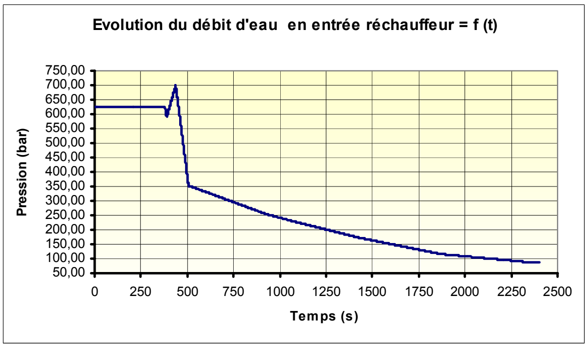


**Figure 9d: Inlet flow of the feed water**

### *4.6.* Results of dynamic simulations

In order to cover the whole transient, the simulation time has been set at 2500 seconds.

Simulation runs were done using Dymola 6.1. The simulation of the scenarios were mostly successful, with only one iteration variable to be fed manually.

The following phenomena are simulated:
- flow reversal,
- local boiling or condensation,
- swell and shrink effect in cavity,
- cavity levels and cavity pressure control.

The model is able to compute precisely:
- the mass flow rate of the steam (at the inlet),
- the mass flow rate of the condensate (drain),
- the distribution of water and steam mass flow rate inside the tubes,
- the thermal power of the water heater and tubes,
- the pressure temperature and specific enthalpy distribution across the network,
- the cavity levels and and cavity pressure.

The results of the simulation runs are given in Figure 10. Figures 10a and 10b show that the results obtained with Dymola are very close to the measured values on site. The outflow drain (condensate) in Figure 10d depends on the way the level is controlled inside the heater.

So, the physical validity of the component model is demonstrated, because we believe that this type of fast transient is likely to extensively validate the physics inside the model as it challenges the water heater in very different operating regimes of the rated operation.


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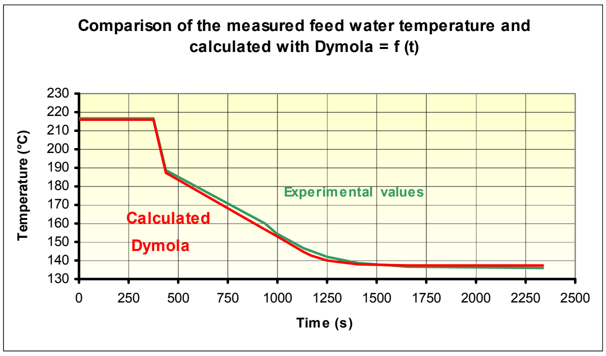


