CFD design and mock up test for heat removal using cylindrical rods mounted on a vertical plate.

Olivier BARDON*, Jérome BELLANGER*, Nasser ZAHRI**

*TN-International, AREVA Group

Abstract

The increasing of burn up spent fuel leads to substantial residual power. To maintain a reasonable cooling time before transport, casks with high dissipative capacity need to be developed.
In order to achieve thermal dissipation of about 70 kW in a 12 PWR fuel cask, the outer surface of the cask must be equipped with high efficiency fins. In general, heat removal from nuclear casks at high heat load is achieved using various types of fins working in natural convection. The fins shapes are usually limited by manufacturing considerations. The improvement of automation associated with the electric capacity discharge process for welding pin cylinders led us to examine the thermal performance of using long copper cylindrical rods to make a large exchange area at the outer surface of the cask. The main technical interest is that it offers good thermal performance regardless of the general orientation of the cask, which is an important point to be addressed after an accidental drop and a question generally raised by safety authorities.
In order to get a quick preliminary design of a rod's density and arrangement to reach a target value of global heat exchange performance, model and tests were conducted with horizontal cylindrical rods welded on flat vertical surfaces. Computational Fluid Dynamic preliminary design was a good aid for matching both vertical and horizontal orientation constraints, as well as assessing relatively good predicted values of global heat coefficient. .

1- Introduction

To dissipate almost 70 kW from high burn-up fuel in a 12 PWR fuel transportation cask, the outer surface of the cask must be equipped with a high efficiency fins surface. As thermal performance being directly related to radial fin length, the main constraint is the maximum diameter available around the cask which can be seen as the maximum air volume available to cool the fins surface. This constraint being setled, it is necessary to find the optimal distribution of fins inside this volume which maximizes the global heat conductance to air. But maximizing performance in a transport configuration may often mean reducing its performances in another configuration (for example vertical positionning after ACT (accidental condition of transport)) and this is a point to be adressed. Fortunately, the criteria are often different in NCT (normal conditions of transport) or ACT: limitation of the heat load is often due to the low temperature criteria of neutron absorbing material in NCT while after ACT and fire scenario this material is often removed for shielding analyses and limitation may be reported on higher temperature criteria like gaskets. This is why a good design would maximize the performance in a tranport configuration while satisfying necessary criteria in ACT.

Knowledge on fin's cask performances is restricted to certain geometries: axial and radial fins. Axial fins are mainly used on storage concepts where this geometry is more fitted to an ascending flow, while circumferential fins are more common on transport cask. However axial fins are sometimes used on transportation casks. This is not an optimal configuration as it tends to slow down heat exchange at the bottom of the cask while the top is correctly cooled. The consequence is an early limitation of the maximum heat load admissible in the transport configuration. The need to find a fin's geometry which is well adapted for each orientation cask (horizontal/vertical) motivates the investigation of a cylindrical rod's geometry. The improvement in the capacity discharged welding process (automatisation) further reinforces on the interest to investigate this novel heat transfer solution.

2- Theoretical and CFD design of rod's arrangement

a. Theoretical results

It is a well known result that there exists an optimum number of fins in a fixed volume that maximizes the global thermal conductance to the air. The main reason for this optimum comes from the competition between heat exchange area and global mass flow rate. For a large area in a fixed volume, the air flow inside the fins drops to nothing so that the global conductance tends to a minimum value. At the opposite extreme, when so few fins are present, the heat exchange area tends to a minimum value corresponding to a vertical plate so that the global conductance also tends to a minimum value. This reveals an existence of a maximum in the global conductance between these two limits. Bar-Cohen and Rosenow in <1> were the first to propose a general correlation for vertical heated parallel plates. Bejan <2> proposed an efficient analytic method to the general problem by solving each asymptotic solution and then intercepting them to find the optimum. The expressions for isothermal cylinders with an equilateral triangular pitch and in laminar convection are:

$$\frac{S_{opt}}{D} = 2.72\left(\frac{H}{D}\right)^{1/3} Ra_D^{-1/4} + 0.263 \qquad (1)$$

Optimal spacing

$$\frac{q_{max}}{D} = 0.448\left[\left(\frac{H}{D}\right)^{1/3} Ra_D^{-1/4}\right]^{-1.6} \qquad (2)$$

