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Convection Concentric Annulus Vertical Cylinders Filling Porous Media

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Abstract

An experimental and theoretical study of convection vertical concentric annulus two cylinders filling with porous medium was carried out . The experimental part was contained manufactured test rig and connected thermocouples then power on and reading temperatures until steady state . The theoretical part was contained solved governing equations by using Fluent software which drawing temperature profiles and stream functions for different ratio (r/R) and Ra . The results contained the relations between Nu with Ra and Nu with ratio (r/R) then experimental Nu with theoretical Nu so there was a good agreement between them.

Introduction

Jack(2006) was presented a results of heat transfer measurements in a differentially heated annulus of fluid for both the non-rotating .The resulting main effects appear to be exerted through a decrease in the thickness.The steady conjugate heat transfer problem in vertical open ended concentric annuli under the laminar natural convection flow regime was presented by El-Shaarawi(1995). A finite difference technique was used to solve the governing equations, Numerical results are presented for a Newtonian fluid of Pr = 0.7 in a fluid annulus of radius ratio 0.5.the effect of solid fluid conductivity ratio on the induced flow behavior and some engineering parameters, such as the heat absorbed and the miring-cup temperature. A finite-difference scheme to solve the transient conjugated heat transfer problem in a concentric annulus with simultaneously developing hydrodynamic and thermal boundary layers presented by El-Shaarawi (1999). The effects of solid-fluid conductivity ratio and diffusivity ratio on the thermal behavior of the flow have been investigated. Numerical results were presented for a fluid flowing in an annulus of radius ratio 0.5 with various values of inner and outer solid wall thicknesses. Dirker(2000) were presented a comparative study of literature involving convective heat transfer in annuli .Which shown that more research is needed in the area of convective heat transfer correlations in concentric annuli,as little agreement are found among existing correlation. A correlation predicting Nu in annuli with ratios ranging from 1.7 to 3.2 was developed experimentally for water as fluid .The correlation has an accuracy of 3% in terms of experimental values for Re range. Projahn & Beer(1985) was studied numerically the various Pr on the laminar convection flow between concentric and vertically eccentric cylinders. Two independent computer programs of the governing equations were used local heat transfer rates are found to depend on the Pr in addition to the Ra dependence. A numerical study of the natural convection heat transfer of cold water ,having the density inversion between two isothermal eccentric horizontal cylinder is studied by Raghavarao & Sanyasiraju (1996).The numerical solution are obtained for Ra ranging between 1000-100000 a Pr 12 and inversion parameter 0,-1,and -2 .The affect of the radius ratio on the flow patterns and heat transfer coefficient was studied by taking the radius ratio as 1.5 and 2 .It is again for Y=-1 ,the minimum heat transfer is observed like in the case of concentric annulus. Numerical solutions of transient natural convection heat transfer problem in horizontal isothermal cylindrical annuli were presented by Hassan & Al-lateef (2007).The results showed that Grashof number was changed with influence of variation in Prandtle number and diameter ratio .Good agreement with previous data was obtained. An experimental and analytical model for natural convection in a horizontal circular annulus with isothermal boundary condition was presented by Teertstra & Yovanovich (2000).Comparisons between the available data and models were carried with specific recommendations concerning. In this work an experimental and theoretical study of convection concentric annulus cylinders filling with porous medium was carried out.The aim of this study to present the effect of ratio (r/R) and heat flux or Ra on heat transfer coefficient with porous medium, then finding a relation between Nu with (r/R) ratio for different Ra and Nu with Ra for different ratio (r/R).

Experimental Part

The experimental investigation was carried out on annulus two cylindrical concentrically rig, especially designed for the present study covering all tests. The test rig was designed and manufactured to fulfill the requirements of the test system for annulus concentric cylinders .The main rig and the apparatus of the system illustrated in fig (1) , which contained of inner steel cylinder (r=0.012,0.02,0.026m) thickness wall (0.002m) and length (0.26m) put on outer C.I cylinder (R=0.12m), thickness wall (0.008m) and length (0.26m) with heater inside the inner cylinder , closed by Gipson and the annulus between heater and inner cylinder filled with sand to ignore the natural convection in heater and getting constant temperature around cylinder. The diameter ratio was used (r/R=0.10,0.16,0.22) but the length was constant . The annulus between two cylinders filling with porous media (Gravel) the thermal properties was determined by Al-Kamil(1989).For measuring temperature12 thermocouples type T were placed longitudinally along the section at points located on the outer surface of inner cylinder for three levels as shown in fig(2).

