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THE INFLUENCE OF FRICTION FACTOR ON THE COMBINED CONVECTIVE AND RADIATIVE HEAT TRANSFER IN A RECTANGULAR DUCT WITH INTERIOR CIRCULAR CORE

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ABSTRACT:

The effect of friction factor on the steady state natural convective – radiative heat transfer in an inclined rectangular channel with concentric circular core is investigated from continuity, momentum and energy equations. These equations are normalized and solved using the Vorticity-Stream function and the Body Fitted Coordinates (B.F.C) methods. The finite difference approach with the Line Successive Over-Relaxation (LSOR) method is used to obtain all the computational results. The (B.F.C) method is used to generate the grid of the problem. A computer program (Fortran 90) is built to calculate the Nusselt number (Nu) and friction factor *f* in steady state flow for thermal boundary condition of constant wall temperature and for aspect ratio AR (0.55-1) and geometry ratio GR (0.1-0.9). The fluid Prandtl number is Pr = 0.7, Rayleigh number ($0 \le Ra \le 10^4$), Reynolds number ($1 \le Re \le 2000$), Optical Thickness ($0 \le t \le 10$), Conduction- Radiation parameter ($0 \le N \le 100$) and Inclination angle ($0^\circ \le \lambda \le 90^\circ$). The results show reasonable representation to the relation between Nusselt number and friction factor with other parameters (AR, GR, and Re, Ra, λ, t and N). Generally, Nu will be increased with increasing Ra, t, N, λ, and Re. In the same time, *fRe* will be increased when Re and GR increase and AR decrease. The effect of radiation on the bulk temperature is concluded by correlation equations.

الخلاصة

تمت دراسة تأثير معامل الإحتكاك على أنتقال الحرارة بالحمل المختلط–المشع لجريان مستقر خلال قناة مستطيلة مائلة مع جزء مركزي دائري بأستخدام معادلة الإستمرارية، الزخم والطاقة. تم تحويل هذه المعادلات الى معادلات بدون وحدات ثم الى صيغة دالة الإنسياب–الدوامية ثم الى صيغة مطابقة الإحداثيات. أستخدمت طريقة الفروق المحددة لإجراء جميع الحسابات باستخدام نظام مطابقة الإحداثيات لبناء شبكة النظام. برنامج فورتران ٩٠ أستخدم لحساب رقم نسلت (Nu) ومعامل الاحتكاك *f* لجريان مستقر ولظروف حرارية محيطية ذات درجة حرارة جدار ثابتة ولنسبة باعية (0.55-1) ولنسبة شكل تتراوح بين(0.1-0.9). رقم برانتل للمائع(Pr = 0.7)، رقم رايلي ($0 \le Ra \le 10^4$) رقم رينولد ($1 \le Re \le 2000$) والسمك البصري ($0 \le t \le 10$) وبارامتر التوصيل– الإشعاع ($0 \le N \le 100$) وزاوية الميل ($0^\circ \le \lambda \le 90^\circ$) . النتائج بينت تمثيل معقول لعلاقة رقم نسلتNu ومعامل الإحتكاك مع البارامترات الأخرى(Ra, λ, t and N) (AR, GR, Re,) . عموما رقم نسلتNu يزداد بزيادة (Ra, t, N, λ, Re) . في نفس الوقت يزداد متوسط حاصل ضرب معامل الإحتكاك *f* برقم رينولد (*fRe*) مع زيادة رقم رينولد Re ونسبة الشكل GR وبإنخفاض قيمة النسبة الباعية AR . تم التوصل الى معادلات لتأثير الإشعاع على درجة الحرارة.

KEY WORDS
Mixed convection, radiation, rectangular duct, circular core, steady state laminar flow, friction factor.

INTRODUCTION:

[Balaji et al, 2005] reported the use of the technique of combining asymptotic with computational fluid dynamics to handle the problem of combined laminar mixed convection and surface radiation from a two – dimensional differentially heated lid driven cavity. The fluid under consideration was air, which is radiative transparent, and all the walls were assumed to be gray and diffuse and have the same hemispherical total emissivity (ε). Correlations were obtained for the weighted average convective Nusselt number that is valid for the entire mixed convection range of Richardson number ($0 \le Ri \le \infty$) and ($0 \le \varepsilon \le 1$).

[Bahloui et al, 2005] studied numerically a mixed convection coupled with radiation in an inclined channel with constant aspect ratio and locally heated from one side. The convective radiative and total Nusselt numbers were evaluated on the cold surface and at the exit of the channel and for different combinations of the governing parameters. The results obtained show that the flow structure is significantly altered by radiation which contributes to reduce or to enhance the number of the solutions obtained.

[Bello-Ochende and Adegun, 2002] also worked on combined convective and radiative heat transfer in a tilted, rotating, uniformly heated square duct with a centered circular cylinder.

[Adegun and Bello-Ochend, 2004] studied numerically the steady state laminar forced and free convective and radiative heat transfer in an inclined rotating rectangular duct with a centered circular tube for a hydro dynamically fully developed flow. The governing equations were solved by using Gauss – Seidel iteration technique subject to give boundary constraints. A thermal boundary condition of uniform wall temperature in the flow direction was considered. The results for local and mean Nusselt number for various values of Reynolds number (Re), Rayleigh number (Ra), Rotational Reynolds number (Ro), Geometric Ratio (GR), Aspect Ratio (AR), Radiation – Conduction parameter (N), Optical thickness (t) and emissivity (ε) were presented. The results show that radiation and rotation enhance heat transfer and that heat transfer from the surface of the circle exceeds that of the rectangle.