Maximum volumetric heat flux

The curve above based on the correlations (1) and (2) lead to a global heat coefficient of about 7 $W/m^2/K^{4/3}$ for a 7-8 mm diameter rod. The optimum equilateral pitch is about 32 mm, i.e a ratio of pitch/diameter about 4 to 5.

b. Arrangement of triangular pitch by CFD calculation

Due to differences in the thermal criteria in NCT and ACT it would be too constraining to design the cask with the same pitch in horizontal and vertical positioning. It would be better to get improved performances from horizontal positioning than vertical orientation. The general equation <1> and <2> indicates that for a rod diameter there is an optimum of the distance between rods but do not indicate the sensitivity of the global conductance in the vicinity of this optimum. This is an important indication for the optimisation of the numbers of rods to be used, which is directly linked to the cost of the solution. Another point is to generalize the study with differential pitches in vertical and horizontal orientations (equilateral configuration is a particular case with the constraint y pitch = $3^{1/2}$ (x pitch)) in order to allow a design to be adapted to different criteria in horizontal and vertical directions.
In order to get quick indication of the optimum geometries, a simplified model is built. This concerns a vertical copper plate of the diameter of the cask (about 2m) in height. Cylindrical copper rods are mounted on the vertical plate.

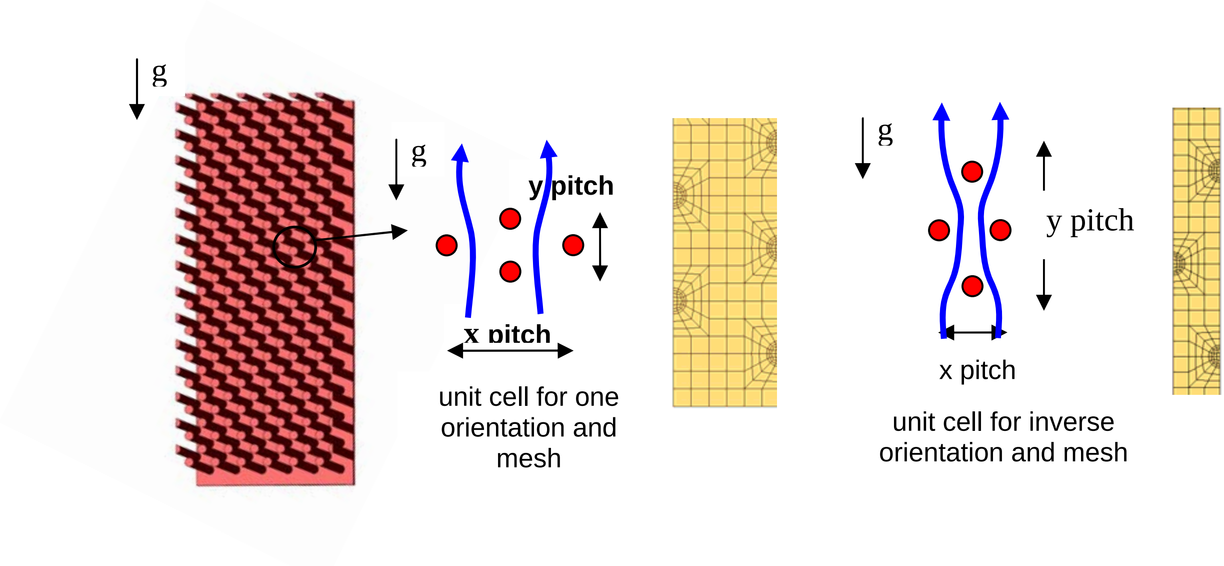

Fig.1. Element geometry for horizontal and vertical orientation

The thermal-fluid code used is ESC developed by MAYA HTT. The model is constructed in 3D. The mesh is obtained from a repetitive volume separated from two fluidic symmetry conditions and then repeated in height to obtain the overall vertical height tested. In order to have the same mesh in the vicinity of the rod, the mesh around the rod is built for the minimum pitch configuration and remains unchanged for the other pitches. The studied parameters are the horizontal pitch to diameter ratio (x pitch)/d and the vertical pitch to diameter ratio (y pitch)/d. The configurations tested are (x pitch)/d = 2 to 10, y pitch = 2 to 6. The rod diameter is 8 mm. The height of the vertical plate supporting the cylinders is 2 m. The model considers the fin rods perfectly welded on a 2 m vertically oriented copper plate of 12 mm thickness. Calculations are performed with classical k/ε turbulent model with a high Reynolds law functions.
The ambient temperature is fixed at 38°C, 2500 W/m2 is imposed on the back of the plate. For the purpose of the study, only convective heat transfer is calculated. We assume in this that radiative dissipation does not change the general conclusions.