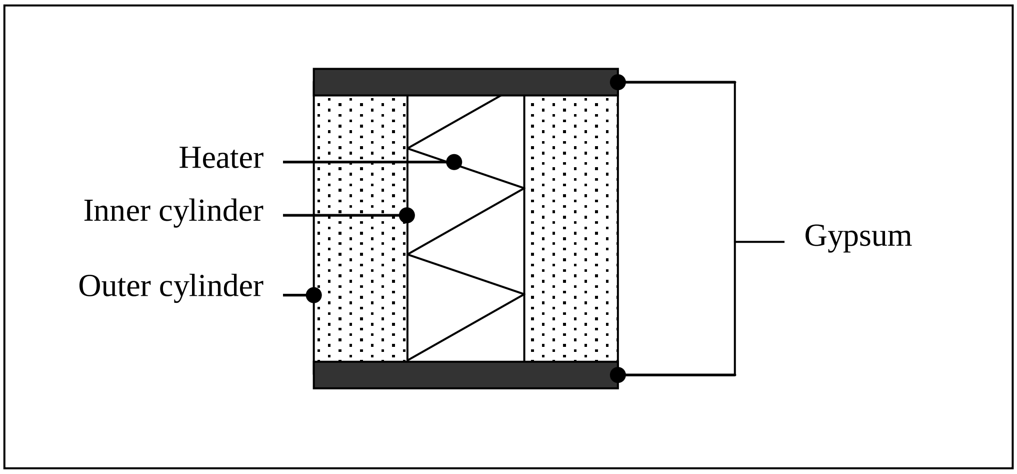


Fig .1: schematic diagram of the test rig

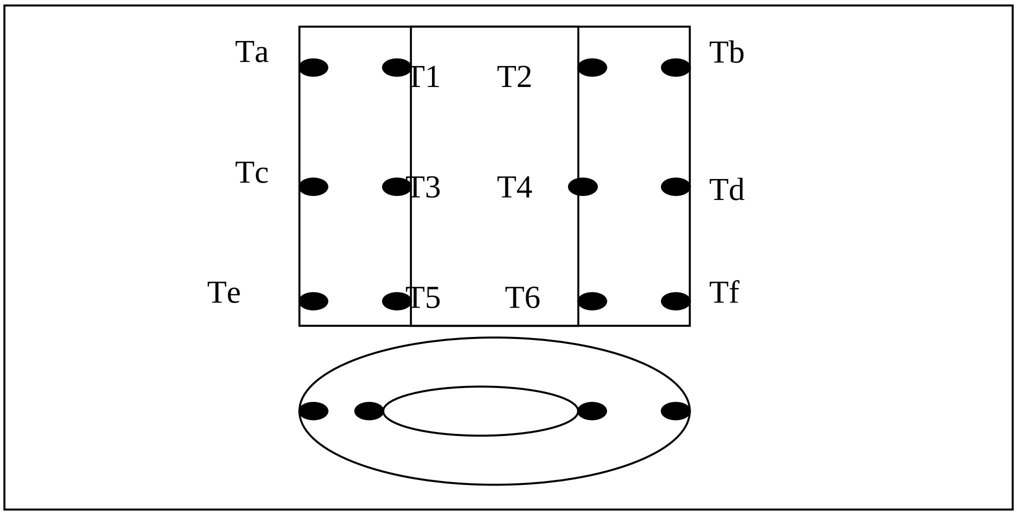


Fig.2:Thermocouples location

1. Measurements system:

The voltage regulator type (HSN 1.25KVA (0-250V)) was used to control the supply voltage to heating elements A voltmeter was used to measure voltage provided to heater(1500w) with accuracy (0.02v) , clamp meter (mode no.266 (100mA+2%)) to measure alternate current passing the circuit . A thermometer (digital multimeter MY-69 accuracy of 0.01^{o}C) was used to measure temperatures by thermocouples .

2.Experimental work:

Assembly two cylinders with heater vertically by steel table and providing with electrical current and reading temperature until steady state .

3.Experimental Calculations:

The calculation of an experimental part was making as following:

$$Q=I*V \qquad \ldots(1)$$

Q(input power) , I=current , V=voltage

$$Q_R = \frac{Ai\,\sigma(T_i^4 - T_o^4)}{[\frac{1}{e_i} + \frac{Ai}{Ao}(\frac{1}{e_o} - 1)} \qquad \ldots(2)$$

σBoltzmann constant =5.67x10^{-8}), e (emissivity), A (area), T (temperature)

$$Qc=Q-Q_R \qquad \ldots(3)$$

$$h = \frac{Qc}{\pi DL(Ti - To)} \qquad \ldots(4)$$

h(convection coefficient) , D(diameter) ,L(length)

$$Nu = \frac{h*D}{k} \qquad \ldots(5)$$

$$Ti= (T1+T2+T3+T4+T5+T6)/6 \qquad \ldots(6)$$

$$To= (Ta+Tb+Tc+Td+Te+Tf)/6 \qquad \ldots(7)$$

$$Tm= (Ti+To)/2 \qquad \ldots(8)$$

$$\beta = \frac{1}{Tm} \qquad \ldots(9)$$

$$\Pr = \frac{\nu}{\alpha} \qquad \ldots(10)$$

ν(kinematic viscosity) α(thermal diffusivity), Pr (prandtl number)

$$Ra^* = \frac{Kg\beta(Ti - To)H}{\nu\alpha} \qquad \ldots(11)$$

K (permeability), g(acceleration) , β (thermal expanded) , H(height)

Theoretical Part

1. Mathematical model:

The governing equations for natural convection heat transfer porous medium problems are the continuity, momentum and energy equations. Consider steady, Newtonian, and incompressible. Heat transfer phenomena are described as follows:

i-Continuity Equation:(Bejan,1984)

$$\frac{\partial u}{\partial x}+\frac{\partial v}{\partial y}=0 \qquad ...(12)$$

ii-Momentum equations:[A.Bejan(1984)]

$$\frac{\partial u}{\partial t}+u\frac{\partial u}{\partial x}+v\frac{\partial u}{\partial y}=\upsilon(\frac{\partial^2 u}{\partial x^2}+\frac{\partial^2 u}{\partial y^2})-\frac{1}{\rho}\frac{\partial p}{\partial x} \qquad \ldots(13)$$

ρ(density) , u(velocity with x-direction) , v(velocity with y-direction)

$$\frac{\partial v}{\partial t}+u\frac{\partial v}{\partial x}+v\frac{\partial v}{\partial y}=\upsilon(\frac{\partial^2 v}{\partial x^2}+\frac{\partial^2 v}{\partial y^2})-\frac{1}{\rho}\frac{\partial p}{\partial y}-g[1-\beta(T-T_O) \qquad \ldots(14)$$

iii-Energy equation: (Bejan, 1984)

$$\frac{\partial T}{\partial t}+u\frac{\partial T}{\partial x}+v\frac{\partial T}{\partial y}=\alpha\left[\frac{\partial^2 T}{\partial x^2}+\frac{\partial^2 T}{\partial y^2}\right] \qquad \ldots(15)$$

2.Solution Using a CFD Package (FLUENT)

FLUENT is a state-of-the-art computer program for modeling fluid flow and heat transfer in complex geometries. In FLUENT, the full set of mass, momentum and energy equations are solved for two- or three-dimensions. FLUENT allow us to include heat transfer within the fluid or solid in our model. When heat is added to a fluid and the fluid density varies with temperature, flow can be induced due to the force of gravity acting on the density variations. Such flows are termed natural-convection flows and can be modeled by FLUENT. The cylinder which equipped by porous media in this paper had been modeled using a commercial computational fluid dynamics package, Fluent, and its accompanying mesh generation software, Gambit. Gambit was used for building the grid as shown in fig. (3), the type of element used was Quadrilateral element. The model uses steady solver formulation. The momentum and energy equations are solved using the first order scheme. The standard discretization is recommended for solving the natural convection . Under relaxation factors are applied in order to control the changes of variable values between successive iterations and avoiding divergence of the solutions.

