Natural Convection in a Square Cavity Filled with Saturated Porous Media and Partially Heated From Below.

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Abstract. This paper represents an experimental study of natural convection heat transfer in a square enclosure filled with saturated porous medium and partially heated from below. Two locations (left and middle of the enclosure) has been studied in the present work to explain heat and temperature distribution inside the enclosure. The experimental results obtained under constant heat flux within the range of (1000-10000 W/m²), and modified Rayleigh number within the range of (0 < Ra_m < 420). The experimental results are presented in the form of temperature distribution, Nussalt and Rayleigh number are plotted versus heat flux, and for both cases. The results were expressed as isotherms, distributions. The location of the heating element has a noticeable effect on the distribution of heat and temperature. The study indicated that the Nusselt number depends on the Rayleigh number and is directly proportional to it. Furthermore, two empirical equations were obtained for the considered study cases illustrated the correlation between the Nusselt and Rayleigh numbers.

Keywords. Natural convection, Porous media, Square cavity.

1. Introduction

In many real-life work fields, such as petroleum technology, mechanical engineering, civil engineering, environmental engineering, and agricultural engineering, the theme of free convective heat transfer in square boxes filled with saturated porous media is important. Due to their potential applications in many sciences and technologies, D A Nield, [1], P H Osthuizen, [2], Z E Heinemann, [3], V Prasad, [4] have become an intensive research subject in the last few decades, studying numerically the two-dimensional steady natural convection in a porous rectangular cavity heated from the side for different values of (width/height) ratio and Rayleigh numbers. Their findings suggested the existence of multicellular flow. Also, depending on the adjusted Rayleigh number (Ra *), the average Nusselt number was maximum in a restricted aspect ratio range (width/height). They defined a criterion for the existence of different flow regimes in terms of aspect ratio and (Ra *). AMisirliglu [5] investigated the steady-state free convection numerically inside a cavity made of two straight horizontal walls and two vertical bent wavy walls. A profile of the cosine curve was presumed to follow the wavy walls. The horizontal walls were kept adiabatic, while isothermal but retained at different temperatures were the bent wavy walls. With the Galerkin Finite Element Method (FEM), the Darcy and energy equations (in nondimensional stream function and temperature formulation) were solved numerically. They observed that for large values of the Rayleigh number (Ra =1000), and moderate values of the aspect ratio and surface waviness (k =0.5), the local Nusselt number from the
vertical walls might even become negative. W Pakdee [6] conducted a study on the transient natural convection flow through a fluid-saturated porous medium in a square enclosure with a convection surface condition. The cavity was insulated, except the top wall was partially exposed to outside ambient. The exposed surface allows the convective transport through the porous medium, generating thermal stratification and flow circulations. It was found that the heat transfer coefficient, Rayleigh number, and Darcy number considerably influenced the characteristics of flow and heat transfer mechanisms. Furthermore, the flow pattern was found to have a local effect on the heat convection rate. The cooling and heating flow directions were the opposite. Owing to greater buoyancy effects, cooling flows were much heavier, suggesting a higher overall convection rate. Relative to the near-wall areas, the heat transfer rate was faster along the vertical symmetric axis. The high Rayleigh number values increased the streamline's intensity, thus increasing the penetration of the downward flow. Deeper into the bottom wall, the temperature stratification penetrated, and the temperature spectrum within the domain was expanded. Therefore, it enlarged the region where convection mode was significant. Small values of Darcy number hinder the flow circulations. Therefore the heat transfer by convection was considerably suppressed. H Oztop [7] investigated the natural convection heat transfer numerically in a partially cooled and inclined rectangular enclosure filled with a saturated porous medium. There was a steady hot temperature on one of the sidewalls, and one of the adjacent walls was partly cooled, while the remaining ones were adiabatic. The simple algorithm was used to implement a finite volume-dependent finite difference method. Rayleigh number (10 ≤ Ra ≤ 1000), heater position core (0.1 ≤ c ≤0.9), inclination angle (0o ≤ φ ≤ 90o), and cooler length (0.25 ≤ w ≤ 0.75) were the governing parameters. He found that the inclination angle was the dominant parameter on the heat transfer and fluid flow and aspect ratio. The following were also found: (1) The heat transfer increased with the increasing the Rayleigh number for all parameters governing the flow and temperature field. (2) Due to the partially cooler plume-like flow did not exist for all cases on the contrary of partially heating, which is mostly studied in the literature.