**Figure 10a: Evolution of the feed water outlet temperature**

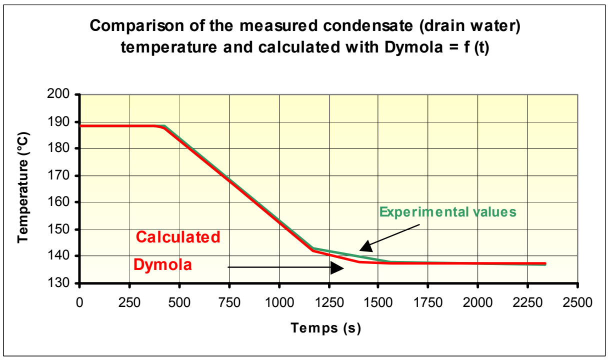


**Figure 10b: Evolution of the condensate (water drain) outlet temperature**

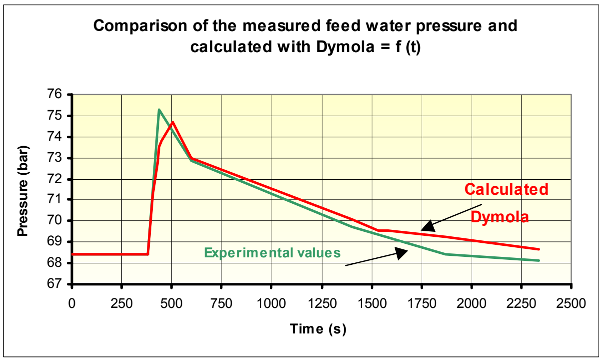


**Figure 10c: Evolution of the feed water outlet pressure**

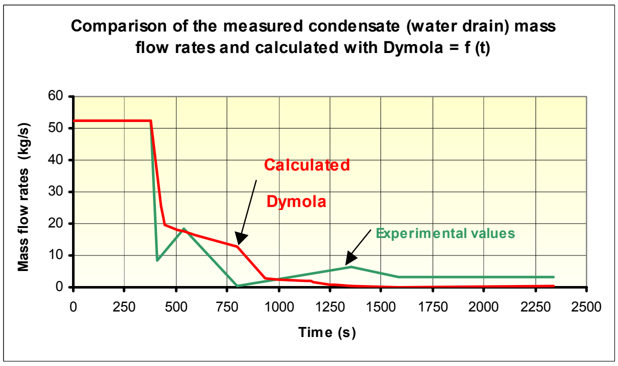


**Figure 10d: Evolution of the condensate (water drain) outlet mass flow rate**

### *4.7.* Validation of the water heater model under flow reversal and zero-flow conditions

The ThermoSysPro library handles flow reversals.

The boundary conditions for the flow reversal scenario are:

- outlet pressure of the steam turbine = 22.733e5 Pa,
- outlet enthalpy of the steam turbine = 2650.6e3 J/kg,
- inlet pressure of feed water = 71.29e5 Pa,
- inlet temperature of the feed water = 454.46 °C,
- inlet mass flow rate of the feed water (t = 0) = 624.97 kg/s,
- inlet mass flow rate of the feed water (t > 2000s) = -200 kg/s,
- outlet enthalpy of the feed water inlet (Q < 0) = 940.e3 J/kg.

Figures 12 and 13 in the Appendix show the results for the scenario of flow reversal in the water heater and the results for the zero-flow scenario.

The possibility of flow reversal and zero-flow in the tube bundle of the component has been experimentally verified. But there are no data available for comparison with the simulation results.

## 5. Conclusion

A new open source Modelica library called ‘ThermoSysPro’ has been developed within the framework of the ITEA 2 EUROSYSLIB project. This library has been mainly designed for the static and dynamic modeling of power plants, but can also be used for other energy systems such as industrial processes, buildings, etc. It is intended to be easily understood and extendable by the models developer.

A new dynamic model of a water heater has been developed using existing elements of ThermoSysPro.

To validate the model, three difficult transients were simulated: the islanding (sudden plant disconnection from the grid), flow reversal and zero-flow inside the water heater.

Regarding the islanding scenario, the simulation results obtained with Dymola are very close to the experimental values measured on site. This transient demonstrates the physical validity of the model at it is fast and challenges the model equations in all operating conditions of the exchanger.

The possibility of flow reversal and zero-flow occurring inside the tube bundle of the module has been experimentally verified and simulated, but no experimental data is available for comparison with the simulation results.

