CONCEPTUAL DESIGN APPROACH OF PASSIVE COOLING SYSTEMS

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ABSTRACT

Active cooling systems are vulnerable to component failures or loss of electrical power. Consequently, to increase the reliability of the system as a whole, adequate redundancy for these components as well as electrical back-up power sources must be provided. The Fukushima Daiichi accident however demonstrates that, despite redundancy and alternative power sources, an extended loss of cooling can occur.

The present paper presents a conceptual design of a passive cooling system with a power removal capacity from 1 kW up to 1 MW. This system, constituted by different fluid loops, can provide adequate confinement in cases it is necessary.

The proposed design is modeled with the FlowMaster simulation tool providing both steady state and dynamic responses. The results show the relationship between the system constraints (heat load and temperature difference between hot and cold source) and the design parameters of the system.

A valuable application of passive systems is spent fuel storage cooling where adequate heat removal must be provided to the fuel elements at all time and confinement is of outmost importance. The proposed approach is applied to the design of the hot cell cooling in which the spent fuel elements of the future MYRRHA facility are handled.

KEYWORDS
Natural circulation, passive cooling system, spent fuel cooling

1. INTRODUCTION

The great advantage of passive cooling systems is their independence from external power sources and active devices. They are generally composed of piping and heat exchangers only. On the other hand, such cooling systems often have poor heat transfer properties that have to be compensated by large transfer areas and/or large temperature differences.

If confinement is needed, the power to be removed is transferred to a coolant fluid which at its turn transfers the heat to the environment (heat sink). Passive cooling systems with different loops are not common or widely studied due to their limited heat transfer capacity and the room needed to install such systems.

The stated problem is to cooldown a volume where a thermal load P is dissipated under a reference temperature (T_{ref}) with a heat sink temperature (T_{cold}) in a passive way, independently of the nature of the thermal limitations. This problem can be defined by:

$$P=R*(T_{ref}-T_{cold}) \tag{1}$$

The goal of the present study is to propose a design for the global thermal resistance R. This design is based on:

- Natural circulation loops which is the basis of the thermal load transport to the heat sink;
- Heat exchangers where the heat load is transferred from one fluid to another.

The present paper proposes a design of a passive cooling system and presents a simulation of its dynamic behavior. It will first focus on the physics of natural circulation in cooling systems and the design basis of heat exchangers (Sections 2 and 3). The methodology to develop the conceptual design of such a cooling system is presented afterwards, as well as the relationship between the design parameters and the criteria to be met (Section 4). The behavior of the cooling system in certain conditions is analyzed by performing steady-state and dynamic simulations.

The suggested approach is applied to produce a conceptual design of a cooling system for the hot cell of the MYRRHA facility [1] [2] in which spent fuel elements are handled and stored (Section 5).

2. NATURAL CIRCULATION MODELLING

The driving force of natural circulation is the density difference in a material as a result of a temperature difference.

The equations modelling the process are:

$$\Delta p_{\mathrm{driving}} = \rho_{cold}(T_{cold}) * g * \Delta h - \rho_{hot}(T_{hot}) * g * \Delta h \tag{2}$$

$$\Delta p_{\mathrm{friction}} = \frac{K q_m^2}{2\rho S^2} \tag{3}$$

$$P = q_m C_p (T_{hot} - T_{cold}) \tag{4}$$

In steady state, the driving head $\Delta p_{\mathrm{driving}}$ is equal to the friction losses $\Delta p_{\mathrm{friction}}$ and the equations (2) to (4) are solved for the temperature difference and the mass flow rate with one of the temperatures and the heat load as parameters.

To be noticed is that the driving head in natural circulation is several orders of magnitude lower than the one in forced circulation. Therefore, the natural flow rate is extremely sensitive to pressure losses. On the other hand, these systems have a positive feedback because any flow perturbation leading to a flow rate decrease increases the temperature difference and therefore the driving head.

Several experiences with the natural circulation of both air and water have been gathered [3-5]. In order to validate the chosen simulation tool, FlowMaster, the experiments presented in reference [3, 4] have been simulated. The comparison between experimental data and results achieved by the simulation is presented in Table 1. The validation has been performed on a broad range of thermal loads from 5 W to 2.5 kW.