T Basak, [8] investigated the natural convection flow in a square cavity filled with a porous matrix numerically. To simulate the transfer of momentum in the porous medium, they used the Darcy Forchheimer model. It was found that the heat transfer was mainly due to the Darcy number conduction regardless of Rayleigh (Ra) and Prandtl (Pr) numbers. They found that the temperature contour lines compressed towards the sidewalls at the beginning of the dominant convection mode and appeared to deform upward. When (Ra=106 and Da≥10-4), the thermal boundary layer was developed near the bottom and sidewalls, and the central regime near the top surface had the least temperature gradient for both uniform and non-uniform heating cases. The local Nusselt numbers at the bottom and side walls represent various interesting heating features. In the bottom wall center, the non-uniform heating exhibits greater heat transfer rates than those for all Rayleigh number regimes with the uniform heating event. H. In a square region filled with a fluid-saturated porous medium with non-uniform internal heating and heated laterally. Saleh, [9] investigated unsteady natural convection. The temperature of the heated wall surface varied sinusoidally with the time of a set mean temperature.

2. Experimental apparatus

The test section is designed and manufactured to fulfill the test system's requirements for a free convection heat transfer. The experimental apparatus consists of: cooling system, test section, measuring devices, and heating element, which is the element to heat the thermal power of this device and the element, Figure 1.

2.1. Test section

The test section is illustrated in Figure (2), which is an enclosure that has a square shape with (54*54 cm) and has a width of (3.4cm) made of fiberglass material. It is filled with spherical glass with a diameter (11.5 mm); the two vertical walls were fully insulated, while the upper horizontal wall was kept at constant low temperature, and the lower horizontal wall was partially heated. The front face of the test section has 25 holes, the hole diameter is 16mm, and their depths are 3.5mm and another hole diameter with 5 mm and distance of 16.5mm. Thermocouples were set in these holes to measure the
temperature in different points of the section of the test, the distance between a thermocouple and another (12.5cm), as well as a hole in the back side of a section of five similar holes test mode at each thermocouple hole and put one of these thermocouples in the center while the remaining four was distributed on the other parts with the distance between them is (50cm). The purpose of these five thermocouples is for measuring the temperatures to compare with those in the front to make sure that a two-dimensional flow. Isolating the test section from the outside by using two layers of insulators.

2.2. Cooling system
The cooling system consists of parallel tank surfaces with dimensions of (54 * 3.4 * 10cm) base made of Al-bras, the purpose of the cooling system cooling the surface of the reservoir to the low and constant temperature during the operation tests.

2.3. Heating element
The heating element is a rectangular piece of Al-bras; its length, height, and width are 27cm, 2cm, and 3.4cm, respectively. The heating holes diameter is 1.25cm along the axis of the heating element heats the heating element by heater mainly consists "of the electrical resistance of nickel-chrome and glass tube heat-resistant diameter amount (1.2cm) and the length (25cm). Glass first tube placed" inside the hole heating element and then placed the electrical resistance inside the glass tube, and then drag it between the ends of the plate and then to the power source.

2.4. Measurement system
The measurement system consists of a measuring device to evaluate the electrical power of the equipped heater, measuring the temperature of a section of testing (30 thermocouples), and measuring devices' temperature heating element.

3. Method of calculation
The practical side of the accounts included diameter glass balls, consisting of the central pore and bulk porosity, as well as "on the physical properties of water and central pore as well as the amount of processed preheated temperature and heat transfer coefficient and Nusselt number and the number of Riley developer.

