Force Convection Heat transfer from a Different Cross Section Cylinder Embedded in Porous Media

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Abstract

This research presents an experimental study of forced convection heat transfer for laminar steady flows in a duct filled with saturated porous media glass balls.The heater model consists of a circular cylinder , square cylinder and triangular cylinder .The experimental work was studied the effect of changing heater section on forced convection heat transfer with selected values of heat supply (2455W/m^2).The experiments were carried out for Reynolds number (1094 $\leq Re \leq$ 1510).The experimental results showed that the surface temperature was highest for circular section type than square section then triangular section.Also, the average Nusselt number increased with increased Reynolds number and the large value for Nusselt number was for circular cylinder than square cylinder then triangular cylinder . The experimental results showed that the best improvement in forced convection heat transfer were when using porous media for the circular cylinder (63%), the Square cylinder (71%) and the triangular cylinder (74%). Also, in the present work, empirical correlations were obtained and comparison was made between the present experimental with the available previous studies a good agreement was obtained.

Keywords: Forced convection, porous media, Heater model

Symbol	*Title*	*Units*
A	Cross section area of the test section	m^2
A_s	Surface area of the heater model	m^2
c_p	Specific heat of air	$J/kg.K$
D_h	Hydraulic diameter	m
D	Diameter of the circle cylinder	m
a	The side length of the Triangle cylinder	m
c	The side length of square cylinder	m
h	The average heat transfer coefficient	$W/m^2.°C$
h_Θ	The local heat transfer coefficient	$W/m^2.°C$
I	Electrical Current	A
ε	Porosity	
k_f	Thermal conductivity of the air	$W/m.K$
k_m	Effective thermal conductivity of the porous media	$W/m.K$
k_s	Thermal conductivity of the spheres (beads)	$W/m.K$
$\dot{m}_{air}$	Mass flow rate of air	kg/s
Q_t	Electrical power	W
q	Heat flux	W
Nu_Θ	The local Nusselt number	-
Nu	The average of Nusselt number	-
Pr	Prandtl number	-
Re	Reynolds number	-
T_{in}	Temperature inlet airflow of test section	$°C$
T_{out}	Temperature outlet airflow of test section	$°C$
T_w	surface temperature of the heater model	$°C$
$T_{w\Theta}$	The local surface temperature of heater model	$°C$
Cy	Circle cylinder	
Sq	Square cylinder	
Tr	Triangle cylinder	
U	Average Velocity in the test section	m/s
V	Electrical voltage	V

1. Introduction

Many industrial processes involve an interaction between fluid and solid. In order to obtain a large ratio of surface area to volume, the fluid may be passed over a packed bed of the solid material. Examples of industrial application involving packed bed are chemical catalytic reactors, direct contact heat exchangers, petroleum reservoirs, geothermal energy, solar energy thermal storage and burying of drums containing heat generating chemicals in the earth **[1].** The heater model was subjected to different air flow rates and constant heat flux. The surface temperature of the heater model was raised to a higher temperature than the surrounding medium and maintained at that temperature thereafter, heat generation from the heater model uniformly diffuse in the radial direction.

Aydin, 2006[2] study the heat transfer in a square duct filled with /without porous media is numerically investigated. To change the heat transfer in the duct a rotating circle cylinder is placed at the center of the duct. The ratio of cylinder diameter to duct height is 0.8; depending on the angular velocity of the circle cylinder the heat convection inside the duct becomes mixed (natural and forced). The Grashof number, Gr, is 10^6, while the parameter defining the heat convection in the duct, $0.0625 \leq Gr / Re^2 \leq 10^2$. The Darcy number in the duct is 10^{-2}, 10^{-3}, and 10^{-4}.

Gazy F. Al-Sumaily, et at., 2012 [3] in this paper forced convection heat transfer from circle cylinder horizontal embedded in porous media of spherical particles under local thermal non-equilibrium condition, solid and fluid thermal conductivity ratio k_r(0. 01 to 10) and Biot number Bi (0.01 to100), the Reynolds number range Re_D(1to 250). The results of the porous media suppress significantly the wakes behind the cylinder and enhance considerably the heat transfer and Increase in $Nu_f$with$k_r$for kr> 10. Also, the increase in Bi decreases Nu_f.

M. A. Eleiwi, 2012 [4] Theoretical and experimental study of heat transfer by forced convection from the cylinder in cross flow embedded in a saturated porous media was carried out. The two equations are solved by finite difference method with constant cylinder surface temperature. The experimental part of this work included the construction of an experimental model composed of cupper cylinder with a (13 mm) in diameter heated internally by an electrical source. The glass spheres with diameter 12 mm. Both the theoretical and the experimental results, the average Nusselt number increase with increase Peclet number, $1.5 < Nu < 14$, $1 < Pe < 10$, The local Nusselt number increase with increase Peclet number $40 \leq Nu_\Theta \leq 170$, $1 \leq Pe \leq 10$ and decrease from highest value at $\theta = 180^o$ and reaches a minimum at $\theta = 0^o$

Aya adnan, 2013 [5]Theoretical and experimental investigations of forced convection heat transfer from a heated flat plate embedded in porous media with a constant heat flux . The experiments carried out to study the effect of Reynolds number ($24118 \leq Re \leq 82208$) on the heat flux ($1000 \leq$ heat flux ≤ 5000 W / m²) and Reynolds number on the local Nusselt number, using fluent program. The Theoretical and experimental results revealed that the local wall temperature increases with the flow, decreases with the Reynolds number and increases with heat flux, but the fluid temperature progressively decreases in the porous media with the vertical direction away from the heated wall, and the results show an increase in local Nusselt number when Reynolds number and heat flux increase.

