Выводы

Показано, что максимальное накопление и отдача теплоты в дисперсном материале двуокиси ванадия происходит при фазовом переходе в условиях адсорбции-десорбции кислорода. Количество запасаемой теплоты зависит от вкладываемой механической энергии при диспергировании исходного оксида ванадия. Численные расчеты показали, что влияние оптических характеристик материала на процесс нагрева усиливается после завершения фазового перехода. Полученные результаты могут быть использованы при создании высокоэффективных преобразователей-накопителей теплоты.

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FLOW AND HEAT TRANSFER IN A PARTITIONED ENCLOSURE

Числово досліджено течію та тепловіддачу при природній конвекції у секційній камері квадратного перерізу при нагріві бокових стінках зі сталою температурою та адіабатних верхніх та нижніх стінках. Розрахунки показали, що зі збільшенням чисел Ra коефіцієнти тепловіддачі суттєво зростають. При збільшенні відстані вертикальних стінок від перегородки середнє число Nu істотно зменшується.

Численно исследованы течение и теплоотдача при естественной конвекции в секционной камере квадратного сечения при прогретых боковых стенках, имеющих постоянную температуру, и адиабатных верхней и нижней стенках. Расчеты показали, что с увеличением чисел Ra коэффициенты теплоотдачи существенно возрастают. При увеличении расстояния вертикальных стенок от перегородки среднее число Nu значительно уменьшается.

Buoyancy driven flow and heat transfer in a partitioned square enclosure having differentially heated isothermal walls and adiabatic horizontal walls were studied numerically. The results show that as the Rayleigh number increases heat transfer rate increases substantially. With using a partition between the vertical walls of the enclosure, average Nusselt number decreases considerable amount.

g – gravitational acceleration;
H – enclosure height;
k – thermal conductivity;
L – enclosure width;
Nu – Nusselt number;
Pr – Prandtl number;
p – pressure;
R – residue;
Ra – Rayleigh number;
r_k – thermal conductivities ratio;
r_w – partition thickness;
T – temperature;
u, v – velocity components in x, y directions;
w – partition thickness;
x, y – coordinate;
α – thermal diffusivity;
β – thermal coefficient of volume expansion;
γ – kinematic viscosity;
η – outward normal variable to the surface;
ρ – density;
ω – vorticity;
ψ – stream function.

Subscripts

a – average;
C – cold;
f – fluid;
H – hot;
p – partition.

Superscripts

$*$ – dimensional quantities.

1. Introduction

Steady state laminar natural convection is an area of interest because of its wide applications in engineering, as comprehensively reviewed by Ostrach [1]. Previous studies related to this subject were mainly concerned with the nonpartitioned enclosure. Recently, the interests of researchers included the partitioned enclosures. Ho and Yih [2] investigated steady state laminar natural convection in an air filled partitioned rectangular enclosure numerically. They found that heat transfer rate is significantly reduced in a partitioned enclosure comparing with that for nonpartitioned enclosure. Tong and Gerner [3] studied the same problem with a thin partition. Numerical results show that placing a partition midway between the vertical walls of an enclosure causes the greatest reduction in heat transfer. The results of Acharya and Tsang [4] for an inclined enclosure with a centrally located partition reveal that inclination angle has a strong influence on the magnitude of the maximum Nusselt number.

Most numerical simulations of natural convection problems in enclosures have used low order finite difference, finite elements and finite volume methods [2–4]. To achieve an acceptable degree of accuracy, low order methods require the use of a large number of grid points. On the other hand, global PDQ method can obtain accurate numerical results with less grid points [5–7].

The aim of the present study is to investigate numerically steady state laminar natural convection in a two dimensional partitioned square enclosure located off-centrally using polynomial based differential quadrature (PDQ) method.

2. Analysis

The study domain is a two dimensional square enclosure with partition as shown in Figure 1. The thickness of the partition was taken fixed and equal to one tenth of the width of the enclosure. The vertical walls of the enclosure were taken at different uniform temperature while the horizontal walls were adiabatic. The conjugate heat transfer boundary conditions were applied at both sides of the partition wall.