Calculations are carried out by fixing the y pitch first and varying x pitch value in the model inside corresponding symmetric vertical plans. Results lead for the same imposed heat flux density at the back of the plate, to different maximum or average temperatures of the plate, which means different performances in global coefficient of heat convection h_g.
Assuming a variation in term of $h_g = A\ \Delta T^{0.333,}$ $A = \Phi_{conv} / \{S_{plate} x(T^*\text{-}Tamb)^{4/3}\}$ with $T^* = T_{max}$ for A_{min} and $T^* = T_{averaged}$ for $A_{averaged}$. "A" has a very slight dependency with T_{film}. "A" can be expressed as:
A = alpha K η where K is the ratio of total developped area / plate area, η is the fin efficiency, and alpha refers to thermophysical properties as alpha =cte $x[\lambda(T)\ (g\beta/\nu\alpha)^{1/3}]$.
The convection efficiency of each solution is then presented by the ratio A/A* where A* is the global coefficient of natural convection for a vertical flat plate of the same height. This ratio directly expresses the global improvement compared to a flat vertical plate of the same height.
The results are presented below.
The curves reveal the existence of a maximum of the conductance for each vertical pitch configuration.

The maximum performance increases when y pitch decreases, but the optimum becomes sharper.
It is interesting to define the optimal path (fig 3 left) for a horizontal position (geometric path of the maxima of the curves) and for this optimal path assess the resulting performance when reversing x with y for the same height (2m).This can be done by extrapolation of the previous results. Curves are presented in Figure 3, which compares performance at optimum for horizontal orientation and resulting performance in the vertical orientation. The curves intersect for y pitch = x pitch = 5.4 for which obviously changing orientation leads to the same configuration and performance

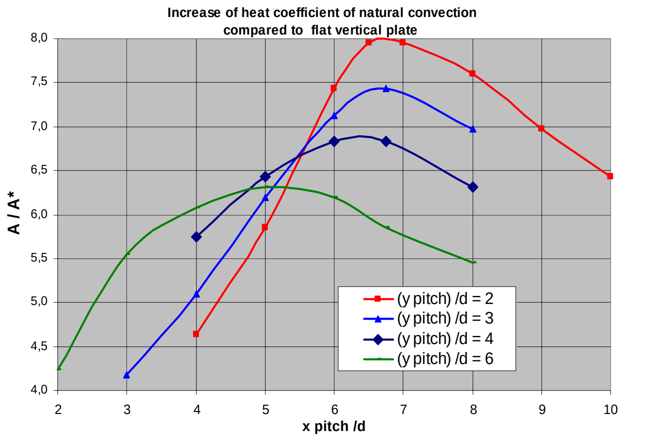

Fig.2. Increase of heat convection coefficient of natural convection (reference = vertical flat plate)

In comparison to this reference point, each increase of performance is almost compensated by a same decrease in the reverse configuration. In the same time, due to the rapid decrease of y pitch, the increase in horizontal performance is accompanied with a rapid increase in the rod's density. The best compromise for the design seems to be between 6 to 7 for (x pitch /d) which corresponds to 4.6 to 3 for (y pitch)/d.