The simulation with FlowMaster takes into account the variation of the fluid properties (density, viscosity, heat capacity …) with respect to the temperature.

The discrepancy between experimental results and simulations is mainly due to the overestimation of pressure losses by FlowMaster. The computed temperature difference is for most of the cases higher than the measured one. Nevertheless, the differences (up to 15%) are not significant and one can conclude that FlowMaster is an appropriate tool for natural circulation analysis.

Table I. FlowMaster validation for natural circulation evaluation

Heat sink Temperature	Measured Temperature difference	Computed Temperature difference	Difference [%]	Thermal load	Reference
25°C	13.5°C	14.3°C	6	5W	[3]
30°C	20.4°C	21.0°C	3	15W	[3]
35°C	24.0°C	24.3°C	1	25W	[3]
20°C	8.5°C	7.2°C	15	2 kW	[4]
30°C	8°C	7.4°C	7	2 kW	[4]
10°C	9.2°C	10.5°C	14	2.5 kW	[4]
20°C	8.5°C	9.1°C	7	2.5 kW	[4]

3. HEAT EXCHANGER MODELLING

The heat exchangers are designed on the basis of the conventional heat transfer equation. The pressure losses in the heat exchanger shall be minimized (minimum velocity). On the other hand, appropriate heat exchange conditions have to be guaranteed. Therefore the heat exchangers are designed as such that the Reynolds number remains above the turbulent limit (~2200 for internal flows).

The global conductance of the heat exchanger is estimated using the following equations:

$$h_{tot} = \frac{1}{\frac{1}{h_{outside}} + \frac{1}{h_{cond}} + \frac{1}{h_{inside}}} \tag{5}$$

$$h_{cond} = \frac{k}{e} \tag{6}$$

$$h_{inside\ /outside} = \frac{Nu_{inside\ /outside} * k_{fluid}}{L_c} \tag{7}$$

$$Nu_{inside} = 0.023 * Re^{0.8} * Pr^{1/3} \tag{8}$$

For the external surface, the heat transfer coefficient has to be estimated based on the type of heat transfer. For finned tubes, this coefficient is estimated at 30 W/m²/K which corresponds to a low air velocity (1m/s).

4. DESIGN APPROACH

4.1. Cooling System General Structure

The general structure of the cooling system described below consists of two cooling sub-systems and three natural circulation loops. This configuration, presented in Figure 1, guarantees the confinement if needed.

The cooling system consists of:

- A circulation loop in the volume to be cooled; in this loop, the heat load is transferred from the heated elements towards an immersed heat exchanger. This immersed heat exchanger has to be located on the top of the cooled volume. The efficiency of this loop is not estimated but it is assumed that no thermal gradient does exist inside this volume. Thus the temperature of the fluid at the outside of the immersed heat exchangers is equal to the average temperature of the volume. This assumption is not unrealistic, especially when the cooled fluid has a high thermal diffusivity (e.g.; air or water with values of $2.3\ 10^{-5}$ and $1.9\ 10^{-5}$ m²/s respectively).
- A first cooling system which consists of a natural circulation loop transferring the heat from one heat exchanger to another. The fluid used has to combine both high heat capacity to maximize the heat transferred with a fixed temperature difference, and a high thermal expansion factor allowing a maximum density difference and hence maximum available driving head. An improvement of the system can be to design it for reflux condensation. This particular natural circulation mode is very efficient as it works without temperature difference and high density difference. The only requirement for the fluid properties is a low saturation temperature that lies between the temperatures of the heat source and heat sink or the next loop.
- A second cooling system which consists of a natural circulation loop transferring the heat load towards the heat sink, in this case the environment, through a heat exchanger which are typically dry air coolers with an extraction based on natural circulation.

