AN EXPERIMENTAL STUDY OF FORCED CONVECTION HEAT TRANSFER IN A VERTICAL PACKED DUCT +

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Abstract:

This paper investigate the effect of tilt angle on heat transfer through (12.5*12.5*100 cm) packed duct experimentally. The duct was at a vertical position ,heated at constant heat flux varying from (0.2 to 0.5 kW/m^2) and the tests were done for Reynolds number (8010 to 18400). The porous duct were packed by (12 piece) of commercial galvanized steel wire mesh and arranged courrgeatly in (V end) shape (zig-zag) at the center line axial with pad porosity (Ɛ=0.99).

The results showed that Nusselt number was increasing with increasing heat flux and Reynolds number. Average Nusselt number obtained in this work at vertical duct was increased up to (150%) when compared to empty duct and up to (40%) when compared to horizontal duct at the same tested condition. This results indicated that heat transfer in vertical duct is enhanced than horizontal duct.

Keywords:Vertical porous duct , heat transfer , wire mesh pad.

دراسة عملية لانتقال الحرارة بالحمل ألقسري في مجرى مسامي عمودي

د.كفاح حامد هلال

المستخلص :.

تشمل الدراسة تأثير زاوية ميل مجرى مسامي مستطيل بأبعاد (12.5*12.5*100 cm)على انتقال الحراة عملياً. زاوية ميل المجرى المسامي (90°) عن الافق (عمودي) وجداره مسخن بفيض حراري ثابت يتغير(0.2 to 0.5 kW/m^2) واجريت التجارب لمدى رقم رينولدز بين(8010 to 18400) .

تتكون الحشوة من 12 قطعة من مشبك سلكي معدني رتبت على شكل (V)عند الاطراف متعرج وتم تثبيتها من الوسط وكانت مسامية الحشوة (Ɛ=0.99) . بينت النتائج ان رقم نسلت يزداد بزيادة رقم رينولدز والفيض الحراري . كذلك بينت النتائج ان رقم نسلت للمجرى المسامي العمودي يكون اعلى بنسبة (40%) عن الوضع الافقي ،ويكون اعلى بنسبة (150 %)عن المجرى الفارغ عند نفس الظروف التشغيلية.

الكلمات الدالة : مجرى مسامي عمودي ، انتقال الحرارة ، حشوة مشبك معدني

Introduction :

The porous media have been used widely in many engineering fields such as heat storage, solid matrix heat exchangers, thermal insulation of buildings,cooling of electronic

+ Received on 24/8/2016 , Accepted on 25/7/2017 .

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equipment and chemical reactors [1]. Wire screen meshes are one type of porous media which used in forced convection heat transfer through duct. Due to high porosity of the media made of wire screen mesh, the specific surface area decreased and the resistance against flow is also decreased.

Ozdemr*etal.* [1]theoretically and experimentally investigated the hydrodynamic and heat transfer characteristics of a porous media a consisting of (20 wire screen meshes) placed under the upper portion of the test section which had dimension of (180*50*500 mm). The hydrodynamic experiments were conducted for the range of Reynolds number (1.5) to (12) based on mean velocity and wire diameter .Correction function of Nusselt number and thermal entrance length were obtained for Peclet number $10 \leq Pe \leq 50$.

Venugopal*etal.*[2] conducted experiments to use a stack of metallic perforated plates filled inside vertical duct for heat transfer enhancement at forced convection. The effect of porosity was investigated to obtain optimum value (ε=0.85) where the average Nusselt number has increased of (4.52) times than empty duct.

Kamath *etal.*[3[studied experimentally mixed convection in a vertical channel containing aluminum metal foams having different number of pores per inch, with high porosity in the range of (0.9) to (0.95) with a view to delineate the flow regimes, experiments were conducted for a wide range of Richardson numbers ($0.05\leq Ri\leq1032$) and Reynolds numbers ($40\leq Re\leq3730$). It was found that pressure drop per unit length increased with increased inlet velocity and decreased porosity. The metal foams were made from (AlSi7Mg aluminum) alloy, having a thermal conductivity (165 W/m.K). The heat transfer coefficient increased as Reynolds number increased and the porosity decreased. The enhancement ratio (the performance of a metal foam case with an empty channel case) was seen to vary from (1.5) to (2.9) for the range of parameters studied.