Mirlatifi. A. M, Ghazal M. 2014 [6] Performed an experimental work to examine the effect of porous media on the heat transfer from a circle cylinder heater of steady-state forced convection. Packed bed consisted of Sphere glasses of diameter 60 mm. Mean Nusselt number ($Nu/Pe^{0.2}$ or $Nu/Pe^{0.33}$) of a circle cylinder is found both for clear air and porous media, it was clearly pointed out that porous media enhanced the heat transfer rate compared to the clear air.

Sherzad, 2000, [7] studied forced convection from cylinder in saturated porous media. An experimental part included measuring temperatures around cylinder and the velocity of the flow that changed from 3 to 50m/s. In the theoretical part, which included momentum, energy equations using Darcy flow model were solved by finite difference method with a constant surface temperature. In this work, a theoretical model of forced convection heat transfer from cylinder embedded in porous media was achieved by FLUENT program to get information concerning the nature of flow and heat transfer about a horizontal cylinder embedded in a porous media. The experimental work studied forced convection in packed bed for a wide range of heat flux and velocity include the measure of temperatures and velocity around cylinder to calculate Nu, Pe, and Re numbers found increase Nu with Pe and Re.

Sobera et al., 2003, [8] performed a systematic study of the airflow around a cylinder sheathed by a second porous cylinder and placed in a perpendicular turbulent air flow. Both numerical and experimental investigations had been carried out for the fluid flow. Heat transfer as a function of the Reynolds number, the Darcy number, the dimensionless air resistance of the porous layer, and the dimensionless distance between the outer and inner cylinder. The largest reduction of heat transfer due to the sheath layer, compared to that for an uncovered cylinder, was found for intermediate values of the Reynolds number. The flow resistance of the sheath layer, and the thickness of the air gap between solid and sheath cylinder, were also found.

Ahmed, 2010, [9] a theoretical and experimental study has carried out of heat transfer by forced convection from the cylinder in cross flow embedded in a saturated porous media. The Theoretical part of the work included solving the standard energy transport equation in porous media regions by FLUENT. There were maximum temperature and minimum velocity at the rear of cylinder Θ=180 but minimum temperature and maximum velocity at the front of cylinder ϴ=0. The stagnant area at ϴ=0 and the separation at ϴ=90 and ϴ=270. The experimental part of this work included the construction of an experimental model composed of cast iron cylinder with a (18 mm) inner diameter and (20mm) outer diameter with a length of(200mm) heated internally by an electrical heater. The cylinder was embedded in a packing of glass ball with diameter (12mm) placed inacross flow wind tunnel. The experimental results revealed that the average heat transfer increased when the Peclets number and Reynolds number increased for steady state condition. The relationship between Nu&Re and Nu&Pe for experimental gives us Nu=alnRe-b and Nu=2Pe-2.6 respectively when a and b are constants depending on Re and Q for 20 $<Q<120$ (Q in watt) and 2000 <Re <3000, 10.087 $<a<27.61$, 66.33 $<b<183.6$.

2. Scope of the Present Work

The literature survey, revealed a number of studies in the field of forced convection heat transfer with plate, and cylinder embedded in porous media. It shows a little number of studies in this field that used in several engineering applications as solar cells, refrigeration systems, thermal storage and heating system...etc. This research, studies the characteristics of forced convection heat transfer with heater circular cylinder, triangular cylinder and square cylinder with and without porous media and making a compare between them.

3. Experimental apparatus and procedure

The experimental set-up is shown photographically in figure (1) and schematically in figure (2) which consists basically of the following elements:-

3-1. Blower

The air forced to flow through duct by centrifugal blower. The maximum speed of the electrical motor is 2900 RPM (3ph, 2.5A, Y), the speed is controlled using voltage regulator (VARIAC, 3ph) to adjust an air flow as required from 0.6 m/s to 3.5 m/s.

3-2. Air Duct

The transfer connected by 20x20 cm^2 iron plate, thickens 2mm and Entrance length 320 cm, this length is to avoid high turbulence level and the flow separation phenomena that can takeplace by connecting the pipe and the duct, a gradual diffusion of flow wasdesigned.

3-3. Test section

The air duct was followed by an iron plate test section; it has the same cross section area of air duct, while the length is 0.4m. The three sides of it were fixed with the duct, whereas the top one was movable to help in filling of the porous elements and change heater model. A mesh wire of (1.5x1.5) mm2 was used to close the ends of the test section and to fix the beads in position; a mesh wire is fixed on the inside walls of the duct by adhesives. The experimental rig was constructed by the researcher in the Lab. at the Mechanical Engineering Department, University of Technology.