1. heat transfer rate: conducted heat transfer from the heating element can be calculated as follows:

\[ Q_{in} = V \times I \times \cos \theta \]

Where \( \cos(\theta) \) ability factor and value (0.92)

2. The heat flux to the surface of the heating element is calculated based on the following equation:

\[ q_{in} = \frac{Q_{in}}{A_s} \]
Where \((A_s)\) is the inner surface area of the heating element and equal to \((W \times L)\)

3. The heat transfer coefficient is computed using the following equation:

\[ h = \frac{q_{in}}{(T_h - T_s)} \tag{3} \]

4. Nusselt number

\[ Nu = \frac{h \times d_p}{k_e} \tag{4} \]

5. Unearthed the physical properties of water \([10], [11]\) When the average water temperature \((T_a)\) which is calculated from the equation

\[ T_f = \frac{(T_h + T_s)}{2} \tag{5} \]

6. Riley number\([11]\).

\[ Ra^* = Da \times Ra = \frac{\beta g K (T_h - T_s) d}{v_f \alpha_f} \tag{6} \]

\[ Ra = Gr \times Pr = \frac{\beta g (T_h - T_s) d^3}{v_f \alpha_f} \tag{7} \]

4. Results and discussion

Results were presented to a number between Riley developer \((0 < Ra \leq 420)\) has also been the representation of the results of the thermal distribution and draw a relationship between Nusselt number and modify Riley number in the form of a graph display also has experimental relations obtained in practice.

5. Thermal distribution

The drawing area temperatures give a vital way to understand and observe the thermal distribution of the sample studied, as in Figures (3) to (5) thermal distribution of temperatures when the heating element in the left was noted the form (3). The temperatures proven to line constitute a stacked line and roughly parallel to the upper wall of the cold and a little deformed so when you heat flux \((1000 \text{w} / \text{m}^2)\) due to the significant effect of heat where it is connected to the impact of convection are very few. Shown in Figure (4) thermal distribution at the heat flux \((3000 \text{w} / \text{m}^2)\) where we note that the stability of the temperature lines start deformation, where this lines is moving to the top of the wall, left almost vertical to the cold surface due to increased buoyancy force and thus increase the impact of any increase convection currents increase thermal flux and this is what we observe in Figures (5) and (6), where increasing heat flux, causing a noticeable change in the thermal distribution toward the heat upwards due to increased buoyant force.

The second case (heating in the middle) is shown in Figure (7) thermal distribution of temperatures when the heating element in the center as seen from the figure that the stability of the temperature lines parallel lines on the hot surface and approaching the rectitude. The closer to the upper surface cold meaning that transmission heat is mostly by conduction only, while Figure (8) shows the deformation stability of temperatures lines any appearance of influence of pregnancy and the end of the process of conduction the reason for the rise of temperatures lines to the top because of the increased strength of buoyancy. Also, it is noted an increase in temperatures to the top of any increase in thermal gradient, as shown in Figures (9) and (10) due to increase the convection currents resulting from the increase in the number of Riley developer because of increasing heat flux and increase the temperature difference.

6. The effect of the number of Riley developer thermal distribution

Figures (11) and (12) show the variation between the Nusselt number and modified Rayleigh number and for two locations (left and middle of the enclosure). The results presented in Figure (11) and (12) show that the heat transfer from the fluid increases with increasing the square cavity. This is due to the increasing domination of convection heat transfer by increasing the fluid buoyancy force.