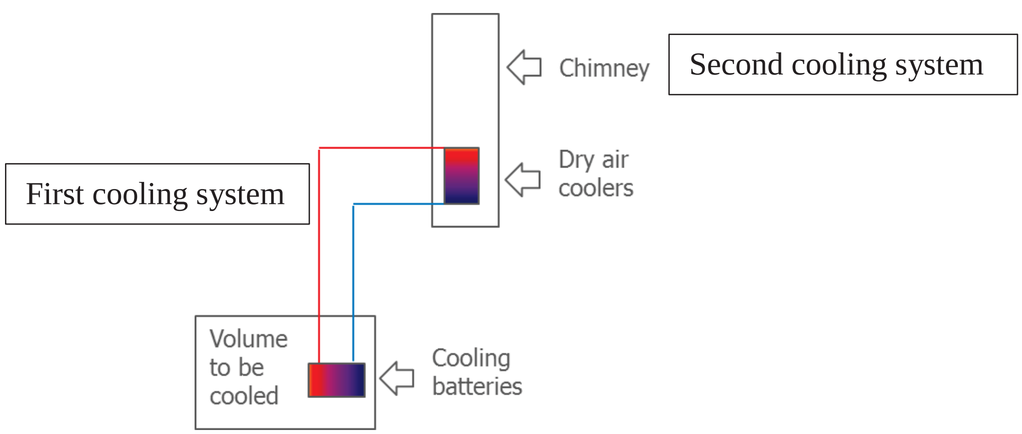

Figure 1. Scheme of the cooling system general structure.

4.2. Temperature Distribution

The temperature distribution between the different systems is a first degree of freedom. It has to be carefully chosen as it will drive the available temperature potential for the two heat exchangers and, therefore, their size. Within the total temperature difference between the heat source and the heat sink, three different temperature potentials are realized: the temperature difference between the heat sink and the coolant, the temperature difference between the two heat exchangers and the temperature difference between the heat source and the coolant.

The temperature distribution is determined as a function of the expected thermal conductance of the heat exchangers, as shown in Figure 2. The average coolant temperature is set close to the volume temperature or close to the air (heat sink) temperature depending on the efficiency of the heat transfer mode. If the cooled fluid is a gas, one can expect that the heat transfer modes at both ends of the cooling chain are similar (30 W/m²/K). In this case the average coolant temperature will be set at half of the total temperature range. If the cooled fluid is a liquid, the heat conductance will be higher at the heat source side of the cooling chain (about 200 W/m²/K). For such a case, it is recommended to set the average coolant temperature at 2/3 of the total temperature range.

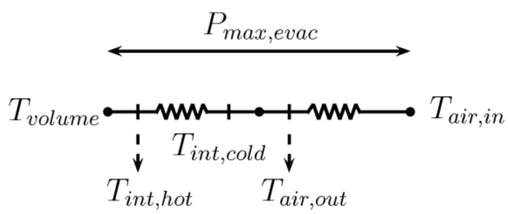

Figure 2. Temperature distribution.

Another attention point is the total temperature range. For common active cooling systems, the global conductance is as high as 10 000 W/m²/K while in natural circulation mode it drops as low as 30 W/m²/K for air heat exchangers, provided that a frontal velocity of 1 m/s is guaranteed. Therefore, for the same heat load and total temperature potential, the heat transfer areas have to be multiplied by a factor of around 300. The passive cooling systems are thus limited to applications with either a relatively low heat load (up to 1 MW) or a high temperature potential. The latter consideration often limits the use of such passive systems to incidental or accidental conditions in which the maximum temperature of the cooled volume is allowed to increase. During normal operation conditions in which the occurring temperature potential is lower, the passive system is supported by active components.

Finally, the cold source (heat sink) temperature has to be minimized in order to increase the total temperature potential. For this, the inertia of the global cooling system and its dynamic behavior is of outmost importance: the higher the thermal inertia, the higher the cut-off frequency of external temperature variations. Passive systems usually have a higher thermal inertia than active cooling systems due to their larger size. This is especially true when the cooled volume is a large liquid volume (e.g. spent fuel pools).

