Natural convection in horizontal cylindrical annuli with fins

D. Alshahrani [a] and O. Zeitoun [b]
[a] Mech. Eng. Dept., College of Eng., King Saud University, Riyadh, Saudi Arabia
[b] Mech. Eng. Dept., Faculty of Eng., Alexandria University, Alexandria, Egypt
ozeitoun@ksu.edu.sa

Natural convection heat transfer between two horizontal concentric cylinders with two fins attached to inner cylinder was investigated numerically using finite element technique. Laminar conditions up to Rayleigh number Ra_i of $5*10^4$ were investigated. Effects of annulus diameter ratio, Rayleigh number, fin length, and its inclination angle on this type of flow were investigated. The conductance thermal resistance of finned annuli was obtained using conduction analysis. The thermal resistance decreases as fin length increases i.e. as the fin is extended within the annulus gap and increases as annulus diameter ratio increases i.e. as annular gap thickness increases. The free convection data were represented in terms of the effective thermal conductivity ratio k_e/k versus Rayleigh number Ra_i. Correlation for thermal conductivity ratio is proposed.

تم دراسة انتقال الحرارة بالحمل الحرارى الطبقى الطبيعى فى الفجوة بين اسطوانتين افقيتين متحدتى المحور فى وجود زعنفتين على الاسطوانة الداخلية عدديا باستخدام طريقة العناصر المحددة. و قد تم دراسة تأثير كل من النسبة بين قطرى الفجوة و رقم رايلى و طول و زاوية ميل الزعنفتين على هذا النوع من السريان. و قد عرضت النتائج على شكل كونتورات لكل من دالة السريان ودرجات الحرارة وتوزيعات الفيض الحرارى على السطح الداخلى للفجوة. تم الحصول على المقامة الحرارية للتوصيل بدراسة التوصيل الحرارى فى الفجوة المزعنفة. تم الحصول على نتائج لانتقال الحرارة فى شكل النسبة بين معامل التوصيل الحرارى المكافىء الى معامل التوصيل الحرارى مع رقم رايلى لنسب اقطار و زوايا ميل و اطوال زعانف مختلفة. تم اقتراح معادلة عامة لحساب النسبة بين معامل التوصيل الحرارى المكافىء الى معامل التوصيل الحرارى.

Keywords: Natural convection, Heat transfer, Laminar, Horizontal, Cylindrical annuli, Fin

1. Introduction

Considerable work was done on investigating natural convection in horizontal annuli without fins, [1-17]. Previous investigation in bare annuli showed limited heat transfer in the annulus. One of the methods used to enhance the heat transfer in the annulus is to equip the surface of the inner cylinder with fins. Thorough literature survey showed that comparatively little work was focused on natural convection in annuli with fins on the inner cylinders. The presence of internal fins alters the flow patterns and temperature distributions that consequently affects heat transfer coefficient in the annuli. Previous researchers [18-22] investigated natural convection in horizontal annuli with fins on the inner cylinder however the fins were symmetric about the vertical axis.

Changing heat transfer rate in fixed-flow geometry of two concentric cylinders can be done through application of some radial fins to the surface of hot cylinder. Chai and Patankar [18] were one of the first to investigate the effect of radial fins on laminar natural convection in horizontal annuli. They considered an annulus with 6 radial fins attached to the inner cylinder. They studied the effects of two fin orientations; the first is when two fins of the six are vertical and the second is when two fins are horizontal. The results indicate that the fin orientation shows no significant effect on average Nusselt number and average Nusselt number increases with increasing Rayleigh number and decreases with increasing fin height. It should be mentioned that Nusselt number alone cannot give indication on increasing or decreasing heat transfer as will be discussed.

Farinas et al. [19] investigated the effect of internal fins on flow pattern, temperature distribution and heat transfer between concentric horizontal cylinders for different fin orientation and fin tip for Rayleigh numbers ranging from 10^3 to 10^6. They employed the two fin orientation used by [18]. The second orientation presents a higher heat transfer rate than that of the first orientation. The fin tip geometry shows insignificant effect on heat transfer. Farinas et al. [20] investigated laminar natural convection in horizontal bare and finned rhombic annulus for different fin numbers and lengths and Rayleigh numbers ranging from 10^3 to 10^7. They concluded that the heat transfer is maximized for a narrow cavity with two longer fins.

Rahnama et al. [21] studied the effect of fins attached to the inner and outer cylinders of annuli on flow and temperature fields in a horizontal annulus. They studied the effect of fin orientation and height attached to the cylinders for Rayleigh numbers less than 10^6. They employed the two fin orientation used by [18] and [19]. Contrary to [19] their results show no effect of fin orientation on heat transfer as reported in [18]. They argued that the results of local and mean Nusselt number prediction show that the fin height has an optimum value for which the heat transfer rate is a maximum. Extending fin height beyond that value reduces heat transfer rate significantly which is due to the blocking effect of long fins on flow recirculation. Rahnama et al. [22] investigated numerically the effect of two horizontal radial fins attached on the inner cylinder on the heat transfer for Rayleigh numbers less than 10^4. They reported that insertion of two fins reduced heat transfer compared to bare annulus.

The objective of the current chapter is to investigate numerically the natural convection heat transfer in horizontal annuli where the inner tube has two 180° apart longitudinal fins of constant thickness. Effects of fin inclination angle to the horizontal, fin height and inner and outer annulus radii will be investigated.

2. Problem description

The physical model of the investigated problem is illustrated schematically in fig. 1. A heated cylinder at T_i and of radius R_i is placed inside another cylinder at T_o and of radius R_o trapping air in the resulting annular cavity. The inner cylinder is placed concentric with the outer cylinder as shown in the figure. Two fins of height of L_f and thickness of d are placed on the inner cylinder. The fin thickness d is kept constant in the current analysis, d = 1 mm. The angle between the two fins is 180°. The two fins are inclined an angle of θ with the horizontal x axis. However to reduce effort in building the geometric models of current problem, the current problem is solved in the coordinate system X and Y shown in figure, where X is the coordinate placed along the fin and Y is the coordinate placed normal to X as shown in figure.

As the inner cylinder is assumed to be hotter than the outer cylinder, a buoyancy induced flow results and natural convection occurs. It is expected that the existence of fins will resist the natural circulation inside the enclosure. The surfaces of the inner cylinder and the two fins are assumed isothermal at T_i and the surface of the outer cylinders is assumed isothermal at T_o, respectively, where $T_i > T_o$.