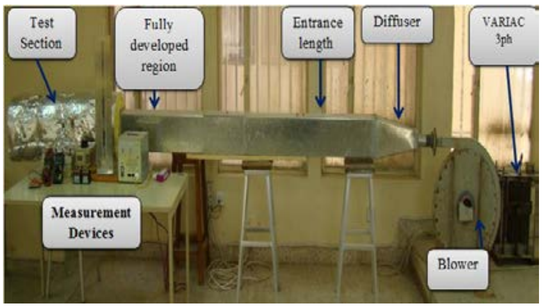


Figure 1: The test rig

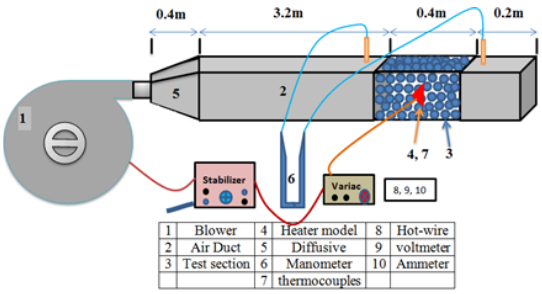


Figure 2: The schematic of the air duct and the test section

3-4. Heater model

The heater element cast iron cylinder (L = 20cm, D_{out} = 1cm) was heated internally by electrical heater , A.C. power supply and external shape made of Commercial copper, the circle cylinder (D_{out}=1.5cm, D_{in}=1cm, L=20cm), the Square cylinder (a=1.5cm,D_{in}=1cm ,L=20cm) and the Triangle (c=2.6cm,D_{in}=1cmL=20cm) as shown in fig(3a,3b). To record the temperature of heater surface (Linked heating system):-

1. Circular cylinder: 12 thermocouples were distributed around the perimeter (4 thermocouples separated 90^0 and 3 thermocouples at one face separated by five centimeter).

2. Square cylinder: 12 thermocouples.

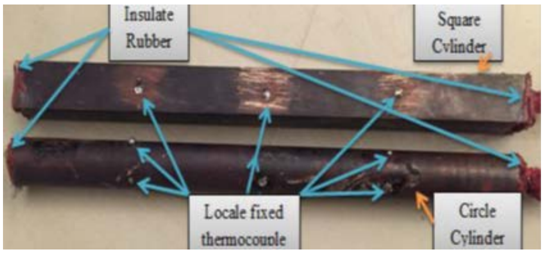
3. Triangular cylinder: 9 thermocouples were distributed three thermocouples at each face separated by five centimeter.

To measure surface heater temperature (Tw), the thermocouples were fixed on heater surface in a small hole of 1mm diameter.

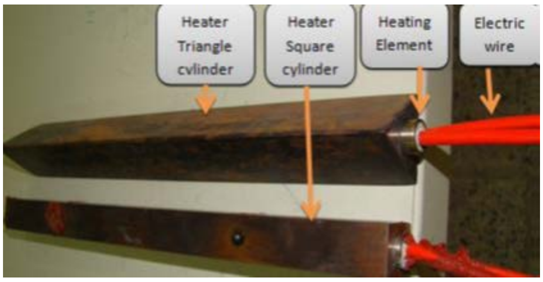
To record the temperature of inlet/outlet of the test section:-

1. Three thermocouples to measure the temperature of inlet airflow (Tin).

2. Ten thermocouples to measure the temperature of outlet air flow, as shown in figure (4).



(a)



(b)

Figure 3: Comparison of the electric heaters a) Square and Circular b) Triangular and Square cylinder

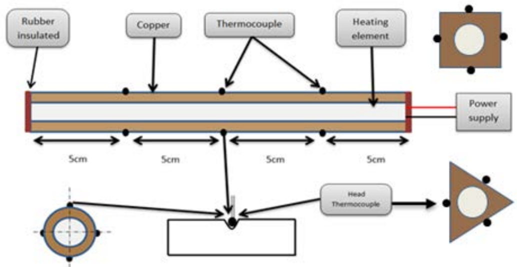


Figure 4: Distribution of thermocouples around heater model

3-5. Flow measurement

The Hot-Wire Anemometers used to measure the average velocity air at of the test section, device type K/J(0.1 m/s).The dimension of local point inside duct determine by "equal area method" [10]

3-6. Electrical devices

3-6-1. Stabilizer

A stabilizer type (DACTRON, 2000 W, A.C.V. Voltage Regulator) was connected in parallel to the power supply to ensure that incoming mains voltage (220 V) with oscillation of 50 Hz with oscillation of (± 1 %).

3-6-2. Variac

A variac type (TDGC) was connected in parallel with the stabilizer to adjust the heater input voltage as required, supplies a voltage (0 – 250)V.

3-6-3. Voltmeter

To measure a voltage supplied to the heater, Digital Multimeter type (FUKE, DT 9205), voltage ranged (0–250V), with sensitivity of (± 0.05V).

3-6-4. Ammeter

To measure the electrical current passing through the heater, Digital Multimeter type (VCTOR, VC 10C+) was connected in series with the heater, current ranged (0-50 A) with sensitivity of (± 5 x 10^{-4} A).

3-7. Experimental Procedure

In this experimental study, the porous media used in the experiments were particle beads of glass spheres with 3mm diameter, the test section was filled with porous media, and the spheres were poured randomly in the test section. At first the blower is turned on for three minutes to reach the steady state operation, and then a flow rate is adjusted to give a suitable Reynolds number ,the air flow fixed by plate design (1/1, 1/4, 3/8, 1/2, 5/8, 3/4) close inlet flow as shown in fig(5). The electrical A.C. power is supplied to the heater model and left it three hours to reach the steady state operation. After steady state has been reached , the readings of surface temperature, inlet/exit air temperature, input voltage, current, and the air flow velocity were measured and Distribution of angles see fig. (6)

