Wasit Journal of Engineering Sciences

Journal homepage: https://ejuow.uowasit.edu.iq

Optimal Design Of The Concentric Annulus For The Solar Collectors And Energy Storage

Amina Hmoud Alikhan[1*], Khaleel Saleem Jebur Al-Ogaili[1], Hadi Obaid Mery[1], Osamah Malik Mohammed[1], and Mohammed Arkan Al-attabbi[2]

Affiliations
[1]Mechanical Engineering Department, Wasit University, Wasit, Iraq.

[2]Midland Oil Company, Ministry of Oil, Baghdad, Iraq

Correspondence
Amina Hmoud Alikhan,
adhaef@uowasit.edu.iq

Received
10-June-2024
Revised
12-July-2024
Accepted
30-July-2024

Doi: 10.31185/ejuow.Vol12.Iss3.570

Abstract

Heat exchangers with uniform heating ensure even heat distribution, improving heat transfer efficiency and lowering thermal stress. It contributes to increasing the effectiveness of solar energy conversion into thermal energy in solar collectors, improving system performance and sustainability. It is essential for enhancing performance in both home and industrial applications because of these advantages. An experimental investigation was conducted to study the natural convective flow between two isothermal concentric cylinders filled with porous media (silica sand) and equipped with annular fins attached to the inner cylinder. Thefins varied in length (Hf = 3, 7, and 11 mm), and the radius ratios (Rr) betweenthe cylinders were also considered during the study. This work examines and discusses the effects of independent characteristics, such as fin height (Hf), radius ratio (Rr), and Rayleigh number (Ra) from 10 to 500, on heat transfer outcomes. As thedistance between the two cylinders expands, the average Nu remains relativelyconstant for low values of Ra. However, as Ra approaches 100, the average Nuincreases. These values also rose when Rr decreased for the hot cylinder and vice versa for the cold cylinder. The average Nusselt number versus Ra was analyzed in graphical form to show the effect of the fin number on the average Nusselt number. The analysis includes a range of fin numbers. It is found that as the amount of fins increases from n = 12 to n = 23 and eventually to n = 45, there is a decrease in the average Nusselt number. The decrease in average Nusselt number can range from 19.2% to 26.6% for the same value of Ra.

Keywords: Natural convective flow; isothermal concentric cylinders; Silica sand; Radius ratios; Heat transfer; Rayleigh number

الخلاصة :تضمن المبادلات الحرارية ذات التسخين الموحد توزيعًا متساويًا للحرارة، مما يحسن كفاءة نقل الحرارة ويقلل الضغط الحراري .ويساهم في زيادة فعالية تحويل الطاقة الشمسية إلى طاقة حرارية في مجمعات الطاقة الشمسية، مما يحسن أداء النظام واستدامته .إنه ضروري لتعزيز الأداء في كل من التطبيقات المنزلية والصناعية بسبب هذه المزايا .تم إجراء بحث تجريبي لدراسة سريان الحمل الحراري الطبيعي بين أسطوانتين متحدتي المركز متساوي الحرارة مملوءتين بوسط مسامي (رمل السيليكا) ومجهزتين بزعانف حلقية متصلة بالأسطوانة الداخلية .تباينت أطوال الزعانف(Hf = 3 ، 7، و11 ملم)، كما تم أخذ نسب نصف القطر (Rr) بين الأسطوانات في الاعتبار أثناء الدراسة .يدرس هذا العمل ويناقش تأثيرات الخصائص المستقلة، مثل ارتفاع الزعنفة(Hf) ، ونسبة نصف القطر (Rr)، وعدد رايلي (Ra) من 0.2 إلى 500، على نتائج نقل الحرارة .ومع اتساع المسافة بين الأسطوانتين، يظل متوسط Nu ثابتًا نسبيًا عند القيم المنخفضة لـ Ra. ومع ذلك، مع اقتراب Ra من 100، يزداد متوسط Nu. كما ترتفع هذه القيم عندما ينخفض Rr بالنسبة للأسطوانة الساخنة والعكس بالنسبة للأسطوانة الباردة .تم تحليل متوسط رقم نسلت مقابل Ra بشكل رسومي لإظهار تأثير رقم الزعنفة على متوسط رقم نسلت .يتضمن التحليل مجموعة من أرقام الزعانف . لقد وجد أنه مع زيادة كمية الزعانف من n = 12 إلى n = 23 وفي النهاية إلى n = 45 ، يحدث انخفاض في متوسط رقم نسلت .يمكن أن يتراوح الانخفاض في متوسط عدد نسلت من 19.2% إلى 26.6% لنفس قيمة Ra.

1. INTRODUCTION

Heat, in essence, is a form of energy, much like the energy we receive from the sun. There are three primary methods by which energy is transferred: through light, sound, and heat. Heat transfer specifically refers to the transmission of energy from a body with a higher temperature to one with a lower temperature. In general, the techniques used to improve heat transfer are classified into two main categories: passive techniques, which include changing the surface geometry, or using additives for improvement while the other category is active techniques which include adding the influence of external power. Due to the importance of the three-dimensional annular channel in various practical applications of solar energy, heat exchangers, and other engineering applications, researchers have presented many practical and numerical studies in the field of improving the heat transfer of the channel, either by passive or active techniques. Some study the effect of the geometry of the inner cylinder, a rectangular, triangular, conical, or square, on the rate of heat transfer; others use nanomaterials with high thermal conductivity [1–4]. In recent years, there has been a noticeable increase in interest in incorporating fins into the active walls of differentially heated cavities to obtain high heat transfer rates. A number of major applications have sparked this attention. Despite numerous studies on two-dimensional simulations of finned cavities, the opportunities given by three-dimensional geometry have yet to be fully explored in numerical research. While there have been various studies on two-dimensional simulations of finned cavities, numerical studies have not fully explored the possibilities offered by three-dimensional geometry [5]. Under the constraints of an imposed heat flux, Fabbri, G. [6] investigated the impact of viscous dissipation on the heat transfer in a finned tube cooled by fluid in laminar flow. Specifically, the changes brought about by viscous dissipation in the ideal finned tube geometries were examined. The results showed that when viscous dissipation is high, finned tube geometries with flatter fin profiles and bigger channels between the fins perform better in terms of heat transmission than those with low viscous dissipation. To anticipate the average heat transfer coefficient and fin efficiency on the fin of annular-finned tube heat exchangers in natural convection for variable fin spacing, Chen and Hsu [7] used the finite difference approach in conjunction with the least-squares scheme and experimental temperature data. The results show that when fin spacing increases, fin efficiency decreases while heat transfer coefficient values increase. Prasad and Kulacki [8] A numerical analysis was conducted to investigate the phenomenon of free convection in a vertical annulus that was filled with saturated porous media. The vertical walls of the annulus were maintained at constant temperatures, while the horizontal walls were insulated. The results revealed that the convective velocities were found to be greater in the upper half of the annulus compared to the lower half. Furthermore, it was observed that the local rate of heat transfer was significantly higher in the vicinity of the top edge of the cold wall. The Nusselt number always increases as the radius ratio increases though. Free convection in an annulus filled with saturated porous media was studied by [9–12] in horizontal and vertical cylinders. The transient free convection in a vertical cylinder annulus filled with fluid-saturated porous media was numerically investigated by Shivakumara and colleagues [13]. In this study, the top and bottom boundaries were maintained adiabatic, while the inner and outer walls were subjected to uniform heating and cooling. The governing equations solution of this problem had been done with finite difference implicit method. The temperature and velocity fields were significantly modified by the radius ratio, and the rate of heat transfer increased with the radius ratio increased. At a high Rayleigh number the curvature effect on heat transfer was insignificant.