## Nomenclature

Symbols

| | |
|---|---|
| $\dot{m}$ | *Mass flow* |
| $\rho$ | *Fluid density* |
| $h$ | *Fluid specific enthalpy* |
| $u$ | *Fluid specific internal energy* |
| $P$ | *Fluid pressure* |
| $T$ | *Fluid temperature* |
| $c_p$ | *Fluid specific heat capacity* |
| $V$ | *Volume* |
| $t$ | *Time* |
| $W$ | *Power* |
| $x_v$ | *Vapor mass fraction in vapor phase* |
| $x_l$ | *Vapor mass fraction in liquid phase* |
| $x_{mv}$ | *Vapor mass fraction in input drain* |
| $\Lambda$ | *Friction coefficient* |
| $\zeta$ | *Friction corrective coefficient* |
| $\Delta x$ | *Tube segment length* |
| $\Delta S$ | *Heat surface exchange of tube segment* |
| $D$ | *Tube diameter* |
| $A$ | *Tube cross section or heat exchange surface* |
| $e$ | *Wall thickness* |
| $\lambda$ | *Conduction coefficient* |
| $K$ | *Heat exchange coefficient* |
| $M$ | *Mass* |
| $h_c$ | *Convective coefficient* |
| $h_{conv1}$ | *Convective coefficient of heat transfer between the condensate and the tube bundle in Pipe 1.* |
| $h_{cond2}$ | *Convective coefficient of heat transfer by condensation between the vapor and the tube bundle in Pipe 2.* |
| $h_{cond3}$ | *Convective coefficient of heat transfer by condensation between the vapor and the tube bundle in Pipe 3.* |
| *ntubes* | *Number of tubes in the bundle* |

Indices

| | |
|---|---|
| $X_i$ or $X(i)$ | *Quantity in volume i* |
| $X_{i:i+1}$ | *Flow between volume i and i+1* |
| $X_e$ or $X^e$ | *Quantity at inlet* |
| $X_o$ or $X^o$ | *Quantity at outlet* |
| $X_l$ | *Quantity relative to liquid* |
| $X_v$ | *Quantity relative to vapor* |
| $X_w$ | *Quantity relative to wall* |
| $X_{ext}$ | *Quantity relative to external side of wall* |
| $X_a$ | *Quantity relative to ambient* |
| $X^{sat}$ | *Quantity relative to saturated phase* |
| $X_{cond}$ | *Quantity relative to condensation* |
| $X_{evap}$ | *Quantity relative to evaporation* |
| $X_{drain}$ | *Quantity relative to drain (condensate)* |
| $X_1$ | *Quantity relative to Pipe 1* |
| $X_2$ | *Quantity relative to Pipe 2* |
| $X_3$ | *Quantity relative to Pipe 3* |
| $X_4$ | *Quantity relative to Pipe 4* |

## References

[1] Bouskela D., Chip V., El Hefni B., Favennec J.M., Midou M. and Ninet J. ‘New method to assess tube support plate clogging phenomena in steam generators of nuclear power plants’, Mathematical and Computer Modelling of Dynamical Systems, 16: 3, 257-267, 2010.

[2] El Hefni B., Bouskela D., ‘Modelling of a water/steam cycle of the combined cycle power plant “Rio Bravo 2” with Modelica’, Modelica 2006 conference proceedings.


DOI
10.3384/ecp12076745

[3] David F., Souyri A., Marchais G., ‘Modelling Steam Generators for Sodium Fast Reactors with Modelica’, Modelica 2009 conference proceedings

[4] El Hefni B., Péchiné B., ‘Model driven optimization of biomass CHP plant design’, Mathmod conference 2009, Vienna, Austria.

[5] El Hefni B., Bouskela D., ‘Dynamic modelling of a combined cycle power plant with ThermoSysPro’, Modelica 2011 conference proceedings

[6] Souyri A., Bouskela D., ‘Pressurized Water Reactor Modelling with Modelica’, Modelica 2006 conference proceedings.

[7] Collier J.G., and Thome J.R., ‘Convective Boiling and Condensation’, Mc Graw-Hill Book Company (UK) limited, 1972 Clarendon Press, Oxford, 1996.

[8] Patankar S.V., ‘Numerical Heat Transfer and Fluid Flow’, Hemisphere Publishing Corporation, Taylor & Francis, 1980.