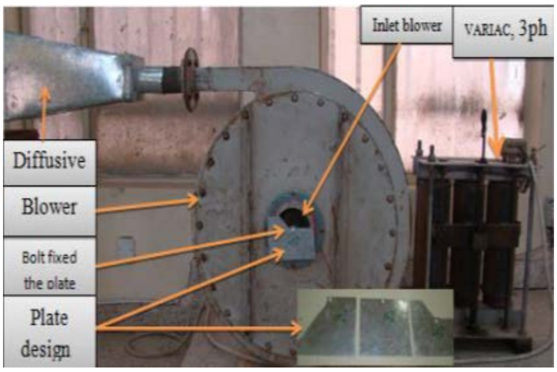


Figure 5: The blower

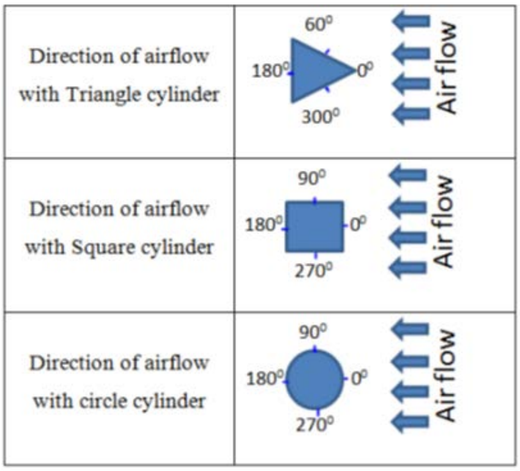


Figure 6: Angle and direction of airflow

4. Method of calculation

A. Calculation of mass flow rate

$$\dot{m} = \rho A U \quad (1)$$

B. Calculation of Heat Transfer to air

$$Q_{air} = \dot{m}\, C_p (T_{out} - T_{in}) \quad (2)$$

C. Calculation of the Surface temperature of the heater T_w

$$T_w = \frac{\Sigma T_{wn}}{n} \quad (3)$$

D. Calculation of temperature mean of air T_{mean}

$$T_{mean} = \frac{(T_{in}+T_{out})}{2} \quad (4)$$

E. Calculation of temperature the membrane over surface heater(T_{film})

$$T_{film} = \frac{(T_{mean}+T_w)}{2} \quad (5)$$

F. Calculation of total power supply to heater model

$$Q_t = V \times I \times \cos\theta \quad (6)$$

G. Heat Losses

$$Q_{loss} = Q_t - Q_{air} \quad (7)$$

H. Calculation of net heat transfer to air

$$q = Q_t - Q_{loss} \quad (8)$$

I. Calculator of average Heat Transfer Coefficient[11]

$$h = \frac{q}{A_s(T_w - T_{mean})} \quad (9)$$

J. Calculation of the hydraulic diameter of heater model (D_h)[12]

$$D_h = \frac{4A_{Circl}}{P_{Circl}} = \frac{4A_{Square}}{P_{Square}} = \frac{4A_{Triangl}}{P_{Triangl}} \quad (10)$$

K. Calculation of effective Thermal Conductivity

$$k_m = \varepsilon k_f + (1-\varepsilon) k_s \quad (11)$$

L. Calculation of Average Nusselt Number

$$Nu = \frac{h\, D_h}{k_m} \quad (12)$$

M. Calculation of Reynolds Number

$$R_e = \frac{D_h\, U\, \rho_{air}}{\mu_{air}} \quad (13)$$

N. Calculate the Surface temperature face of the heater

$$T_{w\theta} = \frac{\Sigma T_w}{3} \quad (14)$$

O. Calculator of local Heat Transfer Coefficient

$$h_\theta = \frac{q}{A_s(T_{w\theta} - T_{mean})} \quad (15)$$

P. Calculation of local Nusselt Number

$$Nu_\theta = \frac{h_\theta D_h}{k_m} \quad (16)$$

Appendix (A) shows a sample of calculation.

5. Properties of porous media

Porosity (ε):- Porosity is the spaces between parts of the solid material containing the fluid passing through or Porosity is the ratio of the pore volume (the volume of the fluid used to fill the medium), to the total volume of the porous media. The porosity can be described mathematically as:-

$$\varepsilon = \forall^\circ / \forall \quad (17)$$

Where $\forall^\circ$ is the difference between the total volume ($\forall$) and the solid matter volume ($\forall p$).

$$\forall^\circ = \forall - \forall p \quad (18)$$

First weighting the sample then finding its volume by dividing mass of sample (mp) to average density (ρp).

$$\forall p = mp / \rho p \quad (19)$$

The sample of glass spheres is added to the graduated cylinder and the volume ($\forall$) of the sample is found from measuring the graduated cylinder, This measurement is repeated several times. The average porosity of the glass spheres (3 mm) is found experimentally to be (0.37)

The thermal conductivity of the glass sphere (w/m. k°) used in the present work was taken from Table (1).

Table (1) Properties of various type of glass [13]

Material	Temp. K	ρ Kg /m³	Cp J/Kg.K	K W/m. K
Glass window	293	2800	800	0.81
Glass window	293	2700	780	0.78
Soda lime	300	2500	750	1.4
glass bead	300	2507	670	0.78

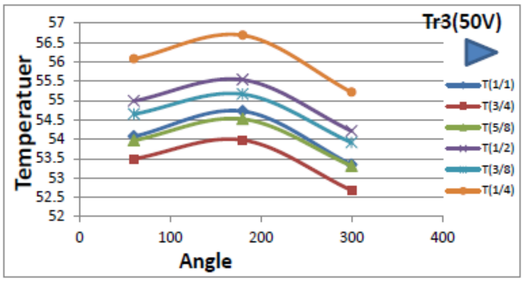
6. Discussion of the results

6-1. Effect of airflow

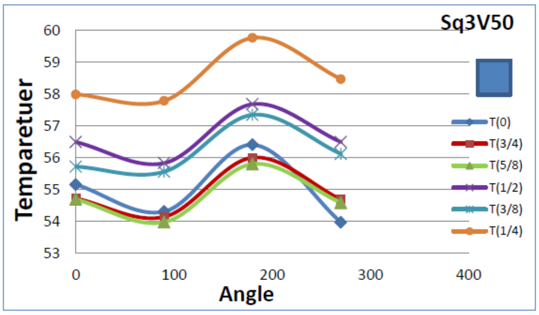
Figure (7) shows the local wall temperature for square and circular cylinder. Figure (7a, b) shows the temperature gradually decreases (0^0-90^0) then gradually increase (90^0-180^0) then gradually decreases (180^0-270^0) with angle for triangular cylinder, figure (7c), the local wall temperature gradually increases then gradually decrease with the increase in the angle. The temperature increases with decreases in Reynolds number. This is because the mass flow rate decreases (Q=m.Cp.ΔT).