The partial differential equations governing the fluid flow and heat transfer in the enclosure include the continuity, the Navier-Stokes and the energy equations. The continuity equation is:

$$\frac{\partial U}{\partial X} + \frac{\partial V}{\partial Y} = 0 \,. \qquad (1)$$

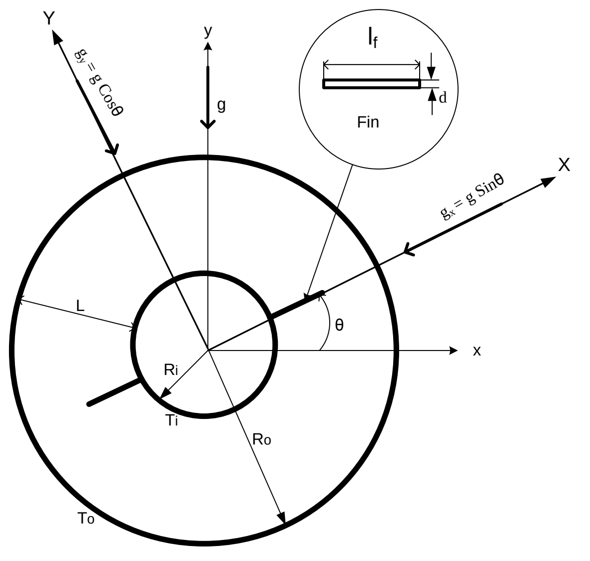


Fig. 1. Physical description and coordinate system of the annulus with fins problem.

The momentum equations in X and Y directions are:

$$\rho U \frac{\partial U}{\partial X} + \rho V \frac{\partial U}{\partial Y} = -\rho g_X - \frac{\partial P}{\partial X} + \mu \left[\frac{\partial^2 U}{\partial X^2} + \frac{\partial^2 U}{\partial Y^2} \right], \quad (2)$$

$$\rho U \frac{\partial V}{\partial X} + \rho V \frac{\partial V}{\partial Y} = -\rho g_Y - \frac{\partial P}{\partial Y} + \mu \left[\frac{\partial^2 V}{\partial X^2} + \frac{\partial^2 V}{\partial Y^2} \right]. \quad (3)$$

The energy equation is:

$$\rho C_p \left[U \frac{\partial T}{\partial X} + V \frac{\partial T}{\partial Y} \right] = k \left[\frac{\partial^2 T}{\partial X^2} + \frac{\partial^2 T}{\partial Y^2} \right]. \quad (4)$$

where U and V are the velocities along X and Y directions, P is the absolute pressure, and g_X and g_Y are the gravitational acceleration components in X and Y directions and they are estimated from:

$$g_X = g \sin\theta , \quad (5)$$

$$g_Y = g \cos\theta . \quad (6)$$

For natural convection flow, the change in density is responsible for the flow and the ideal gas state equation was provided as an input to estimate the fluid density,

$$\rho = \frac{P}{RT} , \quad (7)$$

where P is the absolute pressure. The reference pressure was assumed equal to the standard atmospheric pressure during the current investigation. The properties of air except density are assumed constant and are taken to be the values at the mean temperature,

$$T_m = \frac{T_i + T_o}{2} . \quad (8)$$

The boundary conditions for the current problem are:

1. No slip condition along the surfaces of the inner cylinder and the two fins.
2. No slip condition along the surfaces of the outer cylinder.
3. Isothermal condition at the surfaces of the inner cylinder and the two fins, $T = T_i$.
4. Isothermal condition at the surface of the outer cylinder, $T = T_o$.

The assumption made in 3rd boundary condition is similar to boundary condition used by [18] and [19] and it is based on the assumption that the fin is highly conductive.

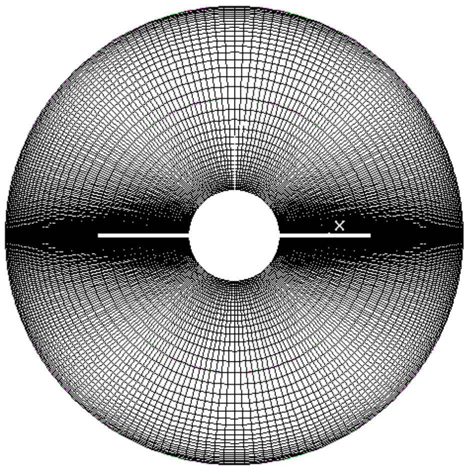
3. Solution procedure

Cosmos-flow plus code, used by Alshahrani and Zeitoun [17], Zeitoun [23] and Zeitoun and Ali [24], which was derived from the SIMPLER solution scheme introduced by Patankar [25], was used to discretize and solve the flow domain in the annuli with fins. Flow plus code is based on Finite element technique. A four node quadrilateral element type was used in solving the current problem. The grid points are not distributed uniformly over the computational domain as shown in fig. 2. They have greater density near surfaces of the inner and outer surfaces of the annulus and the surface of the fins. The spacing expansion and contraction factors of grid distribution along the radial and angular directions were selected 10 and 0.1.

The numerical results were checked for grid independency. The numerical results are obtained with increasing number of nodes till a stage is reached where the results produce negligible changes with further refinement in grid size. The effect of the meshing system on the solution is examined by solving sample cases of the current problem for different meshing systems. Based upon the results of this investigation a grid system of 40×360 is chosen to be used in the following analysis.

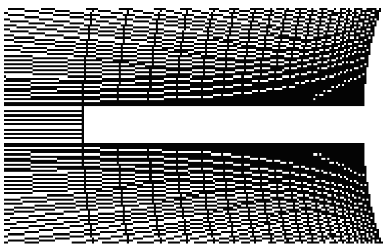
4. Results and discussions

Table 1 shows the input data employed in the present investigation. Natural convection in enclosures of three D_o/D_i ratios, five different angles and three fin height ratios was investigated. Typical samples of stream lines and iso-thermal lines are shown in figs. 3 and 4.



a. Entire domain

b. Grids near outer surface



c. Grids near fin and inner surface

Fig. 2. Grid system.

Table 1
Run conditions

D_o, m	D_i, m	Do/D_i	L_f, %	θ	T_i, K	T_o, K
0.06	0.02	3	0	0°	300.01	300
0.08	0.02	4	25	22.5°	to	
0.1	0.02	5	50	45°	500	
			75	67.5°		
				90°		

4.1. Effect of fin inclination angles

The effects of fin inclination angles on stream lines and temperature fields for temperature difference of 10 °C, L_f = 25% and D_o/D_i = 4 are shown in fig. 3. For fin inclination angle of θ = 90°, there are two symmetric circulation loops. As the fin angles decreases, the fins shifted one loop down and the other up. This shift increases resistance to the flow. This resistance reaches the maximum value at angle of θ = 0°. As a result, circulation velocity decreases and thermal boundary layer gets thick along the upper faces of fins and inner cylinder.