Dyga*etal.*[4] using air and water as a working fluid flow through a channel with and without wire mesh as a porous media to study heat transfer and pressure drop. The increment was (40%) in heat transfer when using wire mesh packing and the energy consumption required for water pumping was higher than using air also water heat transfer is higher than air heat transfer in the packed channel.

Kurian *etal.*[5] investigated the hydrodynamic and thermal performance of brass wire blocks in a vertical channel. The brass wire had a density of (8800 kg/m^3), thermal conductivity of (119 W/m.K) and porosities (0.77,0.88 and 0.85). The wire meshes were arranged side by side to act as a porous block. Experiments in a vertical channel using an isothermal flat plate were conducted for a velocity range (0 – 2 m/s). It showed that the heat transfercoefficient increased by (2 times) as compared to a clear channel case for the same velocity. The used mesh blocks used showed better performance, as compared to lower porosity mesh blocks and other metallic porous media of similar porosity.

Hilal*etal.*[6]investigated experimentally forced convection heat transfer of air inhorisontal porous rectangular duct. The pad consist of (zig –zag) metallic wire mesh insert with porosities (0,97-0,99) and with Reynolds numbers (7682-17323). The Nusselt number in the porous duct was increased up to (144%) for (ε=0.97) and (72%) for (ε=0.99) with comparison empty duct.

The present work analyses the effect of inserting wire mesh in (zig-zag) form insidevertical channel to enhance the convection heat transfer. The experiments are conducted under the same dimension and conditions in Ref.[6] for the purpose of comparison when changing the orientation of the duct willbe changed from the horizontal to the vertical.

Apparatus:

It has been used the same rig in Ref.[6], which illustrated schematically in Fig.(1-a) , photographically in Fig.(1-b) and schematic diagram Fig(2). The system parts are test section, air flow circuit and measuring instrument.

Air is provided by a centrifugal fan to the duct (12.5*12.5 cm) and (90°) fitting to vary the direction of flow. Then(30 cm) length with (5 cm) honey comb rectifier is used to obtain uniform velocity profile and remove eddies.

The test duct with dimension (12.5*12.5*100 cm) is heated by uniform heat flux with heating element consisted of a resistance heater manufactured by (0.5 mm)diameter nickel chrome wire isolated by ceramic spheres. Input electric power input is adjusted by a varice to controlle the heater voltage and current. The velocity of air is measured by anemometer.
Twenty K-type thermocouples are used to measure the temperature during the experiment. Inlet and outlet air temperatures, local temperature of the test duct surface , nine of them is inserted at the center line of the right surface duct and nine thermocouples are inserted in the top surface along the axially center line.

The pad used in the work are commercial galvanized steel screen which (wire diameter 0.76 mm, density 2659 kg/m^3 and thermal conductivity 164 W/m.K) . It was cut to (12 piece) in rectangular form (12.7*12.5 cm) dimension and arranged courrgeatly in (V ends) shape with (8 cm) pitch between two alternate piece at the centerline axial as shown in Fig.(2). The porosity of porous wire pad is calculated the volume ratio of the porous insert to all the screens on and be equal to (Ɛ=0.99).

Air enters the packed duct and when thermal steady state obtained , all temperatures ,air velocity ,heater current and heater voltage are recorded. The energy supplied into electrical heater determined as:

$Q_s = I^2 . R$ --------------------- (1)

The heat transferred into air from heater:

$Q_t = \dot{m}_a * (Cp_o * T_{out} - Cp_i * T_{in})$--------------------(2)

The difference between them is heat loss and in the experiments is not exit (5%) .