Kumari and Nath [14] Found that the annulus filled with a porous medium has the best insulating effectiveness. [15–18] Studied the effect of using porous media on heat transfer characteristics in an annulus enclosure. Yang and Hwang [19] Showed that heat transfer could be improved by using highly thermal conductive porous media. Khanafer and Chamkha [20] Numerically found that the flow and temperature fields are affected by the rotation of the inner wall of a concentric cylinder. It was concluded that the inclusion of porous media enhances heat transfer at high Darcy numbers and that the effect of using porous media increases by increasing the Rayleigh number. Many of the literature in the field of natural convection in saturated porous media were done during the last few decades. Mostly, these studies were achieved using numerical methods of solution to the governing equations. Very few of these studies used the experimental method, as shown in the previous survey of the literature, as concluded in the literature survey natural convection in horizontal and vertical concentric annulus received increasing attention, concentric cylinder configurations can accommodate various types of heat exchange fluids, including gases, liquids, or even phase-change materials, making them versatile for different applications in both heat exchangers and solar energy systems due to their capacity to efficiently transfer heat, ensure uniform flow distribution, and offer versatility and reliability across a range of thermal applications. The effect of geometric shape and the use of porous materials on the heat transfer of two concentric cylinders was studied numerically, and it was concluded that the Nussel number increases with increasing porosity [21] The effect of using hybrid nanomaterials on heat transfer for two concentric cylinders was also numerically studied, and it was found that the use of nanomaterials affects the distribution of heat, velocity, and Nusselt number [22] .To the authors' knowledge, there is no comprehensive experimental work that studied the combined effect of a ctive techniquesrepresented by using different cylinder structures, and passive techniques represented by using porous media to obtain the best thermal performance in the concentric annulus at different operation conditions. In concentric cylinders with annular fins installed on the inner cylinder, a number of variables, including fin length, fin pitch, cylinder radius ratio, and modified Rayleigh number, are studied in order to derive general correlations that can be used to predict the heat transfer rate by natural convection in relation to its use in overhead applications like heat exchangers and solar collectors.

2. EXPERIMENTAL METHOD

A schematic illustration of the experimental apparatus is presented in Figure 1, and Figure 2. It consists of two aluminum outer cylinders with varying diameters that were made to vary the radius ratio and the length of the fins, and ten aluminum inner cylinders, one without fins and the others with different fins. The test section is made up of ten aluminum inner cylinders of varied sizes (27 mm), (260 mm) long, and (5 mm) thick. The test portion is made up of two aluminum outer cylinders with outside diameters ranging from 100 mm to 82 mm, 4 mm thick, and 260 mm long. In order to heat the inner cylinders, an alternating current was passed through a heater located inside the cylinder, while the outside cylinder was exposed to the surrounding temperature, which is 270 K at its lowest. Temperatures at the inner cylinder surface were measured using six thermocouples of type (K). Thermocouples of type (K) were used to measure the temperatures on the surfaces of the inner and outer cylinders; to fit the thermocouples, a number of 1.5 mm-diameter grooves were bored along the cylinder surface. These grooves were arranged as follows: six on the finned inner cylinder's surface and six on the outer cylinder's surface. Figure 2 depicts in full the test section unit and thermocouple setup used to measure temperature distribution. Details of cylinders listed in Table 1.

Table 1 . Details of cylinders

Aluminum outer cylinders	Outside diameters	Thick	Long	Radius ratio Rr=rin/rout
	100,82,70 mm	4 mm	260 mm	0.293, 0.365 and 0.435 mm
Aluminum inner cylinders	Outside diameters	Thick	Long	Fins length
	27 mm	5 mm	260 mm	3, 7 and 11 mm

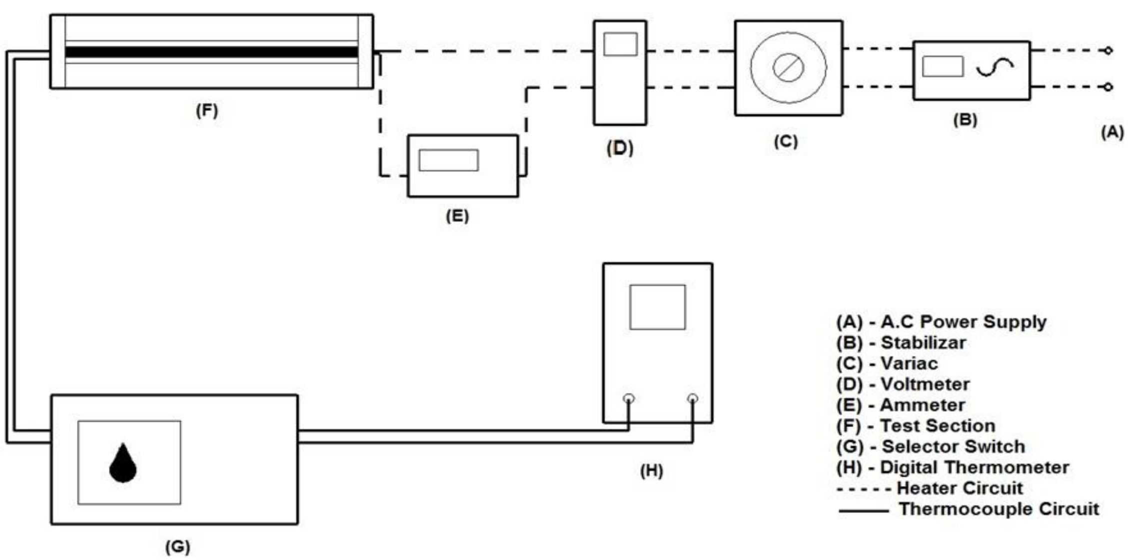


Figure 1 Schematic Diagram of Experimental Apparatus

Figure 2 Real pictures of Inner Cylinder

3. Data reduction

The electric current when passes through a heater, a thermal power is generated and dissipated from the inner cylinder surface, and the total input power (Q**tot**.) supplied to the inner cylinder, all equations are determined from **[Bejan and Nield, 1999]** [23]:

$$Q\mathbf{tot.} = I * V \qquad (1)$$

$$Qtot = Qconv. + Qloss. + Qcond \qquad (2)$$

$$Q_{cond} = \frac{2\pi\ k\ l\ \left(T_1 - T_2\right)}{\ln\left(\frac{r_{out}}{r_{in}}\right)} \qquad (3)$$

$$Q_{loss} = k_t\ A_t \frac{\left(T_{ii} - T_{00}\right)}{L_t} \qquad (4)$$

$$\begin{aligned} Q &= h_i\ A_{in}\left(T_1 - T_2\right) \\ &= h_o\ A_{out}\left(T_1 - T_2\right) \end{aligned} \qquad (5)$$

$Q = Qconv.$ Without fins attached to inner cylinder.

$$Q_{.} = h_o * (\pi\ D_i\ l) * (T_1 - T_2)$$
$$= h_i * \left[\pi\ d_o (l - n\ t) + \frac{\pi}{4}(d_f^2 - d_o^2) * 2n\right] * (T_1 - T_2) \quad (6)$$

Here $Q = Qconv.$ With fins attached to inner cylinder. From equations 5 & 6 hi& ho are calculated, with and without fins on the inner cylinders.