Figure 3 shows the temperature difference frequency response of a large liquid volume (10 000 m³) cooled by a passive cooling system. This frequency response is obtained by applying a variable heat sink temperature as boundary condition and by computing the resulting cooled volume temperature variation. A sinusoidal variation of the heat sink temperature with a variable frequency and fixed amplitude of 20 °C has been chosen. The outcome of this analysis is the percentage of the external temperature variation which is transmitted to the cooled volume as a function of the variation frequency. The results show that the daily temperature variations are cut off by 85%. For the frequency corresponding to a sinusoidal period of 1 day (straight vertical line) only 15% of the amplitude is transmitted. Therefore if the thermal inertia is sufficiently high, these systems should be designed based on an average temperature over a few hours instead of on an external peak temperature.

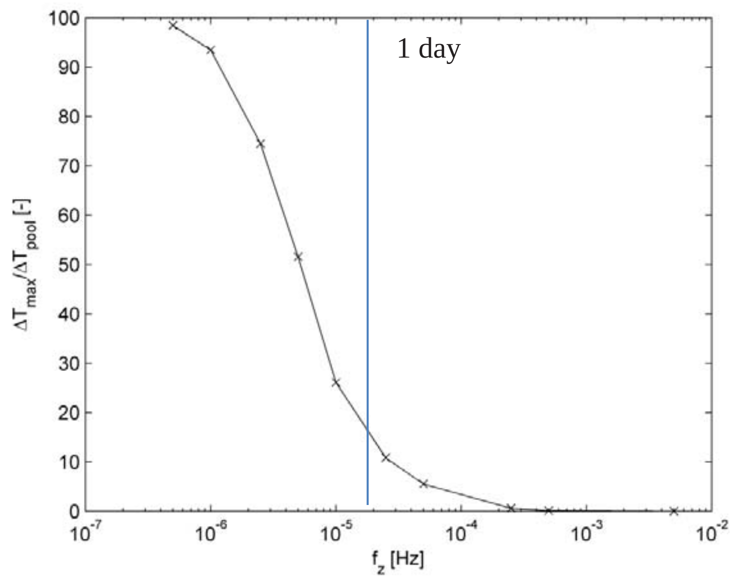

Figure 3. Passive cooling system frequency response.

4.3. Intermediate Cooling System

To show the order of magnitude of the expected coolant temperature differences and system dimensions, Table II gives an example of the operating points and needed piping diameters of a passive cooling system aiming at cooling a certain liquid volume. The fixed parameters are the ambient temperature (20°C), the height difference (10 m) and the heat load (1 MW). When the cooled volume is a liquid, the average temperature of the coolant is set closer to the maximum cooled volume temperature. Only the steady state values are given, therefore the size of the cooled volume is not relevant.

Table II. Example of passive cooling system general dimensions for an ambient temperature of 20°C

Intermediate circuit cold Temperature	Intermediate circuit hot Temperature	Cooled volume Temperature	Diameter for 1 MW
34°C	37°C	40°C	0.45 m
48°C	56°C	60°C	0.25 m
60°C	69°C	80°C	0.22 m
74°C	93°C	100°C	0.14 m

These results show that:

- Such passive systems are able to remove a heat load up to 1 MW provided that the available height difference suffices (10 m). If the available height is lower, the necessary piping diameter will increase;

- Increasing as much as possible the total temperature potential is beneficial. In this example, the piping diameter decreases by a factor of 3 by increasing the total temperature potential by a factor of 4;
- The system is more efficient at high temperatures: by decreasing the flow area by a factor of 9 (diameter by a factor of 3, between 0.45 m and 0.14 m), the coolant temperature difference increases only by a factor of 5.5. In other words, the available driving head for a fixed temperature difference increases with the average temperature as a consequence of the increase of the coolant thermal expansion factor. This is another argument driving the design towards the high temperatures.

4.4. Treatment of uncertainties

Both natural circulation loops and heat exchanger designs determine the value of the system global resistance defined in equation (1). Uncertainties on both the evaluation of the natural circulation flow rate and the heat exchanger efficiency result in a global resistance uncertainty.

In order to compensate for such uncertainties, conservative margins are taken for the pressure losses accounted for in the thermal-hydraulic model and for the heat exchange area in the heat exchanger design.