4.2. Effects of fin height

Fig. 4 shows the effect of fin height on stream line fields for horizontal fins (θ=0) and a temperature difference of 10 °C. For finless annulus, as shown in the figure, there are two circulation loops. For fin height of L_f = 25% and 50%, the circulation loop become thin in the level of fins until they are divided into four loops at a fin height L_f = 75%. The figure also shows that the upper two loops are stronger than the lower loops. The isothermal regions in the bare annulus show that the heat transfer concentrated around the bottom of the inner cylinder. The existence of fins as shown in the figures increases the thermal boundary layer along the bottom part of the inner cylinder and attached fins. For the outer cylinder, the heat transfer intensifies along the upper parts. The existence of fins decreases the boundary layer along the outer cylinder surface. These results indicate that the heat transfer will increase as fin height increases, this will be discussed later.

4.3. Heat flux distributions

The heat transfer from the inner to the outer wall was calculated by integrating the local heat flux along the surfaces of the inner cylinder and fins,

$$Q = \sum\left(\int_{A_i} q_w dA + \int_{A_f} q_w dA\right), \tag{9}$$

where the local heat flux was estimated by applying Fourier's law at the surfaces of the inner cylinder and fins,

$$q_w = -k\frac{\partial T}{\partial n}\Big|_w . \tag{10}$$

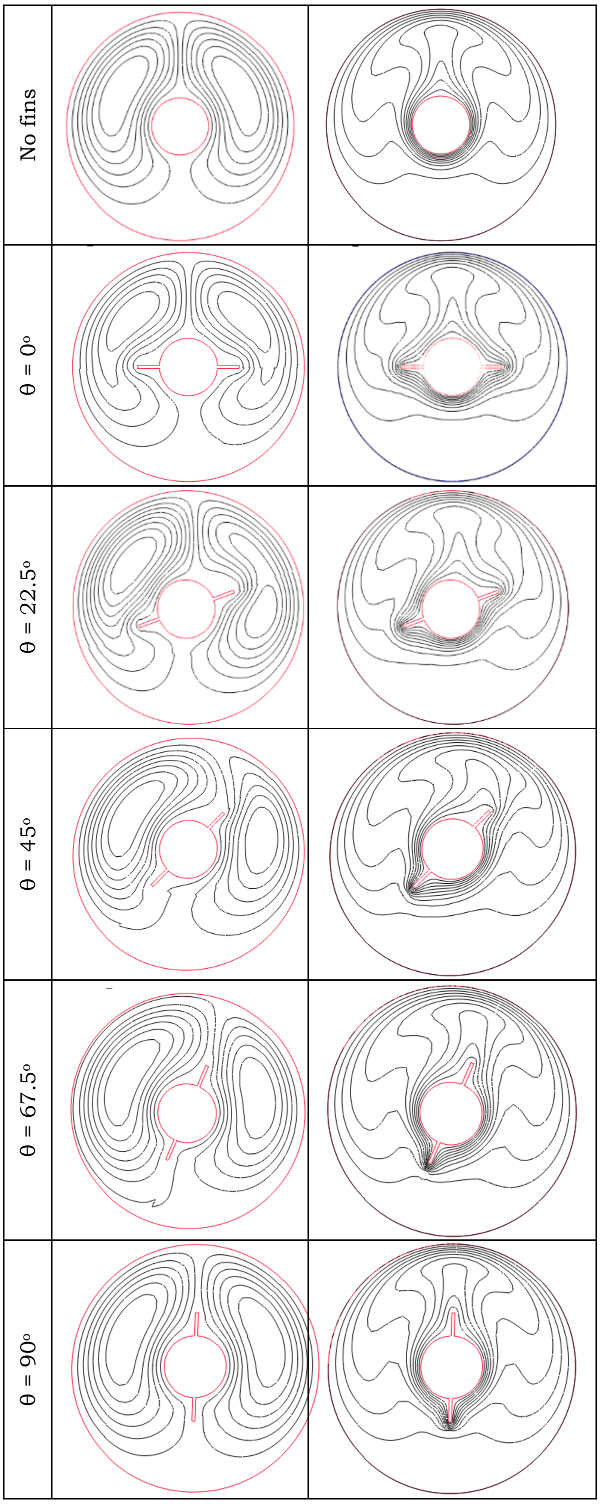


Fig. 3. Stream and isothermal lines at different fin inclination angles for D_o/D_i =4 and ΔT=10K.

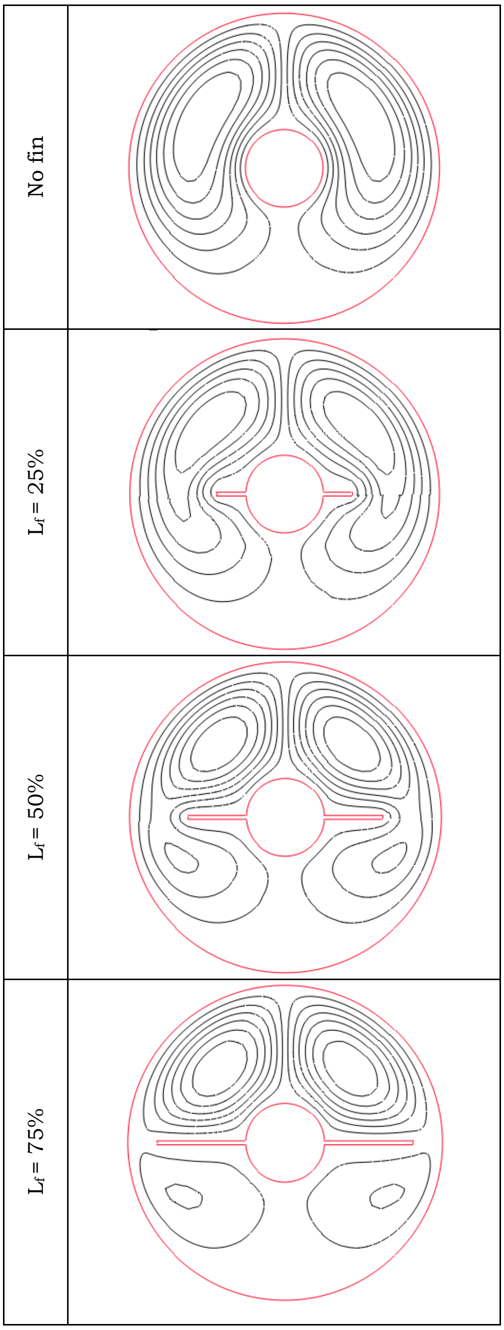


Fig. 4. Effect of fin height on flow pattern for D_o/D_i = 4 and ΔT =10K and θ = 0.

