HEFAT2007
5th International Conference on Heat Transfer, Fluid Mechanics and Thermodynamics
Sun City, South Africa
Paper number: DB1

MIXED-CONVECTION FROM A BUNDLE OF HEATING CYLINDERS IN A CROSS-FLOW AIR-CIRCULATION. EXPERIMENT AND ANALYSIS.

B Duret [1], JC Bonnard [1] and S Bournaud [2]
[1]CEA DEN 17, Rue des Martyrs, F-38054 Grenoble Cedex ,
[2]EDF R&D, Dept. MFEE, 6 Quai Watier BP 49, F-78401 Chatou Cedex,
France,
E-mail : bernard.duret@cea.fr

ABSTRACT

Within the framework of radioactive waste management, the VALIDA program started to provide reliable data for the validation of numerical tools used to model the cooling of spent nuclear fuel containers in dry storage facilities. The design of such facilities implies thermal-hydraulic calculations in order to predict containers and wall temperatures. One has to make sure that these temperatures never exceed critical values. The understanding of mixed-convection flow in realistic conditions and more particularly the interaction between a global cross-flow circulation and local natural convection effects is a key point of these design studies. VALIDA experiments were carried out in this way at CEA on a multi canister configuration (7 rows heated tube bundle mounted vertically) in a special wind tunnel (length:12m, height: 3m width: 2m13) and cooled by a cross-flow air circulation. During the experiments, the air flow rate, the velocity profile and the heating power are controlled and have been adjusted to simulate various thermal-hydraulic conditions. A staggered tubes of seven rows of heated tubes (diameter 0.64m and height 2m) are placed in the wind tunnel, the 18 canisters arrangement use a triangular pitch (P/D = 1.66). Instrumentation includes thermocouples in the air flow, on the cylinders, and on the walls; the wind tunnel is rigged with two air-velocity measurement systems: LDV (Laser Doppler Velocimetry) and PIV (Particle Image Velocimetry). One presents the main experimental results reached with different values of the parameters: air velocity (0.25 to 1 m/s) and power density (300 to 600W/m²). From the downstream air measurements, a visualization of the temperature plume is obtained at different location behind the last tube. Measurements of air velocity are also performed with LDV laser in the air gap above the canisters. All the results show that the flow pattern of air strongly depends on the ratio of the buoyancy to the inertia forces. Convective transfers areas involving predominately forced or natural convection are distinguished thanks to established heat transfer correlations. A dimensionless buoyancy number Bo* is defined to characterize the experimental flow regimes obtained.

INTRODUCTION

This work has been developed within the frame of the French program on “Long Interim Storage” of high-level radioactive waste [1]. One of the main studied concepts is a large hall where cylinders, filled with warm radioactive waste, are stored and cooled by natural-air circulation. In such concepts, one has to demonstrate that wall temperatures never exceed limited values within reasonable safety margins.

The procedure applied to solve thermal-aeraulic problems of a storage facility requires calculations at different sophistication levels: from the simplest one-dimensional calculation to assess the overall performances of a concept (air flow and temperatures) to the most precise homogenous [2] or fine 3-D models for local heat transfer estimation [3]. Whatever the approach, these calculations require reliable experimental data, representative of physical phenomena involved, in order to validate numerical codes simulating cooling cross-flow. Velocity pattern, pressure drop and heat transfer data under steady-state conditions are needed, but few published work exists on heated cylinders networks, especially for those dealing with mixed convection [4].

NOMENCLATURE

D	[-]	Cylinder diameter
g	[m/s²]	Gravitational acceleration
Gr^*	[%]	Modified Grashof number ($g\beta\Delta Tqx^4/\lambda\nu^2$)
H	[m]	Canister height
h	[W/m²K]	Heat transfer coefficient
q	[W/m²]	Heat flux density
L	[m]	Characteristic dimension
Nu	[]	Nusselt number ($Nu= hL/\lambda$)
Pr	[]	Prandtl number ($Pr= \mu Cp/\lambda$)
Re	[]	Reynolds number ($Re= VL/\nu$)
Ri	[]	Richardson number ($Ri= g\beta\Delta TL/V^2$)
T	[°C]	Temperature
V	[m/s]	Air velocity
X	[m]	Distance between 2 consecutive tubes
Special characters		
β	[1/K]	Thermal expansion coefficient
ν	[m²/s]	Cinematic viscosity
λ	[W/m K]	Thermal conductivity
Subscripts		
FC		Forced convection
MC		Mixed convection
NC		Natural convection
l, t		Longitudinal and transverse
w		Wall
x		Local value

THERMAL-AERAULIC BEHAVIOUR OF INTERIM-STORAGE FACILITIES

According to the relative magnitude of the buoyancy forces due to the warming air, compared to the inertia due to its flow velocity, the overall thermal-aeraulic system can involve either natural convection or transverse forced convection, and more likely, a combination of both, i.e. mixed convection [4]. Even for very low Grashof numbers, hot air plumes can occur, arising from areas located in the wake of the tubes and rising to impact the ceiling of the storage area.