5. APPLICATION TO THE MYRRHA HOT CELL COOLING SYSTEM

The presented concept of a passive cooling system is applied to the MYRRHA hot cell cooling system. The hot cell is a confined room where the spent fuel elements are handled and temporarily stored. The atmosphere of the hot cell is almost pure nitrogen. The necessary cooling systems shall remove a heat load of 55 kW with a total temperature potential of 30°C in accidental conditions. The total available height difference for the whole system consisting of an intermediate loop and dry air coolers in a chimney is 60 m.

The first step is to set the height difference distribution and the temperature distribution between the two cooling systems taking into account the considerations presented in 4.2. A height difference of 12 m is allocated to the intermediate loop and of 48 m to the chimney. For the temperature distribution, as the cooled fluid is a gas (nitrogen), the average temperature of the coolant should be set at half of the total temperature range. However, the expected gas velocity over the outer surfaces of the heat exchanger is higher for the dry air coolers, due to the chimney natural draft, and so the heat exchanger efficiency is higher as well. As a consequence the average coolant temperature is slightly shifted towards the cold heat source temperature.

5.1. Intermediate Cooling System

Table III gathers the results obtained for the intermediate loop for different piping diameters. A diameter of 2” induces too high pressure losses. With such a diameter, the intermediate loop’s hot temperature reaches the limit of 70°C for a heat load lower than the design value of 55 kW. Increasing the piping diameter decreases the coolant temperature difference and has thus a positive effect on the heat exchanger size. However, there is a limit to this benefit: to further decrease the temperature difference by one degree, the piping diameter should be multiplied by 2. Therefore, a piping diameter of 8” is selected.

Table III. Example of passive cooling system general dimensions

Piping diameter [in]	Intermediate circuit cold T°C [°C]	Intermediate circuit hot T°C [°C]	Mass flow rate [kg/s]	Heat load [kW]
2	55	70[1]	0.49	26
4	55	62.9	1.98	55
6	55	59.0	3.93	55
8	55	57.4	6.18	55

The pressure loss distribution for the intermediate loop is given in Table IV. The pressure losses must be extremely low to ensure the adequate coolant flow rate. These low pressure losses are obtained by a design leading to low velocities in the different parts of the loop.

Table IV. Pressure loss distribution for intermediate circuit

Device/component	Pressure loss [Pa]
Heat exchangers	237
Lines	49
Others	24
Total	310

5.2. Heat Exchanger Design

As explained above the design of the heat exchanger is based on a Reynolds number leading to turbulent heat exchange conditions. However, the value of this internal heat transfer coefficient is not limiting. The global conductance is driven by the highest heat transfer resistance (i.e. the external one).

For the dry air cooler, the external heat transfer coefficient is estimated at 30 W/m²/K provided that an air velocity of 1 m/s is ensured through the heat exchanger by the natural chimney draft. The design of the intermediate loop results in a temperature potential of 10 K for the heat exchange at the cold side. Therefore, the necessary heat exchange area is estimated at about 180 m².

For the heat exchanger immersed in the cooled volume, the external heat transfer coefficient is estimated at 5 W/m²/K as the heat transfer mode is natural circulation. The design of the intermediate circuit results in a temperature potential of 15 K for the heat exchange at the hot side. Therefore, the necessary heat exchange area is estimated at about 700 m².

5.3. Chimney Design

The chimney design has to ensure both sufficient air flow rate to limit the air temperature increase across the dry air cooler, and a velocity of 1 m/s to guarantee adequate heat transfer conditions. The design parameter is the diameter of the chimney which is set at 1.8 m. The obtained flow rate is 6 kg/s leading to

[1] For this case, the height difference of 12 m does not allow to transfer the requested heat load.

an air temperature increase of 10 K over the dry air cooler and an available driving head for the natural draft of 19 Pa. The pressure loss distribution for the air loop is given in Table V.

Table V. Pressure loss distribution for air circuit

Device/component	Pressure loss [Pa]
Dry air cooler	12
Chimney	3
Chimney outlet/inlet	4
Total	19

5.4. Active Devices for Normal Operating Conditions

The system has been designed for the accidental conditions in which a maximum temperature potential for heat load removal occurs. In normal operating conditions, the temperature potential is divided by a factor of 3 and the heat transfer conditions have to be improved to guarantee the maximum heat load removal without exceeding the temperature criterion in the hot cell.