The average Nusselt number on the inner and outer cylinders is given as [Bejan and Nield, 1999]:

$$Nu = \frac{h_* w}{k_{eff}} \quad (7)$$

The modified Rayleigh number which is defined as **[Bejan and Nield, 1999]**:

$$Ra^* = \frac{g\beta K\ (T1 - T2)(rout - rin)}{\upsilon\ \alpha eff} \quad (8)$$

Where w is the gap width = $rout - rin$

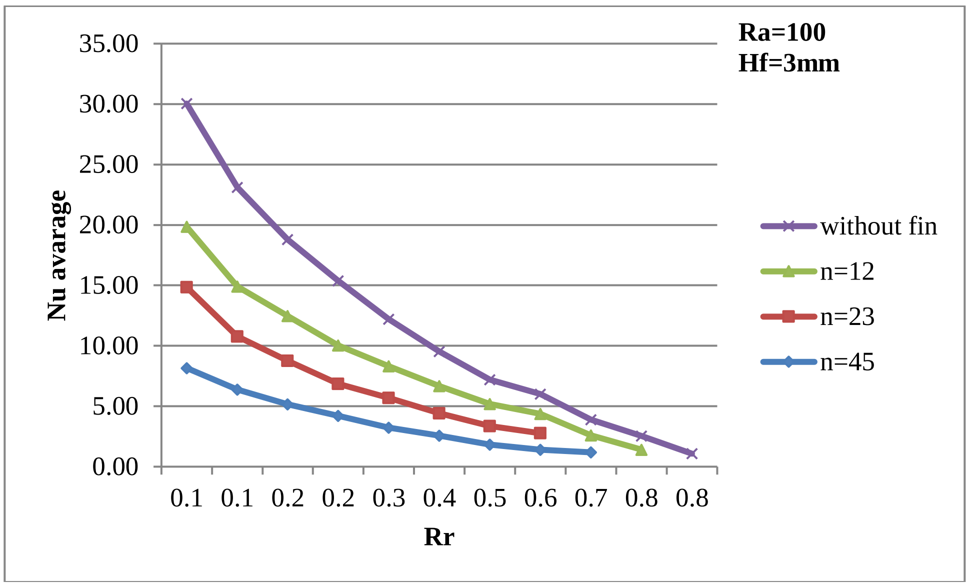
4. RESULT AND DISCUSSION

4.1 EFFECT OF RADIUS RATIO AND FIN LENGTH

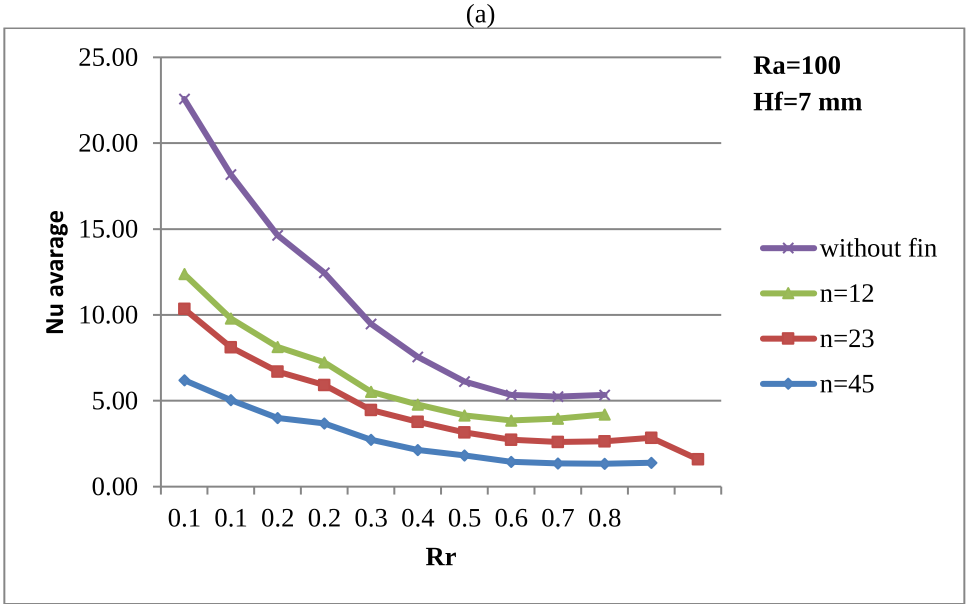
Figure 3 a, b and c show how the radius ratio affects the augmentation rate of heat transfer for the case with and without fins. As shown inthese figures, for Ra equal to 100, when the radius ratio was high, the average Nusselt number values were low, but asthe ratio dropped, they grew with tremendous intensity. Two symmetric convection cells around the vertical midplane characterize buoyancy- driven flow in an annulus surrounded by concentric cylinders. Fluid moves up and down the inner and outer annulus surfaces, respectively, when the annulus inner surface is heated and the annulus outer surface iscooled, as in the current study. The rate of heat transfer decreases as the average temperature of the annulus's inner surface rises. A narrower gap in theannulus increases the resistance to the circulation motion of the convection cells, causing heated air near the inner surfaceto be replaced more slowly by cold air on the outer surface. It is evident that when Ra rises to 500, the curves of Figure 4 a, b, and c indicate a decrease in the radius ratio's influence on the rate of heat transmission. This can be explained by the fact that heat conduction takes over as the primary heat transfer in the fluid layer when Ra drops, making heat convection less important. A comparison of curves indicated that there is a reductionin the average Nu in the presence of long fins as compared with that without fins or with short fins. The reduction in Nu may be ranged between (12.2% - 24.9%). This suggests that if the surface area of the inner cylinder is larger than that of the outer cylinder, the inclusion of annular fins on the inner cylinder's wall will decrease heat transfer and result in the formation of an annulus between the two concentric cylinders. The reason for this reduction is although the annularfins increase the surface area; they also resist the airflow. A decrease in the rate of heat transfer occurs when the reductionin heat transfer coefficient outweighs the increase in surface area. The figure also demonstrate that when Ra raises above 100, Nu falls as Hf rises and Nu lowers when pitch is lowered (by raising fin numbers). The results indicated that When Hf increases from 3mm to 11mm, the average Nusselt number decreases. This determines the appropriate design of the heat exchanger or solar collectors.