The experimental results of the steady airflow heat transfer over the heater model is obtained for constant Prandtl number (0.7), with (50V) at different values of Nusselt number.

In figure (8) effect of airflow change on the Local Nusselt number at the perimeter, constant heater model with different value of the Reynolds number from 1093.877 to1412.466. The Local Nusselt number increase with increases Reynolds number



(a)



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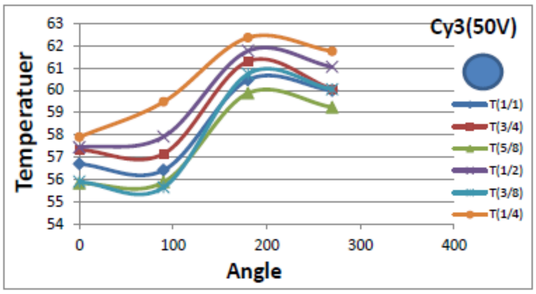


(c)

Figure 7: Temperature Profile (^{o}C) with Angle for different Reynolds number (1094-1510) a) Triangular, b) Square, c) Circular

6-1-1. Equilateral triangle cylinder

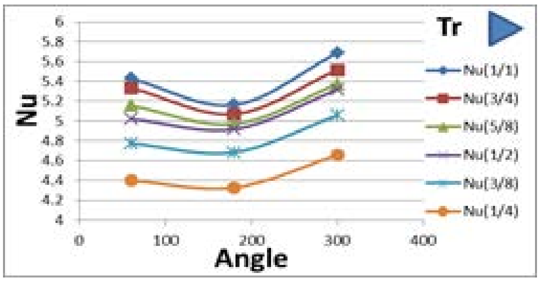
Figure (8,a) shows that, the local Nusselt number gradually decreases then gradually increase with the increase in the angle, the due to upper and lower surface (face) with full cover airflow (at 60^0, 300^0) and small thickness of thermal boundary layer .

6-1-2. Square cylinder

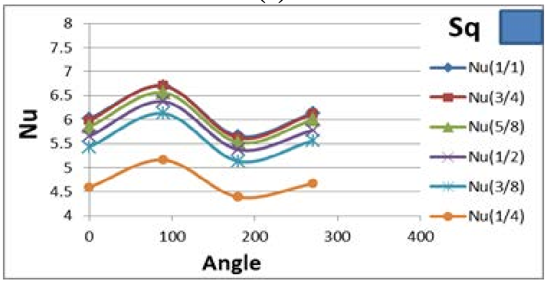
Square cylinder is shown in figure (8,b) with porous media. It shows that, the local Nusselt number increases with angle until point two (900) then decreases with angle to point three (1800) then increases with angle of perimeter to point four (angle 270). This is because that the surfaces are fully covered with airflow. The high mass of airflow, high heat transfer according to equation (Q=m.Cp.ΔT), then increased local Nusselt number.

6-1-3. Circle cylinder

Circle cylinder is shown in Figure (8,b). It shows that, the local Nusselt number gradually increases with the increase in the angle until point two (90^0) then decreases with the increase in the angle (around perimeter) to point three (180^0) then increases with the increase in the angle to point four (2700). This is because, the top and bottom (point two and four) surfaces are fully covered with airflow. The high mass airflow (angle equal 90^0, 270^0), increases the local heat transfer coefficient, hence, increases local Nusselt number according to (h=Q /A.ΔT, Nu=hd/k).



(a)



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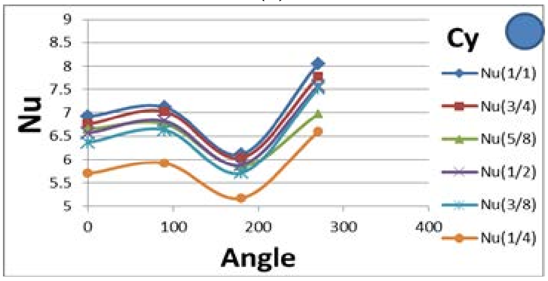

(c)

Figure 8: local Nusselt number with Angle for different Reynolds number change (1094-1510) a) Triangular, b) Square, c) Circular

The direct reading of the local Nusselt number with angle direction is plotted in Figure (9), when comparing the heater cylindrical, heater triangle and heater square, we find that the Local Nusselt number distribution around the triangle cylinder less than the square cylinder and the circle cylinder. The circle cylinder has high local Nusselt number than Square cylinder This is because the geometry of circular cylinder is in contact with stream flow then increases heat transfer, hence, increases local Nusselt number. The triangular cylinder and square cylinder have smaller areas of contact with flow stream.

The variation of average Nu–Re for different cross section cylinders shown in fig. (10), We find that the average Nu for the circular cylinder is larger than the square cylinder and the triangular cylinder, This is because the thickness of thermal boundary layer decreases at circular cylinder.