Local heat flux distribution along the surface of the inner cylinder of a finned annulus is shown in fig. 5 where 0° and 360° angles represent the lower point of the inner cylinder and 180° angle represents the upper point of the inner cylinder. The fin inclination angle is zero for this distribution. The local heat flux distribution for un-finned inner cylinder is also shown in the figure. The results indicate that the local heat flux along inner cylinder is decreased, compared to un-fined cylinder, due to existence of fins. The heat flux along bottom surface of finned cylinder (for 0° to 90° and from 270° to 360°) is higher than upper part. As discussed in [17] the thermal boundary layer is thinner along lower surface. The heat flux reaches very low values at root of fins. However, the reduction in heat transfer along inner cylinder surface due to flow resistance is compensated by the heat transfer from fin surface as shown in fig. 6. Examining the current data shows that increasing fin length at different fin inclination angles increases the total heat transfer. However, the effect of fin height is higher in the low Rayleigh number region, where the conduction heat transfer mechanism is the dominant mechanism. In his region, using fin height of L_f = 0.75 increases heat transfer about 100%. In high Rayleigh number region, where the convection heat transfer mechanism is the dominant mechanism, increasing fin height increases heat transfer but at lower rate due to resistance of fins to the natural circulation flow.

4.4. Thermal conductivity ratio

The data of heat transfer within the annulus were represented in terms of the effective thermal conductivity ratio k_e/k versus Rayleigh number Ra_i, where k_e is the equivalent thermal conductivity for both of conduction and convection in the annulus. The equivalent thermal conductivity k_e can be estimated from:

$$Q = \frac{k_e(T_i - T_o)}{R_{thf}} . \tag{11}$$

Where R_{thf} is the thermal resistance of the finned annulus which was estimated from the conduction heat transfer analysis, conducted by flow plus, within the annulus:

$$Q_{cond} = \frac{k(T_i - T_o)}{R_{thf}} . \tag{12}$$

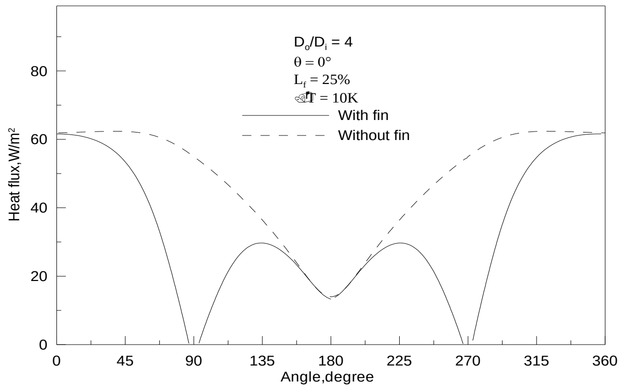


Fig. 5. Heat flux distribution along surface of inner cylinder.

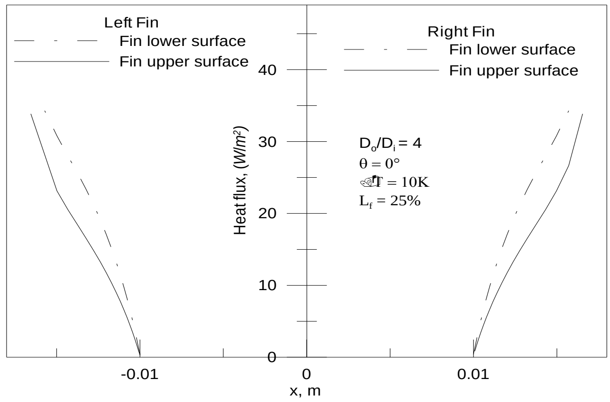


Fig. 6. Heat flux distribution along fin surfaces.

Thermal resistance of the finned annulus is represented versus fin length ratio L_f as shown in fig. 7. It should be mentioned that this resistance is only a function of annulus geometry. As shown in the figure the thermal resistance decreases as fin length increases, i.e. as the fin is extended within the annulus gap, and increases as annulus diameter ratio increases, i.e. as annular gap thickness increases. The ratio between thermal resistance of finned annulus R_{thf} and thermal resistance of un-finned annulus R_{tha} is shown in fig. 8 as function of fin length ratio L_f and annulus diameter ratio. These data were correlated by the following correlation:

$$\left.\frac{R_{thf}}{R_{tha}}\right|_{cor} = 1 + \left(-0.1963\left(\frac{D_o}{D_i}\right) + 0.2705\right)L_f + \left(0.1615\left(\frac{D_o}{D_i}\right) - 0.8001\right)L_f^2 . \tag{13}$$

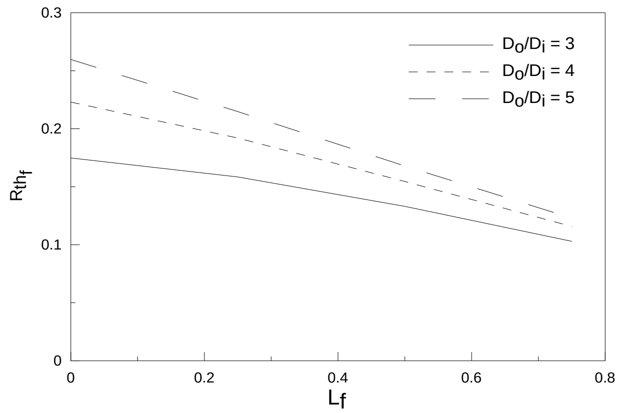


Fig. 7. Thermal resistance of finned annulus.

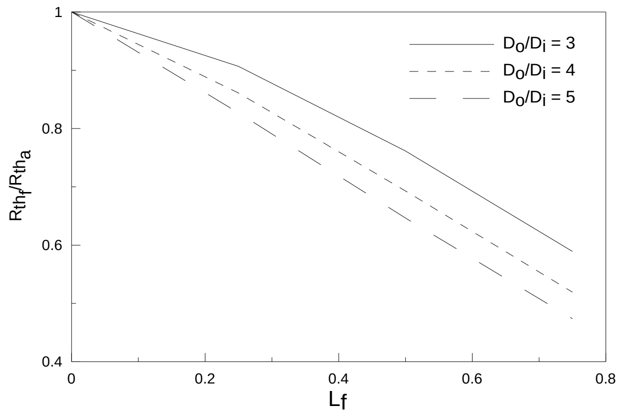


Fig. 8. Thermal resistance ratio of finned annulus.

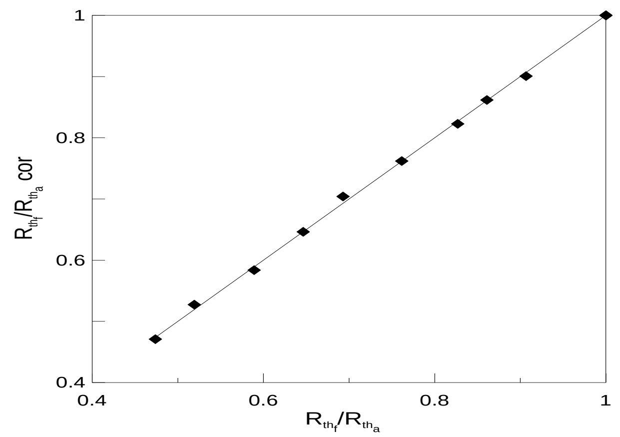


Fig. 9. Comparison between correlation of thermal resistance and CFD data.