4.2 EFFECT OF FIN NUMBER

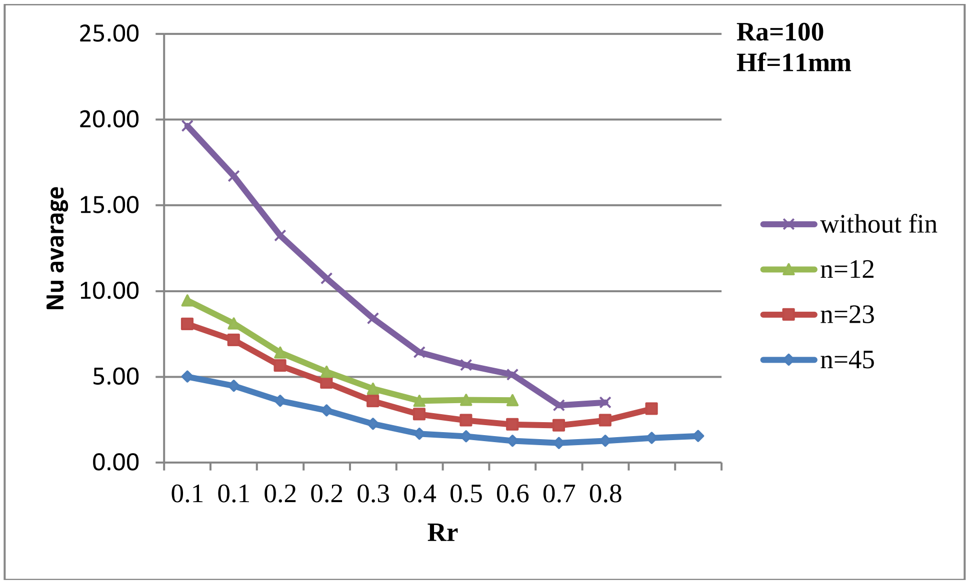
Rayleigh number signified the thermal buoyancy force induced by the differential heating for air in porous media between inner and outer cylinders. This force represents the main mover for fluid flow inside the cavity thereby when the temperature difference ΔT between the inner and outer cylinders increases, the Rayleigh number would be increased; stronger convective currents will be generated between the inner and outer cylinders, and larger heat quantity transported from inner cylinder (heated wall). Figure 5 a, b, c, d, e and f shows the average fluctuation in the Nusselt number on hot and cold cylinders with Ra at various radius ratios (Rr), both with and without fins. According to these graphs, the average Nu grows as Ra approaches 100 and remains almost constant for lower values of Ra for all radius ratios. Due to the two cylinders' greater distance from one another, these values increased as Rr decreased for the hot cylinder and vice versa for the cool cylinder. In the event of a reversal in the scenario, the highest Nu values would correspond to the lowest Rr values. This suggests that at low Ra levels, heat transfer would primarily occur through conduction, while at higher Ra levels, convection heat transfer would become predominant. The maximum Nu value for low Ra values was observed at the highest Rr until Ra reached approximately 100. These graphs demonstrate that when Ra raises above100, Nu falls as Hf rises and Nu lowers when pitch is lowered (by raising fin numbers). The results indicated that When Hf increases from 3mm to 11mm, the average Nusselt number decreases. When Ra decrease is the same value, the average Nusselt number could range between (13.2%, and 23.7 %). The average Nusselt number versus Hf has been analyzed in graphical, form. Figure 6 demonstrates the influence of the fin number, representing the fin pitch, on the average Nusselt number. The analysis encompasses a range of fin numbers. The figure also illustrates that as the quantity of fins rises from n=12 (pitch=19.2 mm) to n=23 (pitch=8.4 mm) and ultimately to n=45 (pitch=3 mm), there is a decrease in the average Nusselt number. A decrease in the average Nusselt number can range from 19.2% to 26.6% for the same value of Ra.



(a)

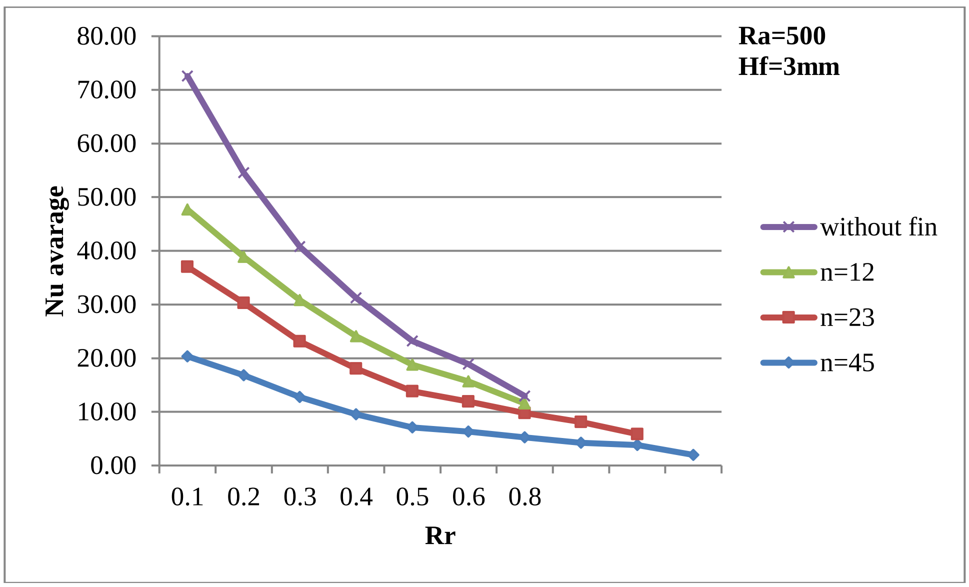


(b)

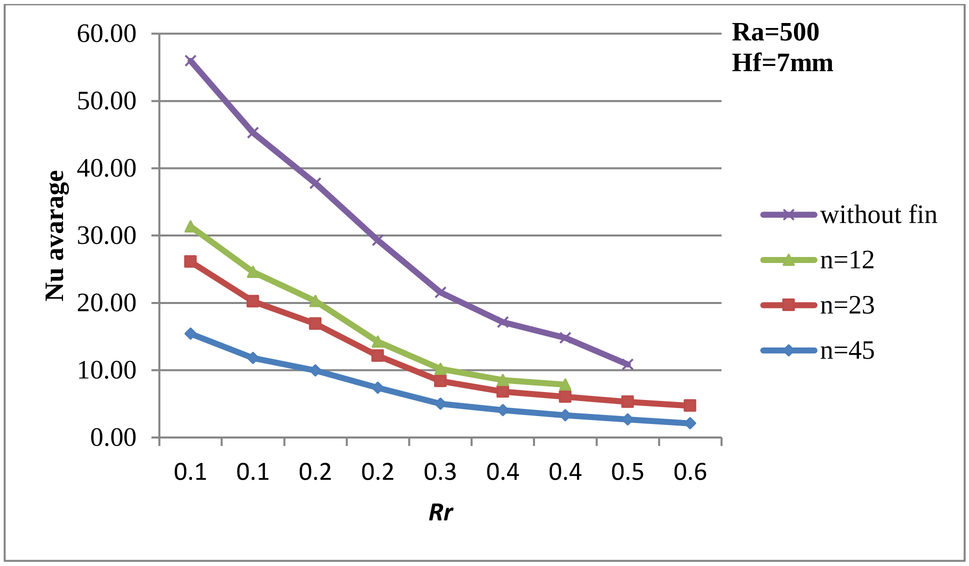


(c)