Actually, the ventilation system of an interim storage facility is designed in order to fit two thermal criteria. For optimum cooling, transverse air-flow circulation, characterized by a low Grashof number, should predominate. The cylinders, filled with warm radioactive waste, and the concrete structures cannot have temperatures exceeding some specific values to ensure a durable operation. Yet, special attention must also be given to compliance with the dry corrosion criterion after the thermal power of the package has decreased. Indeed, dry corrosion criteria require limited cooling when the residual heat in the packages is low, to avoid long-term corrosion

Hence, short-term and long-term requirements are opposed. Since the design of interim storage facilities has to fit both constraints, thermal-aeraulic behaviour of such facilities appears as a key point of investigation. Yet, very few available experiments exist and when available, they largely minimize the effects of natural convection and may not be suitable for an extrapolation to larger scales. Moreover, transient phenomena in turbulent flow (Van Karman instabilities, drag, etc.) have not been measured yet. VALIDA experiments, dealing with an arrangement of heated cylinder mounted vertically in a wind tunnel cooled by a cross-flow air circulation, have been carried out to fill this lack of experimental data.

EXPERIMENTAL FACILITY

The VALIDA mock-up is a special wind tunnel (length:12m, height: 3m width: 2m13) covering any flow configuration needing validation, regardless of the selected vault or cask storage concept.

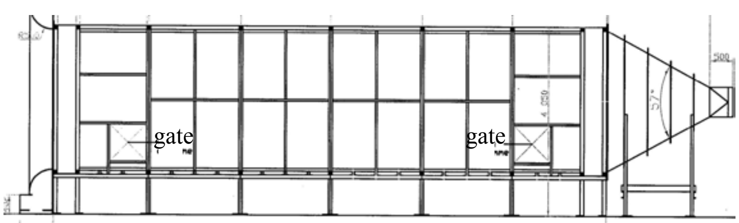

Figure 1 VALIDA wind tunnel

After studying one canister configuration [5,6], a new arrangement of staggered heated tubes (diameter 0.64m and height 2m) have been placed in the wind tunnel. Aeraulic boundary conditions in VALIDA were maintained in a large air stream fitted with pressure loss devices so that the upstream profile could be flat, with a better assessment of the input velocity. This requirement led to the addition of a head loss section at the intake and sheets of anti-turbulence fabric. The anti-turbulence device consisted in two square-pitched wire gauze.

The 18 canisters have been arranged as seven rows with a triangular pitch (P/D = 1.66). For each of them the inner wall is equipped with electrical wires aimed at reaching a uniform and well-known power density, and its external surface is polished up to mirror surface to eliminate radiation effect.

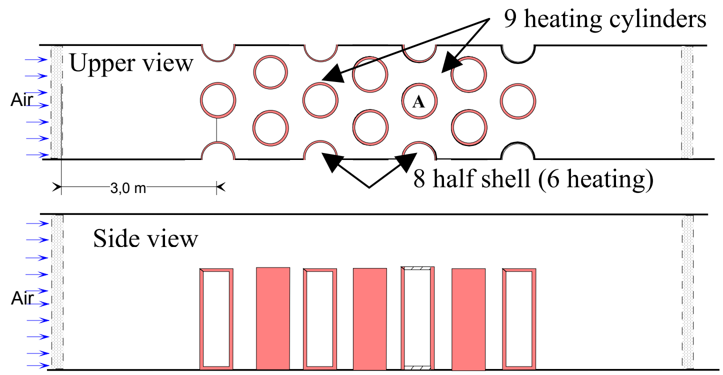

Figure 2 Sketch of the tube bundle

A special cylinder was equipped with thermocouples placed halfway through the thickness in holes having a 1.1-mm diameter, every ten centimetres on a single vertical line. This system, combined to a rotation of the cylinder led to the reconstruction of the whole temperature distribution of the cylinder with a quite fine precision. A movable grid system with 77 thermocouples was settled to investigate air temperatures in the plume. The grid could place the thermocouples from 10cm to 4m distance downstream to the last row. Finally, the instrumentation includes thermocouples in the air flow (90), on the cylinders (84), and on the walls (12); the wind tunnel is rigged with two air-velocity measurement systems: LDV (Laser Doppler Velocimetry) and PIV (Particle Image Velocimetry).