All air properties are find at the average temperature:

$T_{av} = \frac{T_{in}+T_{out}}{2}$ ---------------------- (3)

Heat flux can be calculated from:

$q_w = \frac{Q_t}{A_s}$ -------------------- (4)

Local heat transfer coefficient (h_x) is equal:

$h_x = \frac{q_w}{(\Delta T)_x}$----------------------- (5)

Local air temperature (t_{bx}) :

$t_{bx} = T_{in} - \frac{x}{L} (T_{out}\text{-}T_{in})$ ----------------------- (6)

Then (h_x) become :

$h_x = \frac{q_w}{(t_{sx} - t_{bx})}$--------------------------- (7)

The hydraulic diameter (D_{eq})[7]:

$D_{eq} = \frac{4\,(W*H\,)}{2\,(W+H)}$--------------------------- (8)

The effective thermal conductivity is computed from [8]

$k_e = k_f^{\varepsilon} k_s^{1-\varepsilon}$----------------------------- (9)

The local Nusseltnumber (Nu_x):

$Nu_x = \frac{h_x D_{eq}}{K_e}$---------------------------- (10)

The Reynolds number is :

$Re_D = \frac{U \quad D_{eq}}{v}$------------------------ (11)

The average heat transfer coefficient calculated from :

$h = \frac{1}{L}\int_{x=0}^{x=L} h_x \quad dx$------------------------ (12)

andthe average Nusselt number:

$Nu = \frac{h \quad D_{eq}}{K_e}$------------------------- (13)