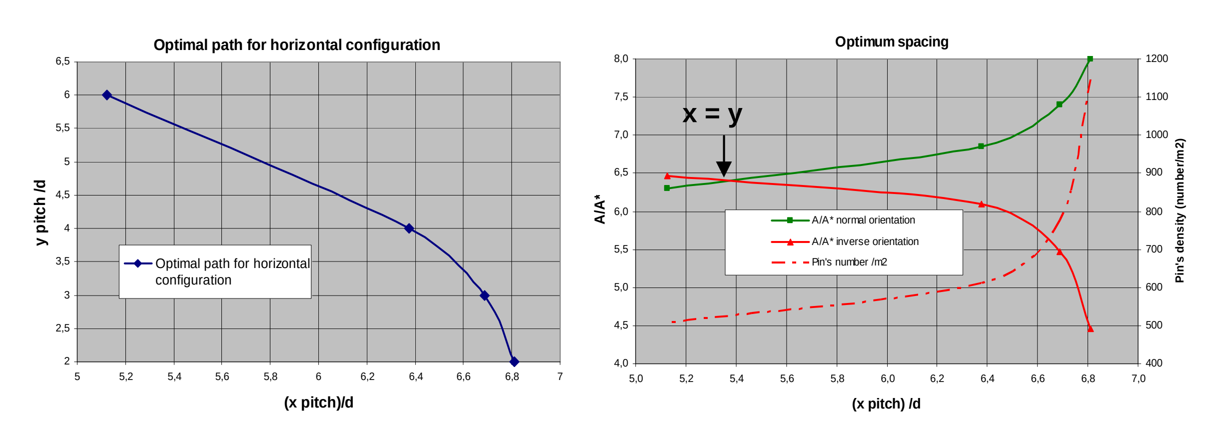

Fig. 3 Optimal path (left) and performances at optimum spacing (right)

For the manufacturing of the mock up we considered the curve (y pitch)/d =3 for which the two configurations close to the optimum (x pitch)/d = 6 and (x pitch)/d = 7 are built.
The expected results for the chosen configurations are presented below.

"A/A*": increase of heat coefficient of natural convection compared to vertical flat plate* *based on T_{max}	(y pitch)/d		
(x pitch)/d		3	6	7
	3		**3.6**	**-**
	6	**4.9**		
	7	**5.6**		

"A/A*": increase of heat coefficient of natural convection compared to vertical flat plate* *based on $T_{averaged}$	(y pitch)/d		
(x pitch)/d		3	6	7
	3		**5.5**	**-**
	6	**7.1**		
	7	**7.4**		

Fig. 4. Calculations results for selected configurations

In term of thermal performance, calculation shows a reduction of about -20% when orientation is changed from horizontal to vertical.

3- Thermal tests

a. Description of the mock up and the instrumentation

The mock up consists of the assembly of four copper plates of about 500 mm x 500 mm and 12 mm thick, to form a continuous surface of 2 m high in the vertical dimension and 0.5 m wide in the horizontal dimension. Fin copper rods are electrically welded at the surface in respect with the

Nevertheless, due to the junction of the plates it cannot be possible to respect the exact heat exchange area used in the calculation. The difference is about -6%. Moreover, standard diameter of the fin pins is 6 mm compared to 8 mm. Calculations have been redone for chosen configurations and show very slight decrease of the global conductance. Configurations are now relatively to the new diameter y pitch/d = 4 and x pitch /d = 8 and 9.3. A special support is made to hold vertically the mock up. The back of the plates is mounted with four flexible heating mats, stuck at the surface. Each one can deliver a 1 kW electrical power which is powered controlled. Inside each plate four 1.5 mm thermocouples are inserted in four holes at 5 mm depth

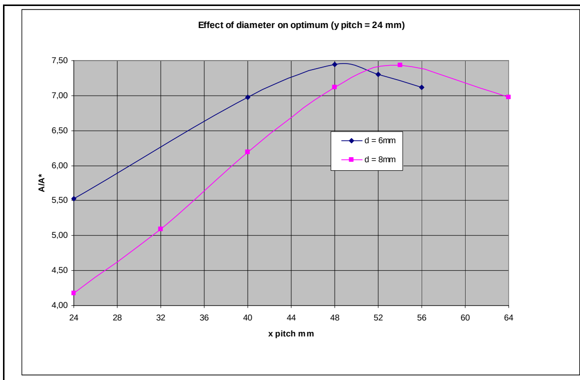

Fig. 5. Effect of diameter on chosen configuration for mock up

To limit heat losses, a 300 mm glassfoam insulation is used at the back of the assembly and promatech is used along the lateral sides. The “promatech insulator” is also designed as a deflector to prevent any lateral air flow. Its inner surface is coated with aluminium foil to avoid lateral radiation losses. For a best estimate of direct thermal radiation loss, the copper plate and fins rods are black coated with a black paint having known high emissivity. Two air temperature sensors are used at the front and the back of the mock up. They are protected against infrared radiation by a metal foil of low emissivity. To complete the analysis, infrared thermography is used and images of the stabilized temperature field are carried out for one case.