The active devices needed are one ventilator blowing nitrogen on the hot cell heat exchanger and one ventilator for the dry air coolers. No pump for circulating the intermediate fluid is provided because the limiting conductance is at the outside of the heat exchangers. Therefore, increasing the intermediate fluid velocity will not significantly improve the heat transfer conditions. In order to meet the hot cell temperature criterion, the nitrogen velocity on the immersed heat exchanger has to be increased up to 0.7 m/s and the air velocity on the dry air cooler up to 5 m/s.

The normal operating point with the active devices available is detailed and compared with the accidental operating point in passive mode in Table VI. The temperature potential at both heat exchangers is reduced by a factor of 5 by improving the heat transfer conditions. The temperature difference for the intermediate loop remains unchanged as no pump is provided. However, the intermediate flow rate increases slightly when switching from active to passive mode because the temperature level increases which improves the natural circulation due to a better thermal expansion coefficient.

Table VI. Operating points for normal and accidental conditions

<table>
<tr><th>Location</th><th colspan="2">Temperature [°C]</th><th colspan="2">Mass flow rate [kg/s]</th></tr>
<tr><th></th><th>Accidental operating conditions</th><th>Normal operating conditions</th><th>Accidental operating conditions</th><th>Normal operating conditions</th></tr>
<tr><td>Hot cell (cooled volume)</td><td>70</td><td>50</td><td colspan="2"></td></tr>
<tr><td>Intermediate loop inlet dry air cooler</td><td>60</td><td>48</td><td rowspan="2">3.9</td><td rowspan="2">3.8</td></tr>
<tr><td>Intermediate loop outlet dry air cooler</td><td>56</td><td>44</td></tr>
<tr><td>Air hot</td><td>50</td><td>42</td><td rowspan="2">5.7</td><td rowspan="2">29.7</td></tr>
<tr><td>Air cold</td><td>40</td><td>40</td></tr>
</table>

5.5. Dynamic Behavior

The cooled nitrogen volume has a low thermal inertia. Hence it is a concern that under external perturbations such as thermal load increase and external temperature variation, the hot cell temperature may exceed temporarily its design basis limit of 70 °C. The two transients selected to assess the behavior in such cases, are an outside temperature step increase by 10°C and a heat load increase from 0 kW to the maximum design heat load.

The analyzed parameter is the hot cell reduced temperature, defined as:

$$T_{reduced} = \frac{T_{hot\ cell} - T_{amb,max}}{T_{hot\ cell\ steady\ state} - T_{amb,max}} \qquad (8)$$

The results of both cases are given in Figure 4 and Figure 5, respectively. Both transients have similar behaviour: the temperature of the hot cell undergoes several oscillations due to the fact that the temperature of the cold and hot legs of the intermediate loop does not increase at the same rate. Therefore the driving head, that is a function of the temperature difference, changes with time and so the intermediate flow and the heat exchanger conductances. However, the inertia of the cooling system consisting of the intermediate loop and the heat exchangers is sufficient to damp these oscillations so that the hot cell steady state temperature at full load and maximum external temperature is never exceeded.

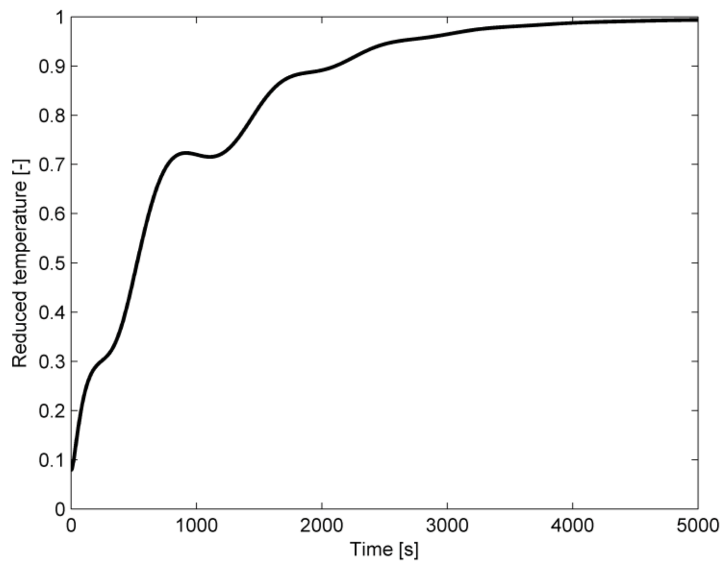

Figure 4. Hot cell reduced temperature evolution for an outside temperature increase of 10°C.

