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SERVICES FOR SCIENCE AND EDUCATION

Natural Ventilation in a Semi-Confined Enclosure Heated by a Linear Heat Source

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ABSTRACT

This work aims to study by numerical computation the natural ventilation in an enclosure with six openings heated by a solid and linear heat source. The study is carried out in steady state for the values of the reduced Rayleigh number Ra* = 6.09 × 107, 9.74 × 107 and 1.61 × 108. The numerical computation device has an aspect ratio H/L = 2.46. The heat source is centered and located near the floor of the enclosure. The openings are symmetrical in pairs and located on the two side walls located at positions x+ = - 0.5 and 0.5. The numerical studies were carried out by DNS considering that the heat source is initially at constant temperature. The finite volume method was used to discretize the conservation equations of mass, momentum and energy of the problem studied. The results show the existence of the thermal plume deviated towards x+ > 0. The fields and temperature profiles and static pressure profiles are non-homogeneous in the enclosure. An asymmetrical phenomenon is observed in the stationary enclosure of the differential static pressure profiles and a heating. The differential static pressure profile located at position x = - 0.5 is different from that at position x+ = 0.5. Fresh air enters the enclosure through openings located below the height z+ = 0.65. The fresh air which enters the enclosure through the opening located at position z+ = 0.59 mixes with the hot gas from the thermal plume and lowers its temperature for Rayleigh numbers of 6.09 × 107, 9.74 × 107 and 1.61 × 108. The hot gases exit the enclosure through openings located in the ceiling of the enclosure. The results of numerical computation of differential static pressure obtained near the position x+ = 0 agree with those of experimental measurements in the literature.

Keywords: Natural ventilation, Heat source, Semi-confined enclosure, Numerical computation

INTRODUCTION

Natural ventilation makes it possible to renew and clean the indoor air of a classroom, a shopping center, a laboratory, a house, etc., presenting [1 – 14] or not [15 – 17] openings. Its aim is to sanitize and provide fresh hygienic air necessary for people, machines, devices, etc., present in these premises by evacuating humidity and pollutants linked to occupation. Air circulation in these rooms takes place without mechanical action such as the fan.

In this study, we will study enclosures or premises presenting several openings with localized point and linear buoyancy sources of these enclosures. In this type of situation, two types of ventilation are distinguished: displacement ventilation [1 -4, 7 – 10, 12, 14] and mixing ventilation [11]. Mixing ventilation occurs when cold air enters the enclosure through one or

more openings with a mass flow rate Q induced by the pressure difference between the inside and outside of the room and mixes with hot gases by lowering their temperature. And that in steady state, the enclosure is almost well mixed at an equilibrium temperature T and with movements of fluid entering and leaving the enclosure at the openings. In displacement ventilation, in steady state, the air inside the enclosure is divided into two zones: a hot zone which can be at constant or stratified temperature and a cold zone at almost constant temperature and equal to or slightly higher than the exterior temperature of the enclosure. The plane of separation of these two zones is called interface.

Depending on the type of ventilation, air movements in the enclosure are created either by temperature differences which generate heating of the enclosure or by density differences which induce pressure differences at inside and at outside the enclosure. These pressure and temperature differences induce a plume inside the enclosure.