The comparison between the prediction of the above correlation and current data of thermal resistance of the finned annulus is shown in fig. 9. As shown in this figure, the prediction of the proposed correlation accurately fits the current data of thermal resistance of finned annulus with a margin of error of ±0.5%.

The data of the effective thermal conductivity ratio k_e/k for the conditions listed in table 1 are shown in figs. 10, 11 and 12. As shown in the figures, this ratio is represented versus Rayleigh number based on the inner cylinder diameter for different annulus diameter ratios, fin length ratios and fin inclination angles. The pure conduction heat transfer limits is $k_e/k = 1$ as shown in the figures. There are three distinct regimes of heat transfer. The first regime is the conduction dominated regime where the effective thermal conductivity ratio $k_e/k = 1$. The third is the convection dominated regime where the curves show strong dependency on Rayleigh number. The second regime which falls between the above mentioned regimes is a transition regime. The data in the figures indicate that the ratio k_e/k decreases as fin length ratio L_f increases. This means that convection effect is decreased as fin length increases. The inclination angle has a significant effect on the thermal conductivity ratio k_e/k, however, the ratio k_e/k increases slightly as the inclination angle increases. The effect of fin on natural convection circulation reaches its minimum at an angle of 90°.

The data of thermal conductivity ratio for each fin inclination angle were correlated individually. Investigation of the data for each inclination angle indicated that they are strongly dependent on the modified Rayleigh number Ra_m introduced by Alshahrani and Zeitoun [17] and on fin height ratio, Ra_m is given by:

$$Ra_m = Ra_i^{1/4}\left[0.1389\left(1-\frac{D_i}{D_o}\right)+0.0927\right]\ln\left(\frac{D_o}{D_i}\right). \tag{14}$$

The data of thermal conductivity ratio k_e/k are correlated in terms of thermal conductivity ratio of un-fined annulus $(k_e/k)_{unf}$ and fin height ratio L_f. The best fit for thermal conductivity ratio of un-finned annulus $(k_e/k)_{an}$ introduced by Alshahrani and Zeitoun [17] is,

$$\left.\frac{k_e}{k}\right|_{unf} = 1 \quad \text{for} \quad Ra_m \le 0.8 \quad \text{and}$$

$$\left.\frac{k_e}{k}\right|_{unf} = 0.0123\,Ra_m^6 - 0.2167\,Ra_m^5 + 1.5514\,Ra_m^4 -$$
$$5.7568\,Ra_m^3 + 11.553\,Ra_m^2 - 10.683\,Ra + 4.5486$$
$$\text{for} \quad 0.8 \langle\, Ra_m \,\langle\, 4.7\,. \qquad (15)$$

The non-linear regression analysis was carried in two steps. First, the regression was conducted for each inclination angle separately. The nonlinear regression analysis was carried out using XLSTAT software. The obtained correlations for five different fin inclination angles are:

$$\frac{k_e}{k} = 1 \quad \text{for } Ra_m \le 0.8 \quad \text{and}$$
$$\left(\frac{k_e}{k}\right) = \left(\frac{k_e}{k}\right)_{unf}^{1/(1+m\,L_f^n)} \quad \text{for} \quad 0.8 \langle\, Ra_m \,\langle\, 4.7. \qquad (16)$$

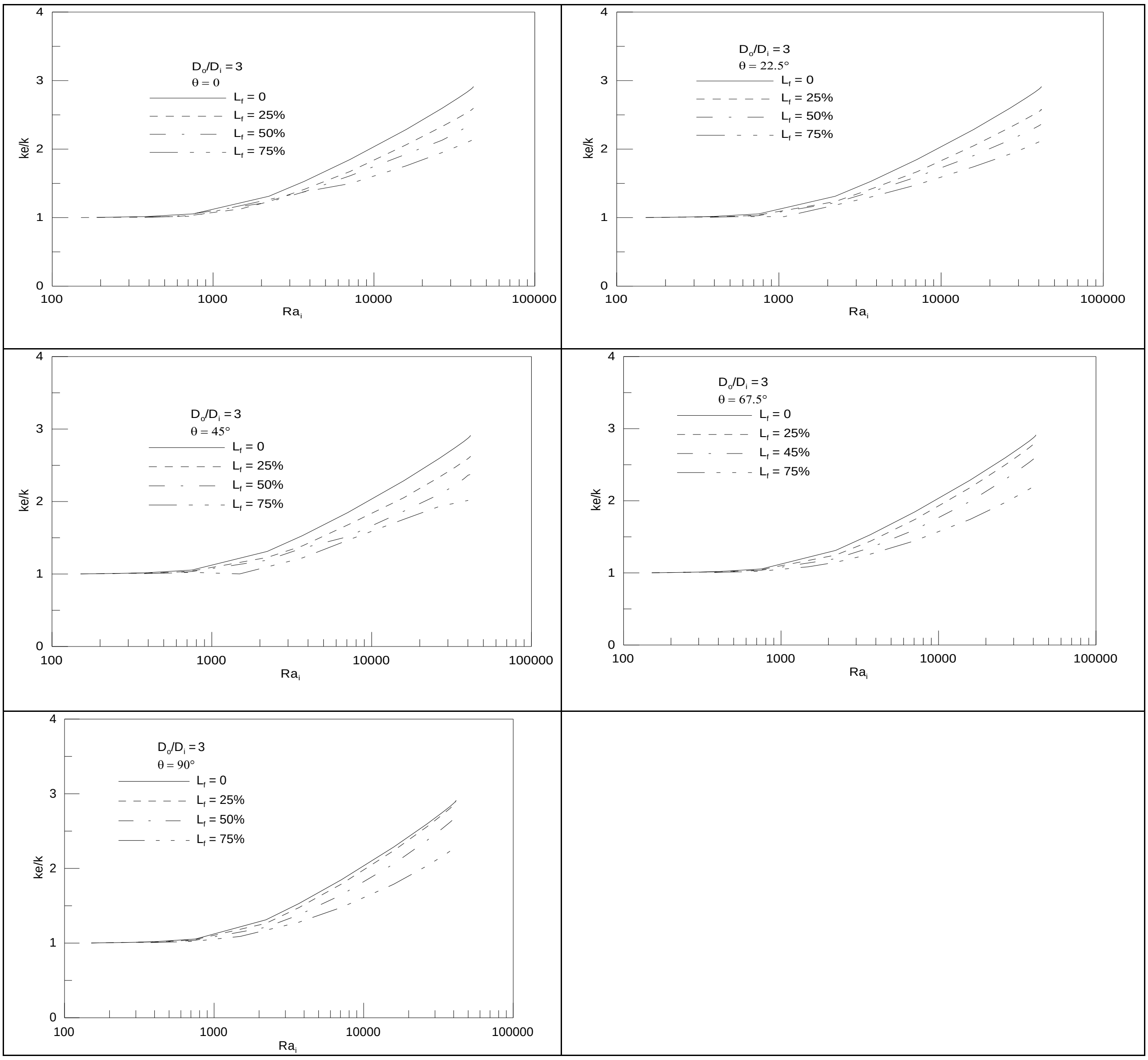


Fig. 10. Thermal conductivity ratio of finned annulus versus Rayleigh number.