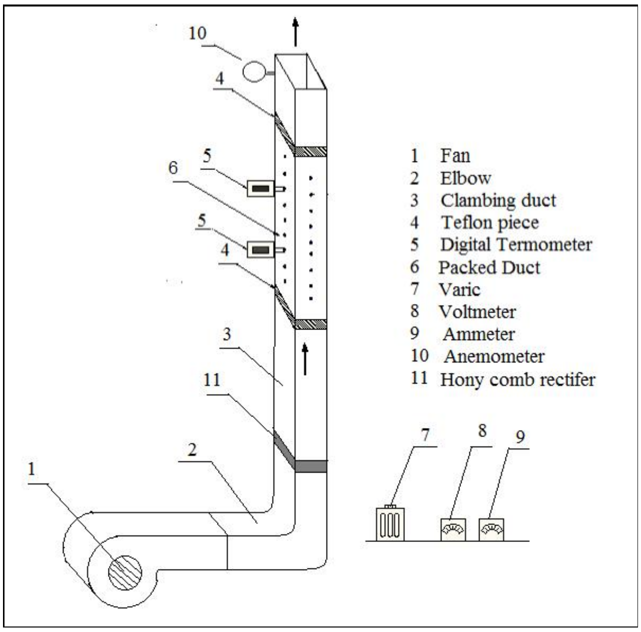


Fig (1-a) Experimental rig

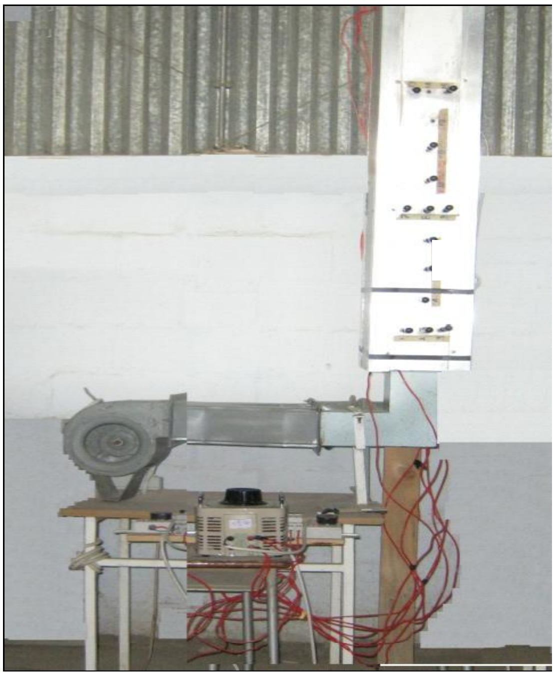


Plate (1-b)photographic of experimental rig

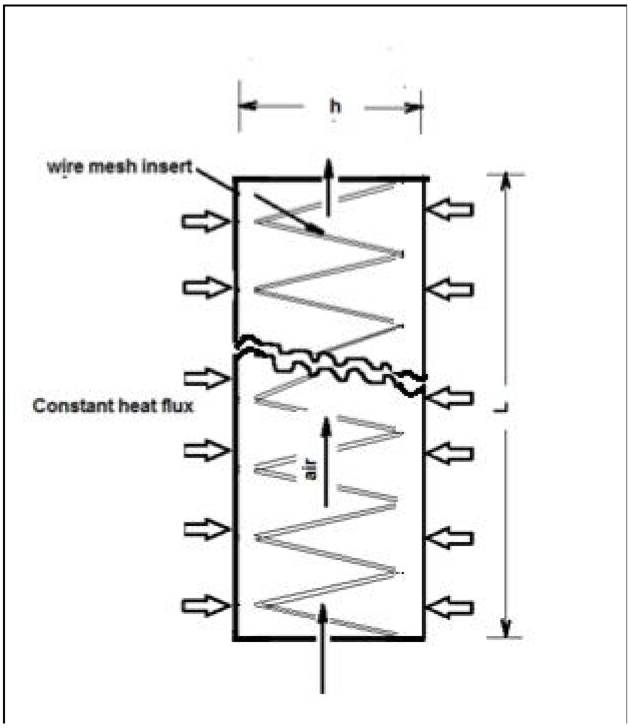


Fig. (2) Schematic diagram

Plate (2) Wire mesh pad

Results and Discussion :

Results are obtained for experiment with heat flux (0.2 kW/m^2- 0.5kW/m^2), porosity (ε=0.99) and Reynolds number (8010 - 18400) at vertical position.

Fig,(3) illustrate wall temp. distributionalong the packed duct. The temperature increases with increasing heat flux and decreasing Reynolds number and the highest temperature ratio (21.4%) when increasing heat flux at (60%) percent and (56.5%) percent decreased in Reynolds number.

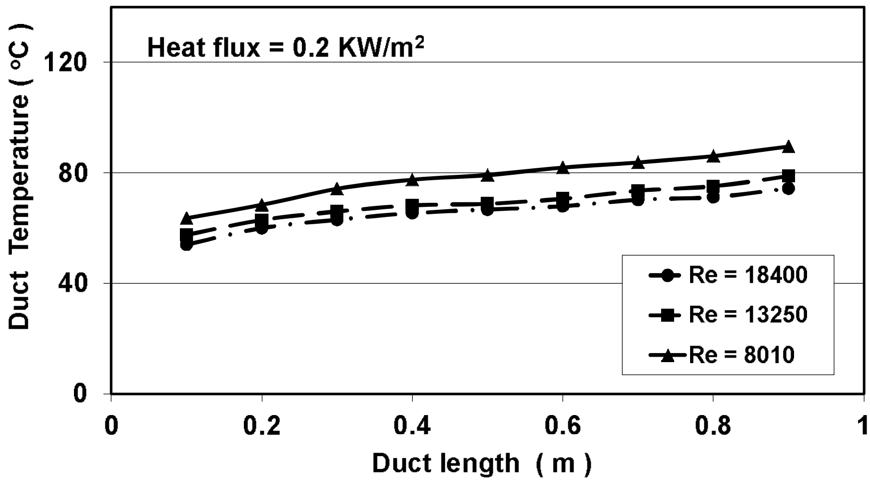


Fig.(3-a)