The objective of this work consists in studying the coupling between fully developed laminar mixed convection and radiation in an inclined rectangular channel including circular core by examining the effect of the emissivity ε, of the flowing fluid and the friction factor effect on the temperature distribution and the flow structure within the channel. A detailed attention will be given to the contribution of the friction to overall heat transfer for different combinations of the governing parameters.

The channel is fixed (not rotating) and the effect of friction factor is studied for thermal boundary condition of constant wall temperature and for ($0.1 \le GR \le 0.9$) ($0.55 \le AR \le 1$), ($1 \le Re \le 2000$), ($0 \le Ra \le 10^4$) and ($0^\circ \le \lambda \le 90^\circ$).

MATHEMATICAL MODEL:

The geometry considered is depicted in **Fig.(1)** with the Cartesian coordinate system employed in solving the problem. It consists of an inclined rectangular channel of finite length including circular core, with an aspect ratio ($0.55 \leq AR \leq 1$). The system is submitted to an imposed flow of ambient air, the flow is occurred between the circular tube and the rectangular duct (Annulus). This annulus is symmetrical about Y-axis ($\partial/\partial x = 0$). The width and height of the channel are L and H respectively. The diameter and radius of tube are D and R respectively and the hydraulic diameter is d:

$$d = \frac{4A}{P^*} = \frac{2(HL - \pi R^2)}{(H + L + \pi R)} \tag{1}$$

The working fluid is assumed absorbing, emitting and the fluid properties are assumed constant except for density variation with temperature resulting in the secondary flows generated by the buoyancy forces. The axial (z) direction shown in **Fig. (1)** is the predominant direction for the fluid flow. The flow is laminar, and viscous dissipation effects are neglected. Axial conduction and radiation are assumed negligible following [Yang and Ebadian, 1991].

Governing Equations:

The governing equations are:

Continuity Equation

$$\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} = 0 \tag{2}$$

Momentum Transport Equation

The momentum transport equations in the x, y and z directions are respectively:

$$u\frac{\partial u}{\partial x} + v\frac{\partial u}{\partial y} = -\frac{1}{\rho}\frac{\partial p}{\partial x} + \nu\ \Delta^2 u \tag{3}$$

$$u\frac{\partial v}{\partial x} + v\frac{\partial v}{\partial y} = -\frac{1}{\rho}\frac{\partial p}{\partial y} + \nu\ \Delta^2 v - \beta\ g\left(T_w - T\right)\ \cos\lambda \tag{4}$$

$$u\frac{\partial w}{\partial x} + v\frac{\partial w}{\partial y} = -\frac{1}{\rho}\frac{\partial p}{\partial z} + \nu\ \Delta^2 w - \beta\ g\left(T_w - T\right)\ \sin\lambda \tag{5}$$

Energy Transport Equation:

In the absence of energy sources and viscous energy dissipation, the energy equation for steady state flow, with radiation incorporated is:

$$u\frac{\partial T}{\partial x}+v\frac{\partial T}{\partial y}+w\frac{\partial T}{\partial z}=\alpha\left(\frac{\partial^2 T}{\partial x^2}+\frac{\partial^2 T}{\partial y^2}\right)+\frac{\sigma\ K_R\varepsilon}{\rho\ c_p}\left(T_w^4-T^4\right) \tag{6}$$

Normalization Parameters:

The variables in the governing equations and boundary conditions are transformed to dimensionless formula by employing the following transformation parameters:

$$X=\frac{x}{d}\ ,\quad Y=\frac{y}{d}\ ,\quad Z=\frac{z}{d}$$

$$U=\frac{ud}{v}\ ,\quad V=\frac{vd}{v}\ ,\quad W=\frac{wd}{v}$$

$$\theta=\frac{T}{T_w}\ ,\quad \frac{\partial p}{\partial z}=-\frac{4\rho\nu^2}{d^3}\mathrm{Re}\ ,\quad \frac{\partial T}{\partial z}=\frac{T_w}{\mathrm{Pr}\,d}\ ,\quad \mathrm{Pr}=\frac{\nu}{\alpha}$$

$$N=\frac{4\sigma\varepsilon T_w^3}{K_R k}\ ,\quad t=K_R d\ ,\quad U=\frac{\partial\psi}{\partial Y}\ ,\quad V=-\frac{\partial\psi}{\partial X}$$

Where d is the hydraulic diameter.

After eliminating the pressure terms in the two momentum equations in X and Y directions by using vorticity-stream function method, the governing equations in the dimensionless form become:

$$\frac{\partial\psi}{\partial Y}\frac{\partial\omega}{\partial x}-\frac{\partial\psi}{\partial X}\frac{\partial\omega}{\partial Y}=\left(\frac{\partial^2\omega}{\partial X^2}+\frac{\partial^2\omega}{\partial Y^2}\right)-\frac{R_a\ \cos\lambda}{\mathrm{Pr}}\frac{\partial\theta}{\partial X} \tag{7}$$

Stream Function Equation

$$-\omega=\frac{\partial^2\psi}{\partial X^2}+\frac{\partial^2\psi}{\partial Y^2} \tag{8}$$

Axial Momentum Equation

$$\frac{\partial\psi}{\partial Y}\frac{\partial W}{\partial X}-\frac{\partial\psi}{\partial X}\frac{\partial W}{\partial Y}=\left(\frac{\partial^2 W}{\partial X^2}+\frac{\partial^2 W}{\partial Y^2}\right)+4\,\mathrm{Re}-\frac{Ra\ \sin\lambda}{\mathrm{Pr}}(1-\theta) \tag{9}$$

Dimensionless Energy Equation

$$\frac{\partial\psi}{\partial Y}\frac{\partial\theta}{\partial X}-\frac{\partial\psi}{\partial X}\frac{\partial\theta}{\partial Y}=\frac{1}{\mathrm{Pr}}\left(\frac{\partial^2\theta}{\partial X^2}+\frac{\partial^2\theta}{\partial Y^2}\right)-\frac{W}{\mathrm{Pr}}+\frac{N\ t^2}{4\,\mathrm{Pr}}\left(1-\theta^4\right) \tag{10}$$