The dimensionless variables are defined as follows:

$$x=\frac{x^*}{L}\ ,\ y=\frac{y^*}{L}\ ,\ x_p=\frac{x_p^*}{L}\ ,\ r_w=\frac{w}{L}\ ,\ r_k=\frac{k_f}{k_p} \qquad (1)$$

$$u=\frac{u^*}{\alpha/L}\ ,\ v=\frac{v^*}{\alpha/L}\ ,$$

$$p=\frac{L^2}{\rho\alpha^2}(p^*+\rho_0 g y^*)\ ,\ T=\frac{T^*-T_C}{T_H-T_C}\ , \qquad (2)$$

where u^* and v^* are the dimensional velocity components, p^* is the dimensional pressure, T^* is the dimensional temperature, ρ is the fluid density and α is the thermal diffusivity of the fluid. Thermal conductivities of the fluid and the partition are k_f and k_p respectively and rk and rw are the thermal conductivities ratio and the dimensionless partition thickness respectively.

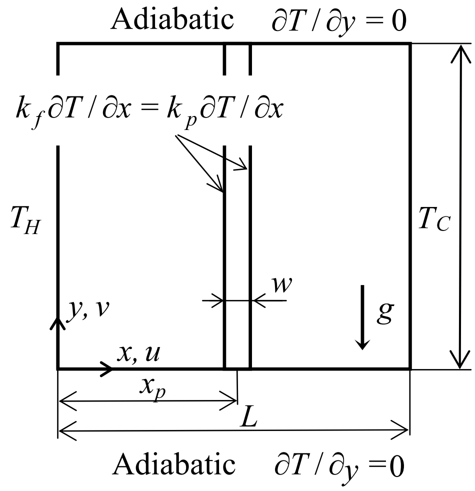


Figure 1. Geometry and the coordinate system.

The flow was assumed to be two-dimensional and laminar. The fluid was assumed to be incompressible, with constant physical properties. The buoyancy effect was taken into account through the Bousinesq approximation. The viscous dissipation terms and the thermal radiation were neglected.

Once the above assumptions are employed, the nondimensional equations describing steady state flow are obtained as follows:

$$\frac{\partial^2\psi}{\partial x^2}+\frac{\partial^2\psi}{\partial y^2}=-\omega \tag{3}$$

$$u\frac{\partial\omega}{\partial x}+v\frac{\partial\omega}{\partial y}=\Pr\left(\frac{\partial^2\omega}{\partial x^2}+\frac{\partial^2\omega}{\partial y^2}\right)+\mathrm{Ra}\,\Pr\frac{\partial T}{\partial x} \tag{4}$$

$$u\frac{\partial T}{\partial x}+v\frac{\partial T}{\partial y}=\frac{\partial^2 T}{\partial x^2}+\frac{\partial^2 T}{\partial y^2} \tag{5}$$

$$\frac{\partial^2 T}{\partial x^2}+\frac{\partial^2 T}{\partial y^2}=0\,. \tag{6}$$

Appearing in equation 4 $\Pr=\gamma/\alpha$ is the Prandtl number and $\mathrm{Ra}=g\beta L^3\Delta T^*/\gamma\alpha$ is the Rayleigh number.

Dimensionless stream function and vorticity are defined as follows:

$$u=\frac{\partial\psi}{\partial y}\,,\ v=-\frac{\partial\psi}{\partial x}\quad \omega=\frac{\partial v}{\partial x}-\frac{\partial u}{\partial y}\,. \tag{7}$$

The related boundary conditions are as follows:

$$\psi(x,0)=0\ ,\ \left.\frac{\partial T}{\partial y}\right|_{x,0}=0\quad \psi(x,1)=0\ ,\ \left.\frac{\partial T}{\partial y}\right|_{x,1}=0 \tag{8}$$

$$\psi(0,y)=0\ ,\ T(0,y)=1\quad \psi(1,y)=0\ ,\ T(1,y)=0 \tag{9}$$

$$\psi(x_p-0{,}5r_w,y)=0\ ,\ \left.\frac{\partial T_p}{\partial x}\right|_{x_p-0,5r_w,y}=r_k\left.\frac{\partial T}{\partial x}\right|_{x_p-0,5r_w,y} \tag{10}$$

$$\psi(x_p+0{,}5r_w,y)=0\ ,\ \left.\frac{\partial T_p}{\partial x}\right|_{x_p+0,5r_w,y}=r_k\left.\frac{\partial T}{\partial x}\right|_{x_p+0,5r_w,y} \tag{11}$$

Physically, there is no boundary condition for vorticity. But an expression can be written from the stream function equation as follows:

$$\omega_{\mathrm{wall}}=-\frac{\partial^2\psi}{\partial\eta^2}\ , \tag{12}$$

where η is the outward normal variable to the surface.