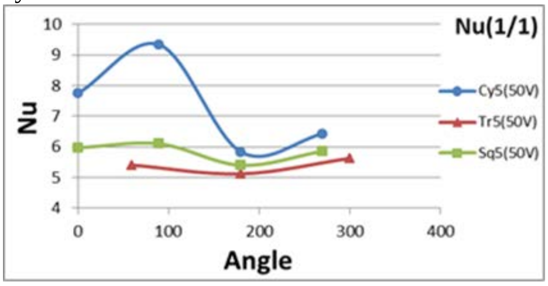

Figure 9: local Nusselt number with Angle for different heater model

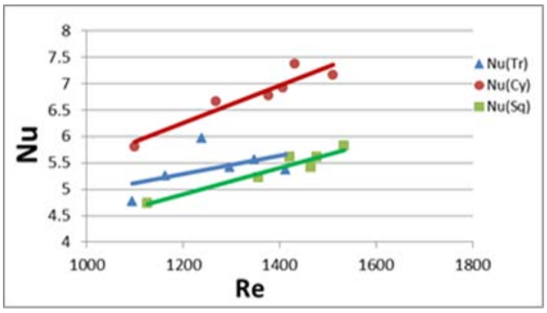

Figure 10: Average Nusselt number with Reynolds number for different heater model

The relationship between average Nusselt number with Reynolds number shown in table (2).

Table (2): correlations of Nu with heater type

Type of heater model	Equation	Re
Triangle	Nu = 2.23ln(Re) - 10.51	1094<Re<1413
Square	Nu = 3.27ln(Re) -18.332	1125<Re<1534
cylinder	Nu = 4.57ln(Re) - 26.146	1093<Re<1510

6-2. Enhancement in heat transfer

Table (3) illustrates effectiveness of porous media in the improved forced convection heat transfer of the circular cylinder, square cylinder and triangular cylinder in the perpendicular horizontal airflow. The average heat transfer coefficient of heater model embedded in porous media is large from free flow for the same air flow rate (mass flow). From circular cylinder the mass flow rate was (0.104kg/s). Also the average heat transfer coefficient was 198.31. While, without porous media, the mass flow rate was (0.109kg/s). The average heat transfer coefficient was (72w/m2.c). The improvement in heat transfer for circular cylinder in porous media, that the percentage of enhancement is 63.7% [E= (hporous-hempty)/hporous], the enhancement for square cylinder is 71.99%, and triangular cylinder is 74%, see table (3).

Table (3): Enhancement in heat transfer

Heater types	Material in test section	Mass airflow kg/s	Surface temperature (Tw) °C	Heat Transfer Coefficient W/m °C	Percentage Enhancement in heat transfer
Circular cylinder	porous	0.1049	56.79126	198.31	63.7%
	Empty	0.1093	80.35501	72.005	
Square cylinder	porous	0.1067	55.83245	174.68	71.99%
	Empty	0.1012	82.30926	48.924	
Triangular cylinder	porous	0.0926	52.23488	172.83	74.06%
	Empty	0.0906	76.12895	44.837	

6-3. Comparison with Previous Work

Figure (10) shows that, the average circular Nu is of similar behavior to that presented in ref. [9] as shown in Figure (11). The average circular cylinder Nusselt number increases with increasing value of the Reynolds number in both cases. However, different experimental results for Nu due to the present work, the heat flux ; the circle cylinder diameter and 1300<Re<2300. While in ref. [9] the diameter circle cylinder 2cm and 2200<Re<2800, as shown in table (4).

6-4. Error analysis

The quantities used to estimate the Nusselt number are subject to certain uncertainties due to errors in the measurement. These individual uncertainties are presented here. The analysis is carried out on the basis of the suggestion made by Robert J. Moffat [14]. Details of error analysis are given in Appendix (B).

Table (4): Comparison of Nusselt number with previous work

Refer.	Nu	Re	a	b
Refer[9]	Nu= alnRe-b	2000<Re <3000	10 < a < 27.6	66.3< b<183
present work	Nu=alnRe-b	1300<Re <2300	1.54< a< 2.31	5.1≤ b≤10.9

7. Conclusions

1-There is max local Nu at angle (90^0 , 270^0) and min local Nu at angle (0^0 , 180^0). The circular cylinder has higher local Nusselt number than square cylinder and triangular cylinder.

2-There is increasing Nusselt number with increases Reynolds number.

3- The average Nu for the circular cylinder is larger than the square cylinder and triangular cylinder.
4-The highest improvement in heat transfer was of porous media for the circular cylinder 63.7%, square cylinder 71 % and triangular cylinder 74 %.
5-Comparison gives a good agreement between the present and previous works

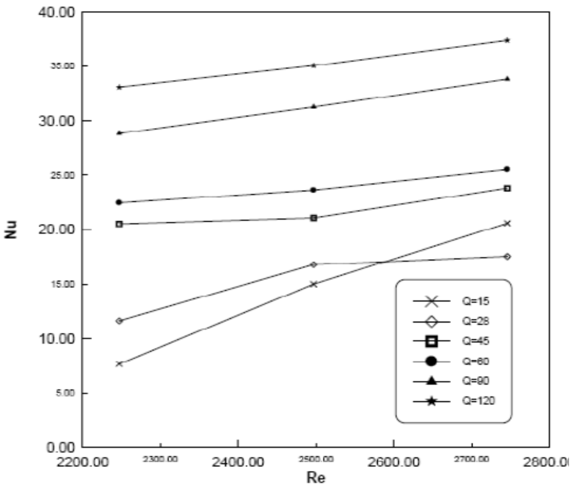

Figure 11: Comparison of Average Nu-Re relation between present work and Ref[9] for circular heater