The boundary conditions applicable to these equations are:
At the inlet of the duct (Z= 0):

$$U = V = \psi = \omega = 0\,,\ \theta = 0.5\ ,\quad W = \frac{\mathrm{Re}}{vd}$$

(2) At the walls:

$$U = V = W = \psi = 0\,,\ \theta = 1$$

$$\frac{\partial \theta}{\partial X} = \frac{\partial \psi}{\partial X} = \frac{\partial \omega}{\partial X} = \frac{\partial W}{\partial X} = 0 \quad \text{(at symmetry line)}$$

NUMERICAL METHODS

Numerical Grid Generation:

The elliptic transformation technique which was originally proposed by [Fletcher, 1988] is applied to generate the curvilinear grid for dealing with the irregular cross sections. The transformation functions $\xi = \xi(X, Y)$ and $\eta = \eta(X, Y)$ are obtained to accommodate the irregular shape by solving the following partial differential equations:

$$\frac{\partial^2 \xi}{\partial X^2} + \frac{\partial^2 \xi}{\partial Y^2} = G(\xi, \eta) \tag{11}$$

$$\frac{\partial^2 \eta}{\partial x^2} + \frac{\partial^2 \eta}{\partial Y^2} = S(\xi, \eta) \tag{12}$$

Where *G* and *S* two functions which are defined to artificially adjust the density of the grid locally. Using the curvilinear grid obtained, the governing eq. (7) to (10) and the boundary conditions are then discretized and solved in the computation domain (ξ, η). In this work, an (81 X 61) grid in the transformed domain $(\xi \text{ X } \eta)$ is adopted. The grid systems have been properly adjusted to be orthogonal locally at the boundaries. The grid generation technique used is standard and well accepted. Therefore, further description about this technique would not give here.
By using this method, the following general equation can be used to generate all the governing –equations (7-10) in computational co-ordinate's formula:

$$J\Gamma\left(\psi_\eta \phi_\xi - \psi_\xi \phi_\eta\right) = \left(\tau\phi_\xi + \varpi\phi_\eta + \alpha_1\phi_{\xi\xi} - 2\beta_1\phi_{\xi\eta} + \gamma\phi_{\eta\eta}\right) + suJ^2 \tag{13}$$

Where ϕ represent the general variable which may be ω, W or θ and su is the source term. Where Γ = 1 for vorticity transport and axial momentum equations and Γ = Pr for energy equation.

Finite Difference Formulation

The three-point central difference formula is applied to all the derivatives. Each of the governing equations can be rewritten in a general form as:

$$ap_{(i,j)}\phi_{(i,j)} = ae_{(i,j)}\phi_{(i+1,j)} + aw_{(i,j)}\phi_{(i-1,j)} + an_{(i,j)}\phi_{(i,j+1)}as_{(i,j)}\phi_{(i,j-1)} + SU_{(I,J)}J_{(I,J)} \quad (14)$$

Where:

$$ap_{(i,j)} = 2\left(\alpha_{1(i,j)} + \gamma_{(i,j)}\right)$$
$$ae_{(i,j)} = \alpha_{1(i,j)} - B$$
$$aw_{(i,j)} = \alpha_{1(i,j)} + B$$
$$an_{(i,j)} = \gamma_{(i,j)} - C$$
$$as_{(i,j)} = \gamma_{(i,j)} + C$$

$$B = \left(J_{(i,j)}\Gamma \frac{\psi_{(i+1,j)} - \psi_{(i-1,j)}}{2} - \tau_{(i,j)} \right) \Big/ 2$$

$$C = \left(- J_{(i,j)}\Gamma \frac{\psi_{(i,j+1)} - \psi_{(i,j-1)}}{2} - \varpi_{(i,j)} \right) \Big/ 2$$

$$SU_{(I,J)} = -\frac{\beta_{1(i,j)}}{2J_{(i,j)}}\left(\phi_{(i+1,j+1)} - \phi_{(i+1,j-1)} - \phi_{(i-1,j+1)} + \phi_{(i-1,j-1)}\right) + su_{(i,j)}J_{(i,j)}$$

In the equations above i and j indicate to the points of the grid in the generalized coordinates ξ and η respectively.

As pointed out in [Anderson, 1984] the Relaxation method can be employed for the numerical solution of the eq. (8). For this study, the LSOR method [Fletcher, 1988 and Anderson, 1984] is used to solve equations (7, 9 and 10). The convergence criterion for the inner iteration ($Error_{in}$) of ψ is 10^{-4} and for the outer iteration ($Error_{out}$) of θ_b is 10^{-10}, where:

$$Error_{in} = 2\left(\alpha_{1(i,j)} + \gamma_{(i,j)}\right)\Delta\psi_{(i,j)} \quad (15)$$

$$\Delta\psi_{(i,j)} = \frac{\psi_{(i,j)}^{it+1} - \psi_{(i,j)}^{it}}{RP} \quad (16)$$

Where RP is the Relaxation Parameter and equal 1.1 and represent the number of iterations. The outer iteration is checked only for θ_b as follow:

$$Error_{out} = \frac{\theta_b^{it+1} - \theta_b^{it}}{\theta_b^{it}} \leq 10^{-10} \quad (17)$$

EVALUATION OF HEAT TRANSFER:

The peripheral heat transfer is defined through the conduction referenced Nusselt number as:

Local Nusselt number

The peripheral local Nusselt number on the walls of the channel is computed from:

$$Nu_L = \frac{-\frac{\partial \theta}{\partial n}\Big|_w}{(1-\theta_b)} \tag{18}$$

Where *n* represent the dimensionless normal outward direction.