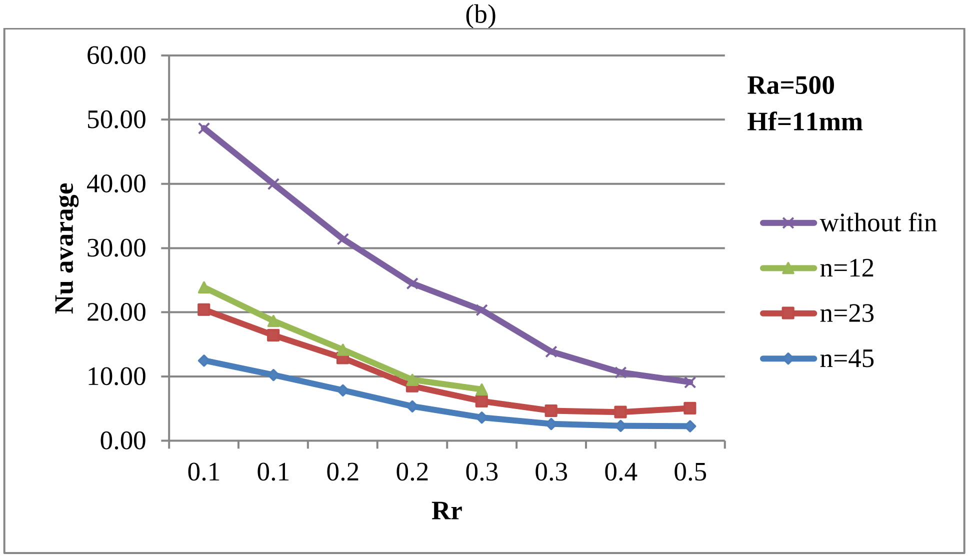
Figure 3 a, b, and c **Variation of Average Nu with Rr at Ra 100, and different Hf, with or without fins.**



(a)

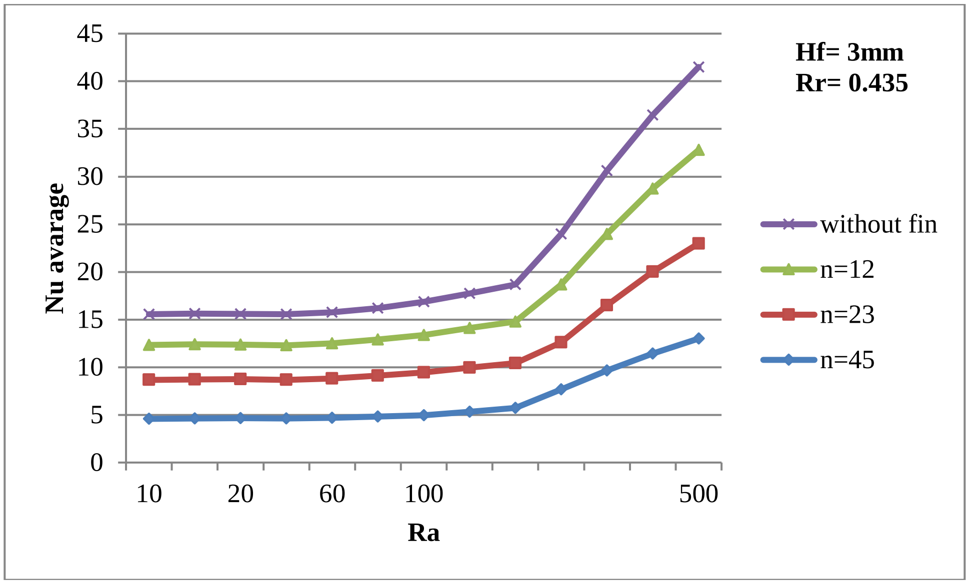


(b)

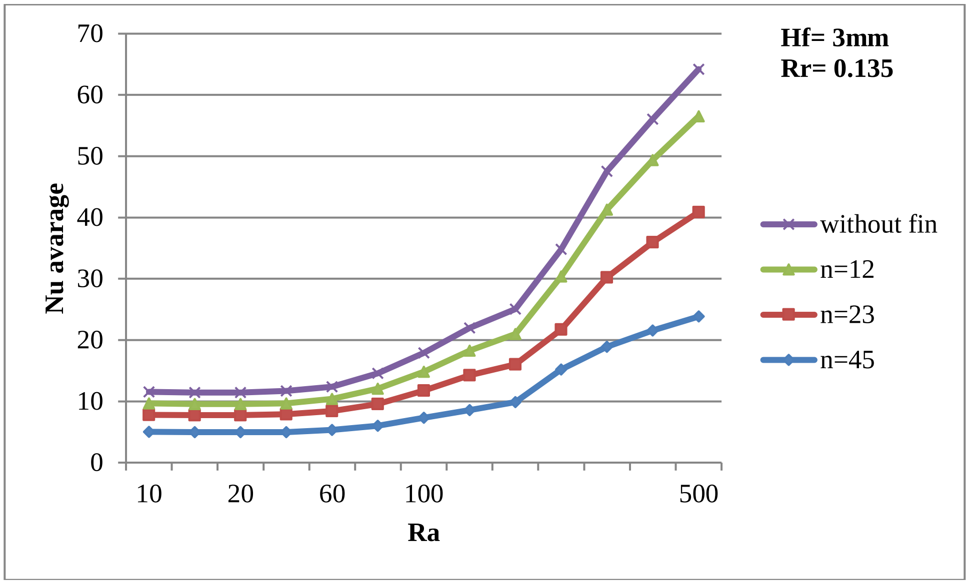


(c)

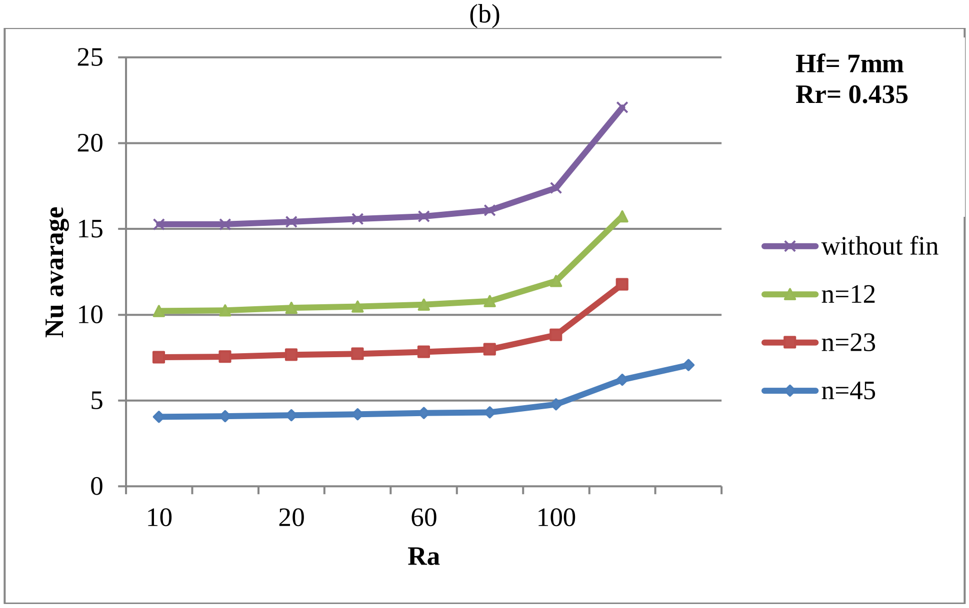
Figure 4 a, b, and c Variation of Average Nu with Rr at Ra 500 and different Hf, with or without fins.