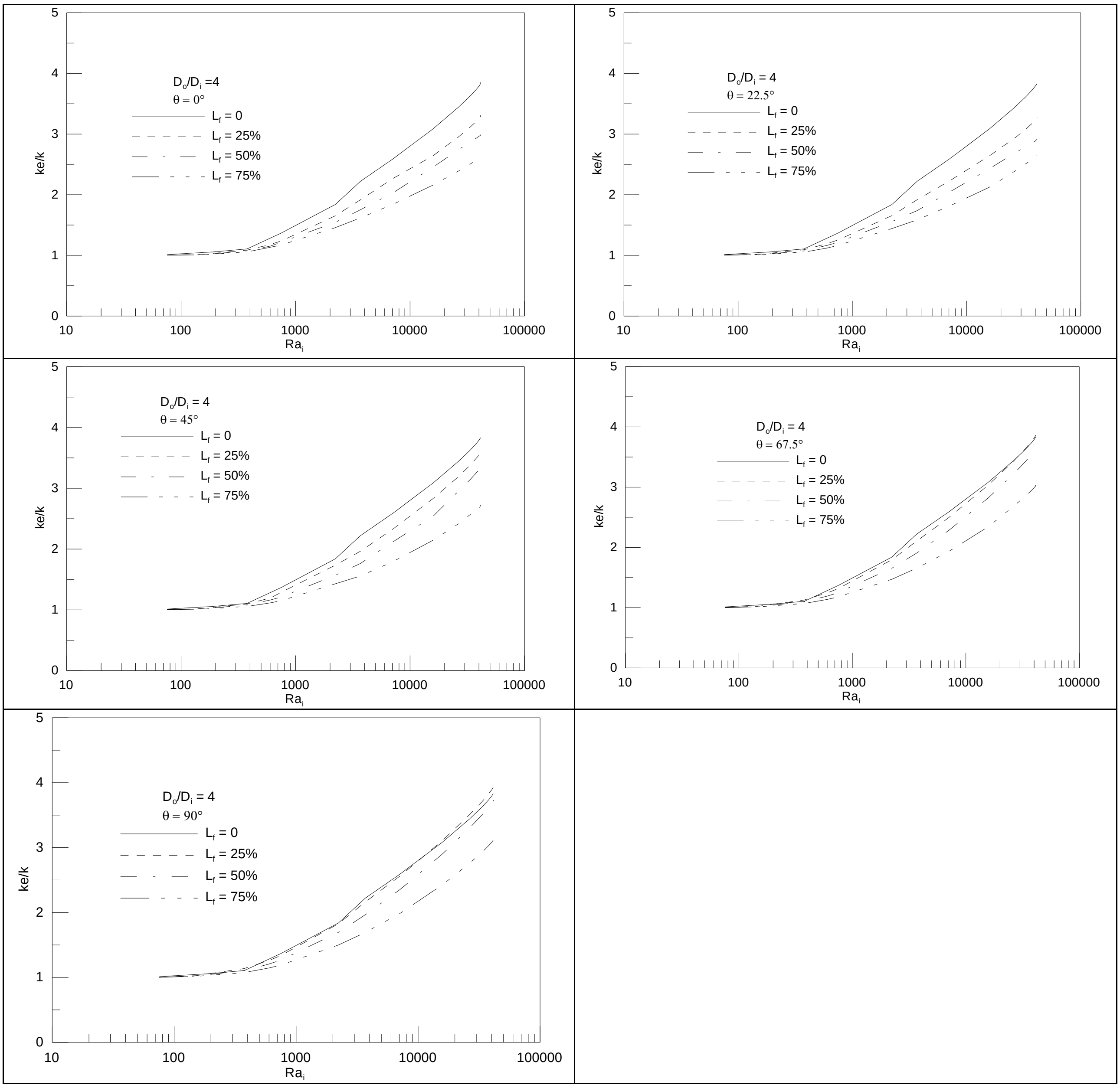


Fig. 11. Thermal conductivity ratio of finned annulus versus Rayleigh number.

The constants m and n for different fin inclination angles are listed in table 2. The regression coefficients R_c of the above fore-mentioned correlations fall in the range from 99.3% to 99.8%. The constants m and n in the above correlation can be represented by the following equations,

Table 2
Regression results

θ	m	n	R_c, %
0°	0.5359	1	99.8
22.5°	0.51877	1.0242	99.8
45°	0.5626	2.2096	99.3
67.5°	0.7145	4.2542	99.6
90°	0.7465	5.0677	99.6