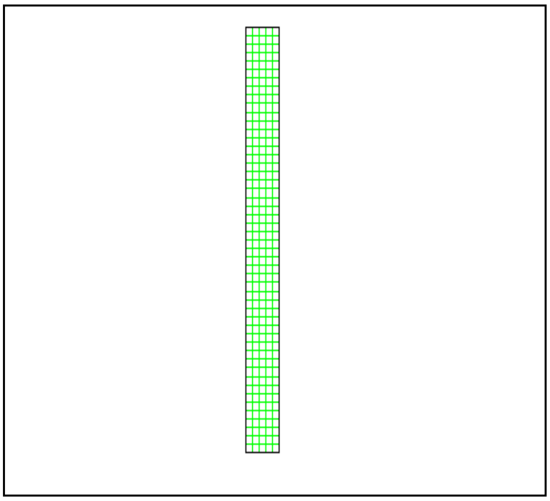


Fig.3:Grid of mesh in gambit

Results and Discussion

The temperature profiles and stream functions were plotted for two heat fluxes(250,400 w/m^2) and 3 different ratio of diameter(r/R=0.10,0.16,0.22). Fig(4-a) showed the temperature profiles for ratio r/R=0.1 , each closed curve in the plots represented a contour of temperature profiles which showed the red lines closed hot side and the blue lines closed cold side and the max. value of temperature was 165^{o}C for heat fluxes 250w/m^2 but the fig.4-b showed the eddy represented the stream function which showed the red lines around the center of gap and blue lines closed walls , the max. value was 0.84 for heat fluxes 250w/m^2. Fig.5-a showed the temperature profiles for ratio r/R=0.10 had the same behavior and the max. value of temperature was 294^{o}C and the fig.5-b showed the stream function which the max. value was 0.044 for heat fluxes (400w/m^2) .Fig(6-a) showed the temperature profiles for ratio (r/R=0.16) had the same behavior and the max. value of temperature was 105^{o}C and the fig.6-b showed the stream function which the max. value was 1.47 for heat fluxes (250w/m^2) . Fig.7-a showed the temperature profiles for ratio (r/R=0.16) had the same behavior and the max. value of temperature was 178^{o}C and the fig(7-b) showed the stream function which the max. value was (0.735) for heat fluxes (400w/m^2).Fig(8-a) showed the temperature profiles for ratio (r/R=0.22) had the same behavior and the max. value of temperature was (91.9^{o}C) and the fig(8-b) showed the stream function which the max. value was (2.0) for heat fluxes (250w/m^2) .Fig(9-a) showed the temperature profiles for ratio r/R=0.22 had the same behavior and the max.

value of temperature was 232°C and the fig(9-b) showed the stream function which the max. value was 4.33 for heat fluxes (400w/m^2) . From these figs.4-a,6-a,8-a and figs.5-a,7-a,9-a noted that the hot area was increased vs. cooled area when increasing the heat flux and gap space , thus figs.4-b,6-b,8-b and figs.5-b,7-b,9-b showed increasing the value of stream function when increasing the heat flux and gap space .Fig (10) showed Nu no. was changed with the ratio (r/R) for different heat fluxes so different Ra no. and the equation between them:

$$Nu=1.7558e^{4.6635(r/R)} \qquad \ldots(16)$$

Fig (11) contained the relation between Nu and Ra for 3 diameter ratio (r/R) showed the increasing of Nu with increasing of Ra and the relation was found between them by the equation:

$$Nu = 1.6264e^{0.0027Ra} \qquad \ldots(17)$$

Note that the increasing of convection heat transfer came from the increasing of heat flux which acting by Ra number.Fig (12) showed the relation between experimental Nu and theoretical Nu which showed a good agreement between them. Note that the behavior of heat transfer experimentally was the same of the behavior of heat transfer numerically.

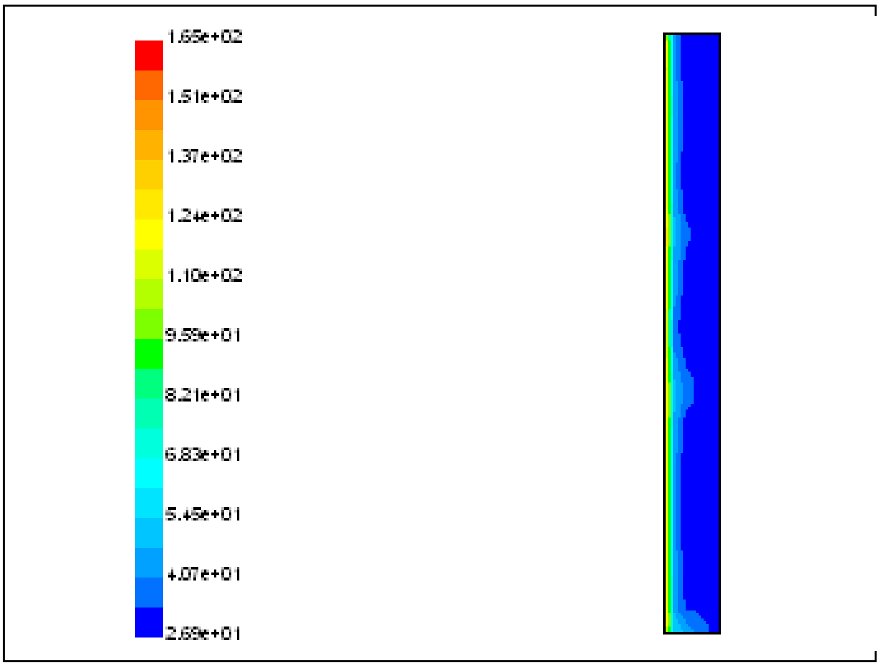


Fig.4-a:Contour of Temperature Profiles for (r/R=0.1) with heat flux (250w/m^2)