EXPERIMENTAL PROCESS

The cylinder environment was representative of the flow modes at study for interim storage. VALIDA test-cases covered mixed-flow regimes around the cylinder with a plume under the ceiling. The power density is 300 or 600W/m^2 for each of the canister, upstream air velocity (0.25 m/s to 1 m/s), could be adjusted in order to cover different flow regimes. Each time a new air velocity was imposed, one checked its profile using PIV and LDV. These velocity measures confirmed the low turbulence rates at the entrance of the test section (in the range of 2 to 3 %). The whole circumference of the A cylinder was explored with 32 radial positions, over a period of 3 to 4 hours for each position. A test-case period could basically last more than 4 days in order to reach thermal steady state. During each test-case, both the instrumented vertical line on the cylinder and the thermocouples on the movable grid in the plume could have their position changed. They respectively recorded the temperature of the A cylinder wall and the temperature of the air at different locations. Therefore, for each test-case, the following parameters were accurately measured :

- Air temperature fields in the plume behind the cylinder
- Cylinder wall temperatures around its circumference
- Others canisters wall temperature

The data acquisition for the grid's thermocouples used a slow recording as for the cylinders but also a fast one (acquisition frequency of 100 Hz), was performed sometimes for a 30- to 90-second duration. Thermocouple certificates of calibration gave an accuracy of ±0.22°C, but taking into account the whole acquisition system, the temperature measurement uncertainties were estimated at ±1°C. Electrical power and air flow are known with a 5% accuracy.

MAIN RESULTS

Six tests have been performed (q=300 and 600W/m² V_{air}=0.25, 0.5, 1m/s). Based on the measurements described above, a mean value was obtained after achievement of thermal stabilization. One obtained thus the whole air and cylinders temperature distributions.

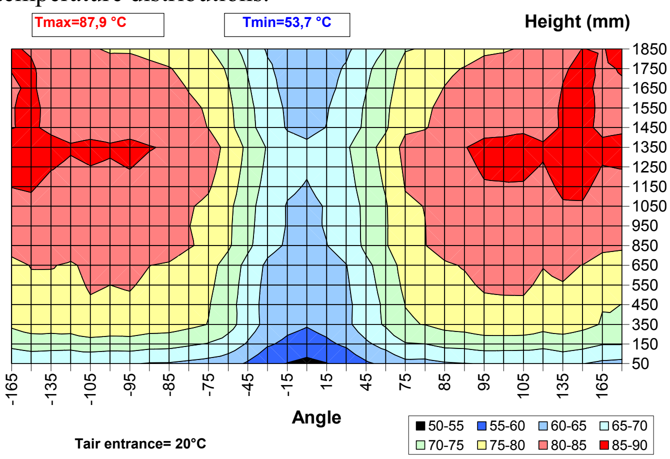

Figure 3 A canister temperature map (q=600W/m² V=1m/s)

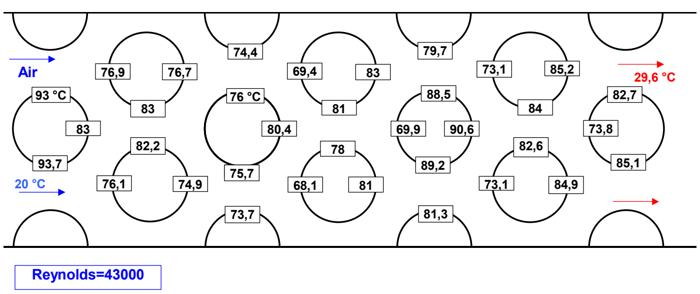

Figure 4 Wall temperature in the bundle (H=1m50)

For every case, a temperature map in the rear plume was measured by placing the instrumented grid at different location. Air temperature fluctuations were also measured (f=100Hz). Then, VALIDA test facility provided a large database for analysis.