The local Nusselt number is given by

$$\mathrm{Nu}=-\frac{\partial T}{\partial\eta}\ . \tag{13}$$

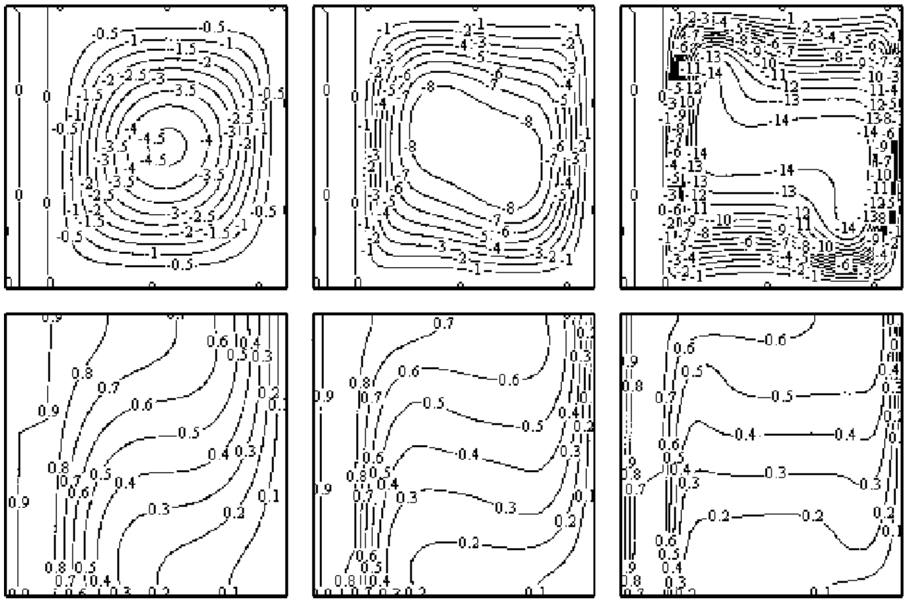
3. Results and Discussions

Numerical simulations have been performed for Pr = 0,71 with Ra varying from 10^3 to 10^6, x_p varying from 0,1 to 0,5. Thermal conductivities ratio and dimensionless partition thickness have been taken constant ($r_k=0{,}01$, $r_w=0{,}1$). Derivatives in the equations have been discretized by PDQ method using a nonuniform grid point distribution given below:

$$x_i=\frac{1}{2}\left[1-\cos\left(\frac{i}{n_x}\pi\right)\right],\quad i=0,1,2,.,\ n_x$$

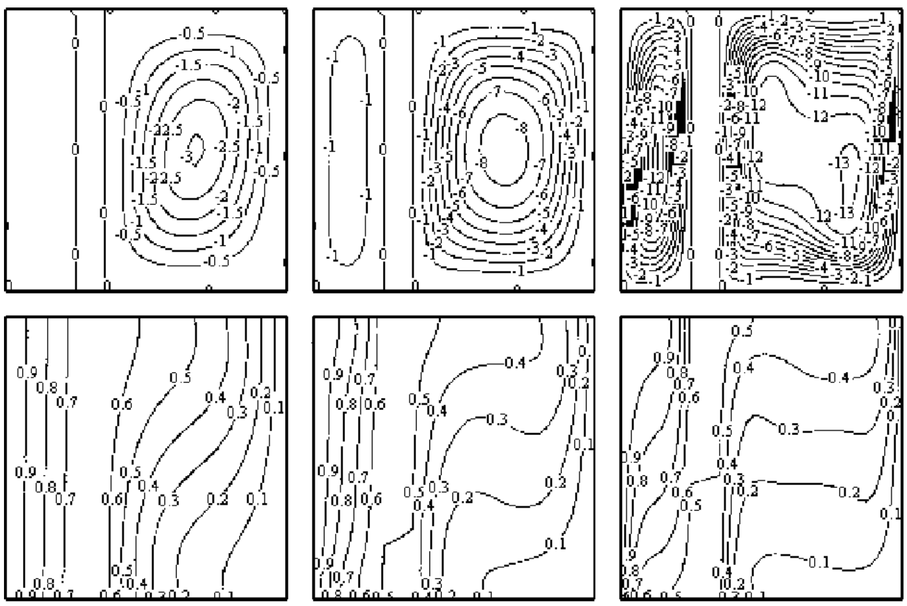
$$y_j=\frac{1}{2}\left[1-\cos\left(\frac{j}{n_y}\pi\right)\right],\quad j=0,1,2,.,\ n_y\ . \tag{14}$$

