

The Research of Heat-Transfer Conductance of Multiple Endoseptums in An Workshop Building

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Abstract: Due to practical applications, convective heat transfer in a square chamber has been an important subject numerically. Also, the management of Heat Transfer Conductance Coefficients (HTCCs) in an enclosing wall has a remarkable influence. Based on handling of regions close to wall, the equation for convective heat transfer coefficient is acquired. Meanwhile, the dimensionless local Nu number is also obtained as well as average Nu number. For the enclosure with convective heat transfer, to affirm the applied model, the influence of first grid point position is analyzed and the position of the first grid point is suggested to be less than 0.5mm.

Keywords: numerical research, heat-transfer conductance, endoseptum

NOMENCLATURE

A	surface area of the blocks [m^2]
b	height of each block [m]
H	height of the total isolated blocks [m]
h	convective heat transfer coefficient [$W m^{-2} K^{-1}$]
h_x	local convective heat transfer coefficient [$W m^{-2} K^{-1}$]
k	kinematic energy of turbulence [$m^2 s^{-2}$]
l	length of each block [m]
T	temperature [K]
T_C	temperature at cold wall [K]
T_H	temperature at hot wall [K]
T_p	temperature at the point [K]
T_{ref}	reference temperature [K]
T_w	temperature at the wall surface [K]
U_0	inlet velocity [$m s^{-1}$]
u	velocity component in x-direction [$m s^{-1}$]
u_r	friction velocity $u_r = \sqrt{\tau_w / \rho}$
v	velocity component in y-direction [$m s^{-1}$]
w	width of the inlet (outlet) [m]
y_p	distance from the first grid to the wall [m]
y^+	$y^+ = u_r y / \nu$ local Reynolds number
<i>Greek symbol</i>	
α	an empirical const
λ	thermal conductivity [$W m^{-1} K^{-1}$]
μ	viscosity [$kg m^{-1} s^{-1}$]
ν	kinematic viscosity [$m^2 s^{-2}$]

ρ	density of air [$kg m^{-3}$]
τ	shear stress [Pa]
<i>Subscripts</i>	
C	cold wall
H	hot wall
l	laminar
m	mixed convection
n	natural convection
p	the first point from the surface
t	turbulent
x	local value
r	friction

1. INTRODUCTION

Due to several applications of practical interest, convective heat transfer in a square chamber has been an important subject numerically. Such applications as electric transmission cable can be modeled by various extensions of this type of geometry. Natural convection heat transfer in an enclosure has been noticed by many researchers [1-3]. Chu carried out a numerical study to investigate the effects of heater size, heater location, and boundary conditions on two-dimensional laminar convection in rectangular enclosures[2]. Hatton and Awbi have shown the existence, in the literature, of the large variation in the values of HTCCs for some internal surface in an enclosure [3].

It is obvious that convective heat transfer and airflow in an enclosure with multiple isolated blocks hasn't got enough attention as it deserves, furthermore, data for the mixed convective heat transfer coefficients of blocks is rather scant.

Our work is to investigate the natural convection, forced convection in an enclosure with multiple isolated blocks as well as mixed convection.

PHOENICS and low Reynolds number model are used. For simplicity, the enclosure is regarded as two-dimensional, namely end wall effects are negligible in a direction perpendicular to the plane of the figure. Firstly, natural convection and forced convection in an enclosure with ten isolated blocks are numerically calculated respectively, with a range of Grashof number and Reynolds number. Secondly, mixed convection in the enclosure with the blocks is calculated numerically. The primary target of this paper is to find how the natural convection and forced convection are 'compounded' to describe mixed convection.

2. PHYSICAL MODEL AND EDGE CONDITIONS

A closed chamber with ten isolated blocks for natural convection is shown as Fig.1(a). A vented chamber with ten isolated blocks is shown as Fig.1(b).