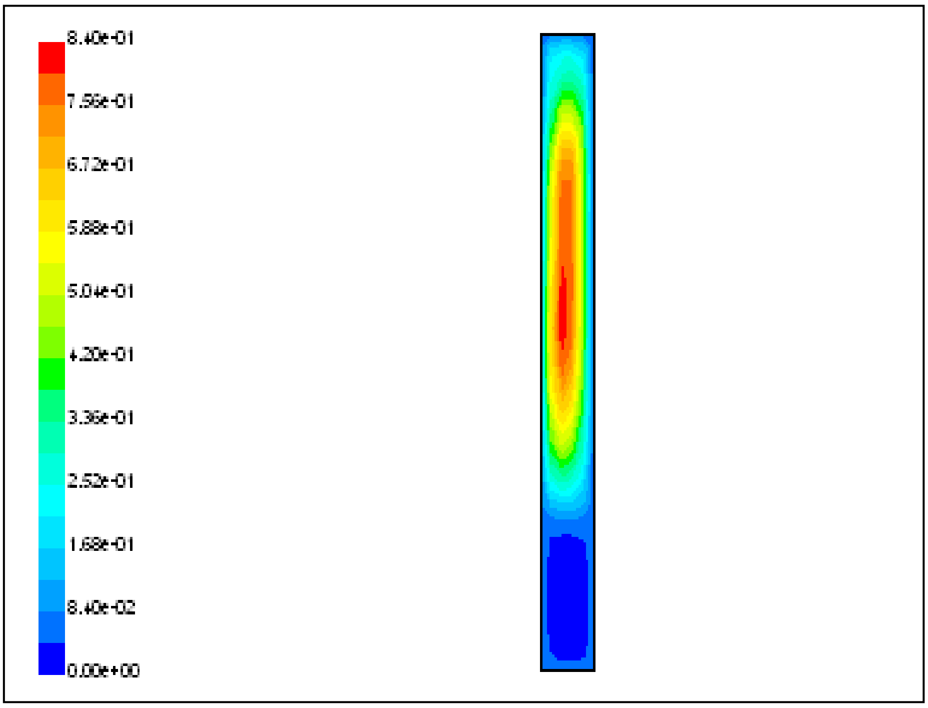
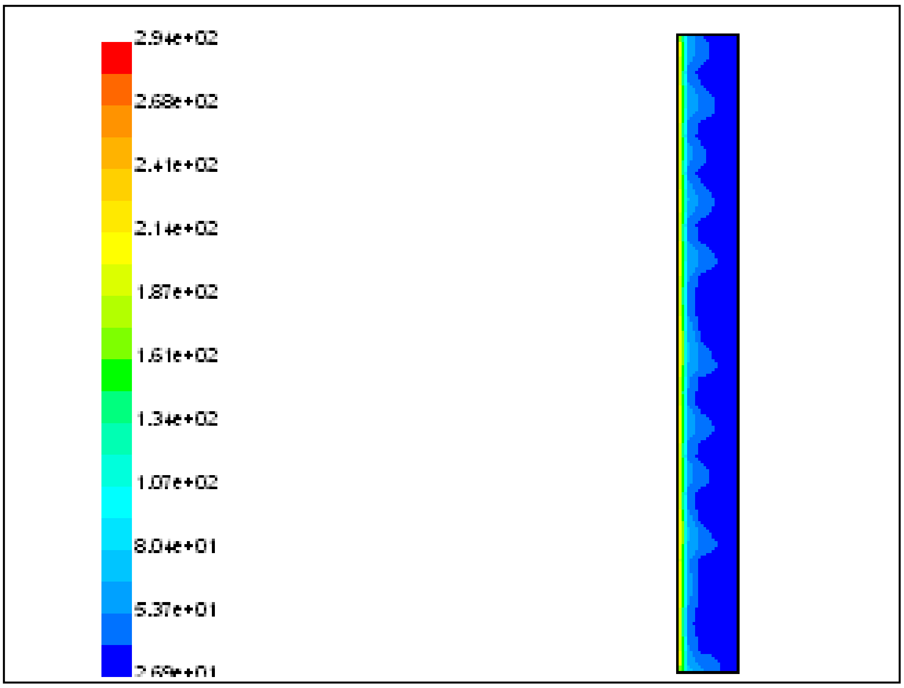
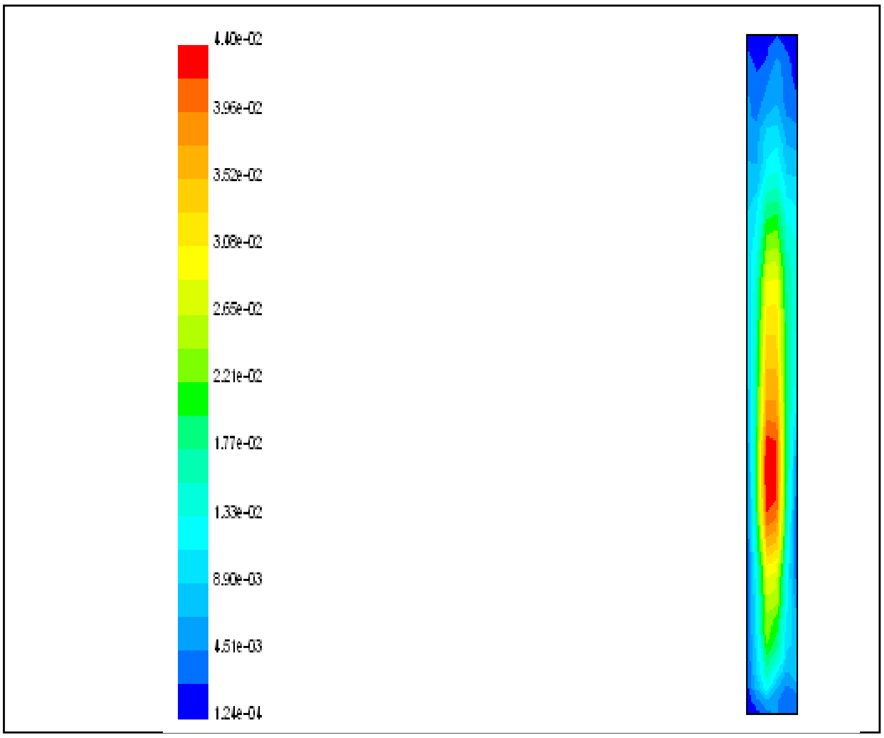


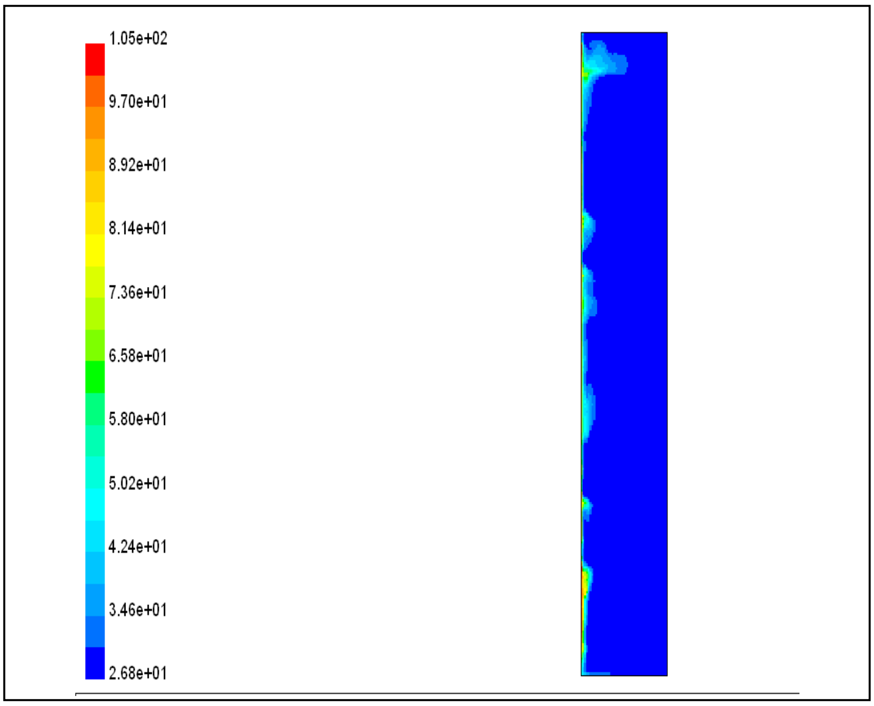
Fig.4-b:Contour of stream function for (r/R=0.1) with heat flux (250w/m^2)