After discretization, governing equations were solved by the successive over relaxation (SOR) method for values of parameters taken into consideration. Variations by less than 10^{-6} over all grid points for all dependent variables were adapted as the convergence criterion. It was found that the minimum mesh size to get the grid independent



Ra = 10^4 Ra = 10^5 Ra = 10^6

Figure 2. Streamlines and isotherms for x_p = 0,1.



Ra = 10^4 Ra = 10^5 Ra = 10^6

Figure 3. Streamlines and isotherms for x_p = 0,3.

solution by PDQ approach is between 20×20 to 40×40 depending on the values of Rayleigh number. Hence, mesh sizes given above were selected in the present study.

For the validation of numerical code, the solution obtained for a nonpartitioned square enclosure has been compared with the benchmark results obtained by Vahl Davis [8]. Particularly, maximum deviation for the average Nusselt number between the present and Vahl Davis [8] results is 0,3%.

Streamlines and isotherms for x_p = 0,1, 0,3, 0,5 are shown in Figures 2–4. It can be observed that the flow on either side of the partition consists of a clockwise rotating circulation. For low Rayleigh number, the heat transfer across the enclosure is evidently conduction dominated, as indicated by the uniformly distrib-

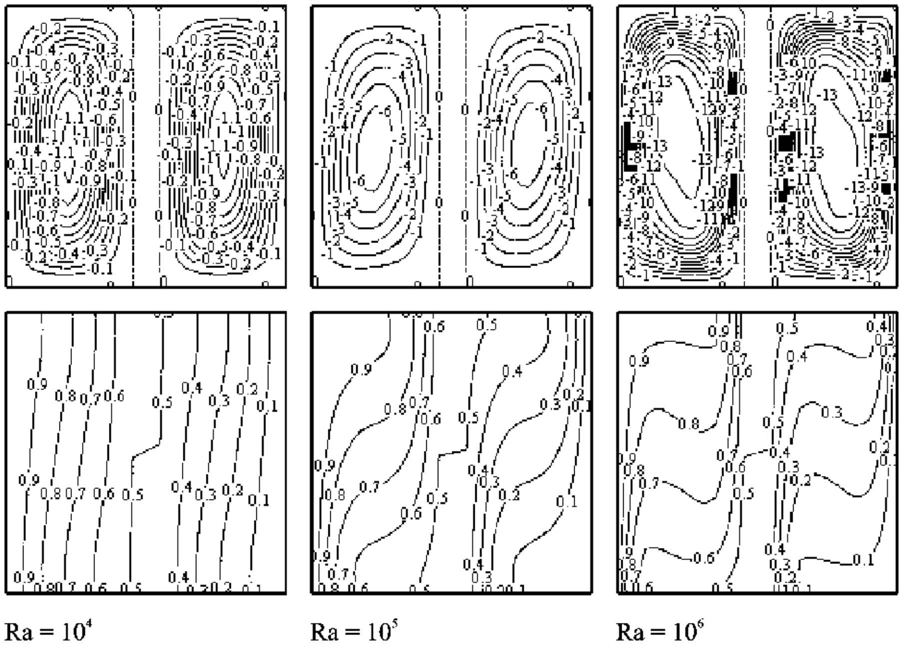


Figure 4. Streamlines and isotherms for x_p = 0,5.