The mean Nusselt number on the wall of the rectangular duct and circular tube is obtained by using Simpson's rule:

$$Nu_{c,r} = \frac{1}{s}\int_s Nu_L \, ds \tag{19}$$

Where s represents the length of the wetted perimeter in the rectangular duct and in the circular tube.

The mean Nusselt number (Nu) is a measure of the average heat transfer over the internal surface of the rectangular duct and the outer surface of the circular configuration. It is computed from the following equation:

$$Nu = C_c \; Nu_c + C_r Nu_r \tag{20}$$

Where, C_C Nu_C is a measure of average heat transfer from the outer surface of the circular core while C_r Nu_r corresponds to heat transfer from of the internal surface of the rectangular duct. Cc and Cr are the perimetric ratios for the heat transfer and are defined as:

$$C_c = \frac{\pi R}{H + L + \pi R}$$

$$C_r = \frac{H + L}{H + L + \pi R}$$

Calculation of Friction Factor

The friction factor is very important, where; the friction surface resistance is determined by it. The friction factor can be obtained as follow:

$$\tau_S = \mu \frac{dw}{dn}\Big|_w \tag{21}$$

The shear stress at the wall may be expressed in term of friction factor *f* as follow [Mohanty A.K. and Viskanta R., 1987] and [Balaji et al, 2005]:

$$f \frac{\rho \overline{w}^2}{2} = \mu \frac{dw}{dn}\Big|_w \tag{22}$$

$$f = \frac{2\mu}{\rho \overline{w}^2} \frac{dw}{dn}\Big|_w$$

By using the dimensionless magnitudes the equation above become:

$$f = \frac{2}{\overline{W}^2} \frac{dw}{dn}\Big|_w \tag{23}$$

$$f \,\mathrm{Re} = \frac{2\,\mathrm{Re}}{\overline{W}^2} \frac{dw}{dn}\Big|_w \tag{24}$$

RESULTS AND DISSCUSION:

Fig. (2) illustrates the effect of Re on *fRe* value with increasing in GR. It is shown that for small value of GR, the effect of Re will be large. This effect results from the increasing of the axial velocity of the flow that resulting from increasing in Re. The increasing in the axial velocity leads to increase the friction between the air and the walls of the channel. For the high values of GR, the effect of Re will be very small because of the increasing in the surface area.

Fig. (3) show the effect of inclination angle (λ = 0, 45 and 90) on the value of *fRe* with increasing in GR. The effect of inclination on *fRe* is constant with variation in GR value, where *fRe* increased with the increasing of GR .

The variation of *fRe* with AR and for different value of λ (λ = 0, 45 and 90)is shown In **Fig. (4)**. The effect of inclination angle will be decreased with increasing in AR, while *fRe* decrease. The inclination of the channel leads to increase the effect of the buoyancy force component towards the flow direction. This effect will be small with increasing the surface area because of the decrease of the vorticity intensity of the flow and the buoyancy force component. The effect of inclination on *fRe* is approximately constant with variation in AR.

The variation in *fRe* with AR is shown in **Fig. (5)**. This figure show that when AR is increased *fRe* will be decreased because of the increase in AR lead to increase the cross section area, and that led to increase the intensity of the air vorticity and the effect of the buoyancy force which then decrease the rate of heat transfer. The effect of radiation is noted for AR < 0.8, but with increasing in AR, the effect of radiation will be neglected.

Fig. (6) shows the variation of *fRe* with Re, where the increasing of Re cause to increase the axial velocity and that in turn cause to increase *fRe* between the air and the walls of the channel The radiation effect on the *fRe* is approximately constant.

The effect of inclination angle on the Nu value with the variation in Re is shown in **Fig. (7).** The inclination effect is very small when Re < 500, but it is increased with increasing in Re above 500.

In **Fig. (8)** the effect of Ra on Nu value is shown with the variation of Re. The effect of Ra is approximately constant with the variation of Re value while Nu decreased with Re increasing.

The variation of Nu with Ra value is cleared in **Fig. (9)**. It is shown that when Ra increased Nu will be increased because of the increasing of the effect of the buoyancy force (natural convection) on the rate of heat transfer. The radiation effect also increased with increasing in Ra because of the heat gain that transform to the air by radiation will be accelerated the flow and the vorticities of air and that lead to increase the rate of heat transfer.

The inclination effect is shown in **Fig. (10)**. This inclination effect will be increased with increasing in Ra. This is resulted from increase the effect of the buoyancy force component towards the flow direction.
A reasonable agreement between the present work results and the previous researches are shown in **Fig. (11).**

Radiation Effect:

the effect of radiation on the bulk temperature will be written in correlations as follow:

$$\theta_b = -12.287 + 13.304 \ GR^{6.2*10^{-3}} \quad \text{(With Radiation)}$$

$$\theta_b = -15.747 + 16.763 \ GR^{5,4*10^{-3}} \quad \text{(Without Radiation)}$$