Hf= 3mm
Rr= 0.435
Nu avarage
45
40
35
30
25
20
15
10
5
0
10
20
60
100
500
Ra
without fin
n=12
n=23
n=45

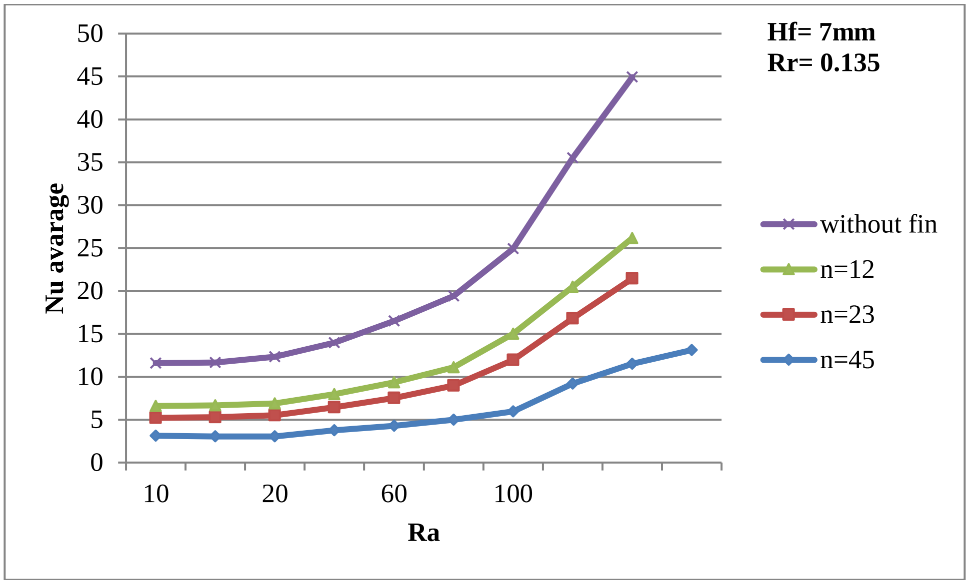
(a)


Hf= 3mm
Rr= 0.135
70
60
50
40
30
20
10
0
Nu avarage
10
20
60
100
500
Ra
without fin
n=12
n=23
n=45

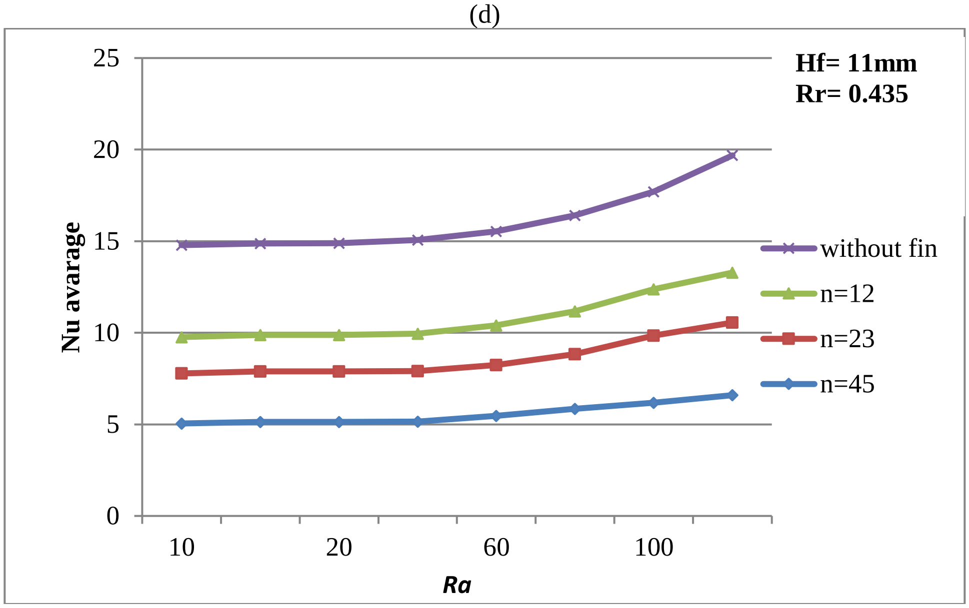
(b)


Hf= 7mm
Rr= 0.435
25
20
15
10
5
0
Nu avarage
10
20
60
100
Ra
without fin
n=12
n=23
n=45

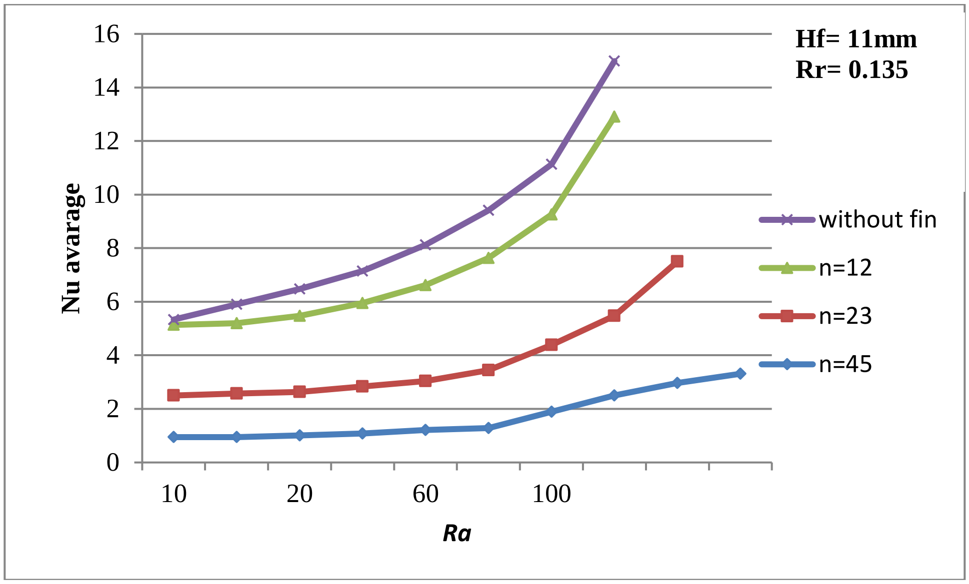
(c)


Hf= 7mm
Rr= 0.135
50
45
40
35
30
25
20
15
10
5
0
Nu avarage
10
20
60
100
Ra
without fin
n=12
n=23
n=45

(d)