EXISTING CORRELATIONS

Using a dimensionless approach and from the basic thermohydraulic equations, we identified dimensionless numbers for mixed convection: Re, Pr and Ri, the Richardson number (the inverse of the Froude number), representing the ratio of the buoyancy forces to the inertial forces. In this regard, the Richardson number can be considered as the ratio $(U_{nc}/U)^2$, U_{nc} being the natural convection velocity and U the transverse one. The initial difficulty is that the characteristic dimension is not the same for Re and Ri, in addition the Richardson number is function of ΔT, the temperature difference between bundle and air, but the latter is unknown before the test. This is the reason why other authors used a global Richardson number, based on ΔT heating across the test section and a characteristic length, L, equal to the height of the room. In our case, after each test, an average difference, ΔT, between bundle and air, can be calculated, from which the classical Richardson number ($Ri= g\beta\Delta TH\ /U^2$) can be derived. Test carried out at the CRIEPI [7] with bundles of small dimensions showed that when Ri<3 (Ri global), the forced convection regime is preponderant and the classical correlations of heat exchange across systems can be applied. If we represent the two approaches to Ri, the table below shows that the extent of the thermohydraulic regions studied in VALIDA is very wide.

			CRIEPI [7] approach		Classical approach	
Test case name M+q(W/m²)+V(m/s)	Re	Gr*	ΔT (°C)	Ri CRIEPI	ΔT (°C)	Ri
M600V1	42000	$3.6\ 10^{13}$	4.3	0.4	56	3.6
M600V0.5	22000	$3.3\ 10^{13}$	8.2	2.8	67	15
M300V1	39000	$2.2\ 10^{13}$	2.4	0.3	30	2.3
M300V0.5	21000	$2.1\ 10^{13}$	4.3	1.6	36	9.
M300V0.25	11000	$2.0\ 10^{13}$	9	13	37	37
M600V0.25	11000	$3.0\ 10^{13}$	17	24	69	66
M600NC		$2.7\ 10^{13}$			93	

Table 1 Tests matrix and associated dimensionless numbers

The wide range of characteristic parameters led to a relevant interpretation. From the above dimensionless numbers, the various hydraulic regimes and the heat transfers have been studied in laminar and turbulent flows by many authors and represented by correlations that we shall now describe. We shall work here on average exchange values.

Forced convection approach

The bibliography relating to forced convection around a bundle of cylinders is very extensive and rather old; we identified a reliable bibliographic survey performed by S Kakac [8]. For an hexagonal system of cylindrical tubes of diameter D, arranged in staggered configuration, the dimensionless pitches $X_t^*=X_t/D$ and $X_l^*=X_l/\ D$ are derived from distance between two consecutive rows (transverse distance X_t and longitudinal X_l). For VALIDA, we have X_t^* =1.65 and X_d^*=1.45.

Grimson [9] carried out tests in which the transverse and longitudinal pitches were modified and obtained correlations depending on X_t^* and X_d^*. Kacak [8] recommends, on the basis of many tests carried out by Zukauskas [10,11], the following correlations, which have the advantage of using analytical formulations for the influence of the transverses and longitudinal spacings.

Correlations (1)	Re_d
$Nu=0.71\ Re_D^{0.5}\ Pr^{0.36}\ (Pr/Pr_w)^{0.25}$	$510^2 < Re_D < 10^3$
$Nu=0.35\ (X_t^*/X_l^*)^{0.2}\ Re_D^{0.6}\ Pr^{0.36}\ (Pr/Pr_w)^{0.25}$	$10^3 < Re_D < 2\ 10^5$
$Nu=0.31\ (X_t^*/X_l^*)^{0.2}\ Re_D^{0.8}\ Pr^{0.36}\ (Pr/Pr_w)^{0.25}$	$2\ 10^5 < Re_D < 2\ 10^6$

The Reynolds number is being calculated at minimum flow area: the values of the VALIDA tests, which vary between 1.1 and 4.2 10^4 with the inlet air velocity are shifted to 1.8 10^4 to 7 10^4. We are thus still in the region of line 2 of the previous table, $10^3 < Re_D < 2\ 10^5$. Recently, Yovanovich [12] obtained a more widely applicable correlation:

$$NuD=C_1\ Re_D^{1/2}\ Pr^{1/3} \qquad (2)$$

with $C_1=(0.61\ X_t^{*0.091}\ X_l^{*0.091})/(1-2\exp(-1.09\ X_t^*)$

Note: the heat exchange develops from the first row up to a thermally and hydraulically stable region. To take this effect into account, the value of the number of Nu must be multiplied by a correction factor, C, which is obtained for a number of rows, Nl, less than 10 :

