MODELLING OF THERMAL EXCHANGE IN AN ENCLOSURE

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Résumé

Notre étude concerne la modélisation des échanges thermiques par la convection dans une cavité. Nous décrivons la méthode expérimentale employée basée sur des flux et des phénomènes d'échange thermique. L'effet de la température des murs sur l'équilibre thermique de la cavité est montré. Ces échanges ayant lieu entre les faces verticales sont produits par l'écoulement. Nous prouverons que, afin de définir ces échanges moyens de convection, la différence de température caractéristique doit absolument prendre en compte la température moyenne de l'air au centre des cavités. Nous donnerons des relations permettant de déterminer exactement cette température après avoir déterminé la température de surface moyenne pour les six parois de la cavité. Nous montrerons le phénomène de la stratification et la pente correspondante.

Mots clés : *La modélisation des échanges thermiques, convection, cavité, stratification.*

Abstract

Our study concerns the modelling of the thermal exchanges by convection in an enclosure. We describe the experimental method used based on fluxes and heat exchange phenomena. The effect of the walls temperature on the thermal equilibrium of the cavity is shown. These exchanges taking place between vertical faces are generated by the flow. We will show that, in order to define these mean convection exchanges, the characteristic temperature difference must absolutely take into consideration the mean air temperature in the centre of cavities. We will give relations permitting to determine accurately this temperature after having determined the mean surface temperature for the six walls of the cavity. We will show the phenomenon of stratification and corresponding slope.

Keywords: *Modelling of the thermal exchanges, convection, enclosure, stratification.*

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ملخص

إن دراستنا تخص نماذج المبادلات الحرارية عن طريق التوصيل داخل تجويف ما. كما شرحنا الطرق التجريبية المستعملة و التي تعتمد على التدفق و ظواهر التبادل الحراري كما تم توضيح تأثير درجة حرارة الجدران على التوازن الحراري للتجويف. إن التبادلات المتحصل عليها بين السطوح العمودية ناتجة عن السيلان. و سنبين بأن الفرق الخاص بدرجة الحرارة تتحصل عليه بأخذ درجة الحرارة المتوسطة للهواء في مركز التجويف، و تم إعطاء العلاقات التي تسمح لنا بحساب درجة الحرارة هذه بصفة دقيقة بعد تحديد درجة الحرارة المتوسطة للسطح للجوانب الست لهذا التجويف. و نبين أيضا ظاهرة التنضيد والانحدار الموافق له.

الكلمات المفتاحية: *نماذج التبادلات الحرارية، التوصيل، تجويف، الإنحدار.*

Several studies in relation with the cavity had been treated. The authors have analyzed the convection and instability phenomena following several steps. H. Manz [1], in his work, compares the overall heat transfer results simulated with a CFD code with correlations derived mainly from experimental data, including laminar and turbulent flow. M. Corcione [2] investigates natural convection in rectangular cavities simultaneously heated from below and cooled from above with several specified thermal boundary conditions at the sidewall. F. Ampofo, and T.G. Karayiannis [3] have measured simultaneously local velocity and temperature at different locations in the cavity using a Laser Doppler Anemometer (LDA) and a micro-diameter thermocouple. Their study aimed to provide highly accurate turbulent convection data, which can provide further insight into the turbulent heat transfer in natural convection in enclosures. M. Prud'homme et al. [4] have studied free convection in a vertical cavity heated from the four walls by uniform heat fluxes, is considered. K.A.R. Ismail et al. [5] gave a solution of free convection problem based on extending the finite element model, proposed by Rice and Schnipke [6]. S.-S. Hsieh et al. [7] gave a direct extension of previous three dimensional steady state experiments [8]. Their study present three–dimensional pictures of the gross features of a time evolving convective pattern in the cavity, for transient buoyancy-induced natural convection for Rayleigh number near of 10^8, but width silicone oil as working medium inside a cavity. They define duration of the thermal layer development (t_b) as a function of Rayleigh number.