# Appendix

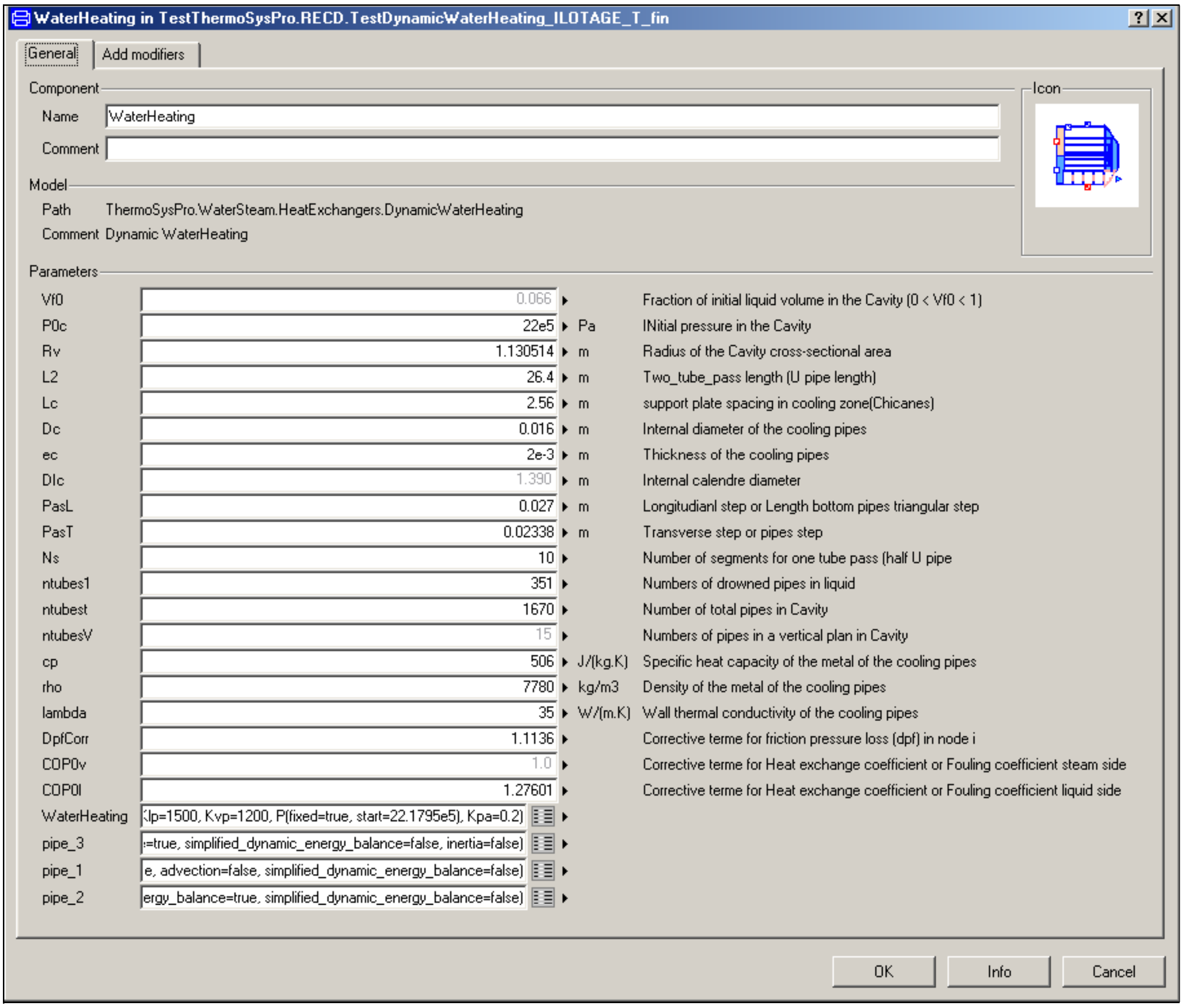


**Figure 11: Data of the model**