We will subsequently focus on enclosures presenting several openings with solid heat sources with a time constant. In this theme, certain experimental, analytical and numerical studies have been carried out, like Kouéni Toko, 2019 [7], who carried out an experimental, analytical and numerical study of the dynamic and thermal fields in 03 enclosures at 02 and 04 apertures, of which 02 have an aspect ratio H/L = 2.46 and 01 an aspect ratio of 0.82. These enclosures are heated by a linear heat source. Two situations of the heat source were studied. In the first situation, the heat source is initially at a constant temperature and higher than the exterior temperature of the enclosure and in the second situation, the heat source is at the exterior temperature of the enclosure. In this study, Kouéni Toko [7] showed that when the heat source is initially at a constant temperature and higher than the outside temperature, in steady state the thermal plume is almost centered. And when the heat source is initially at outside temperature and in steady state, the plume could have 03 positions in the enclosure: centered, deviated to the side of x < 9 and deviated to the side of x > 0. It also showed that when the enclosure has 04 openings, the cold air entry flow rates into the enclosure and the hot gas exit flow rates are greater than with 02 openings. Kouéni *et al.* [8] studied by numerical computation the temperature fields in an enclosure with 20 small openings located at the bottom of the enclosure and on the two side walls. This enclosure is heated by a solid and linear heat source. The speaker has an aspect ratio H/L = 2.476. They showed the impact of the reduced Rayleigh number ($Ra^* = \delta^2 Ra$) on the steady state temperature fields and profiles. As the reduced Rayleigh number increases, the temperature in the hot zone at the top of the enclosure increases. Koueni Toko Christian Anicet [9] used the same experimental setup and showed through a numerical study the influence of the reduced Grashof number (($Gr^* = \delta^2 Gr$)) on the velocity fields, the profiles speed and differential static pressure. At the level of the openings and at the neutral height, the speed along the x axis is zero. And when the static pressure is negative the direction of the speed along the x axis is entering the enclosure is the opposite phenomenon the differential static pressure is positive.

Koueni Toko Christian Anicet *et al.* [10] studied through an experimental study the differential static pressure in 04 enclosure configurations case 1, case 2, case 3 and case 4, heated by a linear heat source. They showed that the profiles, values and neutral height were different when we went from case 1 to case 4 and when the reduced Grashof number increased. Koueni

Toko Christian Anicet [10] studied the thermal fields in case 1 by numerical calculation. He showed that outside the side walls and above the heat source the temperature was almost homogeneous on the same horizontal plane and increased slightly with the height of the enclosure.

The objective of this study is to study by computation numerical the temperature and static pressure difference fields generated in a 06-parallelepiped enclosure distributed two by two at different heights of the enclosure by a stationary buoyancy source. This study was never carried out. We will study the influence of the reduced Rayleigh number on the phenomenon studied. This study is numerical and in steady state.

EXPERIMENTAL DEVICE

The device in Figure 1 was produced by KOUENI *et al.* [10]. This device was used to carry out the numerical computation of our study. It is rectangular and has 06 openings which allow it to communicate with the outside. These openings are symmetrical in pairs, located on the 02 side walls located at positions x+ = - 0.5 and 0.5 of the enclosure. These openings are located at 03 different levels of the enclosure: 02 openings near the floor, 02 at height z+ = 0.59 and 02 at the ceiling of the enclosure. The enclosure is rectangular with aspect ratio H/L = 2.46, length L = 210 mm, width l = 210 mm and height H = 520 mm. The openings have a height h = 34 mm and a width l identical to the width of the enclosure.

The axis system is orthonormal with center 0 and 03 dimensions, that is to say 03 axes: axis 0x, axis 0y and axis 0z. the walls of the enclosure are made of plexiglass of different thicknesses. The ceiling and floor are 14mm thick and the side walls are 8mm thick.

The heat source is located in the center of the enclosure, close to the floor and at position x = 0. It is cylindrical in shape and parallel to the openings located at floor level. Its diameter is D_0 = 20 mm and its length l_1 = 200 mm.

This experimental device is the numerical computation device of our study, but we considered that the base section of the heat source is square with side c = 20 mm instead of circular. And of length l_2 = 208 mm. The heat source is located 2 mm above the floor of the enclosure.

Initially the heat source is at a positive temperature ΔT_0 and different from 0. And the interior of the enclosure is at the outside temperature. 03 values of the reduced Rayleigh number were used for the numerical calculations: 6.09×10^7, 9.74×10^7 and 1.61×10^8.