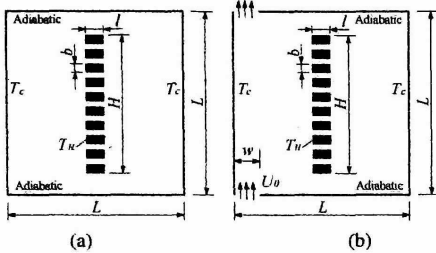


Fig.1: Square enclosure with ten isolated blocks($k=2b$): (a) natural convection, (b) forced convection and mixed convection

Two vertical walls of the enclosure are kept at a constant temperature T_c , the ten isolated blocks are arrayed uniformly in the center of the enclosure, and these blocks are kept at T_H , which is higher than T_c . This makes Grashof number positive. The two horizontal walls are supposed to be adiabatic. As Fig.1(b) shows, the forced conditions are imposed by providing an inlet at the bottom of the adiabatic wall, and a vent facing the inlet at the top. The inlet and outlet's width w is equal to $L/9$. We can study the change in the sign of the forced flow by fixing the inlet velocity at the bottom. The Reynolds number is based on the H , thus the 'height' of the ten blocks is chosen to be the length scale. Grashof and Reynolds numbers determine which of the two mechanisms of flow is predominant. At the same time, the Prandtl number is fixed as a constant of 0.7.

3. SIMULATION USING PHOENICS WITH LAM-BREHMOST K-E TURBULENCE MODEL

3.1 Heat Transfer close to the Wall Region

As we know, a viscous sub-layer is adjacent to wall region, where molecule viscosity cannot be neglected. A low-Reynolds number $k-\epsilon$ turbulence model is relatively applicable to the treatment of the adjacent wall regions. The model in our paper bases on the work of Henkes and Hoogendoorn [4]. Near the

wall region for a two-dimensional edge layer, the total shear stress is expressed by

$$\tau = \tau_t + \tau_l = -\rho u \overline{v} + \mu \frac{\partial u}{\partial y} \quad (1)$$

in the formula, the first item on right-hand side is the fluid turbulent shear stress, while the second item is the fluid laminar shear stress.

In this area, the laminar shear stress is much greater than that of the turbulence, and the transfer of heat is predominated by conduction. The wall heat flux q_w is as following:

$$q_w = -\lambda \frac{\partial T}{\partial y} = -\lambda \left(\frac{T - T_w}{y} \right) \quad (2)$$

λ is the thermal conductivity of the fluid and T_w is the temperature of the surface. According to the Newton cooling expressions, the wall heat flux is given by

$$q_w = h(T_w - T_{ref}) \quad (3)$$

T_{ref} is a reference temperature. Because the heat conected to the fluid is equal to that conducted from the surface, Eqs.(2) and (3) can be combined to give an expression for heat transfer coefficient h as follows.

$$h = \frac{\lambda}{y_p} \left(\frac{T_w - T_p}{T_w - T_{ref}} \right) \quad (4)$$

y_p is the first grid distance from the surface.

The local heat transfer coefficient h_x is determined through Eq.(4) on the surface of each block, so the local Nusselt number can be defined as

$$Nu_x = \frac{h_x H}{\lambda} \quad (5)$$

The average heat transfer coefficient h is obtained by integration Eq.(4) over the surfaces of the blocks, and the average Nusselt number is given by

$$Nu = \frac{hH}{\lambda} = \frac{1}{A} \left(\int_A \frac{h_x H}{\lambda} dA \right) \quad (6)$$

3.2 Affirmance of HTCCs calculation

With PHOENICS and Lam-Brehmost $k-\epsilon$ turbulence model, the natural convective heat transfer coefficient in an enclosure is calculated. A non-uniform fine grid of 135×135 is used for simulation while the position of the first grid point ranges from 0.067mm to 1.155mm. Corresponding to the value of y_p , the range of y_p^+ is from 0.02 to 13.6. The results are studied for the effect of y_p on the calculated value of h and are shown in Fig.2(a).