Fig(5-a):Contour of Temperature Profiles for(r/R=0.1)with heat flux (400w/m^2)



Fig(5-b)Contour of stream function for (r/R=0.1) with heat flux (400w/m^2)



Fig(6-a):Contour of Temperature Profiles for (r/R=0.16) with heat flux (250w/m^2)

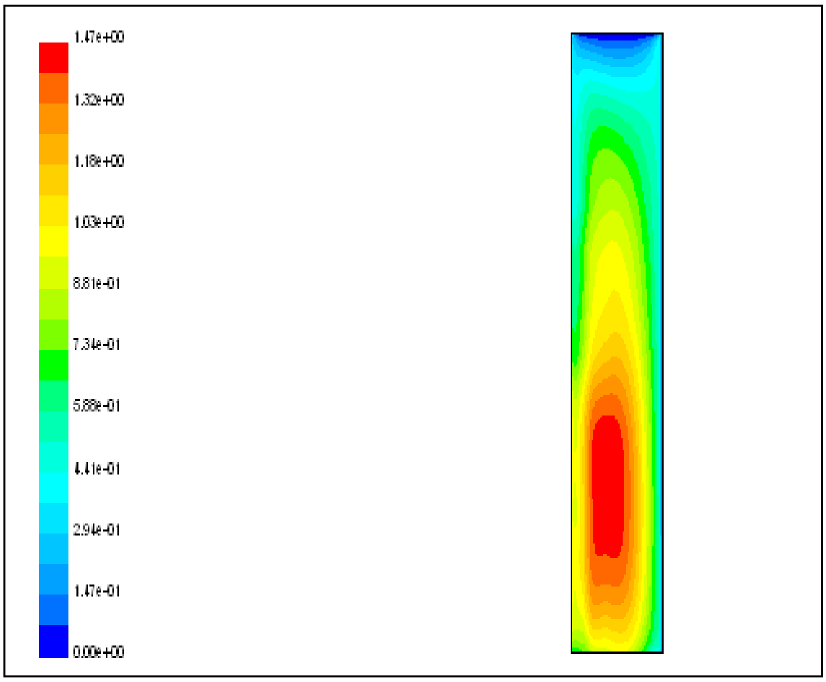


Fig.6-b: Contour of stream function for (r/R=0.16) with heat flux ($250 w/m^2$)

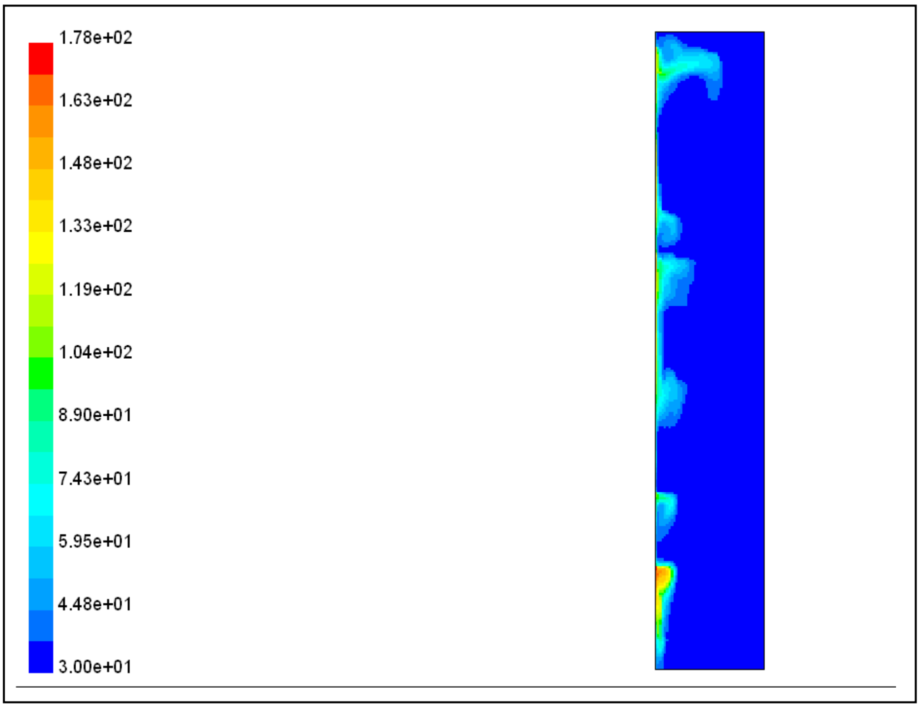


Fig.7-a: Contour of Temperature Profiles for (r/R=0.16) with heat flux ($400 w/m^2$)