Fig. 6 Photos of the mock up thermal test

b. Tests results

Different tests were conducted for each of the following configurations. Note that the reverse configurations were obtained by rotating the four copper plates by 90°. Tests were performed for a 2500 W/m2 heat flux and for 5°C ambient temperature.

List of tests conducted:

		Vertical (y pitch)/d		
		4	8	9,3
Horizontal (x pitch)/d	4		X	X
	8	X		
	9,3	X		

Thermocouples temperature profiles

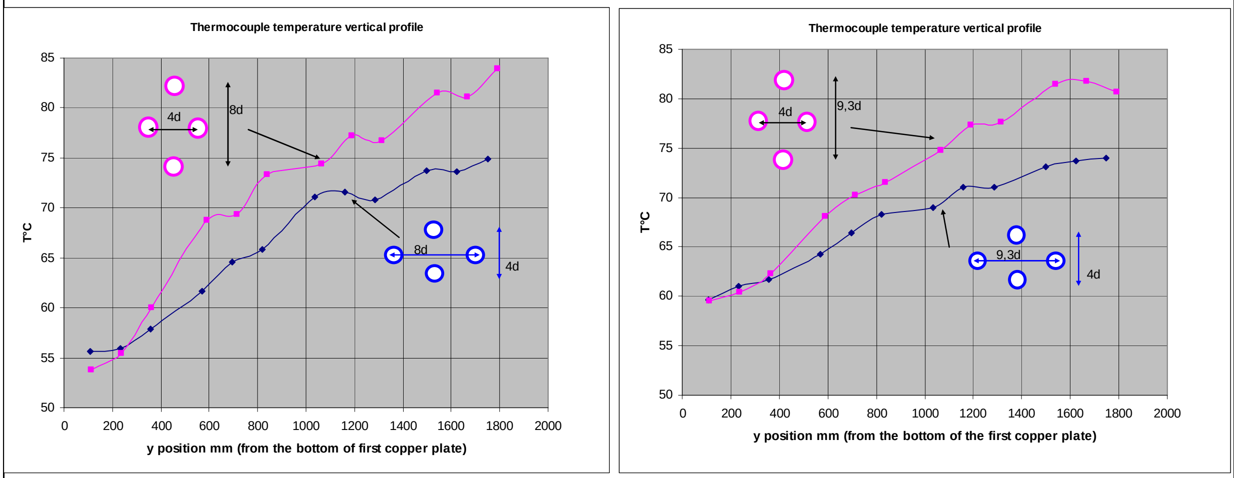

Fig 7 Comparison of test results (4d/8d) left and (4d/9.3) right

The comparison of the performances of the different configurations shows:

- a slight effect for the same vertical (y pitch/d) = 4 to increase horizontal pitch from (x pitch)/d = 8 to 9.3 (comparison of the two curves left and right of figure 7 “diamond” points). It is possible that the optimum is in between,
- slight effect for the same horizontal (x pitch)/d = 4 to increase vertical (y pitch/d) from 8 to 9.3, (comparison of the two curves left and right of figure 7 “square” points)
- an important effect for rotating each configuration : as expected, there is a significant reduction in performances when the smaller pitch in the x horizontal position and the larger pitch is in the vertical position.

Infrared thermography

Infrared thermography was used for the first case (x pitch)/d = 8 / (y pitch)/d = 4 left picture) and the reverse configuration (x pitch)/d = 4 / (y pitch)/d = 8). The thermal profiles show the same difference as the thermocouples (about – 6 to -8°C) of Tmax for the (x pitch/d) = 8 / (y pitch)/d= 4) compared to the reverse one.