The numerical studies have been showed their deficiency at least until the current hour, because they cannot provide reliable and consistent results for values of the Rayleigh number higher than 10^8 to10^9. This limitation requires from the experimental method to determine the exchanges by convection on the faces of these cavities. Besides, these experimental studies provide a necessary database to validate the numerical techniques to get solutions around a range of the Rayleigh number. Therefore the data presented herein three–dimensional numerical code validation and assessment of the two dimensions assumption.

Many correlations of the Nusselt-Rayleigh type (or Nusselt-Grashoff type) have previously been proposed to characterize exchanges by convection [9, 10, 11, 12, 13], based on the temperature difference between the two faces (hot-cold; Tch-Tf). This difference does not takes in to account the temperature of the other faces of the cavity, which are considered by numerical modellers as adiabatic or submitted to a linear evolution of temperature, and assumed by scientists as being close to adiabatic conditions. On the other hand, very few correlations contain the effect of the elongation on heat transfer. The correlations proposing an evolution of the type $Ra^{1/3}$ lead to an independence of the exchange coefficient from elongation.

Our experimental study firstly, relate cavities having an important height and different aspect ratios (A=0,81 and A=6,3), secondly, the conditions on these faces of cavities can be with varied thermal distributions (temperature or flux). They are known with precision. Finally we remedy the representative temperature difference problem by the association of all faces in the global exchange process.

EXPERIMENTAL CONDITIONS AND DESCRIPTION OF THE TWO INSTALLATIONS

The experimental tests that we did for this study have been achieved in two enclosures of different shapes: the first of approximately cubic shape is a cavity whose dimensions are equal to those of an average room and the second test cell, of rectangular shape is elongated in the vertical direction. For these two cavities we conceived an original method of a generalized fluxmeter type [14]. This technique permitted us to impose on walls that were divided into elementary surfaces either a flux or a temperature which it was measured also allowed the control of these quantities in a bi-dimensional scale.

Description of the elongate cavity

This cavity of elongated type (height 2.46m, width O.72m and depth O.385rn) has a shape aspect ratio of A=6.3, Fig.1. Unlike the preceding enclosure, this one has only one fluxmetrical face (2.46xO.72m), with 162 independent heating elements distributed in three identical vertical strips each having 54 elements, Fig.2. The readings of temperature are done in the centre of each of the 54 heating elements except, for the central strip where one also finds three more finely discredited zones. The first zone, where we tripled the number of thermocouples, is situated to mid-height and spreads over six heating elements. We can realise in this zone flux discontinuities and measure with precision the evolution of convective fluxes. The two other zones are located in the immediate neighbourhood of the lower and higher extremities that are often submitted to important temperature gradient due to the nature of the flow and the thermal conditions imposed on the horizontal walls.

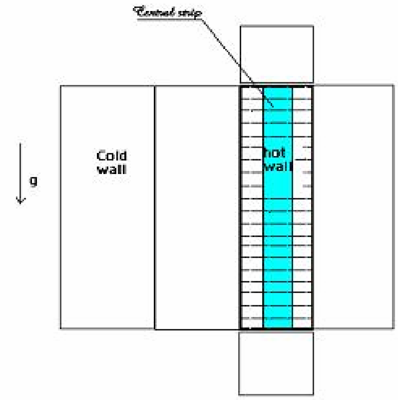

Fig.1: View exploded of the cavity with fluxmetrical hot wall.

Due to a lack of measurement apparatus, flux measurements are only done in the central strip area, Fig.2. The two lateral strips serve, to prevent sides' phenomenona resulting from the conduction between the hot flux meter walls and the lateral walls. There is a comb in this cavity with 10 thermocouples permitting to measure the mean temperature of air in front of flux meter walls, at distances comprised between O.5 and 20 cm. With the help of a pulley device, we can displace this comb vertically along the face and determine the behaviour of air in the boundary layer as well as the stratification in the central core of the cavity. A more complete description of this installation can be found in references [16] and [17].