The expression for the reduced Rayleigh number is:

$$Ra^* = \delta^2 Ra \quad (1)$$

With

$Ra = Gr \times Pr$: Rayleigh number;

$\delta = \frac{\mathrm{L}}{\mathrm{H}} = \frac{1}{2{,}46}$: Inverse of aspect ratio;

$Gr = \frac{g\beta H^3(T_0 - T_{ext})}{\nu^2}$: Grashof number;

$Pr = \frac{\boldsymbol{\nu}}{\boldsymbol{\alpha}}$: Prandtl number.

MATHEMATICAL MODEL AND NUMERICAL METHOD

Mathematical Model

The conservation equations of mass, momentum and energy ((5), (6), (7)) for a viscous compressible fluid developed by Kouéni Toko [7] are representative of the physical problem of this study. To establish these equations, we considered that the regime is stationary. These conservation equations take into consideration the confinement of the enclosure represented by δ which is the inverse of the aspect ratio. The aspect ratio is H/L. In this problem studied, the thermal plume is at the origin of the phenomena present in the semi-confined enclosure. We considered the turbulent thermal plume. The heat source is in natural convection with its environment. The equations for conservation of mass, momentum and energy have been established in natural convection. The heat source is solid and has a time constant. We established the reduced Rayleigh numbers (equation (1)) and reduced Grashof numbers t(equation (9)) to characterize the buoyancy flux created by the heat source when it is at a constant temperature and higher than the external temperature. . The Boussinesq approximation was considered in the term of upward thrust characterized by the density which is a function of the temperature (equation (2)). The materials of the heat source, side walls, floor and ceiling as well as their thicknesses were considered.

$$\rho = \rho(T) \tag{2}$$

Expression (3) of buoyancy is defined by Morton *et al.* [18].

$$\frac{\rho_0 - \rho}{\rho_1} = \beta(T - T_0) \tag{3}$$

With

- ρ_1: Reference density;
- ρ_0: Outside density of enclosure;
- T_0: Outside temperature of enclosure.
- For a gas, β is given by the expression (4)

$$\beta \sim \frac{1}{T_1} \tag{4}$$

T_1: Reference temperature.

The expression of the Boussinesq approximation is $|\delta\rho| \ll \rho_0$ ($\frac{\rho_0}{\rho} \approx 1$)

The dimensionless conservation equations of mass (5), momentum (6) and energy (7) [7, 8 and 11] are:

$$\rho\left(\frac{1}{\delta}\frac{\partial u^+}{\partial x^+}+\frac{1}{\delta}\frac{\partial v^+}{\partial y^+}+\frac{\partial w^+}{\partial z^+}\right)=0 \tag{5}$$

$$\begin{cases} \frac{u^+}{\delta}\frac{\partial u^+}{\partial x^+}+\frac{v^+}{\delta}\frac{\partial u^+}{\partial y^+}+w^+\frac{\partial u^+}{\partial z^+}=-\frac{1}{\delta}\frac{\partial P^+}{\partial x^+}+\frac{1}{Gr^{*1/2}}\left(\frac{\partial^2 u^+}{\partial x^{+2}}+\frac{\partial^2 u^+}{\partial y^{+2}}+\delta\frac{\partial^2 u^+}{\partial z^{+2}}\right) \\ \frac{u^+}{\delta}\frac{\partial v^+}{\partial x^+}+\frac{v^+}{\delta}\frac{\partial v^+}{\partial y^+}+w^+\frac{\partial v^+}{\partial z^+}=-\frac{1}{\delta}\frac{\partial P^+}{\partial y^+}+\frac{1}{Gr^{*1/2}}\left(\frac{\partial^2 v^+}{\partial x^{+2}}+\frac{\partial^2 v^+}{\partial y^{+2}}+\delta\frac{\partial^2 v^+}{\partial z^{+2}}\right) \\ \frac{u^+}{\delta}\frac{\partial w^+}{\partial x^+}+\frac{v^+}{\delta}\frac{\partial w^+}{\partial y^+}+w^+\frac{\partial w^+}{\partial z^+}=-\frac{1}{\delta}\frac{\partial P^+}{\partial z^+}+\frac{1}{Gr^{*1/2}}\left(\frac{\partial^2 w^+}{\partial x^{+2}}+\frac{\partial^2 w^+}{\partial y^{+2}}+\delta\frac{\partial^2 w^+}{\partial z^{+2}}\right)-T^+ \end{cases} \tag{6}$$