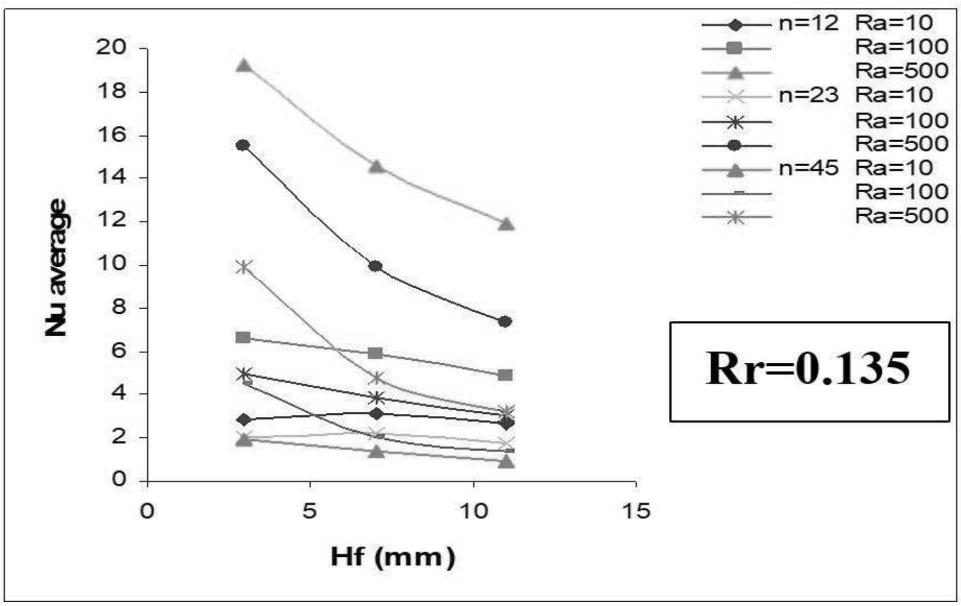

Hf= 11mm
Rr= 0.435
25
20
15
10
5
0
Nu avarage
10
20
60
100
Ra
without fin
n=12
n=23
n=45

(e)

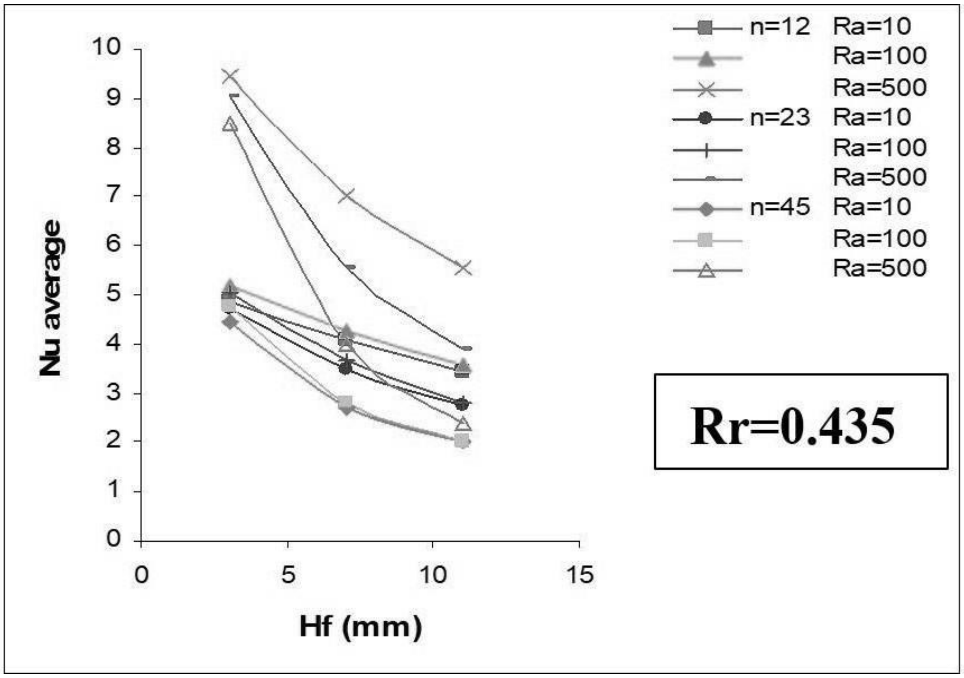


(f)

Figure 5 a, b, c, d, e, and f Variation of Average Nu with Ra at the different Rr, and Hf, with or without fins.



(a)



(b)

Figure 6 a, b Variation of Average Nu with Hf at differnt Ra, Rr and, and number of fins.

5. CONCLUSION

Heat exchangers with uniform heating ensure even heat distribution, improving heat transfer efficiency and lowering thermal stress. It contributes to increasing the effectiveness of solar energy conversion into thermal energy in solar collectors, improving system performance and sustainability. It is essential for enhancing performance in both home and industrial applications because of these advantages. The investigation focused on the analysis of three-dimensional steady-state laminar natural convection within concentricannuli containing porous media and featuring fins connected to the inner cylinder, with the possibility of fins being present or absent. A consistent low temperature was maintained on the annulus outer surface and a constant high temperature was maintained on the annulus inner surface. The goal of the studies is to determine how the rate of heat transfer is affected by various fin numbers (n =12, 23, and 45), fin lengths (Hf=3, 7 and 11 mm), and modified Rayleighnumbers ($10 \leq Ra \leq 500$). The average Nusselt number versus Ra was analyzed in graphical form to show the effect of the fin number on the average Nusselt number. The analysis includes a range of fin numbers. It is found that as the amount of fins increases from n = 12 to n = 23 and eventually to n = 45, there is a decrease in the average Nusselt number. The decrease in average Nusselt number can range from 19.2% to 26.6% for the same value of Ra.

The study yields several significant findings that can be inferred as follows:

1- If the surface area of the inner cylinder is not greater than that of the outer cylinder, heat transfer increases with the length of the fin at the same Rayleigh number and number of fins, unless the heat is retained in the porous media.
2- The average Nu number is almost constant for low Ra and obviously increases with an increase in themodified Rayleigh number, according to the results, for all values.
3- With an increase in Rr, a noticeable reduction in the average Nusselt number for Ra 100 is observed.
4- An inner cylinder without fins is the best design for the heat exchanger; 45 fins, each measuring 11 mm inlength, are the optimal design for the solar collectors and energy storage system.