Description of the measurement method

On the hot wall of the cavity, we want to achieve two thermal condition types:

- Either a uniform temperature
- Or a density of uniformly exchanged by convection

To achieve these conditions, the cavity's wall is heated by Joule effect using a big number of resistances distributed so that to assure while controlling the intensity of the current crossing them, either an uniform temperature, either a density of flux uniform convected. Therefore, this method allows determining directly the convected flux. If we assume the fluxmeter to be equivalent to the resistances, we can write the following heat equilibrium:

$$P_{tot} = \phi_{conv} + \phi_{cond} + \phi_{ray}$$

P_{tot}: dissipated electric Power in the heating resistances
ϕ_{conv} : convected flux
ϕ_{cond}: conducted flux (losses through the cavity's wall).
ϕ_{ray} : radiated flux (towards the other surfaces)

From the measured power and the calculated conducted and radiated fluxes, obtained from temperature measurements we can determine convected fluxes. This fluxmétrical method has the advantage to permit to establish of a temperature distribution representation and a simultaneous measurement of fluxes convection on a very high number of areas in the hot wall. Therefore, this method does require precise temperature gradient determination of cavity's wall. It is sufficient to use a thermocouples comb to determine both fluctuations and average temperatures inside the thermal boundary layer. The choice of a large cavity for temperature and flux measurements is justified by the need to reach high Rayleigh numbers (of the order of 10^{10}) which characteristic of buildings. A height (H = 2.46 m) was studied because it corresponds to that of an average sized room. Another reason behind this choice is the constraigning character of the other methods (investigation and calculus).

Configurations of temperature studied

Results that we present in this study are relative to the very varied configurations that we organised in five categories:

-the first category represents the configurations where walls are heated so as to get the isothermal faces. The objective is to obtain some relatively simple results for analysis and which we can use for the validation of certain codes of calculation.

-the second category corresponds to configurations where the two vertical walls facing each other are isothermal and the four other faces are adiabatic with the condition

$$\varphi_{cv} = 0 \qquad \varphi_{cd} + \varphi_{rd} = 0$$

It allows a better comprehension of convective flows.

-the third category corresponds to an elongated cavity where the vertical face is uniformly heated. The strong insulation behind this wall and the weak radiative losses at its front (weak emissivity) lead in this case to a condition of nearly uniform convected flux.

-the fourth category is the one where the wall is subject to quite varied flux distributions along its height (hotter areas or non-heated areas).

Finally, several experiments with varied difference temperature have been carried out in each category.

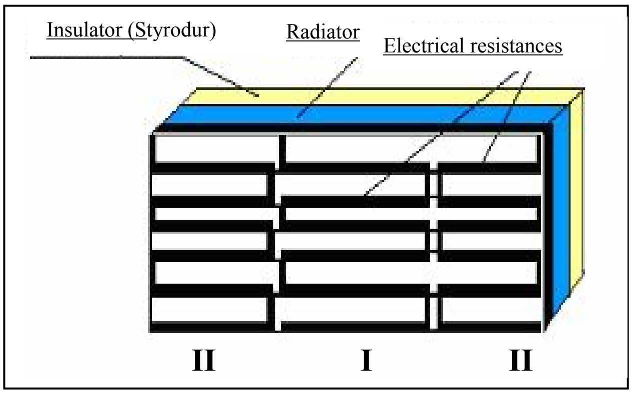

Fig.2: Heating element including a central zone (I) and two lateral zones (II) of electrical resistances.

RESULTS AND DISCUSSION

Choice of the reference temperature for modelling the average flux of convection on the vertical faces

In established thermal regime and with the hypothesis of air that is transparent to infrared radiation, the equilibrium of convective transfers on the six faces must be nil and can be written as follows:

$$\sum_{i=1}^{6} \phi_{cvi} = 0 \tag{1}$$

In case we simply model the convective transfers by the means of average transfer coefficients on each face and a unique reference temperature, we can write that equilibrium as follows:

$$\sum_{i=1}^{6} h_i . S_i . (T_{pi} - T_{ref}) = 0 \tag{2}$$

Supposing that the transfer coefficients have the same value on each face, we can define a reference temperature

$$T_{ref} = \frac{1}{S_t} \sum_{i=1}^{6} T_{pi} . S_i = T_{ms} \tag{3}$$

This appears to be a good correlation in Figure 4 where the cavity's central air temperature (T_c) is given as a function of the average surface temperature (T_p) of the six faces (T_{ms}). We have tried to better define experimentally equation (3) by distinctly characterizing transfers on the ceiling, the flooring, both motive vertical faces and both lateral ones. We have established for the elongated cavity the best correlation (least squares method) between the measured central cavity's air temperature and the faces temperatures. We obtained:

$$T_c = 0.5 * T_{vm} + 0.4 * T_l + 0.08 * T_{pd} + 0.04 * T_{pr} \tag{5}$$

Or broadly speaking and without dimension:

$$\frac{T_c}{\Delta T_m} = 0.5 * \frac{T_{vm}}{\Delta T_m} + 0.4 * \frac{T_l}{\Delta T_m} + 0.08 * \frac{T_{pd}}{\Delta T_m} + 0.04 * \frac{T_{pr}}{\Delta T_m} \tag{5'}$$

With $\Delta T_m = (T_{ch} - T_{ms})$

Where T_{vm} is the motive vertical faces temperature, T_l is the lateral faces temperature, T_{pd} is the ceiling temperature, T_{pl} is the flooring temperature according to the relation (2):

$$T_{ref} = \frac{1}{h_{glob} . S_t} (h_{vm} . S_{vm} T_{vm} + h_l * S_l * T_l + h_{pd} * S_{pd} * T_{pd} + h_{pr} * S_{pr} * T_{pr}) \tag{6}$$

From the experimental results, we get the following reports for coefficients of exchanges:

$$h_{vm} / h_{glob} = 0.85 , hl / h_{glob} = 1.24 , h_{pd} / h_{glob} = 0.70 ,$$
$$h_{pr} / h_{glob} = 0.32$$

These ratios are close to 1, which justifies the use of the simplified hypothesis that we made to establish equation (3). We can see, however, an influence on the cavity's central temperature; this influence is more important on the lateral faces and is weaker on the ceiling and the motive vertical faces. We can see all that in Figure 4 because the most distant points of the bisecting line correspond to settings where horizontal faces temperatures are exceptional.

The mean surface temperature is a very good modelling parameter of the temperature at the centre of the cavity and therefore of the air equilibrium if differences are weak (as in the case of construction) or if the intermediate walls have nearly adiabatic behaviour. This temperature can easily be obtained numerically because it depends directly on the boundary conditions of the problem. That is why it was considered as a reference temperature. A similar approach was adopted in references [20] and [21]. Their results concern a small cavity full with water. They defined a temperature of reference as being the mean surface temperature of the four vertical faces. This is not adequate in certain configurations they did not study.

These points correspond to a variety of different situations, a hot face with five cold faces, five hot faces with a cold face, a hot face and a cold face with four faces at intermediate temperatures, the floor and the ceiling being hot or cold. The nature of the flow induced by these configurations is appreciably different. Results described in detail in reference [16, 19] showed the very complex influence of temperature configurations on the flow characteristics (level of turbulence, localization of the transition zone, etc...), and the absence of tendency curve.

Table 1: Flux convected in order of Rayleigh number. Flux Convected/(T_{ch}-T_{ms}).

Case	C: hot wall F: cold wall M: mean configuration	C.(Ra)^{0,25} C=	C.(Ra)^{0,33} C=	C.(Ra)^{B} C=	Constant B=
C1	CCCFFC	1,09	0,172	0,21	0,322
C2	FCFFFC	1,01	0,15	0,631	0,27
C3	CCCFCC	0,887	0,143	1,124	
C4	MCMFFF	Insufficient	Point	number	
C5	FCFFFF	0,824	0,122	0,14	0,324
C6	CCCFCF	0,817	0,122	0,82	0,25
C78	Floor very	cold		0,0015	0,52

The table presents a correlation of the convection fluxes according to the Rayleigh number. The function is power law of $\Phi = C.(Ra)^{B}$. We observe under the different cases of configurations (C1, C2), the constants to take different values. The objective is to classify this law, either in the category with B=0.33, either in the category with B=0.25. It results a constant C. This constant in the same way changes for every configuration, as shown in the Table 1. (columns 3 and 4). Otherwise taken, the law can appear coherent with other different constants C and B, as it appears in the column 5 and 6 . The configuration C4 does not make it possible to define its constants clearly, because the number of necessary points of experiments is not sufficient. Finally the configuration C78 appears to obey a completely different law. The floor in this case is very cold. The fig.3 expresses the flux density evolution of convection according to the temperature difference, but also according to three intervals of the Raleigh number as follows: $6*10^9 \div 1*10^{10}$, $1*10^{10} \div 2*10^{10}$, $6*10^9 \div 4*10^{10}$