Row	1	2	3	4	5	6	7	10
Correction factor C	0.7	0.75	0.83	0.9	0.92	0.95	0.97	1

Natural convection approach

The heat transfer in natural convection in an infinite medium have often been studied, in particular around vertical plates or cylinders. In the case of great height, we are in a turbulent regime and the heat transfer is directly linked with the temperature difference between wall and external air or with the power density in case of the uniform heat flux (UHF). In case of uniform wall temperature (UWT), much correlations have been obtained : Mc Adams [13] Churchill and Chu [14]. For VALIDA, we apply an uniform power density Vliet and Ross [15] propose the following equations (local values):

$$Nu_x=0.55(Gr^*_x\ Pr)^{0.2} \quad \text{for } 10^5 < Gr^*_x\ Pr < 10^{13} \qquad (3)$$

$$Nu_x=0.17(Gr^*_x\ Pr)^{0.25} \quad \text{for } 10^{13} < Gr^*_x\ Pr < 10^{16} \qquad (4)$$

Remark : $Gr= g\beta\Delta TL^3/\nu^2$, by extension $Gr^*_x= g\beta\Delta Tqx^4/\lambda\nu^2$, with q= flux density in W/m² , H the cylinder height and x the considered location. As an average value we have:

$$Nu_m=1.25\ \ Nu_H\ (Nu_H=0.55(Gr^*_H\ Pr)^{0.2}) \qquad (5)$$

$$Nu_m=Nu_H \qquad (Nu_H=0.17(Gr^*_H\ Pr)^{0.25}) \qquad (6)$$

More recently, Aydin [16], summaries the knowledge in the case of natural convection and proposes a correlation depending on the Prandtl number:

$$Nu_x=C_1\ (Ra^*_x\ Pr/(C_0+Pr))^n \qquad (7)$$

Laminar flow $C_0=0.67$ $C_1=0.63$ and $n=1/5$

Turbulent flow $C_0=191$ $C_1=219$ and $n=1/4$

From tests carried out in air and with turbulent flow ($2\ 10^{13} < Gr^* < 1.7\ 10^{14}$), Miyamato [17] proposes the correlation: $Nu_x=0.104\ Ra^{*\,0.272}_x$ (8)

EXPERIMENTAL CONVECTIVE HEAT TRANSFER

From the measured temperatures, electrical flux supplied and emissivity knowledge, one can evaluate, for each test, the heat transfer coefficients hmin (minimum) and hm (hmean), over the total surface of the A cylinder, located at row 5 during the series of experiments. Classification in decreasing order of powers (the properties of the air are taken at 20 °C) is given in table 2.

				Canister A			
Test	Psp (W/m²)	Vm (m/s)	Re	Tmin (°C)	Tmax (°C)	hmin (W/m²°C)	hm (W/m²°C)
M600V1	584	0.98	42000	53.7	87.9	8.8	11
M600V0.5	580	0.51	22000	59.3	107.5	6.9	9
M300V1	304	0.91	39000	40	57.1	8.1	10
M300V0.5	298	0.50	21000	42.2	67.1	6.4	8
M300V0.25	301	0.25	11000	43.2	71.1	5.9	7.2
M600V0.25	581	0.25	11000	58.8	115	6.3	8.4
M600NC	587	0		88.6	114	5.8	6.3

Table 2 Tests matrix and main measurements on A canister

It is noticeable that for natural convection the characteristic dimension is the height, while for forced convection it is the diameter, it is decided, in this part, to work only with the heat exchange coefficient and not with the Nusselt numbers, NuH or NuD.

The azimuth temperature profile measured on natural convection test confirms the axial symmetry of the heat transfer. Above a height of 30 cm, a constant wall temperature is obtained up to the top of the cylinder and the exchange coefficient is about 6W/m²°C. According to Van Vliet [15], the laminar-turbulent transition begins for a value of Ra* (Gr*Pr) between 2 10^{12} and 4 10^{13} and the regime is fully developed at 1 10^{14}; the test shows that, on the one hand, the turbulent regime is obtained earlier and the exchange is higher (+20%) than given by Van Vliet's approach. We then represented the measured local value, Nux, as a function of Ra* and the semi-empirical curves for the turbulent regime obtained by Van Vliet, Miyamoto and Aydin.