$$\frac{u^+}{\delta}\frac{\partial T^+}{\partial x^+}+\frac{v^+}{\delta}\frac{\partial T^+}{\partial y^+}+w^+\frac{\partial T^+}{\partial z^+}=\frac{1}{(PrRa^*)^{1/2}}\left(\frac{\partial^2 T^+}{\partial x^{+2}}+\frac{\partial^2 T^+}{\partial y^{+2}}+\delta\frac{\partial^2 T^+}{\partial z^{+2}}\right) \tag{7}$$

Where

$$x^+=\frac{x}{L},\ y^+=\frac{y}{L},\ z^+=\frac{z}{H},\ u^+=\frac{u}{U_0},\ v^+=\frac{v}{U_0},$$

$$w^+=\frac{w}{U_0},\ T^+=\frac{T-T_{ext}}{T_0-T_{ext}},\ \text{et}\ P^+=\frac{P}{|P_{max}|}$$

With

$$U_0=\sqrt{\frac{P_{max}}{\rho_\infty}} \tag{8}$$

The equation (8) is the maximum velocity air inlet cool into the enclosure.

The reduced Grashof number is:

$$Gr^*=\delta^2 Gr \tag{9}$$

Numerical Method

Direct numerical simulation (DNS) was used to solve the equations of the physical problem of our study. It uses the finite volume method. The mathematical equations for solving our physical problem are discretized [19] and form a system of algebraic equations. The discretization scheme used to solve the momentum conservation equations is the QUICK scheme [20]. To solve the pressure, we considered the "Weighted Body Force" scheme. Spalding and Patankar established the SIMPLE algorithm. This algorithm makes it possible to correct the pressure on a non-uniform mesh. The temperature and pressure solutions were obtained by the commercial name FLUENT calculation code. This code is based on the execution of a Gauss Seidel linear system algorithm based on the AMD multigrid algebraic method. The resolution of the equations in the steady state is done by iteration by considering the conduction in the side walls, floor and ceiling.

The mesh considered is regular fine non-uniform. This mesh makes it possible to have a stable solution close to reality. The calculation stops after the convergence of the temperature and the speed. To have a stable solution and reduce the temperature and pressure calculation times, we studied the impact of the number of meshes (Table 1) on the temperature and pressure solution (Figure 4). According to Figure 4 the optimal mesh is the coarse mesh (Table 1). The mesh has been tightened and smoothed at the level of strong temperature gradients to have a solution close to reality.

Boundary Conditions

Initially, the temperature of the heat source is at a constant temperature and higher than the temperature outside the enclosure and the static pressure difference between the inside and outside of the enclosure is zero at the level of the enclosure. openings. A natural convection phenomenon is produced at the heat source. 03 reduced Rayleigh numbers of the heat source were considered: 6.09×107, 9.74×107 and 1.61×108. The boundary conditions of the heat source, side walls, floor, ceiling and at the openings are shown in Figure 3. It appears from this Figure that a Newman condition is applied on the side walls and the ceiling. A constant temperature condition is imposed on the heat source and the floor. A constant pressure condition is imposed on the openings. On the heat source, the side walls, the floor and the ceiling, the axial, vertical and longitudinal velocities are zero.

RESULTS AND DISCUSSION

The study of temperature fields, temperature profiles, pressure profiles as a function of the reduced Rayleigh number Ra*, as well as the comparison of the pressure profiles obtained by numerical computation with those experimental measurements carried out by Koueni Toko [10] is presented in this part. 03 values of Ra* were chosen: 6.09×10^7, 9.74×10^7 et 1.61×10^8.