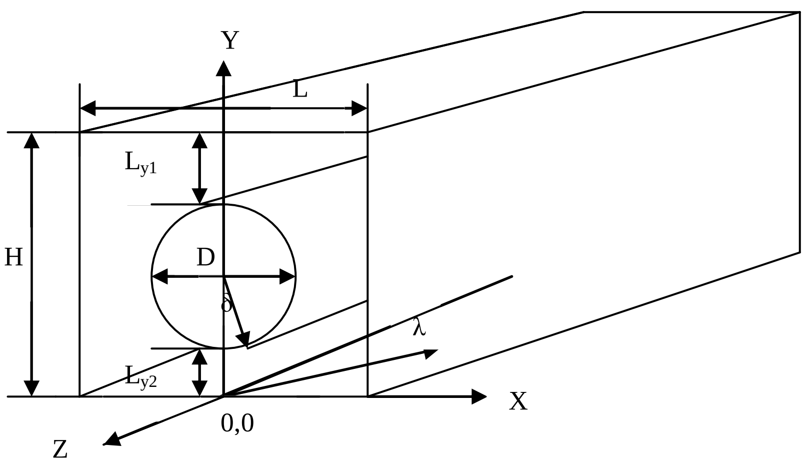

Fig.(1) Schematic of the Problem Geometry

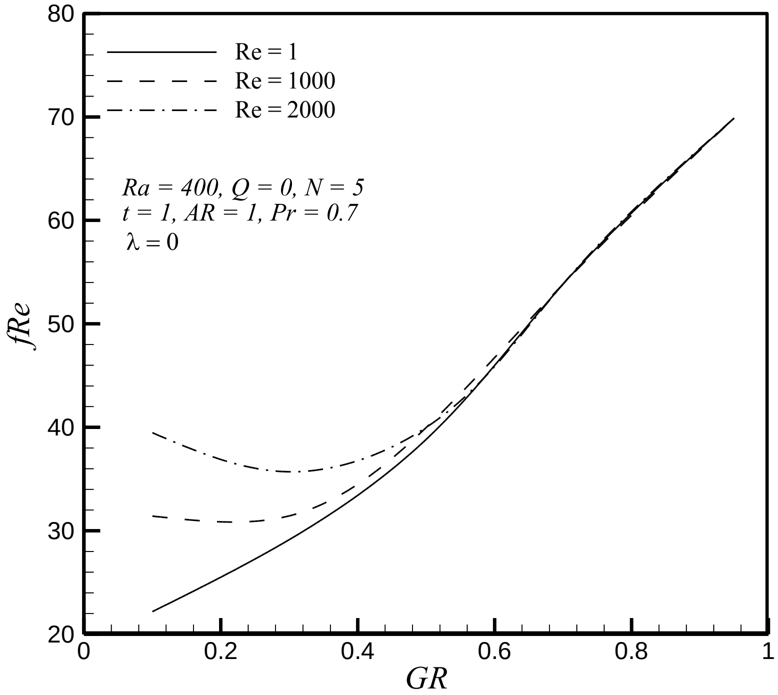

Fig. (2) The Variation of *fRe* with GR for Different Values of Re

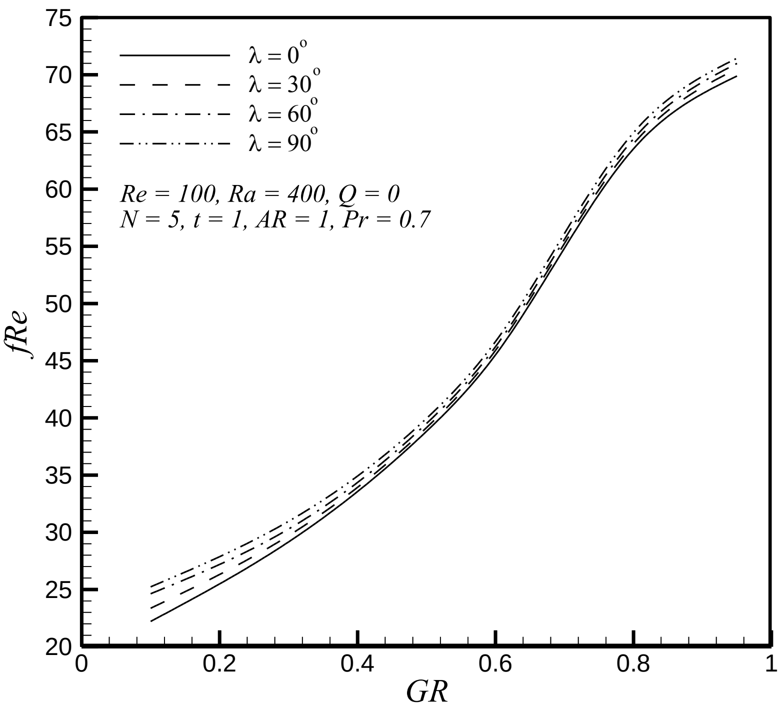

Fig. (3) The Variation of *fRe* with GR for Different Values of λ

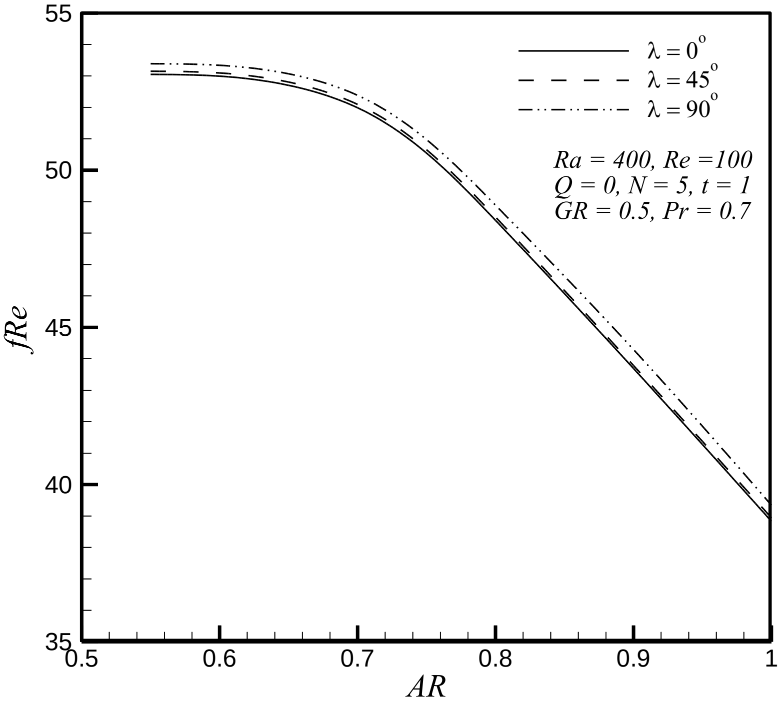

Fig. (4) The Variation of *fRe* with AR for Different Values of λ

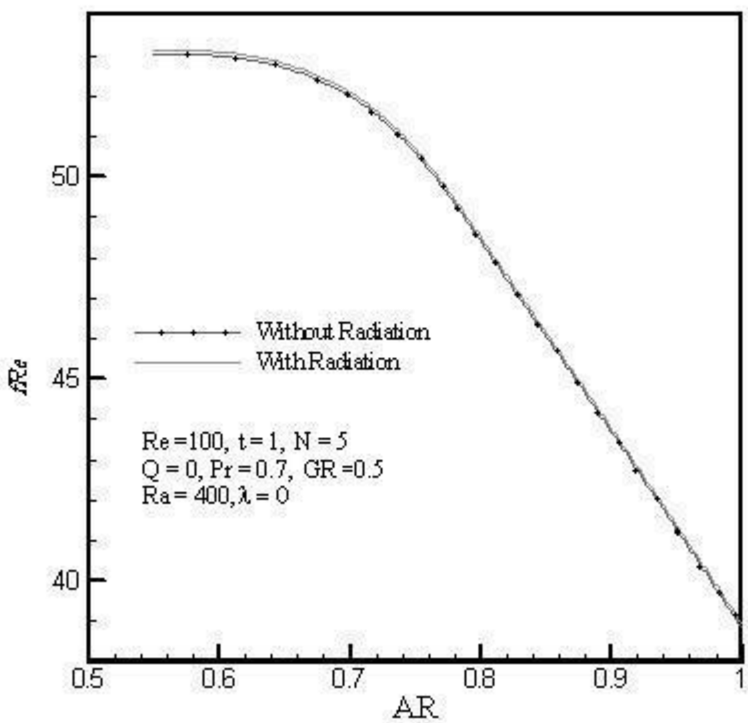

Fig. (5) The Variation of *fRe* with AR

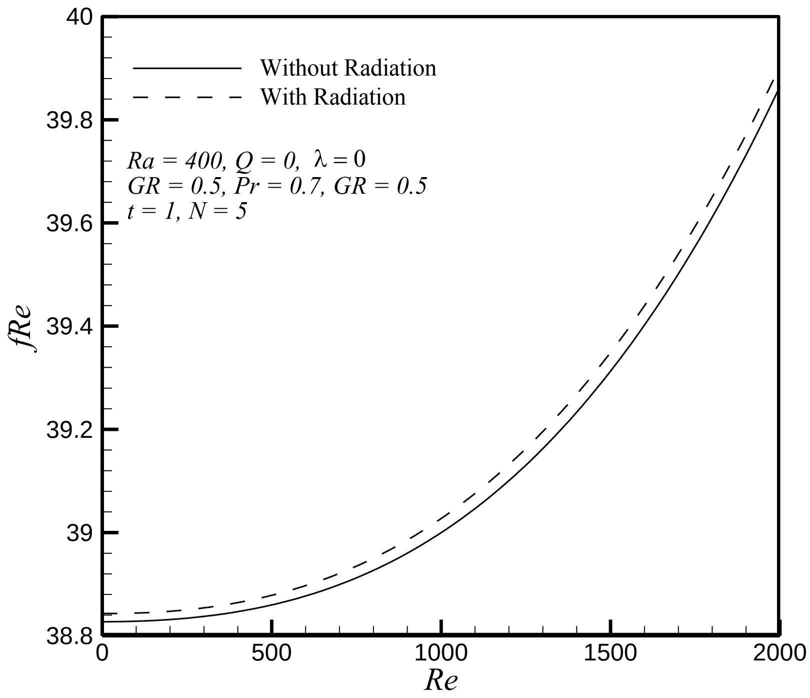

Fig. (6) The Variation of *fRe* with Re

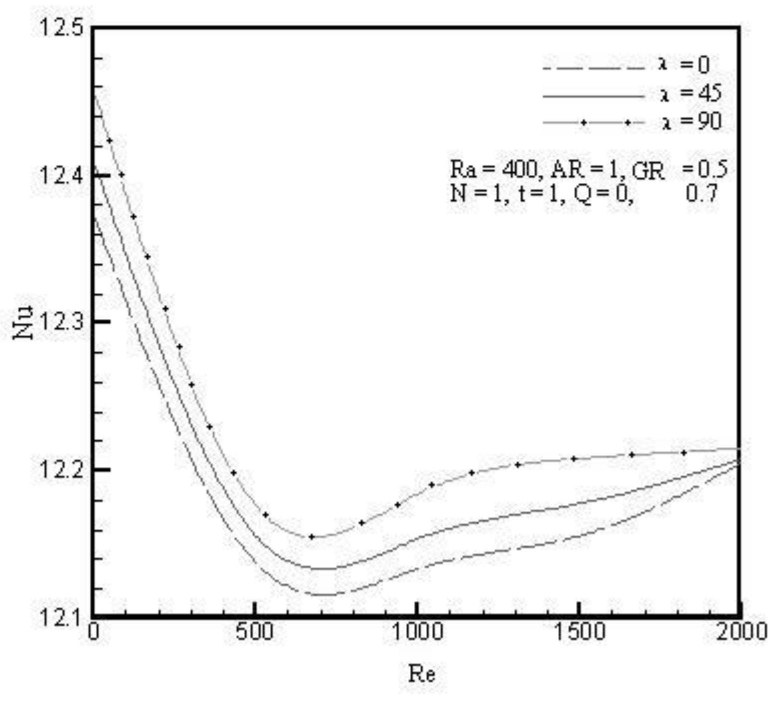

Fig. (7) The Variation of Nu with Ra for Different Values of λ

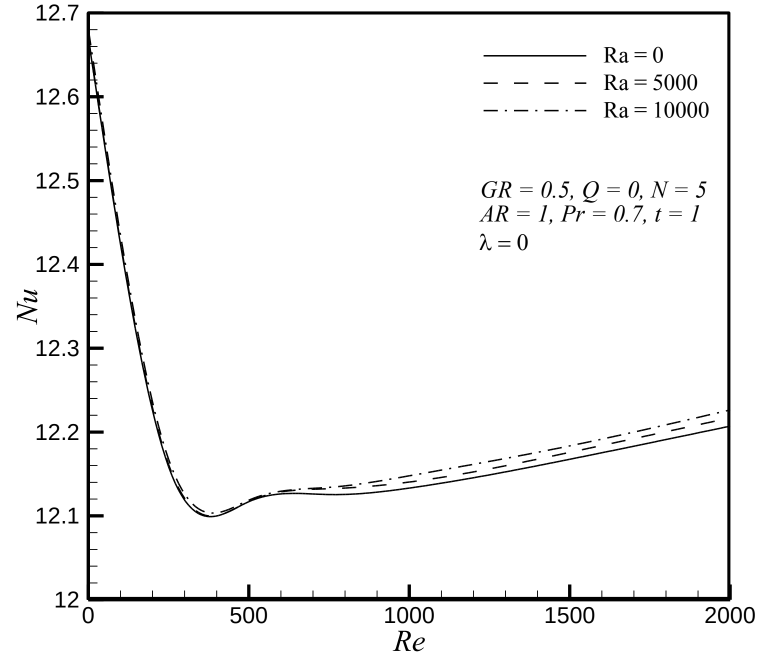

Fig. (8) The Variation of Nu with Re for Different Values of Ra

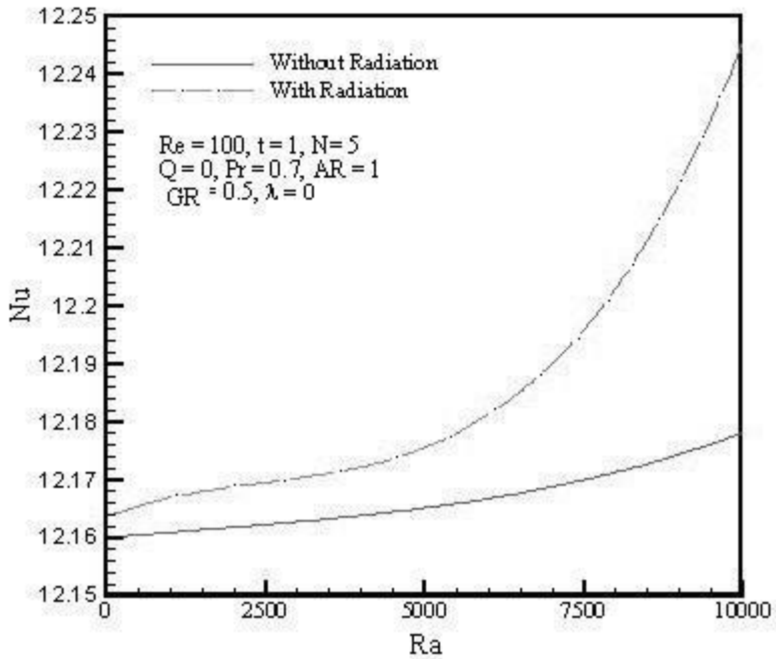

Fig. (9) The Variation of Nu with Ra

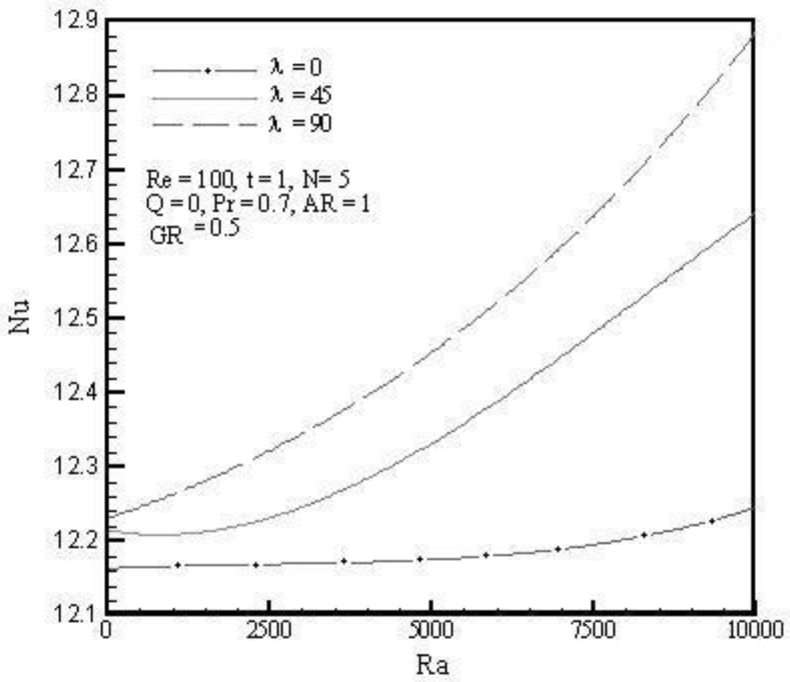

Fig. (10) The Variation of Nu with Ra for Different Values of λ

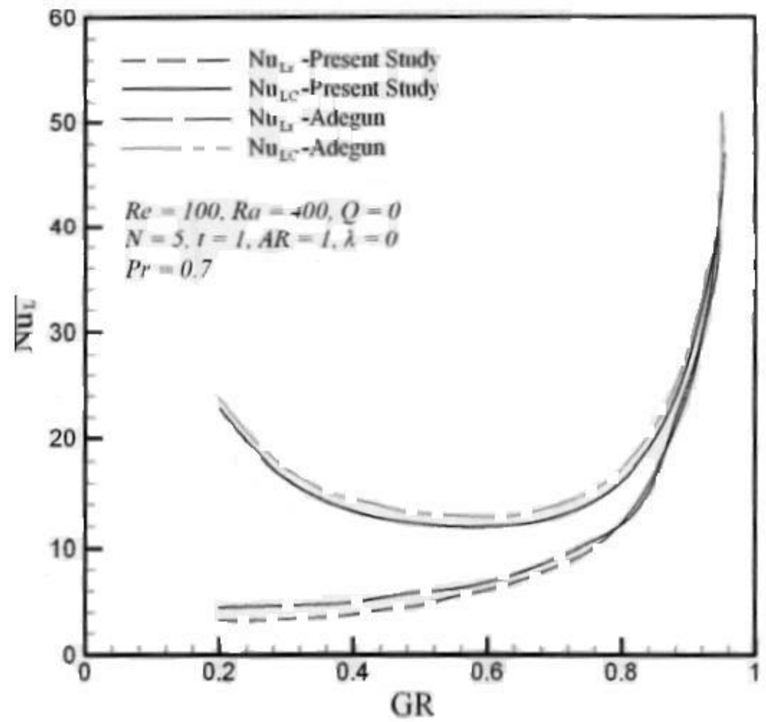

Fig.(11) Comparison Results (With Radiation)

CONCLUSIOS:

From the present work results and for the channel that described previously, the following conclusions can be obtained:

(1) *fRe* increase with the increasing of GR and Re and not affected by the change of channel inclination from horizontal to vertical.

(2) *fRe* decrease with increasing of AR, while it is constant with the variation of λ and not affected by the radiation.

(3) The direct effect of velocity variation is on the sub-channel average mass flow rate, which in turn controls the *fRe* value through eq. (24) and buoyancy reduces frictional resistance.

(4) For Re < 500 Nu decrease with Re increasing, but when Re is greater than 500, Nu will be increased. Nu increased with the increasing of Ra and λ.

(5) The radiation effect on the bulk temperature is known by making a correlation equations and this effect show that θ_b will be increased.

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NOMENCLATURE

Symbol	Description	Units
AR	Aspect ratio (AR = L /H)	-
Cp	Specific heat at constant pressure	J/kg.K
D	The diameter of circular tube	m
d	The hydraulic diameter of the channel	m
f	*Friction factor* $\left(f = \frac{2}{W^2}\frac{\partial W}{\partial n}\Big|_w\right)$	-
fRe	*The mean friction factor multiplying by Re*	-
GR	Geometry ratio (GR = D/H)	-
H	The height of the rectangular duct	m
J	Jacobean of direct transformation	-
K_R	Volumetric absorption coefficient	m^{-1}
L	The width of the rectangular duct	m
N	Conduction-radiation parameter($4\sigma\varepsilon T_w^3/K_R$ k)	-
Nu	The mean Nusselt number	-
Nu_{LC}, Nu_{Lr}	The local Nusselt number in the circular and rectangular walls respectively	-
p	Air pressure	N/m^2
P	Normalized air pressure	-

P^*	Wetted perimeter	m
Pr	Prandtl number ($Pr = \nu/\alpha$)	-
R	The circular tube radius	m
Symbol	Description	Units
Ra	Rayleigh number ($Ra= \beta\ g\ d^3\ T_w/\ \alpha\ \nu$)	-
Re	Reynolds number [$Re = (-d^3/4\rho\nu)(\partial P/\partial z)$]	-
T	Air temperature	˚C
T_w	Wall temperature	˚C
T_b	Bulk temperature	˚C
t	Optical thickness($t = K_R\ D_h$)	-
u, v, w	The velocity components in x, y and z direction respectively	m/s