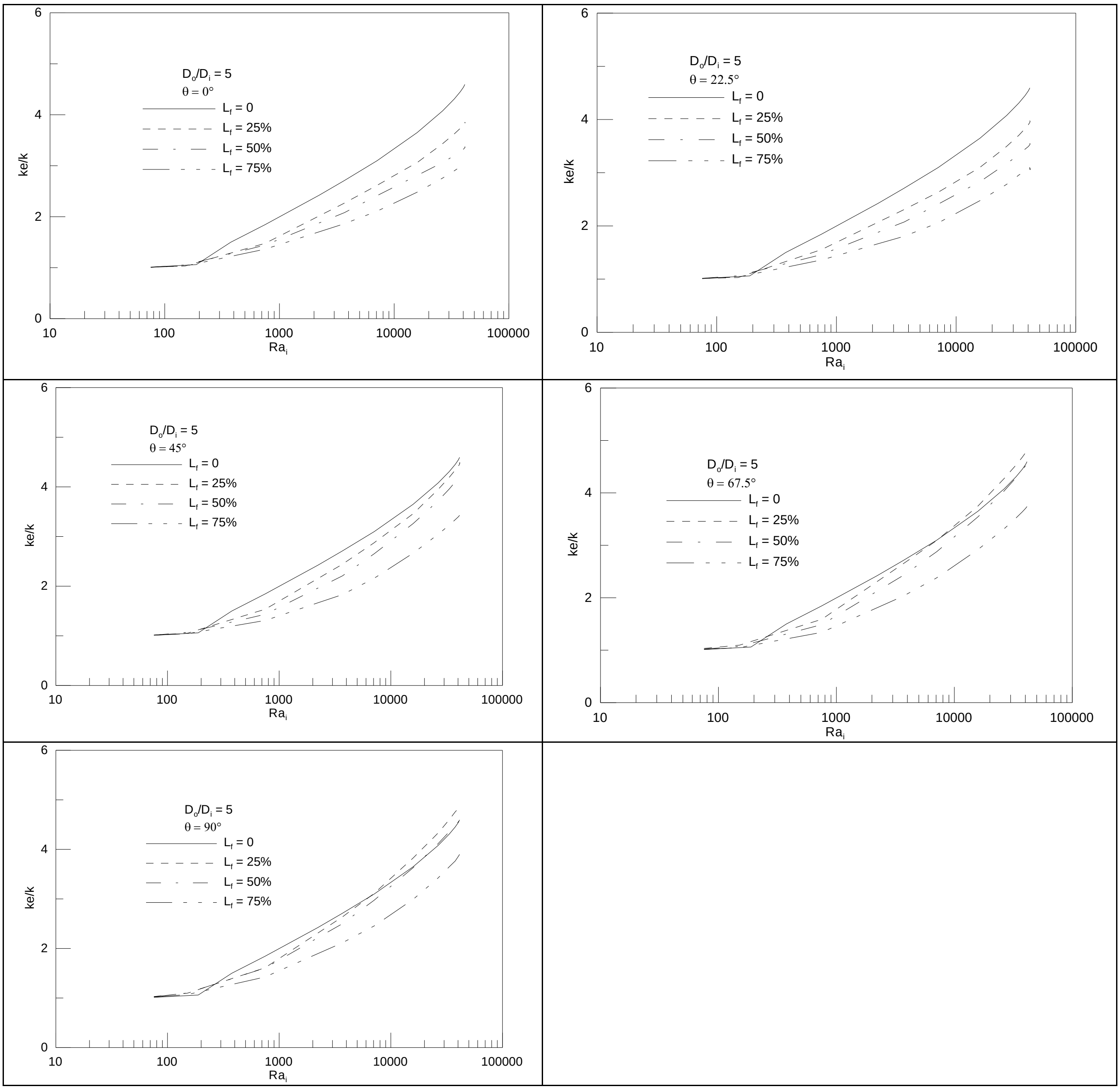


Fig. 12. Thermal conductivity ratio of finned annulus versus Rayleigh number.

$$m = -4.467 \times 10^{-8}\theta^4 + 6.717 \times 10^{-6}\theta^3 - 2.348 \times 10^{-4}\theta^2 + 1.629 \times 10^{-3}\theta + 0.5359\,, \quad (17)$$

$$n = -2.907 \times 10^{-7}\theta^4 + 3.483 \times 10^{-5}\theta^3 - 1.74 \times 10^{-4}\theta^2 + 9.331 \times 10^{-3}\theta + 1\,. \quad (18)$$

The correlation coefficient of above correlations is R_c = 1. Comparison between predictions of above correlations and the numerical data of thermal conductivity ratio is shown in fig. 13. As shown in the figure, the prediction of the proposed correlation fit the current data of thermal conductivity ratio of finned annulus with a margin of error of -6 to +15%.

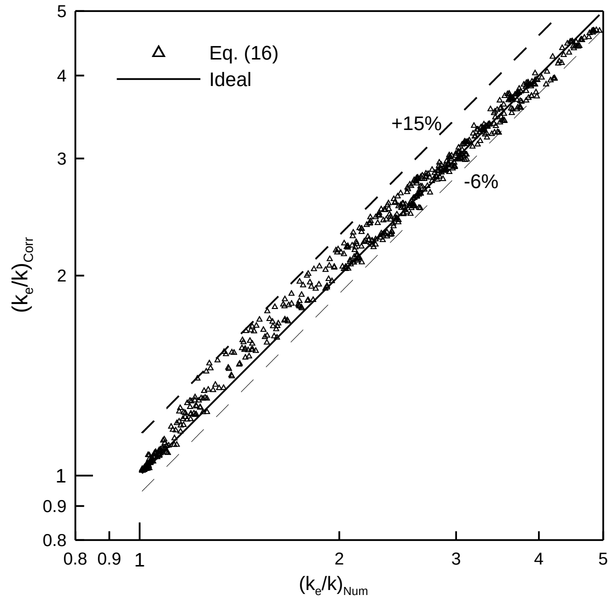


Fig. 13. Comparison between predictions of proposed correlation and current numerical data.

5. Conclusions

Natural convection heat transfer between two horizontal concentric cylinders with two fins attached to the inner cylinder was investigated numerically using finite element technique. Laminar conditions up to Rayleigh number Ra_i of 5×10^4 were investigated. Effects of annulus diameter ratio, Rayleigh number, and fin length and its inclination angle on this type of flow were investigated. The thermal resistance of finned annuli was obtained using the conduction analysis. The thermal resistance decreases as fin length increases i.e. as the fin is extended within the annulus gap and increases as annulus diameter ratio increases i.e. as annular gap thickness increases. A correlation for thermal resistance of fined annuli was obtained. The free convection data were represented in terms of the effective thermal conductivity ratio k_e/k versus Rayleigh number Ra_i. The ratio k_e/k decreases as fin length ratio L_f increases. This means that convection effect is decreased as fin length increases. The inclination angle has weak effect on thermal conductivity ratio k_e/k. Correlation for thermal conductivity ratio is proposed, the predictions of the proposed correlation fit the current data of thermal conductivity ratio of finned annulus with an error of -6 to +15%.

Nomenclature

A_i is the surface area of inner cylinder, m^2,
A_f is the surface area of fins, m^2,
C_p is the specific heat, J/kg K,
D_i is the diameter of the inner cylinder, m,
D_o is the diameter of the outer cylinder, m,
d is the fin thickness, m,
Gr_i is the Grashof number based on inner cylinder diameter, $g\,\beta(T_o\text{-}T_i)\,D_i^3\,/\,\nu^2$,
g is the gravity acceleration, m/s^2,
g_x is the gravity acceleration component along X coordinate, m/s^2,
g_y is the gravity acceleration component along Y coordinate, m/s^2,
k is the thermal conductivity of fluid, W/m K,
k_e is the effective thermal conductivity, W/m K,
L_f is the fin length ratio, $l_f\,/\,(R_o\text{–}R_i)$,
l_f is the fin length, m,
P is the pressure, Pa,
Pr is the Prandtl number,
T is the temperature, K,
T_i is the temperature of inner cylinder and attached fin, K,
T_o is the temperature of outer cylinder, K,
T_m is the mean temperature, K,
R is the gas constant, J/kg K,
R_c is the correlation coefficient,
R_i is the annulus inner radius,
R_o is the annulus outer radius,
R_{tha} is the thermal resistance of unfinned annulus,
R_{thf} is the thermal resistance of finned annulus,
Ra_i is the Rayleigh number based on inner cylinder diameter, $Gr_i\,Pr$,
Ra_m is the modified Rayleigh Number, Eq. (14),
Q is the heat transfer, W,
Q_{cond} is the conductive heat transfer, W,
q_w is the local heat flux, W/m^2,
U is the velocity in X direction, m/s,
V is the velocity in Y direction, m/s,
X is the coordinate along fin, m,
x is the horizontal coordinate, m,
Y is the coordinate normal to fin, m, and
y is the vertical coordinate, m.