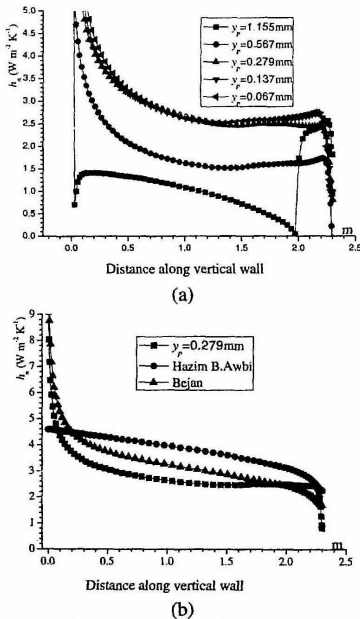


Fig.2: (a) Effects of the first grid positions on h , (b) comparisons with data obtained by Awbi [3] and Neiswanger [5]

As we can see in Fig.2(a), when $y_p = 1.155\text{mm}$ and $y_p = 0.567\text{mm}$, the calculated value of h is greatly different from other cases. It probably results from the fact that the position of the first point locates in the buffer zone. The results of rest three cases accord with each other well. the value of y_p should not be more than 0.5mm , but this value is not valid universally. In Fig.2(b), the calculated h for the case of $y_p = 0.279\text{mm}$ ($y^+_p = 6.3$) is compared with other literatures [5,6]. The current results of h is a little lower than those two results, but agree very well with Bejan's results. In the mass, our results are receivable.

4. THREE CASES FOR CONVECTIVE HEAT TRANSFER WITH TEN ISOLATED BLOCKS IN AN ENCLOSURE

4.1 Natural, forced and mixed convection

The convective heat transfer in the enclosure with ten isolated blocks of the following three cases is simulated with PHOENICS together with the Lam-Bremholt $k-\epsilon$ turbulence model:

1. Natural convection;
2. Forced convection;
3. Mixed convection.

For all these cases, Eq.(4) is used to obtain the local surface HTCCs. In CFD calculations, the results are dependent upon grid number, and the grid-independent solutions must be obtained first.

4.2 Natural convection

In order to get the grid-independent results, a series of calculations with different set of grid are performed. The number of grids is various and ranges from 65×75 to 135×155 . After comparisons, the number of non-uniform grid is selected as 85×135 . With the Grashof number in the range of 1.0×10^9 to 1.0×10^{10} , the solution of natural convection flow and heat transfer in an enclosure with ten internal isolated blocks is researched. The value of y^+_p on each surface of the blocks is between 0.5 to 5.6, so the first point is in the viscous sublayer.

Local and average HTCCs, Nusselt numbers are obtained with Eqs.(4)–(6) and the calculated results. Under a constant Prandtl number of 0.7, the natural convective heat transfer coefficient can be expressed in a dimensionless form: $Nu_n = c(Gr)^m$, where c and m are two constants. According to the numerical results, the Nusselt number for natural convection is given as

$$Nu_n = 0.379 Gr^{0.265} \quad 1 \times 10^9 \leq Gr \leq 1 \times 10^{10} \quad (7)$$

4.3 Forced convection

Concerned with the heat transfer between a moving fluid and a solid surface, forced convection is caused by external means. Suppose the effect of natural convection is neglected, the forced convective heat transfer coefficient of air may be expressed in a dimensionless form: $Nu_f = c(Re)^m$. The non-uniform grid number was selected as 65×115 in the calculation, and the Reynolds number is various and ranges from 5.0×10^4 to 1.5×10^5 . The range of y^+_p on each surface of the blocks is from 1.8 to 8.6. According to the numerical results, the Nusselt number is as follows:

$$Nu_f = 0.231 Re^{0.657} \quad 5 \times 10^4 \leq Re \leq 1.5 \times 10^5 \quad (8)$$

4.4 Mixed convection

Combining the effect of natural to forced convection, mixed convection is very complicated and has known little about it so far. Commonly, the Archimedes number is applied as a crucial parameter to judge which kind of effect is predominant. The Ar number is expressed as:

$$Ar = Gr / Re^2 \quad (9)$$

when $Ar \ll 1$, the heat transfer mechanism is predominated by forced convection and by natural convection as $Ar \gg 1$. When $Ar \approx 1$, both effects can not be negligible.