4- Comparison to calculation

a. Comparison in switching pitches

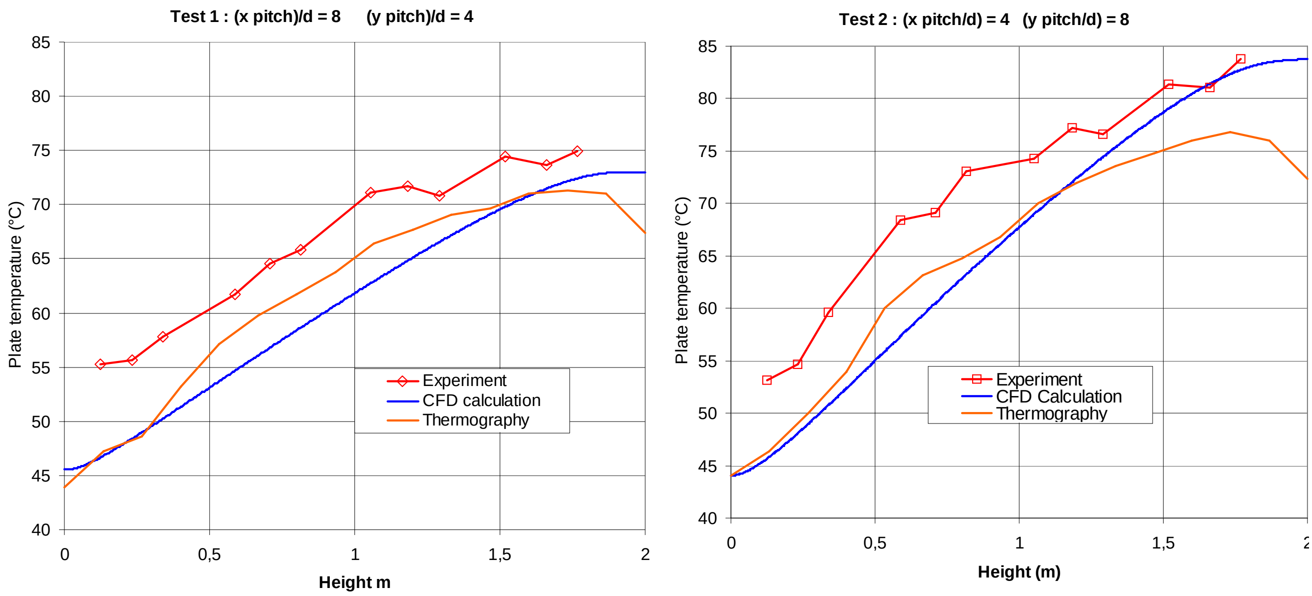

Fig. 7 thermal profile of test 1 and test 2

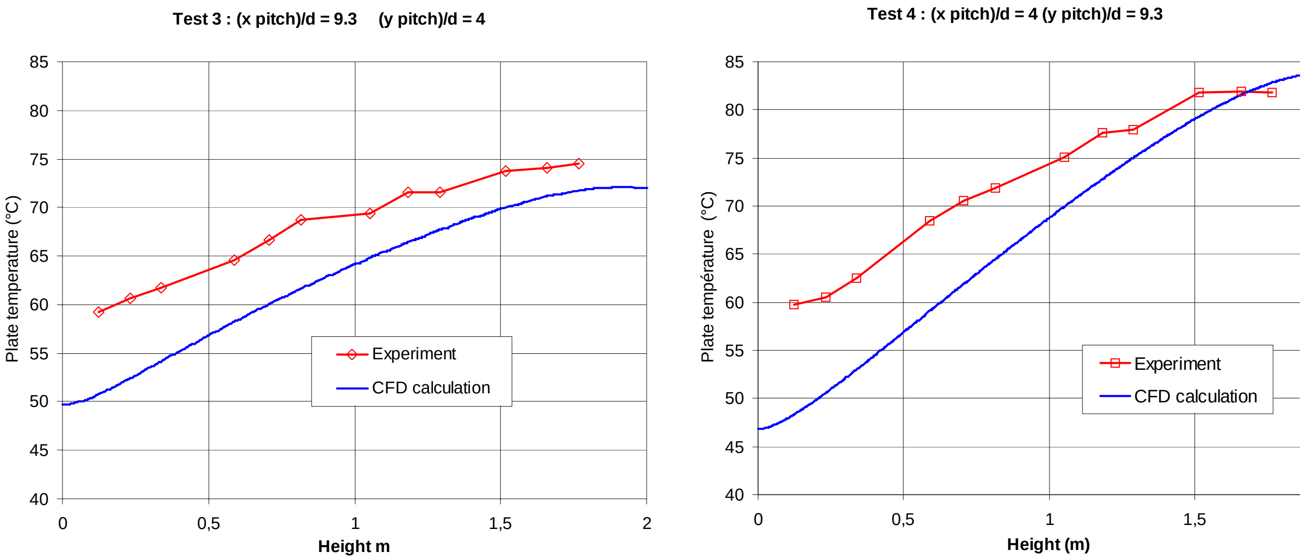

Fig. 8 thermal profile of test 3 and test 4

The CFD calculation over-estimates the heat convection exchange even if the global tendency is correct. The reversed situation shows clearly the increase in temperature. Both calculation and experiment show this effect even if CFD calculation seems to over estimate the effect (+8°C at mid height for CFD compared to +5°C for experiment)

b- Heat convection coefficient