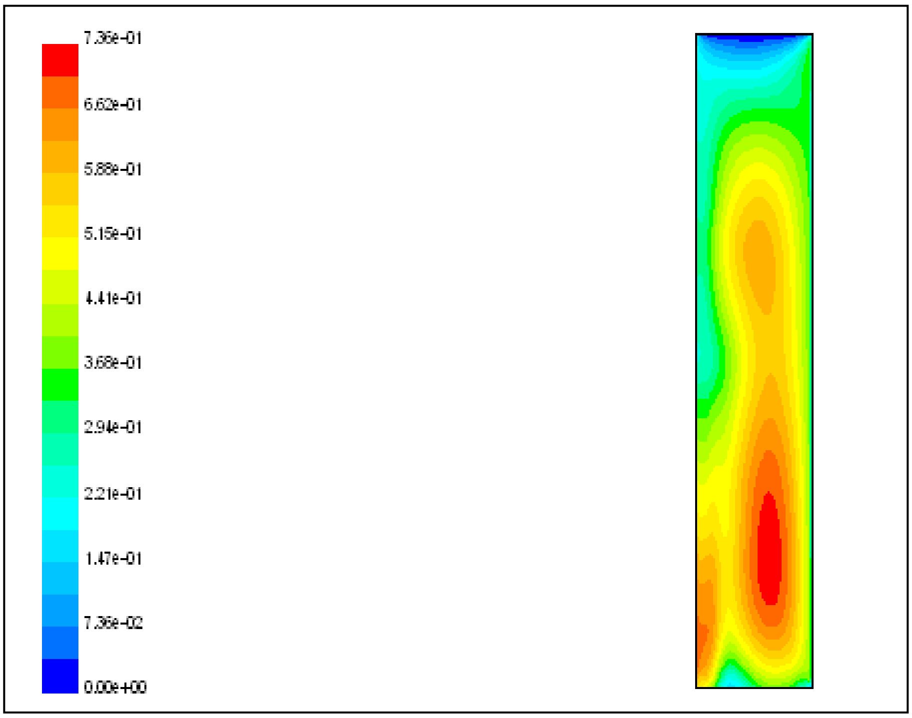


Fig.7-b:Contour of stream function for(r/R=0.16)with heat flux(400w/m^2)

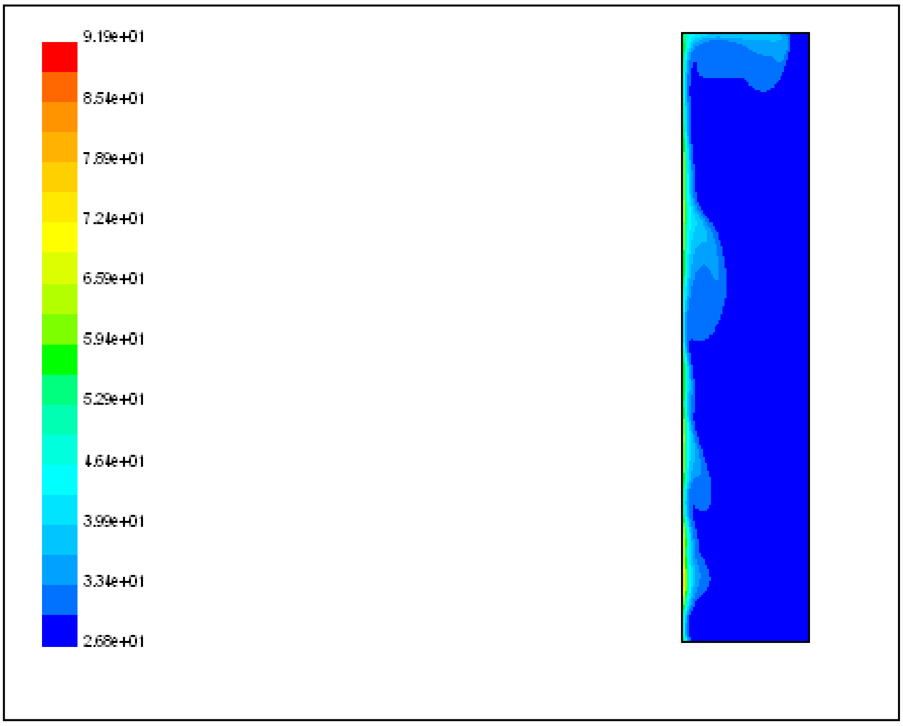


Fig.8-a:Contour of Temperature Profiles for(r/R=0.22)with heat flux (250w/m^2)

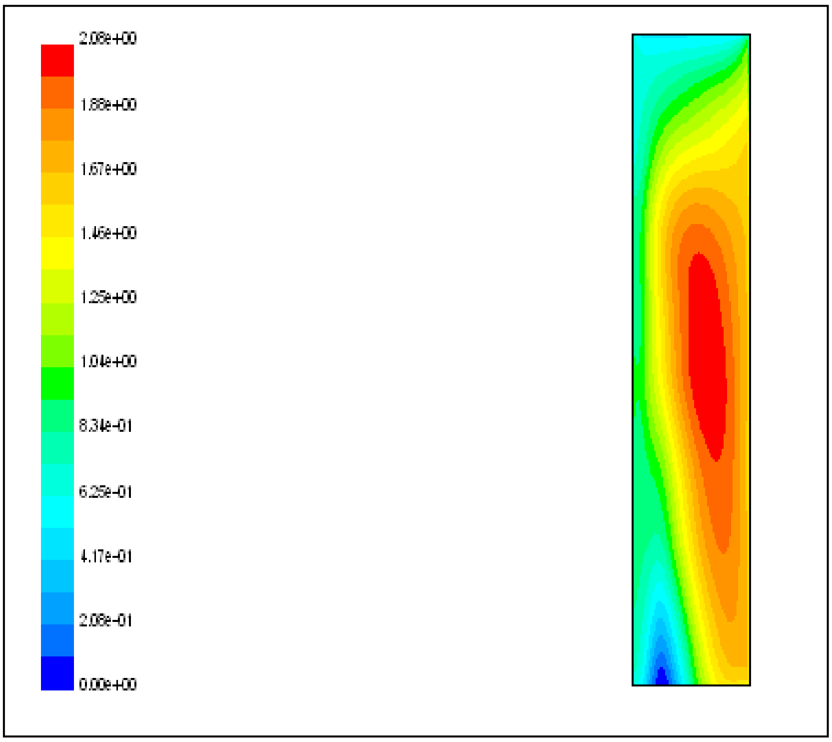


Fig.8-b:Contour of stream function for (r/R=0.22) with heat flux (250w/m^2)