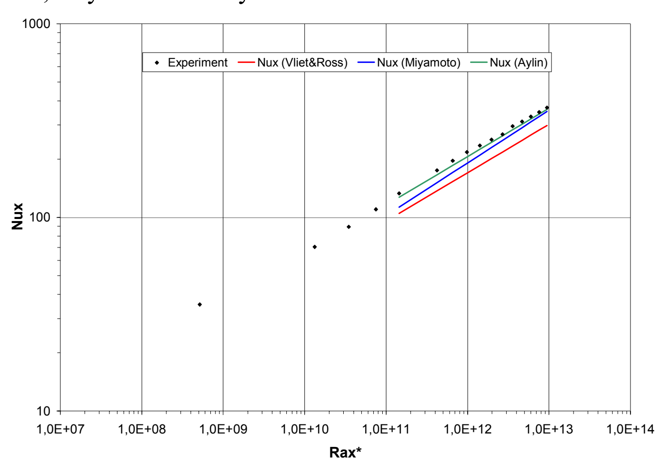

Figure 5 Natural convection test analysis

The experimental points lie in a line with a slope of 0.25, characteristic of turbulent flow. Knowing that Miyamoto's experiments were carried out for Gr*>1.5 10^{13}, the agreement with our results is excellent. The generalization proposed by Aydin seems to be the one that best fits the VALIDA tests in natural convection. In our case, the correlation becomes:

$Nu_x = 0.219\ (Ra^*_x\ Pr/(0.191+Pr))^{1/4}$ leading to

$$Nu_x = 0.206\ Ra^{*\,0.25}_x \quad (9)$$

It is possible to show the effect of transverse convection by comparing all the tests in the case of natural convection. This effect is observed on each canister and an illustration is given figure 6, for the test Multi600V1 at 1.50m height.

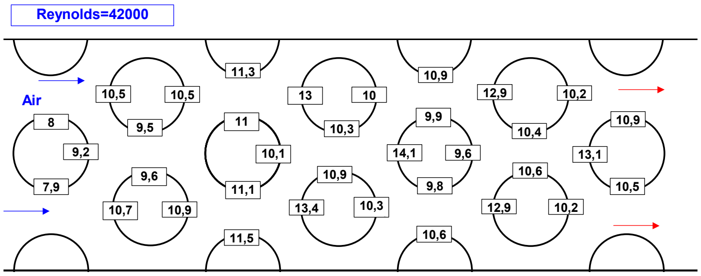

Figure 6 Heat transfer coefficient repartition

It is now possible to analyze the developments of all the heat exchanges obtained during the 6 tests and for all the canister in the system.

Note: To obtain the heat coefficient values on the other tubes and without measurements of air temperature in the system, an interpolation was done between the inlet temperature of the air and the average temperature measured on the downstream grid at 1.50m height.

COMPARISON OF TESTS AND CORRELATIONS

To evaluate the exchanges expected in forced convection, we need to know the average airflow crossing the system. For that, LDV (laser Doppler velocimetry) measurements have been performed to evaluate a ratio between the upper flow (between ceiling and cylinder) and the total flow. The value of Qgap/Qtotal (%) has been measured for each boundary condition (P from 0 to 600W/m² and V=0.25 to 1m/s), and ranges between 43 to 52% for row 1 and 60 to 70% for row 7.

So, the ratios h_{exp}/h_{NC} and h_{exp}/h_{FC} can be calculate on all canister. This work was carried out for all the tests. The final step was to establish a classification of the types of flow and show their dependence on a buoyancy parameter.

The existence of a transitional regime between forced and natural convection was observed. A dimensionless number, also referred to as "Buoyancy Number", was used to characterize the different regimes experimentally obtained. This number is defined as $Bo^* = (Gr^*_x / Re_D{}^k)$ with k=16/5 to take turbulent effects into account. This number is traditionally found in all UHF type (Uniform Heat Flux) mixed convection problems with an exponent that sometimes varies between different authors [5]. The constituents of this buoyancy number are known "a priori". It is noticeable that this is a local number that depends on the row (use of the maximal velocity between consecutive row) and the height. The progressive establishment of the forced convection exchange must taking into account with a correction factor depending on the row number. The figure 7 shows a comparison between experimental and calculated heat transfer coefficient.

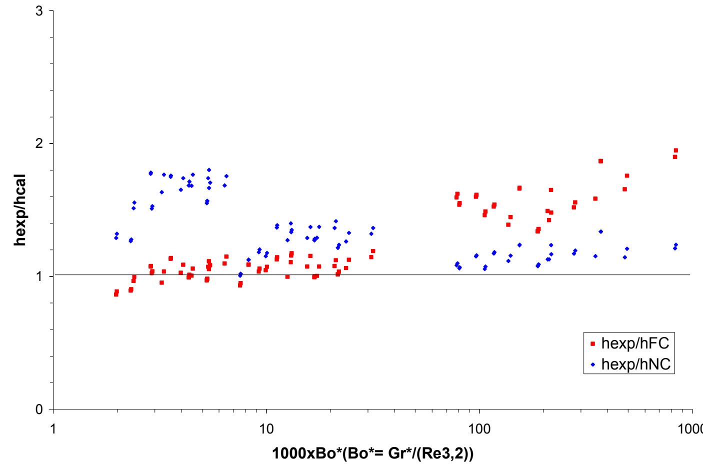

Figure 7 Heat transfer ratio evolution

For values of Bo* from 10^{-3} to $3\ 10^{-2}$, the exchanges can be calculated (as an initial approach) using a forced convection type correlation (for which the flows crossing the system need to be known). Above 0.08, a natural convection type calculation gives excellent results.