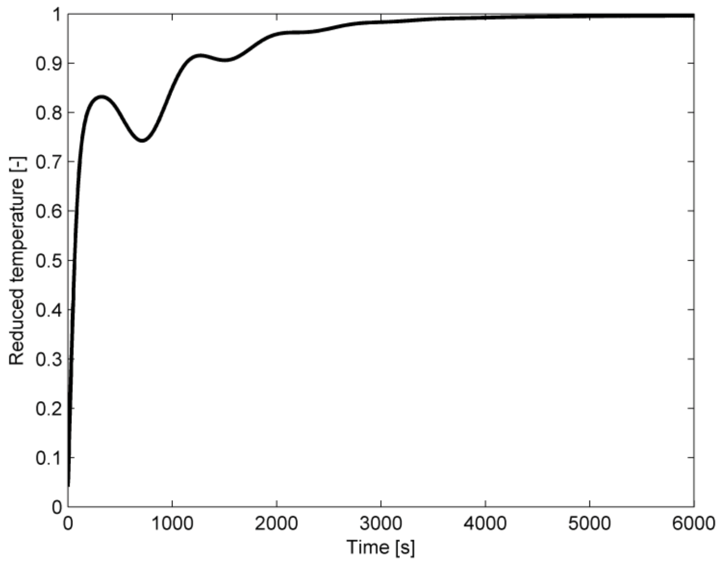

Figure 5. Hot cell reduced temperature evolution for heat load step.

6. CONCLUSIONS

The present paper presents a preliminary design approach for passive cooling systems with two loops. These systems provide adequate cooling and if needed, confinement. Therefore, these systems are well-suited for nuclear applications, especially the cooling of spent fuel elements for which the heat load is manageable.

These systems present different benefits such as their independence of an external power source and their high inertia making them less sensitive to boundary condition variations. As drawbacks, their large dimensions (heat exchange area or height difference) or their low heat removal capacity (up to 1 MW) in case of smaller dimensions are mentioned.

It has been shown that their dimensions can be significantly decreased by increasing the temperature potential, i.e. the difference between the maximum allowable cooled volume temperature and the maximum external temperature. These systems are therefore most valuable for accidental conditions. The heat exchange conditions have to be improved by active devices to meet the normal operating conditions that impose a more stringent criterion on the maximum temperature of the cooled volume.

To ensure a proper working, the design of the heat exchangers is crucial: they have to combine low pressure losses and a large heat transfer area. These constraints lead to a system with large dimension for which the fluid velocities are low.

Finally, the developed approach has been applied to the cooling system of the hot cell of the MYRRHA facility. The outcomes of this analysis are the general dimensions of the intermediate cooling loop, the heat exchangers and the dry air coolers chimney. It confirms the feasibility of such a system with reasonable dimensions provided that the heat exchangers with the needed minimum pressure losses and maximum heat transfer capacity are available.

NOMENCLATURE

C_p	Heat Capacity	[J/kg/K]
e	Thickness	[m]
g	Gravity Constant	[9.81 m/s²]
h	Conductance	[W/m²/K]
k	Conductivity	[W/K/m]
K	Pressure Loss Coefficient	[-]
L_c	Characteristic Length	[m]
Nu	Nusselt Number	[-]
P	Heat Load	[W]
Pr	Prandtl Number	[-]
q_m	Mass flow rate	[kg/s]
R	Thermal resistance	[W/K]
Re	Reynolds Number	[-]
S	Area	[m²]
T	Temperature	[K]
Δp	Pressure difference	[Pa]
Δh	Height difference	[m]
ρ	Density	[kg/m³]

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