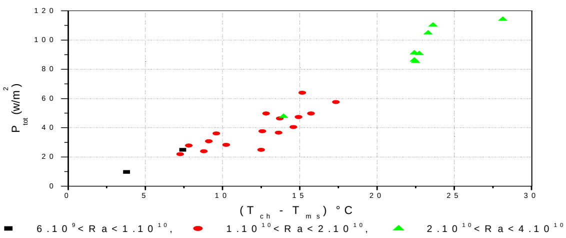

Fig.3: The density of flux with regard to the difference

Fig.4 shows the agreement of the temperatures measured in the cavity core (T_c) and the weighted average value by surfaces of the all temperatures of the other walls. This is materialized by the appurtenance of the all points to the same bisecting axis. This is equivalent of saying that the (T_{ms}) temperature is very representative of the cavity hearth temperature (T_c).

The fig.5 expresses the flux density evolution of convection according to the temperature difference, but also according to three configurations: one to 5 hot faces and a cold face (CCCFCC), one to 5 cold faces and a hot face (FCFFFF) and finally with intermediate faces between the hot face and cold face (MCMFMM).

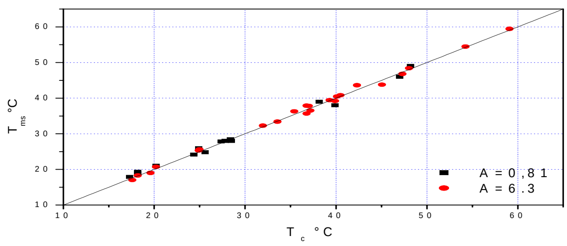

Fig.4: Evolution of the mean surface temperature (T_{ms}) versus the air temperature (T_c) at the centre.

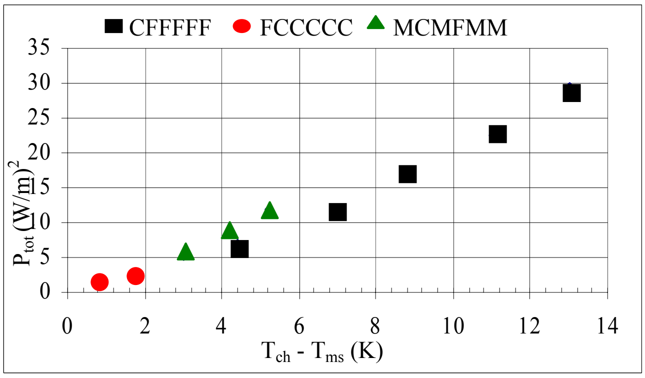

Fig.5: The density of flux with regard to the difference (T_{ch}-T_{ms}).

STRATIFICATION

The variation of the middle temperature outside the boundary layer was drawn, according to the height of all experiments. This middle temperature was determined at nine levels of measurement along the hot plate, while averaging was all values recorded on the last three thermocouples of the comb. These last three thermocouples are located with certainty outside the boundary layer.Fig.6.

In the all studied cases, the evolution was quite linear on an appreciable height. In the higher and lower parts of the cavity the evolution was disturbed by presence of the horizontal walls. In the case of a very hot floor this disturbance was important. It occurs on the first two or three levels, the temperature being then appreciably constant. In the other cases disturbance is limited to very narrow zones.