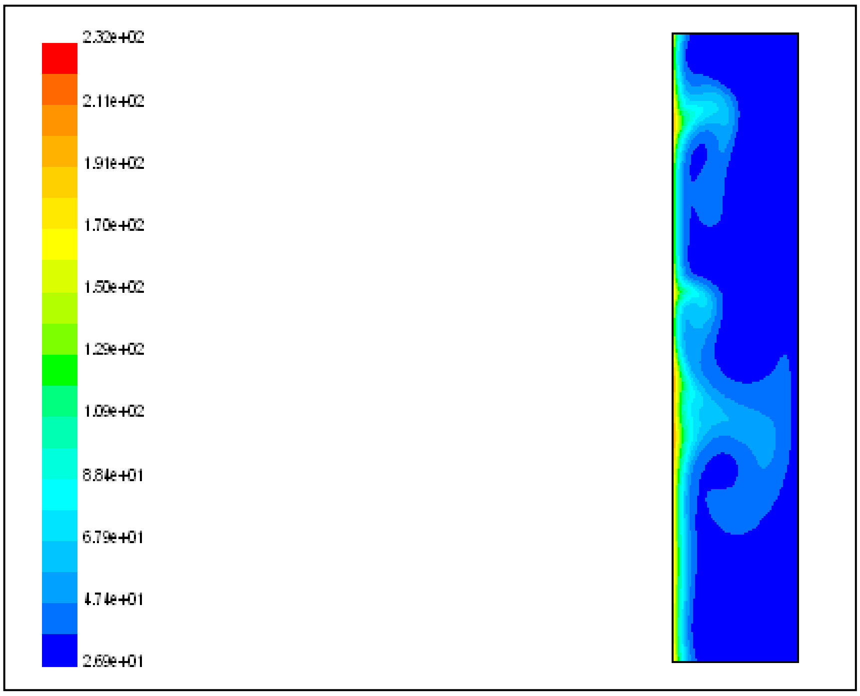


Fig.9-a:Contour of Temperature Profiles for (r/R=0.22) with heat flux (400w/m^2)

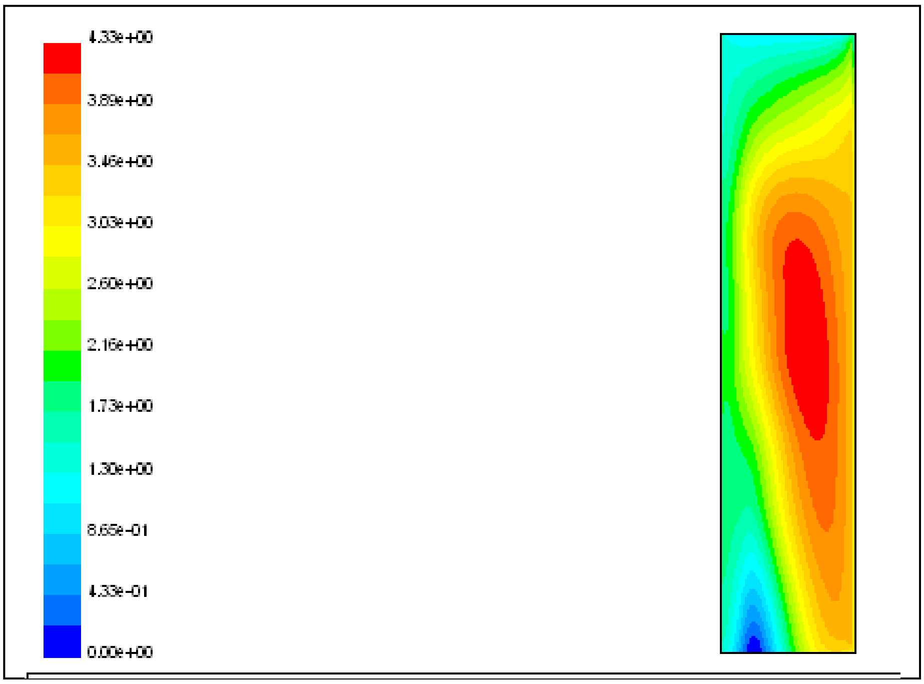


Fig.9-b:Contour of stream function for (r/R=0.22) with heat flux (400w/m^2)