Greek symbols

β is the thermal expansion coefficient, K^{-1},
ν is the kinematic viscosity, m^2/s,
μ is the viscosity, Pa s, and
ρ is the fluid density , kg/m^3.

Subscripts

unf is the unfinned annulus,
cond is the conduction,
conv is the convection,
f is the finned annulus,
i is the inner cylinder, and
o is the outer cylinder.

References

[1] C. Liu, W.K. Mueller, and F. Landis, "Natural Convection Heat Transfer in Long Horizontal Cylindrical Annuli", ASME International Developments in Heat Transfer, Paper #117, pp. 976-984 (1961).
[2] U. Grigull and W. Hauf, “Natural Convection in Horizontal Cylindrical Annuli” Proceeding of the 3rd International Heat Transfer Conference, Vol. 2, pp. 182-195 (1966).
[3] C. Lis, "Experimental Investigation of Natural Convection Heat Transfer in Simple and Obstructed Horizontal Annuli", Proceeding of the 3rd International Heat Transfer Conference, Vol. 2, pp. 196-204 (1966).
[4] E.H. Bishop, C.T. Carley and R.E. Powe, “Natural Convection Oscillatory Flow in Cylindrical Annuli”, Int. J. Heat Mass Transfer, Vol. 11, pp. 1741-1752 (1968).
[5] M. Itho, T. Fujita, N. Nishiwaki and M. Hirata, “A New Method of Correlating Heat Transfer Coefficients for Natural Convection in Horizontal Cylindrical Annuli,” International Journal of Heat and Mass transfer, Vol. 13, pp. 1364-1368 (1970).
[6] T.H. Kuehn and R.J. Goldstein, “An Experimental and Theoretical Study of Natural Convection in the Annulus Between Horizontal Concentric Cylinders”, J. Fluid Mechanics, Vol. 74, Part 4, pp. 695-719 (1974).
[7] T.H. Kuehn and R.J. Goldstein, “Correlating Equations for Natural Convection Heat Transfer Between Horizontal Circular Cylinders”, Int. J. Heat Mass Transfer, Vol. 19, pp. 1127-1134 (1976).
[8] G. Raithby and K. Hollands, “A General Method of Obtaining Approximate Solutions to Laminar and Turbulent Free Convection Problem,” Advances in Heat Transfer, eds. T.F. Irine Jr. and J.P. Hartnett, Academic Press, pp. 265-315 (1975).
[9] R. Kumar, “Study of Natural Convection in Horizontal Annuli”, Int. J. Heat Mass Transfer, Vol. 31 (6), pp. 1137-1148 (1988).
[10] J.S. Yoo, “Natural Convection in a Narrow Horizontal Cylinder Annulus: Pr ≤ 0.3”, Int. J. Heat Mass Transfer, Vol. 41, pp. 3055-3073 (1998).
[11] J.S. Yoo, “Prandtl Number Effect on Bifurcation and Dual Solution in Natural Convection in Horizontal Annulus”, Int. J. Heat Mass Transfer, Vol. 42, pp. 3279-3290 (1999).
[12] L. Crawford and R. Lemlich, "Natural Convection in Horizontal Concentric Cylindrical Annuli", Industrial and Engineering Chemistry Fundamentals, Vol.1 (1), pp. 260-264 (1962).
[13] B. Farouk and S.I. Güçeri, "Laminar and Turbulent Natural Convection in the Annulus Between Horizontal Concentric Cylinders", J. Heat Transfer, Vol.104, pp. 631-636 (1982).
[14] P.M. Kolesnikov and V.I. Bubnovich, “Non-Stationary Conjugate Free-Convective Heat Transfer in Horizontal Cylindrical Coaxial Channels”, Int. J. Heat Mass Transfer, Vol. 31 (6), pp. 1149-1155 (1988).
[15] C. Kim and T. Ro, "Numerical Investigation on Bifurcative Natural Convection in an Air-Filled Horizontal Annulus", J. Heat Transfer, Vol. 116, pp. 135-141 (1994).
[16] S.M. ElSherbiny and A.R. Moussa, “Natural Convection in Air Layers between Horizontal Concentric Isothermal Cylinders”, Alexandria Engineering Journal, Vol. 43, pp. 297-311 (2004).

[17] D. Alshahrani and O. Zeitoun, “Natural Convection in Horizontal Cylindrical Annuli”, Submitted for Publication in Alexandria Engineering Journal (2005).
[18] V. Patankar and J. Chai, “Laminar Natural Convection in Internally Finned Horizontal Annuli” Numerical Heat Transfer, Part A, Vol. 24, pp. 67-87 (1993).
[19] M.I. Farinas, A. Garon and K. Saint-Louis, “Study of Heat Transfer in a Horizontal Cylinder with Fins”, Rev. Gén. Therm., Vol. 36, pp. 398-410 (1997).
[20] M.I. Farinas, A. Garon, K. St-Louis and M. Lacroix, “Study of Heat Transfer in Horizontal Bare and Finned Annuli”, Int. J. Heat Mass Transfer, Vol. 42, pp. 3905-3917 (1999).
[21] M. Rahnama, M. Mehrabian, S. Mansouri, A. Sinaie and K. Jafargholi, "Numerical Simulation of Laminar Natural Convection in Horizontal Annuli with Radial Fins”, Proc. Instn. Mech. Engrs., Vol. 213 Part E, pp. 93-97 (1999).
[22] M. Rahnama, M. Farhadi and M. Mehrabian, “Effect of Inner Cylinder Radius on Natural Convection in a Horizontal Annulus with Two Radial Fins”, Proceeding of the Ninth Asian Congress of Fluid Mechanics, Isfahan, Iran (2002).
[23] O. Zeitoun, “Natural Convection from a Vertical Plate in a Horizontal Cylinder”, Int. J. of heat and Technology, Vol. 23 (1), (2005).
[24] O. Zeitoun and M. Ali “Natural Convection Heat Transfer from Isothermal Horizontal Rectangular Ducts”, Submitted for Publication in Alexandria Engineering Journal (2005).
[25] S.V. Patankar, Numerical Heat Transfer and Fluid Flow, McGraw Hill (1980).

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