Considering the vertical plate tests with uniform heat flux (L=2.9 m with height H=3.03 m) carried out in air by Sieber [19], the authors proposed the following semi-empirical local exchange correlation (z being the altitude and x the length):

$$Nu_x\ Re_x^{-4/5}\ /0.025 = \{\ 1 + [7.067\ (x/_z)(Gr^*_z\ /\ Re_x^{\ 16/5}\ )^{1/4}\ (T_w/T_{ex})^{0.295}]^{3.2}\ \}^{1/3.2} \quad (9)$$

Obsiously, this type of formula can obviously not be used but it can serve as a source of inspiration. We modified it and obtained (with our nomenclature) :

$$Nu_{MC} = Nu_{FC}\ \{\ 1 + [1.7(Gr^*_x / Re_D{}^2)^{1/5}\ ]^3\ \}^{1/3} \quad (10)$$

The comparison with the tests (figure 8) shows that this approach is relevant.

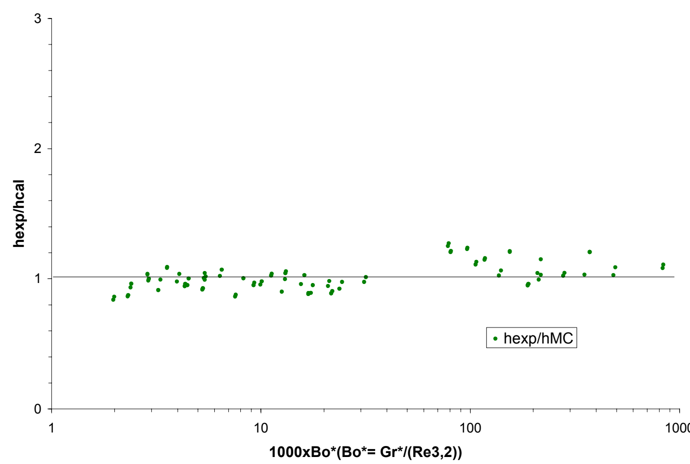

Figure 8 Heat transfer versus Bo*

The minimum convective exchange seems to be well-approximated using this method.

CONCLUSION

The test campaign carried out on the VALIDA test facility provide a large thermohydraulic data base for a good scientific approach to the mixed convection phenomena that occur around a system of heating tubes of large dimensions. This database should enable to understand and predict what happens in a real vault or cask storage of radioactive waste.

The boundaries conditions of the performed tests are well controlled, which allows fine analysis of the measurements. Various values of power density and airflow were tested, covering a wide thermoaeraulic range 11 $10^3 < Re_D < 42\ 10^3$ and a Richardson number (CRIEPI approach) from 0.3 to 23. The tests will continue by studying the air thickness effect below the ceiling (from 1 m down to 0) and measuring the air temperature between the various canisters.

The correlations found in the literature have been investigated and lead to evaluate the average and local heat transfer coefficients in each tests. In order to determine the part played by forced convection, it was necessary to know the airflow crossing the bundle, which was measured using laser doppler velocimetry (LDV). These measurements showed the flow through the system as a function of the considered row, so that the maximum velocity values could be found at each row, which are required to apply the correlations used for forced convection.

Using this dual approach, it has been possible to classify the various tests by using a buoyancy number $Bo^* = Gr^*_x / Re_D^{3.2}$ which is classically found in UHF type mixed convection cases. This number can be determined before the test, as it uses the power density in place of a bundle-air temperature difference as in the Richardson number. The main conclusions are:

For values of Bo* from 10^{-3} to 3 10^{-2}, the average heat exchange can be calculated using a forced convection type correlation.

For Bo*>0.08, a natural convection type calculation gives excellent results.

The impact convective heat transfer is always above the previous one. It has been shown that the formula (hmax-h)/hmin (%)= 80-7.5 Ln(Bo*) represents a good approximation to the reduction of this effect.