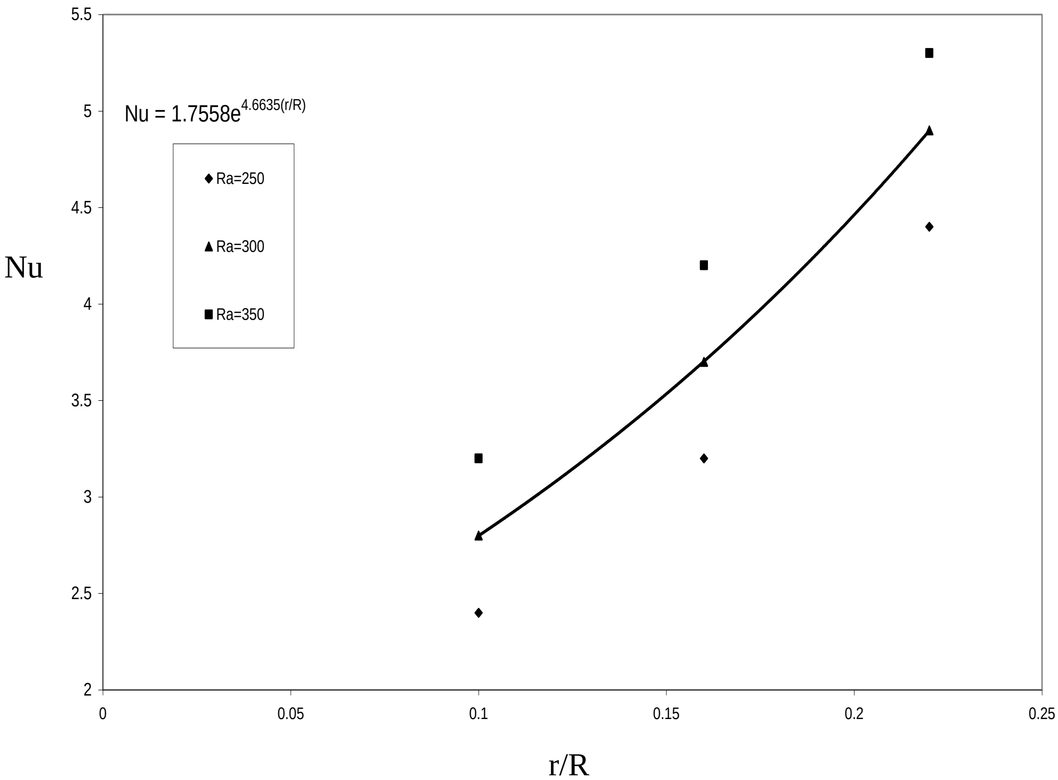


Fig.10: Nu with ratio (r/R) for different Ra

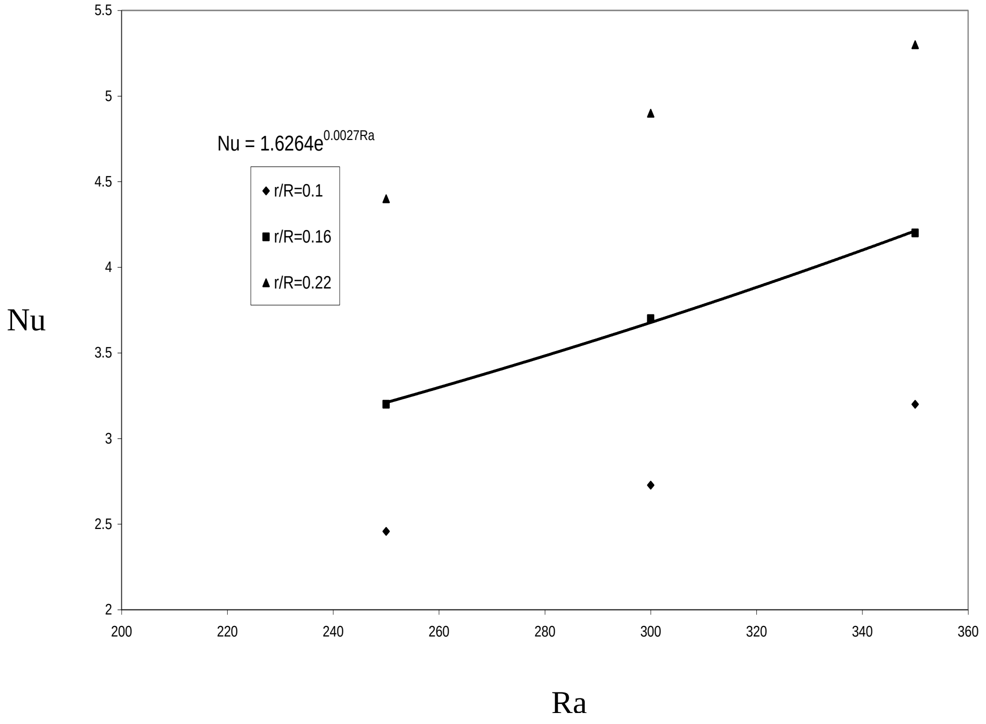


Fig.11: Nu with Ra for different ratio (r/R)

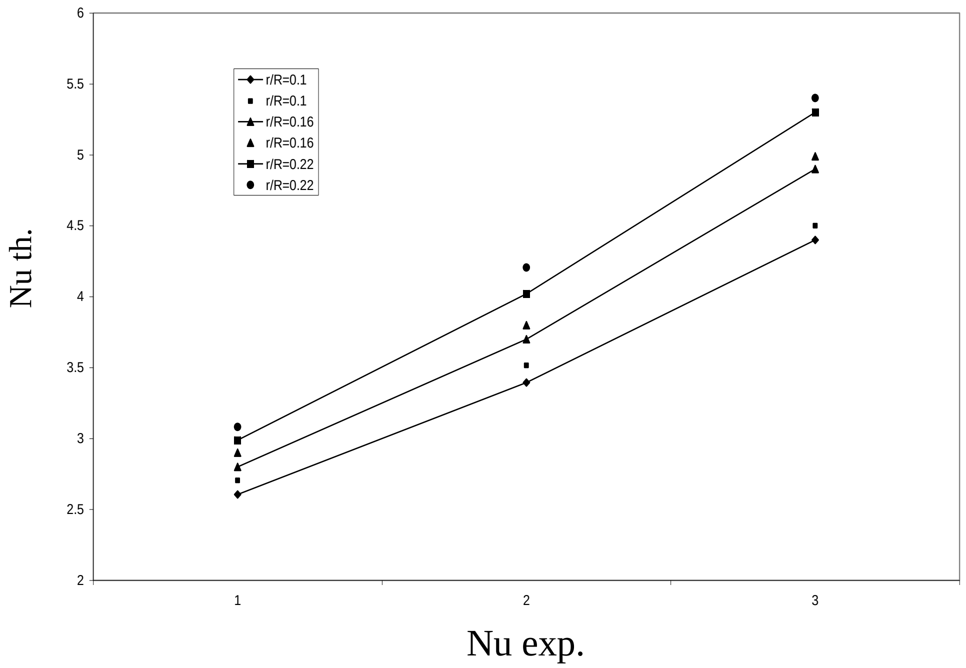


Fig.12: The relation between theoretical Nu and experimental Nu

Conclusions

1.The behavior of temperature profiles were the same for any ratio of diameter and different heat fluxes.

2.The behavior of stream functions were the same for any ratio of diameter and different heat fluxes.

3.The relation between Nu and the ratio (r/R) was :

$$Nu = 1.7558e^{4.6635(r/R)}$$

4.The relation between Nu and Ra was:

$$Nu = 1.6264e^{0.0027Ra}$$

5.There was a good agreement between experimental and theoretical studies, which showed the experimental part proved that the numerical part was true with a few errors. And the experimental was correlated with numerical studies.

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Table .1: lists of Symbols and units

Symbol	meaning	unit
A	Area	m^2
E	emesivity	
H	height	m
I	current	A
K	permeability	m^2
K	Thermal conductivity	W / m.^{o}C
Nu	Nusselt number	-
Q	Thermal Power	W
R	Outer cylinder radius	m
R	Inter cylinder radius	m
Ra	Raleigh number	-
Ti	Temperature of internal cylinder	^{o}C
To	Temperature of external cylinder	^{o}C
U	Velocity in x direction	m/s
V	voltage	V
V	Velocity in y direction	m/s
α	Thermal diffusivity	m^2/s
β	Thermal expansion	1/^{o}C
ν	Kinematics viscosity	m^2/s

الحمل الحر خلال فجوة مملوءة بوسط مسامي بين اسطوانتين عموديتين متحدتي المركز

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الخلاصة

أجريت دراسة عملية ونظرية للحمل خلال فجوة مملوءة بوسط مسامي بين اسطوانتين عموديتين متحـــدتي المركز. تضمن الجزء العملي تصنيع مقطع الاختبار وتثبيت المزدوجات الحرارية ثم تشغيل المنظومة وقـــراءة درجات الحرارة حتى الوصول إلى حالة الاستقرار. أما الجزء النظري فقد تضمن حـــل المعـــادلات الحاكمـــة باستخدام برنامج Fluent والذي تمكن من رسم التوزيع الحراري ودالة التدويم لنسب (r/R) مختلفة وعدد Ra . النتائج تضمنت ربط علاقة بين عدد Nu وعددRa وكذلك Nu وr/R كما تضمنت العلاقة بين عدد نسلت العملي وعدد نسلت النظري وكان هناك توافق جيد بين الدراستين .