8. Reference

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Appendix(A)

Sample of data test: (square cylinder)

<table>
<tr><td>No. of heater face</td><td colspan="3">Data,Thermocoupel on heater face °C</td><td>Locl face</td><td colspan="5">Data,Thermocoupel of airflow</td></tr>
<tr><td>Face 1</td><td>55.9</td><td>56.9</td><td>57</td><td rowspan="2">Front</td><td colspan="2" rowspan="2">Inlet air °C</td><td colspan="3" rowspan="2">Outlet air °C</td></tr>
<tr><td>No.Th.</td><td>6</td><td>9</td><td>4</td></tr>
<tr><td>Face 2</td><td>55.6</td><td>57.9</td><td>56.7</td><td rowspan="2">Top</td><td rowspan="2">41.1</td><td rowspan="2">41.1</td><td rowspan="2">41.9</td><td rowspan="2">41.3</td><td rowspan="2">40.4</td></tr>
<tr><td>No.Th.</td><td>7</td><td>17</td><td>2</td></tr>
<tr><td>Face 3</td><td>57</td><td>57.6</td><td>57.4</td><td rowspan="2">rear</td><td colspan="5" rowspan="4">Local measurement velocity nine point inside duct (m/s) [average Velosity=1.355 m/s]</td></tr>
<tr><td>No.Th.</td><td>18</td><td>11</td><td>3</td></tr>
<tr><td>Face 4</td><td>56.1</td><td>56.7</td><td>57.1</td><td rowspan="2">bottom</td></tr>
<tr><td>No.Th.</td><td>1</td><td>3</td><td>8</td></tr>
<tr><td>voltage</td><td colspan="2">60</td><td>volt</td><td rowspan="3"></td><td colspan="2">1.1</td><td colspan="2">2.1</td><td>1.8</td></tr>
<tr><td>Amp</td><td colspan="2">0.506</td><td>A</td><td colspan="2">1</td><td colspan="2">0.9</td><td>1.1</td></tr>
<tr><td>ΔP(Δh)</td><td colspan="2">33.4</td><td>cm</td><td colspan="2">1.4</td><td colspan="2">1.2</td><td>1.6</td></tr>
</table>

Sample of Calculation:

1. Calculation of inlet temperature of air:
   T in =(Tin1+Tin2+Tin3)/3= 42.12 ^{o}C
2. Calculation of outlet temperature of air:
   T out =(Tout1+Tout2+Tout3)/3= 42.49 0C
3. Calculation of mean temperature of air:
   T_{mean}=(Tin+Tout)/2= 42.31 ^{o}C
4. Calculation of properties of air flow (ρ, Cp) at mean temperature from ref. [11]:
ρ_{air}=1.1224 kg/m^3, Cp_{air} =1.0067 kJ/kg.k
5. Calculation of mass flow rate:

$\dot{m}=\rho AU$=1.1224×0.04×1.7=0.0763 kg/s

6. Calculations of Heat Transfer to air:

Q_{air}=0.0763×1006.7×(42.49−42.12)=28.42 W

7. Calculation of the surface temperature of the heater:

Tw=(T1+T2+T3+T4+...…+T10+T11+T12)/12

$T\omega$=(56.26+56.34+57.25+57.32+56.76+56.12+57.78+57.07+59.07+58.83+57.94+57.04)÷12 =57.31 ^{o}C

8. Calculation of temperature of the membrane above heater surface:

T_{film} =(Tmean+Tw)/2 = 49.81^{o}C

9. Calculation of the physical properties of air at a film temperature [11]:

k_{air} =0.02624 W / m.^{o}C,

μ=1.8462x10^{-5} kg /m.s

10.The total power supply to heater (Square cylinder) can be calculated by

$Qt=V\times I\times\cos\theta$=50×0.456×1= 22.8 W.

11. Average Heat Transfer Coefficient is calculated by:

h=22.8/0.01182(57.31−42.31) =128.595W/m^{2} ^{0}C

12. Calculation of effective thermal conductivity

$km = \varepsilon kf+(1-\varepsilon)ks$

km=0.37×0.026+(1-0.37)×0.78 =0.50102W/m. ^{o}C

13. Calculation of the average surface temperature

$T_{w\Theta1}$=(T11+T8+T12)/3 =(56.76+57.07+57.34)/3=57.05 ^{o}C

14. Calculation of average Heat Transfer Coefficient for face

$h_{\Theta1}$= Qt/($T_{w\Theta1}$−T_{mean})

$h_{\Theta1}$= (22.8/ (0.0118×(57.05-42.31)) = 131.0877W/m2. ^{o}C

15. $Nu_{\Theta1}= h_{\Theta1}D_h/k_m$ =(131.0877×0.015)/0.50102) = 3.993

Appendix(B): Error Analysis

The result R is generally calculated in some way from the data, Xi. Thus we can write:

$R= R(X1, X2, ..., Xi)$ (20)

$\delta RXi = \partial R\partial Xi\times\delta Xi$ (21)

$\delta R=[\Sigma(\partial R\partial Xi\times\delta Xi)2\ ]^{1/2}$ (22)

The uncertainty interval (S) in the result can be given as:

$SR=[(Rx1Sx1)^2+(Rx2Sx2)^2+\cdots(RxiSxi)^2]^{0.5}$ (23)

In dimensionless form:

$SRR=[(\partial R\ SX1/\partial X1\ )^2+(\partial R\ SX2/\partial X2\ )^2+\cdots(\partial R\ SXi\ /\partial Xi\ )^2]^{0.5}$ (24)

Where: $RXi=\partial R/\partial Xi$

The (Nu) is given by the equation :

$Nu=hd/k=Qd/As\Delta Twk$

$=\dot{m}Cp\Delta Tmeand/As\Delta Twkeff$ (25)

V=1.7 m/s

A= 0.04 m^2

kf =0.026 W/m.K , ks=0.78 W/m.K

$\dot{m}=V\times\rho\times A$=1.7×1.122×0.04=0.0763 kg/s

$As=4aL$=4∗0.015∗0.197=0.0118 m^2 (26)

To calculate the uncertainty Nusselt number:

From equation (25):

$\partial Nu\partial\dot{m})\Delta T_{mean},\Delta Tw=Cp\Delta T_{mean}Dh/As\Delta Twk$ =1006.7×0.378×0.015/0.0118×15×0.463 = 69.65 $(\partial Nu\partial\Delta T_{mean}),\Delta Tw=\dot{m}CpDh/As\Delta Twk$ =0.0763×1006.7×0.015/0.0118×15×0.463 =14.059

$(\partial Nu\partial\Delta Tw)\Delta T_{mean},\dot{m}=\dot{m}Cp\Delta T_{mean}D_h/A_s\Delta Tw2k$ =0.0763×1006.7×0.378×0.015/0.0118×152 ×0.463 = 0.354

Then, the uncertainty Nusselt number will be as in equation :

$SNu=[(\partial NuS\dot{m}/\partial\dot{m})^2+(\partial NuS\Delta Tmean/\partial\Delta Tmean)^2+(\partial NuS\Delta Tw/\partial\Delta Tw\ )^2]^{0.5}$

Where:

$S\dot{m}=\pm0.05333\ m/sV\times\rho\times A$ $=\pm0.05333\ m/s1.7\times1.122\times0.04$ $=\pm\ 0.001407$

$S\Delta T_{mead}=\pm0.005oC/\Delta T_{mean}=\pm0.005oC/0.378$ $=\pm0.01322$

$S\Delta Tw=\pm0.05oC/\Delta Tw=\pm0.05oC/15=\pm0.0033$ $=[(68.17*0.0014)^2+(14.05\times0.135)^2+(0.346\times0.00333)^2]^{0.5}$ =±0.21 %

Relative error = (SNu/Nu)

Relative error = (0.21/5.7) = 0.0365

Table (B-1) Instrument and Uncertainty

No.	Name of Instrument	Range of instrument	Variable measured	Least division in measuring instrument	Min. and max. values measured in experiment	Uncertainty (%)
1	Thermocouple	0-300 oC	Wall Temperature Tw	0.1	45-70	± 0.05
2	Thermocouple	0-200 oC	Temperature mean $Tmean$	0.1	40-48	± 0.005
3	Voltmeter	0-250	Voltage	0.1	50-80	± 0.05
4	Ammeter	0-5	Current	0.1	0.4-0.7	± 0.005
5	Manometer	0-90 cm	Height of H2O	1 mm	(22-30) mm	± 0.13
6	Hot-wire Anemometer	0-30 m/s	Velocity airflow	0.1 m/s	0.4-3.5	± 0.05333

أنتقال الحرارة بالحمل القسري من أسطوانات مختلقة المقطع مطمورة في وسط مسامي

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الخلاصة

يقدم البحث الحالي دراسة عملية لإنتقال الحرارة بالحمل القسري لجريان طباقي مستقر خلال مجرى مربع الشكل ابعاده (0.2×0.2) m^2وبطول (0.4m) مملوء بوسط مسامي مشبع .وان الحشوة المسامية المستخدمة تتكون من كرات زجاجية متجانسة القطر,المصدرالحراري عبارة عن اسطوانة دائرية المقطع وثاني ذات مقطع مربع والثالث ذات مقطع مثلث جميعها مصنوعة من مادة النحاس والكل منها مزود بمسخن كهربائي ,تم وضعها في منتصف الحيز ,بينماجميع الجدران معزولة حراريا .
التجارب العملية تضمنت دراسة تاثير تغيرشكل المقطع للاسطوانه المسخنه على انتقال الحرارة بالحمل القسري حيث اجريت التجارب لمدى من رقم رينولد.($1094 \leq Re \leq 1510$)حيث اظهرت النتائج ان درجة حرارة سطح الاسطوانه الاكبر قيمة كانت للمقطع الدائري ثم المربع ثم المثلث اقل قيمة وان النسبه المؤيه لاعظم فرق. (14%) ,ان معدل رقم نسلت يزداد مع زيادةعدد رينولد بثبوت الفيض الحراري وان اعلى قيمه لرقم نسلت يكون للاسطوانه ثم المربع ثم المثلث واعظم فرق 18% للمقطع المربع و اعلى فرق للمقطع المثلث .15%.
افضل تحسين لمعامل أنتقال الحرارة بالحمل القسري وجد عند استخدام المادة المسامية للمقطع الدائري 63% وللمقطع المربع 71% والمقطع المثلث 74% , وكذلك فقد تم في هذا البحث استنتاج علاقات ارتباطية لكل حاله(تجريبية) و مقارنتها مع البحوث السابقة المتوفره لدينا وكانت بنفس النمط لكل حاله.

الكلمات المفتاحية: الحمل القسري ، الوسط المسامي ، نماذج مسخنات