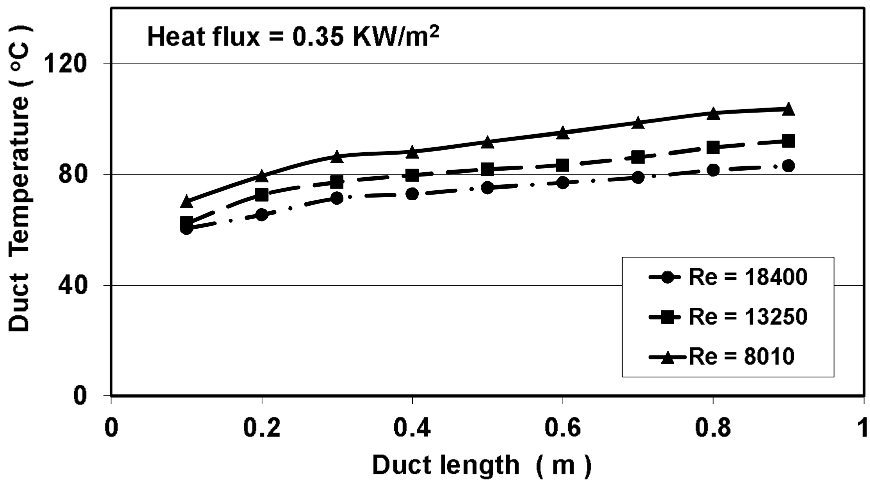
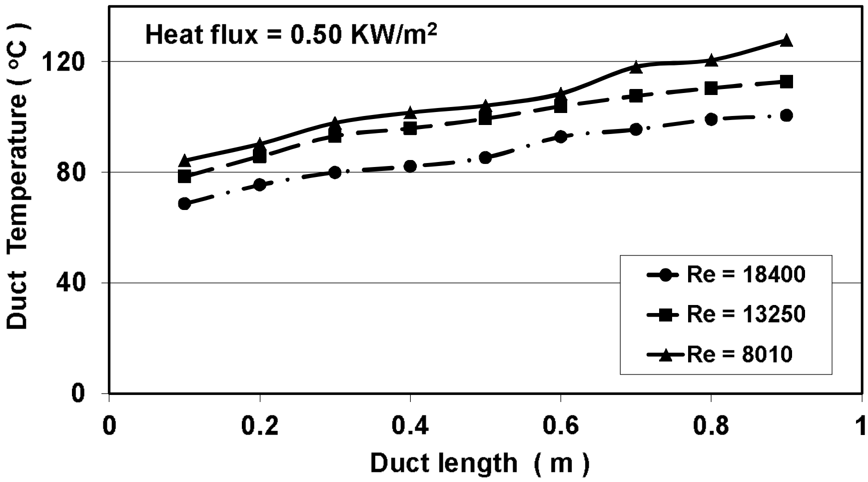


Fig.(3-b)



Fig(3-c)

Fig.(3) Surface duct temperature with the variation of Reynolds number and heat flux for wire mesh pad

Fig.(4) show local Nusselt number in test duct. It is noted that increasing(Re) yield to increasing turbulence in duct due to presence of wire mesh pad and decreasing thickness thermal boundary layer. There is a narrow passage to pass air near the surface duct because the pad does not extend from center to internal surface duct. In the passage air is pushes strongly leading to increase heat transfer from hot transfer surface.

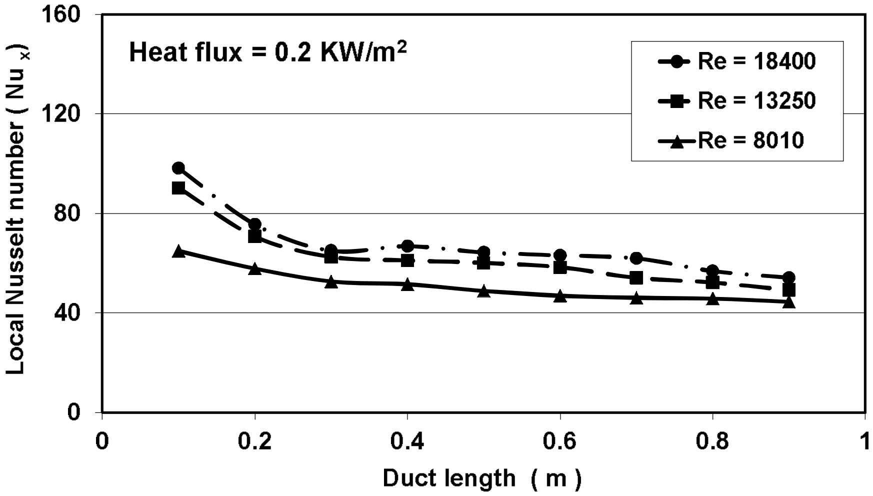


Fig.(4-a)