In this study, the combined effect of natural and forced convection on the convective heat transfer coefficient has been researched. Archimedes number is varied from 0.1 to 4.6. CFD simulations are performed with a grid of 85×135 . The range of y^+_p on each surface of the blocks is from 1.5 to 7.9. The dimensionless Nusselt number versus Archimedes number is shown in the following figure.

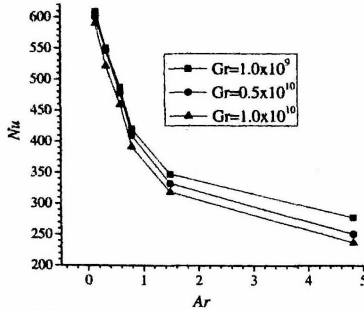


Fig.3: Variation of the Nusselt number due to the Archimedes number

In Fig.3, for the same Archimedes number, the difference in Nusselt number is very small with Grashof number, which indicates that the forced convection dominates and the Nusselt number varies with Reynolds number. when $Ar < 1$, the Nu number is at a higher level and it is opposite while $Ar > 1$, which then indicates that the natural convection dominates and the Nusselt number varies apparently with Grashof number. Both two effects are important in this case, and then the mixed flow is modeled by "blending" the influences of the buoyant and mechanical forces using Churchill-Usagi approach [7]. In this approach, a general expression for correlating heat transfer process in which asymptotic solutions (X and Y) are known for small and large values of the independent variable.

$$Z = (X^\alpha + Y^\alpha)^{1/\alpha} \quad (10)$$

α is the Churchill-Usagi blending coefficient, a constant should be selected to best represent the data of Z .

When the buoyant and mechanical forces are both significant and assisting for the mixed convection, the HTCCs can be calculated as

$$h_m = (h_n^\alpha + h_f^\alpha)^{1/\alpha} \quad (11)$$

A series of numerical simulation for mixed convection are presented in this paper, in these cases the buoyant and the mechanical forces are both assisting, with the Archimedes number varying from 0.1 to 5.0. Using the approach mentioned above, the expression of Nusselt for mixed convection is defined as

$$Nu_m = (Nu_n^\alpha + Nu_f^\alpha)^{1/\alpha} \quad (12)$$

$$= [(0.379Gr^{0.265})^\alpha + (0.231Re^{0.657})^\alpha]^{1/\alpha}$$

Compared with the results for pure natural and forced convection, the value α is found to be from 3.0 to 3.8. With the average of 3.0 and 3.8, the Nusselt number for mixed convection is given by

$$Nu_m = (Nu_n^{3.4} + Nu_f^{3.4})^{1/3.4} \quad (13)$$

$$= [(0.379Gr^{0.265})^{3.4} + (0.231Re^{0.657})^{3.4}]^{1/3.4}$$

Comparisons of the Nusselt number is shown in Fig.4. The maximal error of the Nusselt number is 12% approximately. It is obvious that Eq.(13) predicts results of Nu well, thus it accurately combines the effect of natural and forced convection.

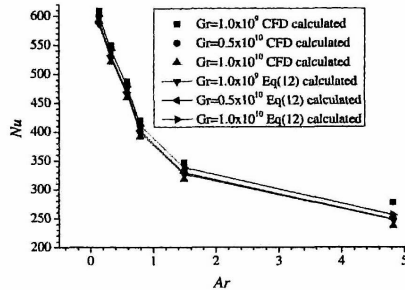


Fig.4: Comparisons of the Nusselt number

5. CONCLUSION

Based on handling of regions close to wall, the equation for convective heat transfer coefficient is acquired. Meanwhile, the dimensionless local Nu number is also obtained as well as average Nu number.

For the enclosure with convective heat transfer, to affirm the applied model, the influence of first grid point position is analyzed and the position of the first grid point is suggested to be less than 0.5mm

The dimensionless Nusselt numbers are derived by the obtained results for both the natural and forced convection. For the case of mixed convection, the HTCCs are established by blending those for natural and forced convection using the Churchill-Usagi approach, which is a general expression combines the asymptotic solutions of independent HTCCs into the mixed convection by using a Churchill-Usagi blending coefficient of 3.4.

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