The contribution of radiation is removed to the energy balance to get the average convective flux. A heat loss of about 5% maximum is considered. Global coefficients of natural convection for experiments are then deduced by a thermal balance with the measured temperature.

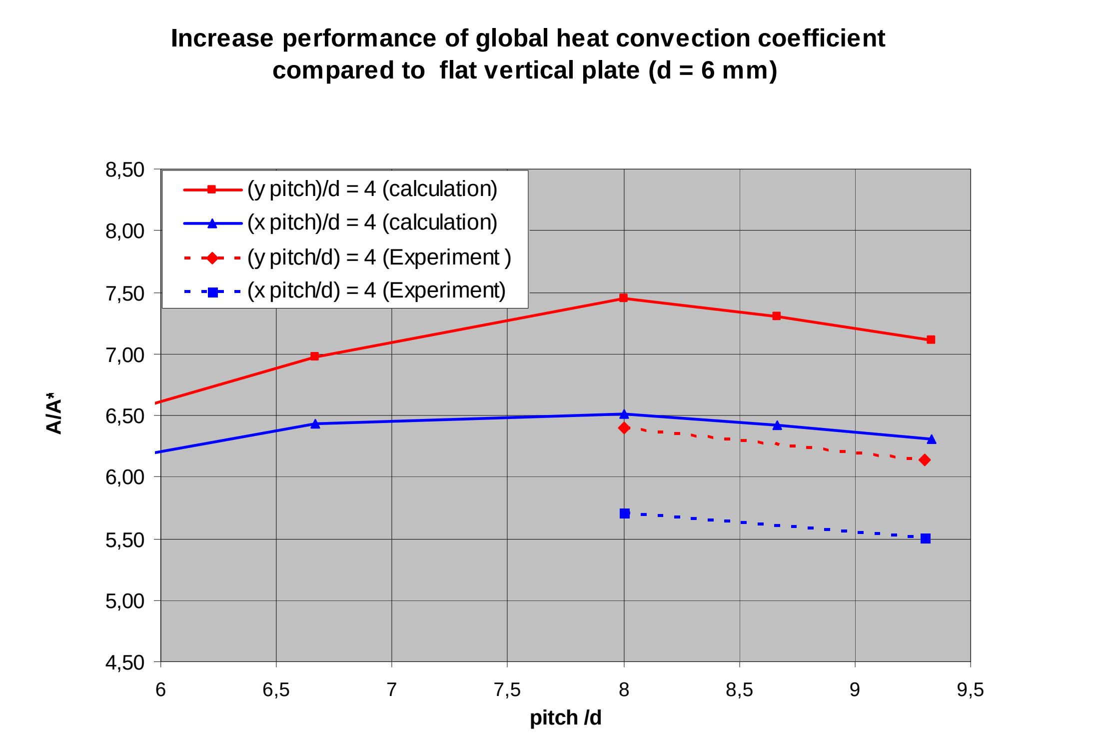

Fig. 9 Comparison with calculation of A/A*

Comparisons of calculations with tests show:
- almost 15% overestimation of heat transfer coefficient “A” in the calculation,
- a very similar slight variation of the coefficient “A” compared to pitch over 8d meaning the maximum is very close,
- a similar effect of reduced performances when switching horizontal and vertical position, although effect is larger in the calculation (30% of effect compared to 20% in experiments).