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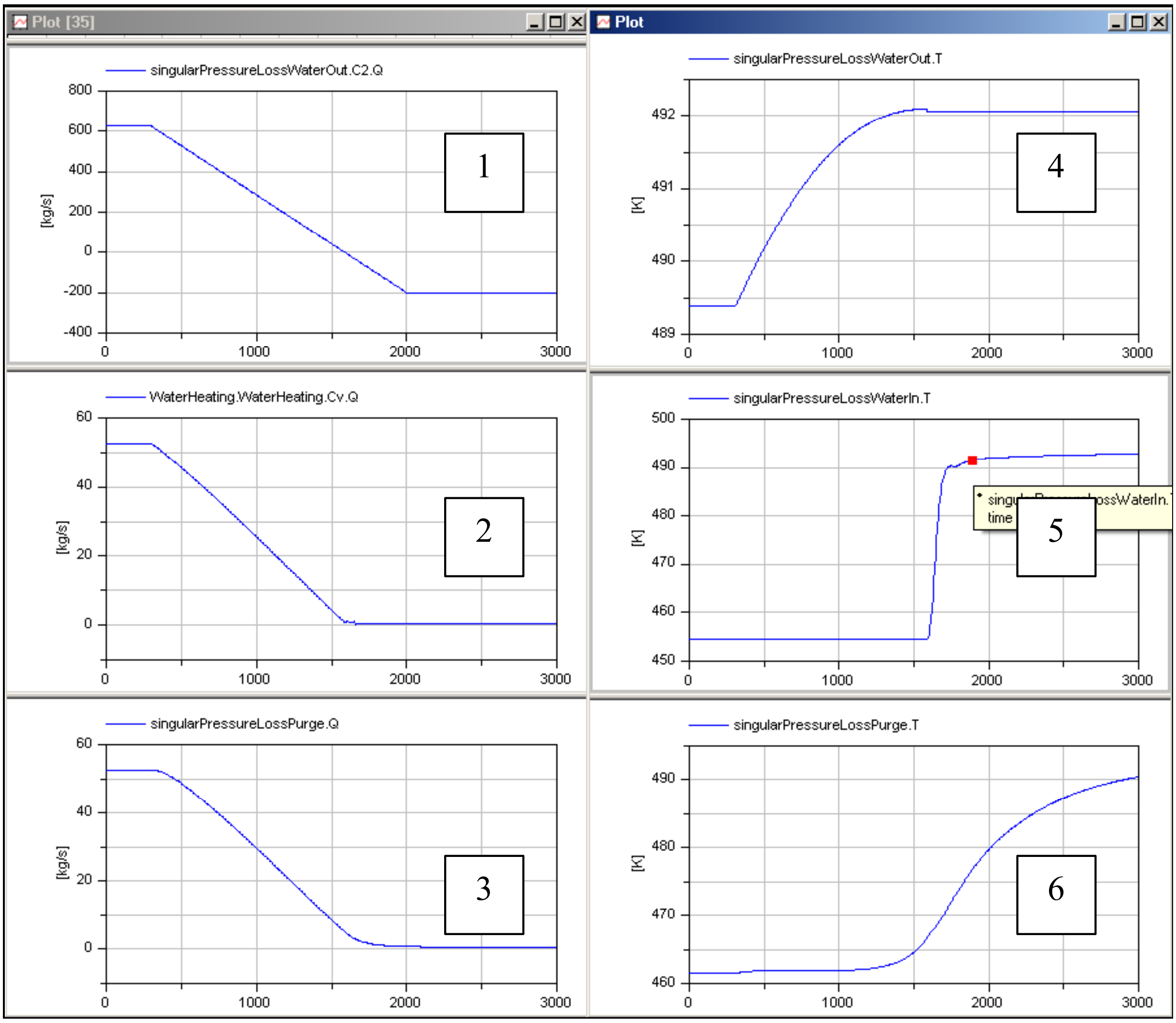