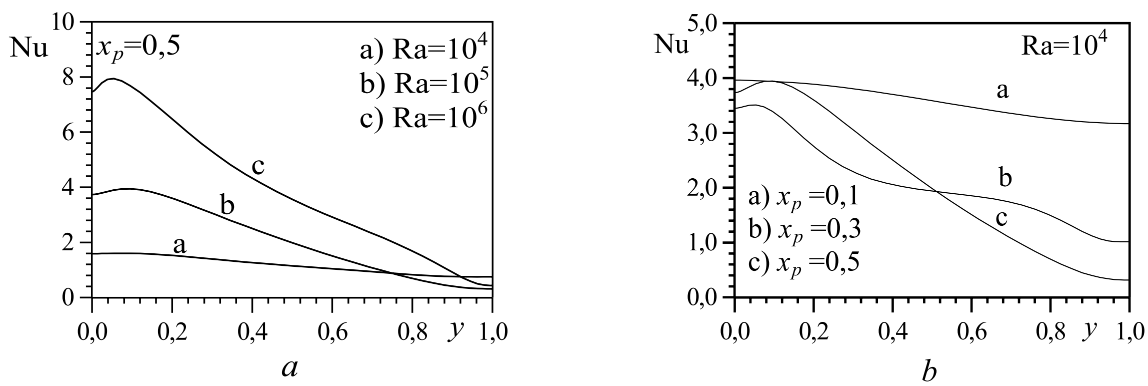


Figure 5. Variation of the local Nusselt number for a) x_p = 0,5, b)* Ra *= 100000.

uted isotherms. As the circulation becomes stronger with increasing Rayleighh number, isotherms undulate remarkably and thus the effect of convection is pronounced. As x_p increases, circulation on the right side weakens while strengthens on the left side.

The local Nusselt number along the hot surface of the enclosure is presented in Figure 5. The decrease in Nusselt number in the upward direction occurs due to the gradual heating of the fluid, in contact. Besides, as the fluid moves upward along the hot surface, a progressively increasing boundary layer thickness offers increasing resistance to heat flow. For low values of Rayleigh number and x_p, local Nusselt number exhibits more uniform structure because of weak convection.

The variation of average Nusselt number with Ra and x_p is given in Figure 6 along with the values for nonpartitioned case for comparison. As it is seen from the Figures, using a partition between vertical walls leads to a substantial reduction in average Nusselt number especially for high values of Rayleigh number. With increasing Rayleigh number, average Nusselt number increases due to the strengthening effect of circulation on convection. Average Nusselt number decreases with increasing x_p and takes its minimum value for the case that partition in the middle of the enclosure for relatively low values of Rayleigh number. For high Rayleigh numbers the average Nusselt number takes its minimum value for smaller values of x_p.

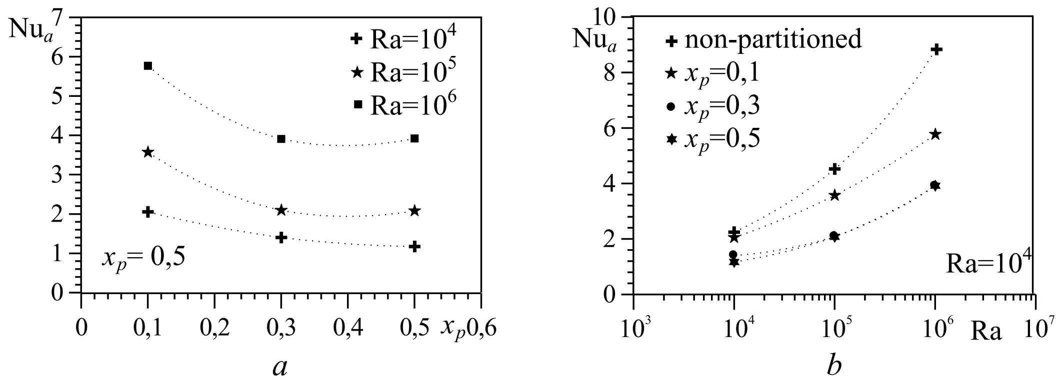


Figure 6. Variation of the average Nusselt number for a) x_p = 0,5, b)* Ra*=100000.