Influence of the Reduced Rayleigh Number on Temperature Fields

Figure 5 presents the temperature fields obtained by numerical computations at positions y+ = 0 and for the values of the reduced Rayleigh number Ra* of 6.09×10^7, 9.74×10^7 and 1.61×10^8. This figure shows that the thermal plume divides the semi-confined enclosure into two temperature zones with almost identical values: a small zone located at positions x+ > 0 and a large zone located at the level of the floor of the enclosure at positions x+ < 0. Therefore, the phenomenon resulting heated is asymmetrical in the enclosure. We also observe in this figure that the thermal plume reaches the ceiling of the enclosure where it is destroyed by shear when the values of the reduced Rayleigh number are 6.09×10^7, 9.74×10^7 and 1.61×10^8.

Figure 6 presents the temperature fields in a semi-confined enclosure located at positions x+ = - 0.5 and x+ = 0.5 and at y + = 0 and for the reduced Rayleigh numbers Ra* of 6.09×10^7, 9.74×10^7 and 1.61×10^8. From this figure it appears that the temperature fields at position x+ = 0.5 are different from those at position x+ = 0.5. For values of z+ between 0.15 and 1 and for values of Ra* between 6.09×10^7 and 9.74×10^7, the temperature values are higher at x+ = 0.5 than at x+ = - 0.5. For z+ values between 0 and 0.15, the temperature values are lower at position x+ = 0.5 than at position x+ = - 0.5. For Ra* =1.61×10^8, we observe the same phenomenon when the values of z+ are between 0.25 and between 0 and 0.25. This may be due to the fact that the thermal plume is strongly deviated towards the x+ > 0 side. This figure also shows that the

thermal plume reaches the ceiling of the enclosure where it is destroyed by shear and descends towards the x+ < 0 where it is mixed with the air entering the enclosure from the side of the openings located atposition x+ = - 0.5. We also observe an asymmetrical phenomenon at the side walls of the enclosure located at positions x+ = - 0.5 and x+ = 0.5.

Influence of the Reduced Rayleigh Number on Temperature Profiles

Figure 8 presents the temperature profiles obtained by numerical computations in the semi-confined enclosure at the positions x+ = - 0.5, - 0.25, 0, 0.25, 0.5 and y+ = 0 and for the values of the reduced Rayleigh number $Ra^* = 6.09 \times 10^7$, 9.74×10^7 and 1.61×10^8. This figure shows that the temperature values obtained at positions x+ = - 0.5, - 0.25 and for values of z+ between 0.2 and 0.4 are lower than those obtained at positions x+ = 0.25, 0.5. For values of z+ between 0.4 and 1, the temperature values obtained at position x+ = 0.5 are higher than those obtained at positions x+ = - 0.5, - 0.25 and 0.25. The temperature values obtained at these latter positions are almost identical. In this figure, we also observe that the thermal plume reaches the ceiling of the enclosure close to the side wall located at the position x+ = 0.5 where it is destroyed by shear and descends towards the x+ < 0 where it is mixed with the air entered the enclosure from the side of the openings located on the side wall at position x+ = - 0.5 and entrained again by the thermal plume. We observe that the temperature profiles are asymmetrical in the semi-confined enclosure. For z+ the values between 0 and 0.2. The temperature profiles are almost identical at positions x+ = - 0.5, - 0.25, 0.25, 0.5. When the values of z+ are between 0 and 0.065, the temperature values obtained at the positions x+ = - 0.5, - 0.25, 0.25, 0.5 are almost identical to that of the external environment and much lower than those of the position x+ = 0 (Figure 7). This can be explained by the entry of fresh air into the enclosure through openings located at floor level. Initially, the thermal plume is centered. From z+ > 0.1 it deviates towards x+ > 0 (Figures 7).