The slope of the stratification $\left(\frac{\partial T}{\partial z}\right)$ has been determined by a linear regression. It is given for all manipulations. To be able to compare the different experiments, a non-dimensional expression was defined in relation to the difference temperature Δt between hot and cold faces and the height H of the cavity, one can note then:

$$A_c^* = \frac{\partial T}{\partial z} * \frac{H}{\Delta T}$$

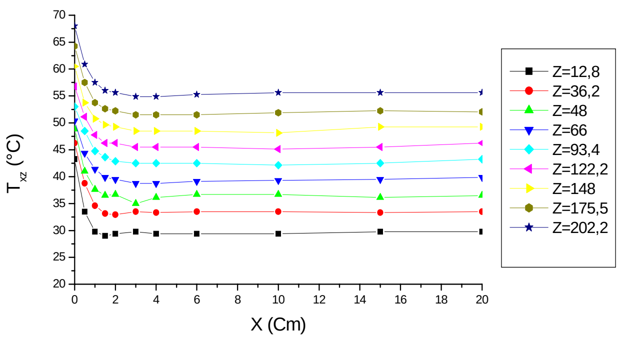

Fig.6: Temperature profiles and stratification.

CONCLUSION

The experimental correlation showed that it is possible to determine with good precision the value of the air temperature at the centre of the cavity when we take into account the mean surface temperature of all faces (T_{ms}).

The expression of the temperature difference defined with regard to Tms lead to results which obeyed the different configurations (uniform temperatures on faces, faces with imposed flux, vertical faces including some cold or hot zones, etc.) a law of the type

$$\phi = C * \left(T_{ch} - T_{ms}\right)^B$$

The distribution of temperatures on the six sides of the cavity has a big influence on the average temperature of the center of the cavity and the stratification across the latter. The temperature of the geometrical center of the cavity can be determined with a satisfactory precision in many cases as long as the average temperature of surfaces is taken into consideration. While taking the average temperature of surfaces. Once parameters have been identified, a model has been established for cavity having an important shape ratio as follows:

$$T_c = 0.5 * T_{vm} + 0.4 * T_l + 0.08 * T_{pd} + 0.04 * T_{pr}$$

This simple modelling allows calculating a satisfactory value of the central temperature of a cavity.

For this type of elongated cavity, the slope of stratification can be written as follows:

$$A_c^* = \frac{\partial T}{\partial z} * \frac{H}{\Delta T}$$

In three groups according to the gradient of temperature between the flooring and the ceiling. This representation allows us to point out that:

- The non-dimensional character permits to become liberated from the influence of the temperature gradient.
- The non-dimensional value doesn't depend on the mode of the heating (constant temperature or constant flux)
- The studied configurations can gather around three values of the stratification according to the gradient of temperature between the flooring and the ceiling, a value of A_c^*:

A_c^* = 0.24 for configurations C1 and C2

A_c^* = 0.4 for configurations C3, C4, C5 and C8

A_c^* = 0.75 for the configurations C6 and C7

Nomenclature

A=H/L: elongation of the cavity;
H: height of the cavity;
L: width of the cavity

Gr: number of Grashoff $Gr = g\beta(T_{ch} - T_{ms})H^3 / \vartheta\alpha$

h; h_{glob}; h_l: coefficient of convection exchange; global of the cavity; on the two lateral vertical faces. h_{pd}; h_{pr}; h_{vm}: coefficient of convection exchange on the ceiling; on the flooring; on the two active vertical faces

P_{tot}: Injected electric strength in the flux metre (W/m^2)
Φ_{cd}; Φ_{cv}; Φ_{rd}: flux by conduction; convection; radiance
Ra=Gr.Pr: number of Rayleigh.
Pr: number of Prandtl
S_i : surface of each of the six faces
S_l: mean surface of the two lateral vertical faces
S_t: total surface of the six faces of the cavity
S_{vm}; T_{vm}: mean surface and mean temperature of the two vertical generating faces.
Spd: surface of the ceiling; Spr: surface of the flooring
T_c: Temperature of air to the centre
T_{ch}; T_{fr}; T_l: mean temperature of the hot vertical face; of the cold vertical face; of the two lateral vertical faces.
T_{ms}: mean surfacical temperature of all faces of the cavity.
T_{ref}; T_{pd}; T_{pr}: temperature of reference; of the ceiling; of the flooring.
B: thermal expansion coefficient
φ_{cd} ; φ_{cv} ; φ_{rd} : density of flux by conduction; convection; radiation.
υ: kinematics viscosity.
T_{xz}: Middle température at level z and distance x.
$\Delta T = T_{ch} - T_{fr}$

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