7. Conclusions

We can summarize what we got from the results of the process by the conclusions of the following points:

1. The thermal flux a significant impact and clear on the process of heat transfer during natural convection in porous media since higher heat flux leads to an increase in the modified Rayleigh number and increases the Nusselt number of two cases.
2. By drawing a relationship between the Rayleigh number and the Nusselt number has been inferred experimental equations that can be used successfully to predict the thermal behavior of the flow inside the space, as shown below:
   a- Heating from left \( Nu = 1 + 0.31 \times Ra_m^{0.356} \)
   b- Heating from the middle \( Nu = 1 + 0.4 \times Ra_m^{0.38} \)

3. Thermal distribution varies within different depending on the space heating element site.

4. The drawing area temperatures gives an important way to understand and observe the thermal distribution inside the enclosure.

5. The modified Rayleigh number is directly linked to the Nusselt number and for different heat flux values since the increased modified Rayleigh number leads to increasing the Nusselt number of two cases, so the Nusselt number is a powerful function of the number of Riley developer.

Figure 3. Experimental thermal distribution for temperatures at heat flux (1000 w/m\(^2\)) when heating element at left.

Figure 4. Experimental thermal distribution for temperature at heat flux (3000 w/m\(^2\)) when heating element at left.

Figure 5. Experimental thermal distribution for temperatures at heat flux (6000 w/m\(^2\)) when heating element at left

Figure 6. Experimental thermal distribution for temperatures at heat flux (10000 w/m\(^2\)) when heating element at left
Figure 7. Experimental thermal distribution for temperatures at heat flux (1000 w/m$^2$) when heating.

Figure 8. Experimental thermal distribution for temperatures at heat flux (3000 w/m$^2$) when heating.

Figure 9. Experimental thermal distribution for temperatures at heat flux (6000 w/m$^2$) when heating element at middle.

Figure 10. Experimental thermal distribution for temperatures at heat flux (10000 w/m$^2$) when heating element at middle.

Figure 11. Modified Rayleigh number & Nusselt number when heating element at left.

Figure 12. Modified Rayleigh number & Nusselt number when heating element at middle.
8. References

[1] D A Nield, A Bejan 2006 Convection in Porous Media (third ed., Springer, New York)
[2] P H Osthuiizen and D Naylor 1999 An Introduction to Convective Heat Transfer Analysis (Mc Graw-Hill Companies)
[3] Z E Heinemann 2005 Fluid Flow in Porous Media (University of Montan, Petroleum Engineering Department) vol 1 p 39
[4] V Prasa and F A Kulacki 1984 Convective Heat Transfer in a Rectangular Porous Cavity- Effect of Aspect Ratio on Flow Structure and heat Transfer (Journal of Heat Transfer) vol 106 pp 158–165
[5] Ay Misirlioglu, A C Baytas and I Pop 2005 Free Convection in a Wavy Cavity Filled with a Porous Medium (Journal of Heat and Mass Transfer) vol 48 pp 1840–1850
[6] W Pakdee, and P Rattanadecho 2006 Natural Convection in Porous Enclosure caused by Partial Heating or Cooling (The 20th Conference of Mechanical Engineering Network of Thailand, 18-20 October 2006, Nakhon Ratchasima, Thailand)
[7] H Oztop 2006 Natural Convection in Partially Cooled and Inclined Porous Enclosures (Journal of Thermal Sciences) 46 pp 149–156
[8] T Basak, S Roy, T Paul and I Pop 2006 Natural Convection in a Square Cavity Filled with a Porous Medium: Effects of Various Thermal Boundary Conditions (Heat and Mass Transfer) vol 49 pp 1430–1441
[9] H Saleh, I Hashim and N Saeid 2010 Effect of Time Periodic Boundary Conditions on Convective Flows in a Porous Square Enclosure with Non-Uniform Internal Heating (Transp.Porous Med) vol 85 pp 885–903
[10] Muyassar E Ismaeel 2011 Heat Transfer in a Square Porous Cavity With Partial Heating and Cooling for Opposite Vertical Walls (Al-Rafidain Engineering) vol 19
[11] Prakash Chandra and V V Satyamurty 2011 Non-Darcian and Anisotropic Effects on Free Convection in a Porous Enclosure (Transp Porous Med) vol 90 pp 301–320