In Figure 7, at position x+ = 0.5 and at the level of the opening located at height z+ = 0.59, we observe a drop in temperature due to the entry of fresh air into the enclosure. After this opening, the temperature values increase with the increase in the height of the enclosure until near the ceiling where the thermal plume is destroyed by shear.

Influence of the Reduced Rayleigh Number on Differential Static Pressures

Figure 9 presents the differential static pressure profiles obtained by numerical computations in the semi-confined enclosure at the positions x+ = - 0.5, - 0.25, 0, 0.25, 0.5 and y+ = 0, and for the values of the reduced Rayleigh number $Ra^* = 6.09 \times 10^7$, 9.74×10^7 and 1.61×10^8. It appears from this figure that the profiles and the rates of differential static pressure obtained at the positions x+ = - 0.5, - 0.25, 0 are different from the profiles and the rates of differential static pressure obtained at the positions x+ = 0.25, 0.5. The profile with differential static pressure at position x+ = 0.5 at several neutral heights (height for which the static pressure difference is zero) while the other profiles have a neutral height. At positions x+ = - 0.5, - 0.25, 0, 0.25, and for openings located at floor level and at position z+ = 0.59, the values of the differential static pressure are negative and at openings located at the ceiling of the enclosure, they are positive. Depending on the values of the reduced Rayleigh number, the values of the neutral height take values between 0.65 and 0.87. The values of the differential static pressures obtained in the semi-confined enclosure at the positions x+ = - 0.5, - 0.25, 0, 0.25, 0.5 are different. The negative

values of the differential static pressure at the openings located on the floor and at the position x+ = 0.59 and for the values of Ra* = 6.09×10^7, 9.74×10^7 and 1.61×10^8, show the entry of fresh air inside the enclosure. The absolute values of the static pressure values at the openings located on the floor at position x+ = 0.5 are lower than those at position x+ = - 0.5. Consequently, the air inlet flow rate into the enclosure is higher at the openings located at position x+ = - 0.5 than that of the openings located at position x+ = 0.5. The hot gases exit the enclosure through openings located in the ceiling. Asymmetrical phenomena of differential static pressure are observed in the enclosure.

Comparison of Differential Static Pressure Obtained by Numerical Computations with Experimental Measurements

Figure 10 compares the differential pressure measurements [10] with the numerical calculations obtained at the position x+ = 0 and for the reducedRayleigh numbers Ra* = 6.09×10^7, 9.74×10^7 and 1.61×10^8. This figure shows that the static pressure profiles have almost the same appearance. We observe a good agreement between the experimental measurements obtained by KOUENI *et al.* [10] and the numerical calculations obtained at the position x+ = 0 when the reduced Rayleigh numbers are 6.09×10^7, 9.74×10^7 and 1.61×10^8.

CONCLUSION

At the end of this study on natural ventilation in a semi-confined enclosure heated by a linear heat source, where it was a question of showing the influence of the reduced Rayleigh number Ra* on the temperature fields, the temperature profiles and differential static pressure profiles. And to compare the results of numerical computations of the static pressure differential with the experimental measurements obtained by KOUENI *et al.* [10]. The numerical computation device is a semi-confined enclosure comprising 06 openings in pairs, symmetrical and distributed on the two walls lateral located at positions x+ = - 0.5 and 0.5. 02 openings are located on the floor of the enclosure, two others at position z+ = 0.59 and 02 openings at the ceiling level. The numerical simulation method used to solve the conservation equations of mass, momentum and energy of our problem is DNS (Direct Numerical Simulation) considering that the heat source is initially at constant temperature and solid. with positive time constant and different from 0. This method uses the finite volume method as a technique for discretizing conservation equations.