Calculation

“A/A*”: increase of heat coefficient of natural convection compared to vertical flat plate* *based on $T_{averaged}$		(y pitch /d)		
		4	*8*	*9.3*
(x pitch /d)	*4*		**6.5**	**6.3**
	8	**7.45**		
	9.3	**7.1**		

Experiment

“A/A*”: increase of heat coefficient of natural convection compared to vertical flat plate* *based on $T_{averaged}$		(y pitch /d)		
		4	*8*	*9.3*
(x pitch /d)	*4*		**5.7**	**5.5**
	8	**6.4**		
	9.3	**6.1**		

Fig 10

The over estimate of the calculation compared to the real test may come from several explanations:
- from the heat losses which are not exactly precisely known but can be estimated to be less than 5%
- from the code itself: mesh sensitivity has not been fully investigated,
- from a potential thermal contact resistance between rods and plate which could explain the difference in temperature between thermography and thermocouples (the 5°C mean difference between front and back measurement seems large for the copper plate) and could explain the higher heat transfer coefficient in the calculation. It was shown in the analysis of certain cut welded junctions, that some areas were not correctly welded.

However, the relative agreement of the calculations and the measurements is acceptable for the validation of the thermal design.

5- Conclusions and further developments

In order to define package solution for either horizontal or vertical cask orientation, a thermal design using long copper rods mounted radially on the outer surface of the cask is investigated.

Computational fluid dynamic (CFD) is used to assess the performances in a large range of pitches in order to define the optimal configurations matching constraint criteria in normal transport condition (horizontal configuration) or in accidental transport condition (vertical configuration).
As expected, each curve at y pitch fixed reveals the existence of a maximum of heat performance with x pitches. Relative to the reference case y=x pitch configuration with the same performances in the both orientation, the gain obtained in improving horizontal performance leads to almost the same reduction in vertical orientation. The best configuration must be chosen with respect to both criteria (horizontal / vertical).
The CFD analysis was compared to real thermal tests which confirm the general trends in the calculations. While the CFD analysis over predicts the global heat convection coefficient, the general tendencies and the influence of orientation were nearly identical. An adjustment of the CFD model compared to the experiments would be useful for a better prediction.

Further development of this design will be carried out through industrialization of the solution.
To further this study, a cylindrical mock up will be built in order to confirm the previous result with a more realistic geometry and manufacturing process.

REFERENCES

<1> A. Bar-Cohen and W. M. Rohsenow, Thermally optimum spacing of vertical, natural convection cooled, parallel plates, J. Heat Transfer 106, 116 - 123 (1984)

<2> A. Bejan, Convection heat transfer, Wiley

NOMENCLATURE

alpha : coefficient of natural convection $W/m2°C^{4/3}$
A : global coefficient of natural convection $W/m2°C^{4/3}$ $= \varphi_{conv} / \Delta T^{4/3}$
A* :global coefficient of natural convection $W/m2°C^{4/3}$ for a vertical flat plate of the same height
h: coefficient of heat convection ($W/m^2.K$)
h_g : global coefficient of heat convection (W/m2K)
d or D : rod's diameter mm
H : height (m)
g : gravitational acceleration
K : coefficient of surface increase
Ra : Rayleigh number $(g\beta/\nu\alpha)\Delta T\ H^3$
S : surface (m^2)
T : temperature (°C ou K)
Tamb : ambient temperature (°C ou K)
Tfilm : film temperature = (Tamb+T)/2
ΔT : temperature difference (°C ou K)

Greek symbols
α : thermal diffusivity ($=\lambda/\rho Cp$)
β : coefficient of thermal expansion (K^{-1})
ε : emissivity
Φ : thermal power W
φ : flux density W/m^2
λ : thermal conductivity (W/m.K)
ρ : density (kg/m^3)
σ : Stefan Boltzmann constant ($5{,}674.\ 10^{-8}\ W/m^2.K^4$)
ν : cinematic viscosity (m2/s)
η : Fin or rod efficiency