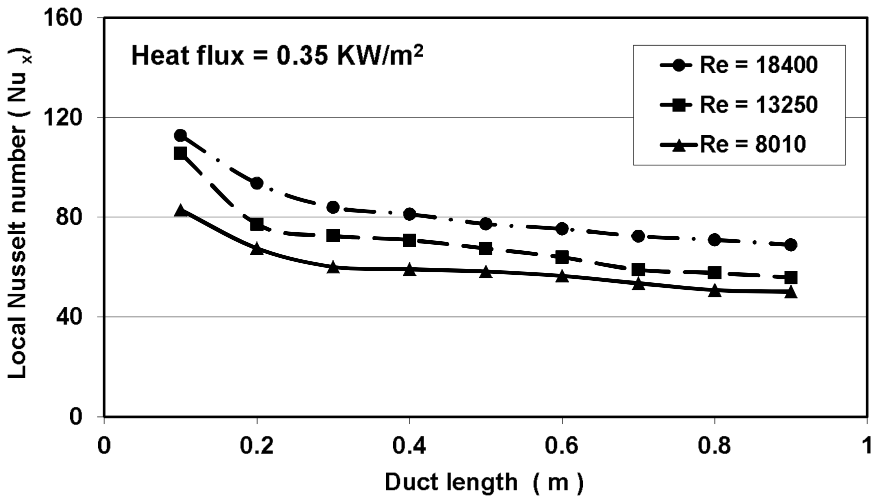


Fig.(4-b)

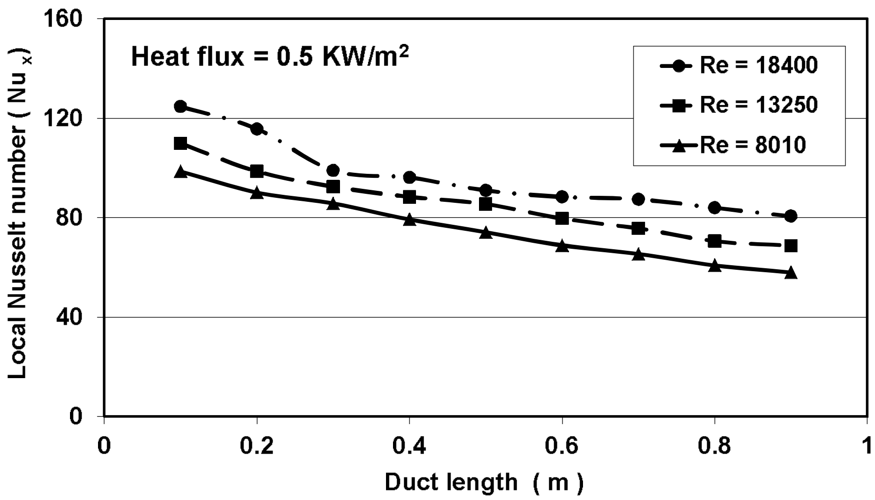


Fig.(4-c)

Fig.(4) Variationof the local Nusselt number along the duct length

The values of average Nusselt number are plotted in Fig.(5) and resulting correlation is :

$Nu = 7.9696 \; Re^{0.2179}$ -------------------------- (14)

To compare the Nusselt number for vertical position with wire mesh pad and (Nu) for empty duct which obtained from Ref. [7]:

$Nu = 0.023 \; Re^{0.8} Pr^{0.4}$ ---------------------------- (15)

Fig.(5) show the average Nusselt number in test duct and empty duct against Reynolds number, It is noted that the wire mesh pad enhanced heat transfer at (150%) times than clear duct.