REFERENCES

1. eikholeslami, M. Gorji-Bandpy, and D. D. Ganji, “Natural convection in a nanofluid filled concentric annulus between an outer square cylinder and an inner elliptic cylinder,” Sci. Iran., vol. 20, no. 4, 2014.
2. M. Sheikholeslami, R. Ellahi, M. Hassan, and S. Soleimani, “A study of natural convection heat transfer in a nanofluid filled enclosure with elliptic inner cylinder,” Int. J. Numer. Methods Heat Fluid Flow, vol. 24, no. 8,2014, doi: 10.1108/HFF-07- 2013-0225.
3. A. A. Mehrizi, M. Farhadi, and S. Shayamehr, “Natural convection flow of Cu- Water nanofluid in horizontalcylindrical annuli with inner triangular cylinder using lattice Boltzmann method,” Int. Commun. Heat Mass Transf., vol. 44, 2013, doi: 10.1016/j.icheatmasstransfer.2013.03.001.
4. A. S. Dogonchi, M. A. Sheremet, D. D. Ganji, and I. Pop, “Free convection of copper–water nanofluid in a porous gap between hot rectangular cylinder and cold circular cylinder under the effect of inclined magnetic field,” J. Therm. Anal. Calorim., vol. 135, no. 2, 2019, doi: 10.1007/s10973-018-7396-3.
5. R. L. Frederick and S. G. Moraga, “Three-dimensional natural convection in finned cubical enclosures,” Int. J. Heat Fluid Flow, vol. 28, no. 2, 2007, doi: 10.1016/j.ijheatfluidflow.2006.03.005.
6. G. Fabbri, “Effect of viscous dissipation on the optimization of the heat transfer in internally finned tubes,” Int. J. Heat Mass Transf., vol. 47, no. 14–16, 2004, doi: 10.1016/j.ijheatmasstransfer.2004.03.007.
7. H. T. Chen and W. L. Hsu, “Estimation of heat transfer coefficient on the fin of annular-finned tube heat exchangers in natural convection for various fin spacings,” Int. J. Heat Mass Transf., vol. 50, no. 9–10, 2007, doi: 10.1016/j.ijheatmasstransfer.2006.10.021.
8. V. Prasad and F. A. Kulacki, “Natural convection in a vertical porous annulus,” Int.J. Heat Mass Transf., vol. 27, no. 2, 1984, doi: 10.1016/0017-9310(84)90212-6.
9. V. Prasad And F. A. Kulacki, “Natural Convection In Porous Media Bounded By Short Concentric Vertical Cylinders.,” In American Society Of Mechanical Engineers (Paper),1984.
10. M. Corcione, M. Cianfrini, and A. Quintino, “Optimization of laminar pipe flow using nanoparticle liquid suspensions for cooling applications,” in Applied Thermal Engineering, 2013. doi: 10.1016/j.applthermaleng.2012.07.029.
11. V. Prasad, “Numerical study of natural convection in a vertical, porous annulus with constant heat flux on the inner wall,” Int. J. Heat Mass Transf., vol. 29, no. 6, 1986, doi: 10.1016/0017-9310(86)90180-8.
12. V. Prasad And A. Chui, “Natural Convection In A Cylindrical Porous Enclosure With Internal Heat Generation.,” In American Society Of Mechanical Engineers, Fluids Engineering Division (Publication) Fed, 1987. Doi: 10.1115/1.3250806.
13. I. S. Shivakumara, B. M. R. Prasanna, N. Rudraiah, and M. Venkatachalappa, “Numerical study of natural convection in a vertical cylindrical annulus using a non- Darcy equation,” J. Porous Media, vol. 5, no. 2, 2002,doi: 10.1615/jpormedia.v5.i2.20.
14. M. Kumari and G. Nath, “Unsteady natural convection from a horizontal annulus filled with a porous medium,” Int. J. Heat Mass Transf., vol. 51, no. 19–20, 2008, doi: 10.1016/j.ijheatmasstransfer.2008.01.030.
15. M. H. Saleh and A. M. Katea, “Laminar Free Convection In Horizontal Annulus Filled With Glass Beads AndWith Annular Fins On The Inner Cylinder,” J. Eng., vol. 19, no. 08, pp. 999–

1018, 2023, doi: 10.31026/j.eng.2013.08.06.
16. M. H. S. A. H. Dhaef, “Heat Transfer in Horizontal Annulus Saturated with Copper Nanofluid under Various Boundary Conditions,” Int. J. Sci. Res., vol. 4, no. 4, pp. 2141–2147, 2015, [Online].Available: https://www.ijsr.net/archive/v4i4/15041501.pdf
17. A. H., “Heat Transfer in Enclosure of Composite Material with Fibers Arranged in Various Geometrical Arrays,” Int. J. Comput. Appl., vol. 125, no. 8, pp. 16–24, 2015, doi: 10.5120/ijca2015906117.
18. M. H.Saleh and A. H. Dhaef, “Heat Transfer in Inclined Enclosure of Silica Aerogel/Glass Fiber Composite Material,” Int. J. Comput. Appl., vol. 117, no. 8, pp. 5–12, 2015, doi: 10.5120/20572-2969.
19. Y. T. Yang and M. L. Hwang, “Numerical simulation of turbulent fluid flow and heat transfer characteristics in heat exchangers fitted with porous media,” Int. J. Heat Mass Transf., vol. 52, no. 13–14, 2009, doi: 10.1016/j.ijheatmasstransfer.2009.02.024.
20. K. Khanafer and A. J. Chamkha, “Mixed convection within a porous heat generating horizontal annulus,” Int. J. Heat Mass Transf., vol. 46, no. 10, 2003, doi: 10.1016/S0017-9310(02)00483-0.
21. M. ShX. Liu, D. Toghraie, M. Hekmatifar, O. A. Akbari, A. Karimipour, and M. Afrand, “Numerical investigation of nanofluid laminar forced convection heat transfer between two horizontal concentric cylinders in the presence of porous medium,” *J. Therm. Anal. Calorim.*, vol. 141, no. 5, pp. 2095–2108, 2020, doi: 10.1007/s10973-020-09406-3.
22. H. Hanif, M. R. Ilias, Z. Iqbal, S. Shafie, S. E. Alhazmi, and M. M. Alqarni, “Heat transfer in hybrid nanofluid flow between two coaxial cylinders,” *Case Stud. Therm. Eng.*, vol. 59, no. April, p. 104327, 2024, doi: 10.1016/j.csite.2024.104327.
23. Nield D. A. and Bejan A., “Convection in Porous Media”, Springer-Verlag, New York, 1999.

Letter	Description	Units
A_{in}	The surface area of the inner cylinder	m^2
A_{out}	The surface area of the outer cylinder	m^2
A_t	The surface area of Teflon piece	m^2
d_f	Fin diameter	m
d_i	Inner diameter of the inner cylinder	m
d_o	Outer diameter of the inner cylinder	m
D_i	Inner diameter of the outer cylinder	m
D_o	Outer diameter of the outer cylinder	m
g	Acceleration due to gravity	m/s^2
h_i	The convection heat transfer coefficient on the inner cylinder (hot surface)	W/m^2K
h_o	The convection heat transfer coefficient on the outer cylinder (cold surface)	W/m^2K
I	Current	A
$K_{eff.}$	Effective thermal conductivity of the porous media	W/m K
l	Cylinder length	m
L_t	Distance between thermocouples of Teflon piece	m
n	Number of fins	-
$Q_{cond.}$	Heat loss by conduction	W
Q_{loss}	Heat loss by Teflon piece	W
r_{in}	Radius of the inner cylinder	m
r_{out}	Radius of the outer cylinder	m
t	Fin thickness	m
T	Temperature	K
T_{ii}, T_{oo}	Inside and outside surface temperature of Teflon piece	K
T_1	Temperature of the inner cylinder surface	K
T_2	Temperature of the outer cylinder surface	K