U, V, W	The normalized velocity components in x, y and z direction respectively	-
x, y, z	The physical coordinates of the channel	m
X, Y, Z	The dimensionless physical coordinates of the channel	-

GREEK SYMBOLS

Symbol	Description	Units
α	Thermal diffusivity	m^2/s
$\alpha_1, \beta_1, \gamma, \varpi, \tau$	The coefficient of transformation of BFC	-
β	Coefficient of thermal expansion	1/K
ε	Emissivity	-
ξ, η	Coordinates in the transformed domain	m

θ	Dimensionless air temperature	-
θ_b	Dimensionless bulk temperature	-
θ_w	Dimensionless wall temperature	-
$\theta_C.\theta_r$	Dimensionless temperature of the circular tube and rectangular duct walls respectively	-
Symbol	Description	Units
λ	Inclination angle	°
ν	Kinematics' viscosity of air	m^2/s
ρ	Air density	kg/m^3
σ	Stefan Boltizmann constant	W/m^2K^4
τ_s	*Shear stress*	N/m^2
ψ	Dimensionless air stream function	-
ω	Dimensionless air vorticity	-

Superscripts & Subscripts

Symbol	Description	Units
b	Bulk	-
C	Circular tube	-
g	Generation	-
(i, j)	Grid nodes in X and Y directions	-
L	Local	-
r	*rectangular duct*	-
w	*Wall*	-
°	*Degree*	-
‾	*Average*	-