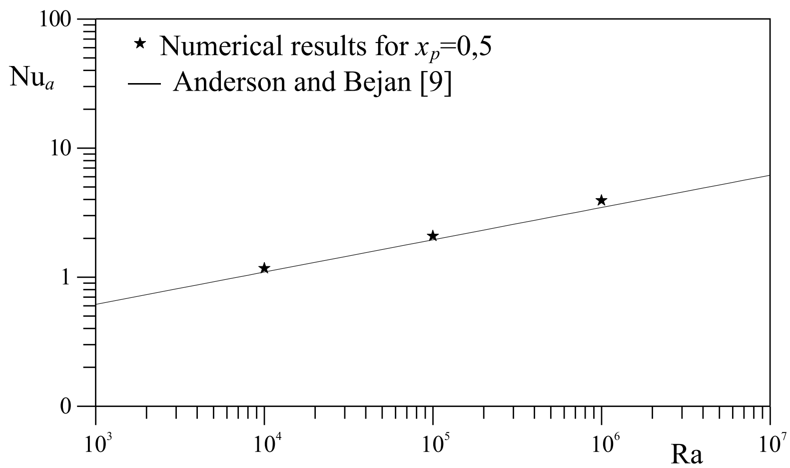


Figure 7. A comparison for the average Nusselt number.

A correlation for average Nusselt number for present configuration was given by Anderson and Bejan [9] based on their experimental results as follows:

$$\mathrm{Nu}_a = 0,201\mathrm{Ra}^{0,276}(N+1)^{-1/4}. \qquad (15)$$

The comparison of the results with correlation given above is presented in Figure 7. As it is seen from the Figure 8, the numerical results for average Nusselt number are in good agreement with those of the correlation given by Anderson and Bejan [9].

Conclusion

Buoyancy driven flow and heat transfer has been investigated numerically by the PDQ method. The computational results show that the PDQ method can be used successfully in the numerical simulation of the natural convection in partitioned enclosures. For the high values of Rayleigh number, average Nusselt number takes the higher values. With using a partition between the vertical walls of the enclosure, average heat transfer rate decreases considerable amount.

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НЕСТАЦИОНАРНАЯ ЛАМИНАРНАЯ СМЕШАННАЯ КОНВЕКЦИЯ В ВЕРТИКАЛЬНОМ ПЛОСКОМ КАНАЛЕ ПРИ ПОПУТНЫХ НАПРАВЛЕНИЯХ ПОТОКОВ

Подано результати числового дослідження нестаціонарної ламінарної змішаної конвекції у плоскому вертикальному каналі для супутних напрямів потоків. Результати числового моделювання тепловіддачі порівняно з даними, отриманими у Литовському енергетичному інституті, і дають задовільну збіжність на початку канала, тобто у зоні стійкої ламінарної течії.

Представлены результаты численного исследования нестационарной ламинарной смешанной конвекции в плоском вертикальном канале для попутных направлений потоков. Результаты численного моделирования теплоотдачи сравнены с экспериментальными данными, полученными в Литовском энергетическом институте, и показывают хорошее совпадение в начале канала, т.е. в зоне устойчивого ламинарного течения.

The results on the numerical modeling of the unsteady laminar mixed convection in the vertical flat channel for aiding flows are presented in this paper. The results of heat transfer modeling are compared to the experimental data obtained at the Lithuanian energy institute and show good agreement in the region of the stable laminar flow.

b – ширина канала;
Bo – параметр термогравитации, Bo = Grq/Re;
d_e – гидравлический диаметр, $d_e = h_b/(h + b)$;
Grq – число Грасгофа по тепловому потоку; $Gr_q = g \cdot \beta \cdot d_e^4 \cdot q_w/\nu^2 \cdot \lambda$;
g – ускорение свободного падения;
h – высота плоского канала;
Nu – число Нуссельта, $Nu = \alpha d_e/\lambda$;
p – давление;
q – плотность теплового потока;
Re – число Рейнольдса, $Re = u_b d_e/\nu$;
t –время;
T_b – среднемассовая температура;
u – среднемассовая скорость;
x – расстояние от начала обогрева;
x, y – декартовы координаты;
α – коэффициент теплоотдачи, $\alpha = q_w/(T_w - T_b)$;
β – коэффициент объемного расширения;
λ – коэффициент теплопроводности;
ν – кинематический коэффициент вязкости;
μ – динамический коэффициент вязкости.

Индексы

b – среднемассовые;
cr – критический;
in – на входе;
w – на стенке;
x, y – по координатам.