**Figure 12: Results for the flow reversal scenario**

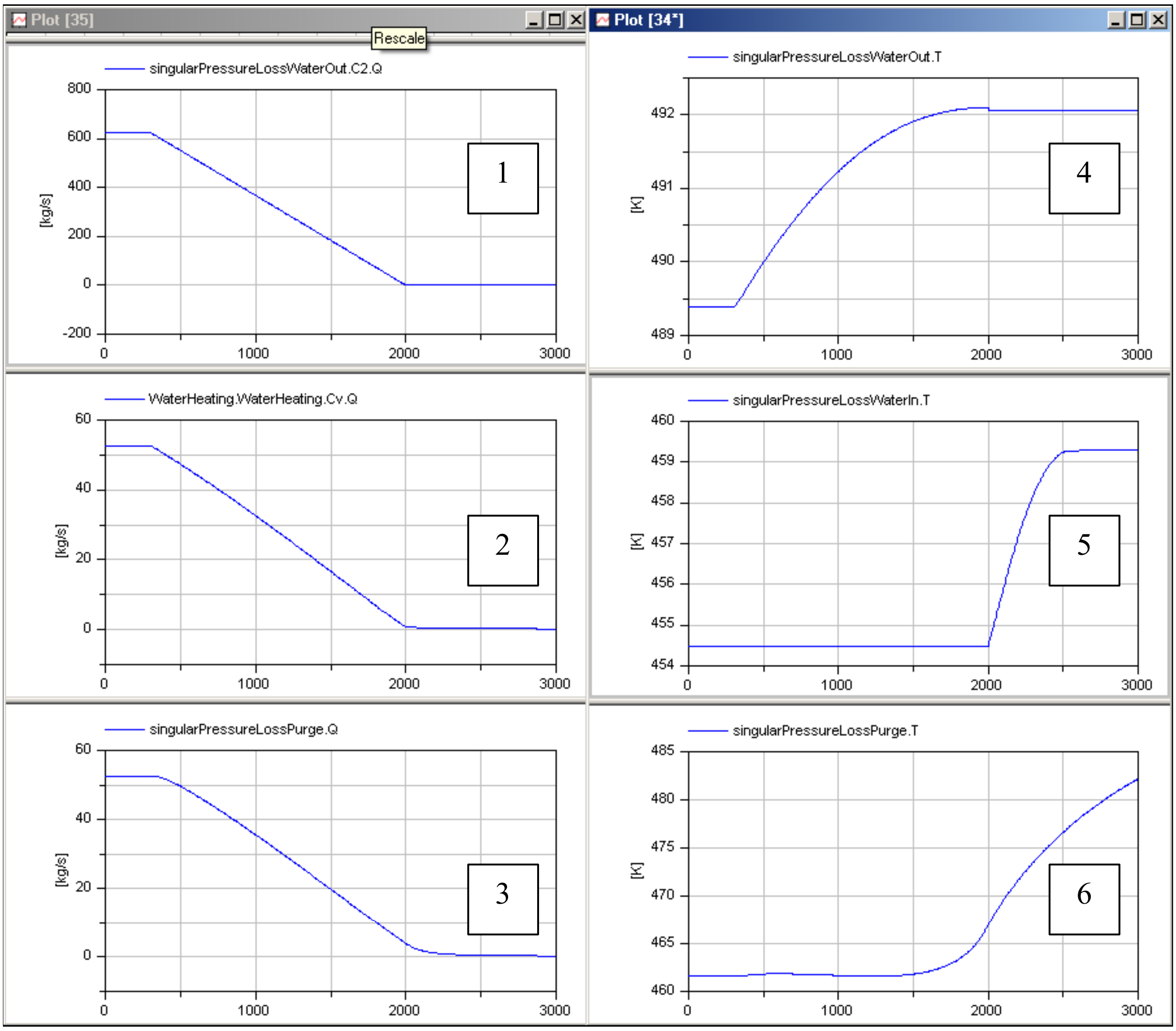


**Figure 13: Results for the zero-flow scenario**

With:
1 - Evolution of the inlet mass flow rate of the feed water,
2 - Inlet mass flow rate of the steam (corresponding to the steam turbine outlet),
3 - Outlet mass flow rate of the water (output drain),
4 - Outlet temperature of the feed water (pipes),
5 - Inlet temperature of the feed water (pipes),
6 - Outlet temperature of the water (output drain).


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