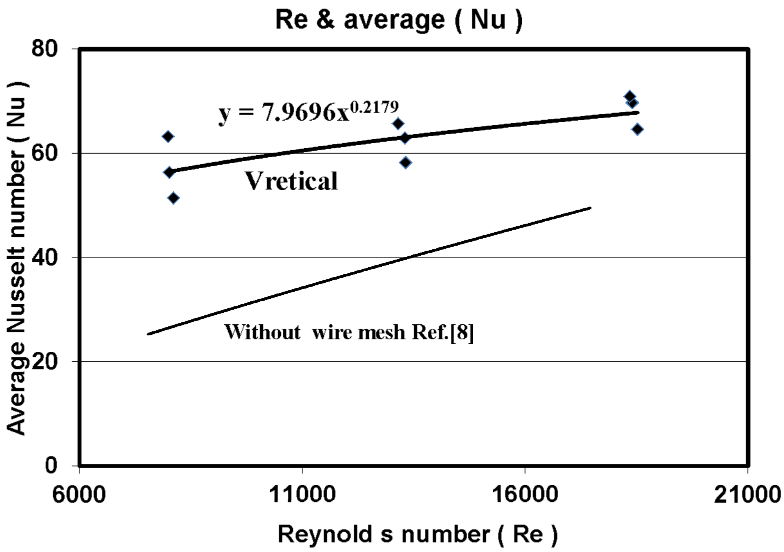


Fig.(5) Average Nusselt number as a function of Reynolds number for wire mesh pad and clear duct

To know the effect of duct inclination angle from horizontal to vertical, Fig(6) show that (Nu) at vertical position is higher than horizontal from Ref.[6] at (40 %) times according to the forces of buoyancy driver flow have the same direction (vertical) of air flow which resulting enhance heat transfer through packed duct.

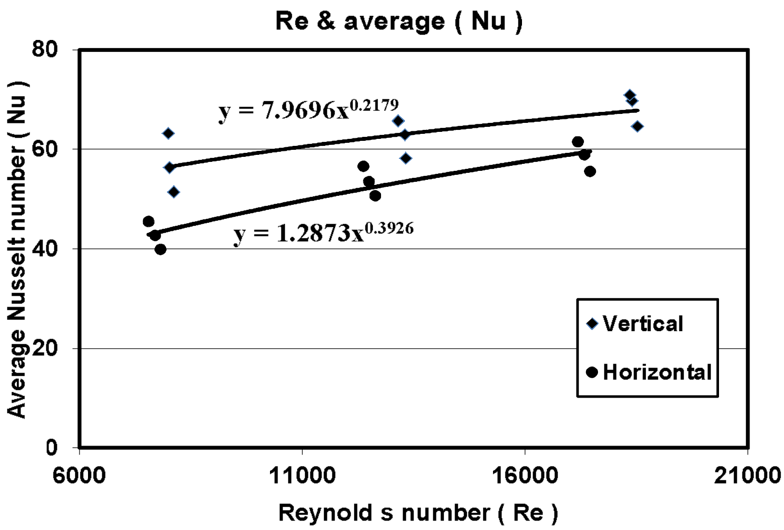


Fig.(6) Average Nusselt number as a function of Reynolds number for packed test duct at vertical and horizontal position

From above, the results show that the heat transfer in vertical position is better than horizontal position at the same conditi]on.

Nomenclature

A_S	Duct surface area	m^2
Cp_o	Air specific heat at constant pressure at outlet air temp.	kJ/kg.K
Cp_i	Air specific heat at constant pressure at inlet air temp	
D_{eq}	Hydraulic diameter	m
h_x	Local heat transfer coefficient	$W/m^2.K$
h	Average heat transfer coefficient	$W/m^2.K$
H	Duct height	cm
K_f	Air thermal conductivity	W/m.K
K_e	Effective thermal conductivity	W/m.K
K_s	Wire mesh thermal conductivity	W/m.K
L	Duct length	cm
Nu_x	Local Nusselt number	-
Nu	Average Nusselt number	-
$\dot{m}_a$	Air mass flow rate	Kg/s
Pr	Prandtl number	-
q_w	Duct heat flux	W/m^2
Q_t	Heat transfer rate to air	W
Q_s	Heat transfer rate supplied from heater	W
Re_D	Reynolds number	
T_{av}	Average air temperature	^0C
T_{in}	Inlet air temperature	^0C
T_{out}	Outlet air temperature	^0C
T_{sX}	Local duct surface temperature	^0C
T_{ax}	Local air mean temperature	^0C
U	Air velocity	m/s
W	Duct width	cm
X	Duct length from duct enterence	cm
ε	Porosity	-
ϑ	Kinematic viscosity of air	m^2/s

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