More generally, the comparison between tests and calculations shows that a mixed convection type exchange: $Nu_{MC} = Nu_{FC} \{ 1 + [1.7(Gr^*_x / Re_D^{.2})^{1/5}]^3 \}^{1/3}$ is usable for all rows.

The initial interpretation of the VALIDA multi-canister tests seems promising and quantitative results have been obtained. In parallel with this physical analysis, 3-D calculations already carried out for single bundle tests [2,3,5,6] are in progress for this 7-row case of VALIDA (LES , RANS porous medium..), the final objective being first to validate the tools and then to use the models to determine the dimensions of real vault and cask storage.

REFERENCES

[1] Lagrave H, Duret B, Gaillard JP, Thermo-aeraulics of high level waste storage facilities, *European Nuclear Conference*, Dec 2005

[2] Berthoux M., 2001, Interim Storage Modelisation. Methodology and Computation Using TRIO, *CEA/LETS 2001-156*

[3] Archambeau Fr., Méchitoua, N., Sakiz, M., Code_Saturne : A Finite Volume Code for the Computation of Turbulent Incompressible Flows - *Industrial Applications, International Journal on Finite Volumes*, 2004

[4] Chen T.S., Armaly B.F; 1987, Mixed convection in external flow, *Handbook of Single Phase Convective heat transfer part 14) John Wiley & Sons*

[5] Duret, B., Bonnard, JC., Chataing, T., Colmont, D., VALIDA MOCK-UP - Experimental results on turbulent heat transfer in mixed-convection for one large vertical heating cylinder in air cross-flow., *Turbulence, Heat and Mass Transfer (THMT06),* Dubrovnik, Croatia, September 25-29,2006

[6] Benhamadouche, S., Bournaud, S., Duret, B., Clement, Ph., Lecocq, Y., Large Eddy Simulation of mixed convection around a vertical heated cylinder cooled by a cross-flow air circulation., 2006,, *Conference on Modelling Fluid Flow Conference on Modelling Fluid Flow (CMFF'06),* Budapest, Hungary, September 6-9, 2006

[7]Heat removal characteristics of vault storage system with cross flow for spent fuel – K Sakamoto, T Koga,M Wataru (CRIEPI) – *Nuclear Engineering and Design* 195 (2000) 57-68

[8] S Kakac ,Convective Heat Transfer in cross flow 6-31 *Handbook of Single Phase Convective heat transfer,* John Wiley & Sons Interscience Publication 1987

[9] Grimison ED "Correlation and Utilisation of New Data on flow resistance and Heat Transfer for Crossflow of Gases over Tubes Banks" *Trans. ASME 59,583-594,*1937

[10] Zukauskas, A Heat transfer from tubes in cross flow.. *Adv. Heat Transfer Vol 8 pp 93-160*,1972

[11] Zukauskas, R. Ulinskas. Heat transfer in tube banks in crossflow. *Hemisphere Publishing Corporation* 1988.

[12] Yovanovich M.M., Khan W.A. Convection Heat Transfer From Tube Banks in Crossflow. Analytical Approach. *43rd Aerospace Sciences Meeting & Exhibit*, 10-12 January 2005, Reno, Nevada

[13] WH Mc Adams *Heat Transmission*, 3rd ed Mc Graw Hill, New York (1954)

[14] SW Churchill and HHS Chu Correlating Equations for Laminar and turbulent free convection from a vertical plate, *Int Heat and Mass Transfer, vol 18 pp 1323-1329*, 1975

[15] G.C Vliet and DC Ross, Turbulent Natural Convection on Upward and Downward Facing Inclined Geat Flux Surface, *J. Heat Transfer*, Vol 94 pp 549-555,1975

[16] G.C Vliet and K Liu, An experimental study of turbulent natural convection boundary layers, *J.Heat Transfer*, vol 91,p 517, 1969

[17] O Aydin, L Guessous Fundamental correlations for laminar and turbulent free convection from a uniformly heated vertical plate, *Int J of Heat and Mass Transfer 44* (2001) 4605-4611

[18] M. Miyamoto, H Kajino and J Kurima, Development of turbulence characteristics in a vertical free convection boundary layer, *Proceedings of the 7th International Heat Transfer Conference*, vol 2, 1982, pp 323-328

[19] D.L. Siebers, R.G Schwind. Experimental mixed convection *Heat Transfer from a Large Vertical Surface in a Horizontal Flow* Rep No HMT-36, Thermosciences Div, Dept of Mech Eng Stanfordv University Feb 1983.