The results show that the thermal plume is deviated towards the x+ > 0 side and close to the side wall located at the position x+ = 0.5 when z+ is between 0.4 and 1. This ascending plume reaches the ceiling where it is destroyed by shear. The hot gases from the destroyed thermal plume descend back into the enclosure where they are carried again by the plume towards the top of the enclosure. This asymmetrical heating causes the difference in differential static pressure profiles in the enclosure. We observe an asymmetric situation in stationary temperature and differential static pressure profiles. With a flow of air entering the enclosure through the openings located on the floor and at the position z+ = 0.59. The hot gases exit the enclosure through openings located in the ceiling of the enclosure. The differential static pressure profiles in the enclosure are non-homogeneous. The air entering the enclosure through the opening on the side of x+ = 0.5 and at the position z+ = 0.59, mixes with the hot air

from the enclosure and lowers the temperature in the thermal plume. At the top of the enclosure and outside the thermal plume, temperatures are weakly stratified.

Several studies have already highlighted this type of behavior between a thermal plume placed near vertical walls. Koueni Toko [7], Bouafia and Daube [21], Ilie *et al.* [22], Jaluria and Himasekhar [23], Jaluria [24]. According to Koueni Toko [7], this behavior is linked to the Couanda effect which induces a deviation of the thermal plume towards the side walls of the enclosure.

The comparison of experimental measurements of differential static pressure [10] with the results of numerical calculations is satisfactory.

In perspective, we will move the heat source at floor level and vary its diameter and carry out numerical calculations and experimental measurements of the temperature and pressure fields in unsteady and stationary situations.

Nomenclature

Lowercase:

- l: Width Enclosure, m
- x: Longitudinal coordinate, m
- y: Vertical coordinate, m
- z: Transverse coordinate, m
- u: Horizontal component of the velocity, $m.s^{-1}$
- v: Vertical component of the velocity, $m.s^{-1}$
- w: Transverse velocity component, $m.s^{-1}$
- *cte:* Constant
- h_i: Heat exchange coefficient inside the enclosure, $W.m^{-1}.K^{-1}$
- h_o: Heat exchange coefficient outside the enclosure, $W.m^{-1}.K^{-1}$

Capital Letters:

- L: Enclosure length, m
- H: Enclosure height, m
- ΔT_0: Temperature difference between the heat source and the outside, K
- T: Temperature inside the enclosure, K
- T_0: Temperature of the heat source, K
- U_0: Maximum inlet gas velocity in the enclosure, K
- P: Static pressure inside enclosure, Pa
- P_{max}: Maximum static pressure inside enclosure, Pa
- P_{total}: Total pressure, Pa
- $P_{total,\ inside}$: Total pressure inside the enclosure, Pa
- $P_{total,\ outside}$: Total pressure outside the enclosure, Pa
- $T_{outside}$: Temperature outside the enclosure, K
- T_p: Side wall or ceiling temperature, K
- T_∞: Temperature away from side wall or ceiling inside or outside the enclosure, K

- S_1: Section of the enclosure ceiling, m^2
- S_2: Section of the side walls of the enclosure, m^2

Greek Symbols:

- ν: Kinematic viscosity of air, $m^2.s^{-1}$
- ρ: Density, $kg.m^{-3}$
- β: Coefficient of thermal expansion, K^{-1}
- ρ_{ext}: Density of gases outside of the enclosure, $kg.m^{-3}$
- φ_p: Heat flux to the side walls and ceiling of the enclosure, $W.m^{-1}$
- λ: Thermal conductivity of the walls in Plexiglas, $W.m^{-1}.K^{-1}$

Dimensionless Numbers:

- *Ra:* Rayleigh number
- *Pr:* Prandtl number
- *Gr:* Grashof number

Special Characters:

- *Ra*:* Reduced Rayleigh number
- *Gr*:* Reduced Grashof number
- *x+:* Dimensionless longitudinal coordinate
- *y+*: Dimensionless vertical coordinate
- *z+:* Coordinated transverse to size
- *u+:* Horizontal component of the dimensionless speed
- *v+:* Vertical component of the dimensionless speed
- *w+:* Transverse component of the dimensionless speed
- *P+:* Static pressure to size
- *T+:* Temperature to size
- δ: Aspect Ratio

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