Mansoura Engineering Journal, (MEJ). Vol. 29, No. 1. March 2004. M. 34

HEAT TRANSFER FOR PULSATING FLOW IN A HORIZONTAL CYLINDER PARTIALLY FILLED WITH A POROUS MEDIUM

"إنتقال الحرارة لسريان نبضى داخل إسطوانة أفقية مملوءة جزئيا بوسط مسامى"

Hesham M. Mostafa *, G. I. Sultan ** and M. G. Mousa ***
* Higher Technological Institute, Tenth of Ramadan City, Egypt.
** Faculty of Engineering, Mansoura Univ., Mansoura, Egypt.
***Faculty of Education, Industrial Branch, Damietta, Mansoura Univ., Man., Egypt.
Email: Mgmousa@mans.edu.eg

خلاصة البحث:

فى هذا البحث أجريت دراسة معملية لإنتقال الحرارة لسريان نبضى لمائع يمر داخل إسطوانة أفقية ساخنة ومملوءة جزئيا بوسط مسامى. السطح الخارجى للإسطوانة معرض لبخار مشبع يجعله عند درجة حرارة ثابتة والمائع الذى يمر داخل الإسطوانة هو الماء. الذى يمر مباشرة من خزان على إرتفاع ثابت إلى داخل الإسطوانة فى حالة السريان المنتظم أما فى حالة السريان النبضى فإنه يمر أولا على مولد للنبضات قبل دخوله الإسطوانة. يمكن التحكم فى عدد النبضات وذلك للحصول على ترددات مختلفة. الوسط المسامى المستخدم لملئ الإسطوانة هو كرات من الصلب. تم تصميم وتنفيذ دائرة إختبار معملية لإختبار تأثير السريان النبضى عند ترددات مختلفة على كمية الحرارة المنتقلة بالمقارنة بالسريان المنتظم وذلك عند معدلات تدفق مختلفة للماء المار داخل الإسطوانة وهى فارغة ثم وهى مملوءة جزئيا ثم وهى ممتلئة تماما بالوسط المسامى. وقد تم تزويد الدائرة بأجهزة قياس لرصد البيانات الخاصة بدائرة الإختبار من قياس لدرجات الحرارة والضغط والتدفق وتردد السريان النبضى والإنخفاض فى الضغط. وقد تم حساب رقم نوسلت المتوسط ومعامل انتقال الحرارة بالحمل الجبرى للسريان النبضى والمنتظم وذلك فى مدى ظروف التشغيل المختبرة. حيث أجريت التجارب عند قيم مختلفة لرقم رينولدز من ٤٠٠ حتى ٢٠٠٠ مع تغيير الفيض الحرارى من ١٠ كيلووات/م^{٢} حتى ٦٠ كيلووات/م^{٢} لكل من السريان المنتظم والسريان النبضى لترددات مختلفة تصل إلى ٥ هيرتز وذلك لنسب إمتلاء للإسطوانة بالوسط المسامى تتراوح من صفر (الإسطوانة فارغة) حتى ١ (الإسطوانة مملوءة تماما).

وقد أظهرت النتائج أن رقم نوسلت المتوسط للسريان النبضى والمنتظم يزداد مع زيادة رقم رينولدز وكذلك الإنخفاض فى الضغط يتزايد مع زيادة نسبة الإمتلاء. أيضا للسريان المنتظم رقم نوسلت يزداد مع زيادة نسبة الإمتلاء للإسطوانة بالوسط المسامى وذلك عند قيم ثابتة لكل من الفيض الحرارى ورقم رينولدز. أما فى حالة السريان النبضى فإن تغيير رقم نوسلت مع نسبة الإمتلاء للإسطوانة بالوسط المسامى يكون غير تقليدى. حيث لوحظ أنه عند نسبة إمتلاء حوالى ٠,٣٥ فإن رقم نوسلت للسريان النبضى تزيد قيمته مقارنة بالسريان المنتظم كما أن الإنخفاض فى الضغط عند هذه النسبة يكون قيمته مناسبة مقارنة بنسب الإمتلاء الأعلى من ٠,٣٥ وقد لوحظ أنه لنسب الإمتلاء الأعلى من ٠,٣٥ أن رقم نوسلت للسريان النبضى يقل مقارنة بالسريان المنتظم كما أن الإنخفاض فى الضغط تكون قيمته كبيرة. ولذلك يمكن إعتبار قيمة نسبة الإمتلاء ٠,٣٥ هى القيمة المثالية لإمتلاء الإسطوانة بالوسط المسامى وأيضا عند هذه القيمة كانت القيمة المثالية لتردد السريان النبضى هى ٢ هيرتز وذلك فى مدى ظروف التشغيل المختبرة. وقد تم إستنتاج صيغة رياضية لرقم نوسلت المتوسط كدالة فى رقم رينولدز وتردد السريان النبضى فى مدى ظروف التشغيل المختبرة. وكذلك تمت مقارنة النتائج المعملية التى تم الحصول عليها مع نتائج الأبحاث السابقة حيث كانت نتيجة المقارنة مرضية.

Abstract

Forced convection heat transfer for pulsating flow inside a horizontal hot cylinder partially filled with porous medium is experimentally investigated. The outer surface of the tested cylinder is exposed to saturated steam to maintain its surface at constant wall temperature. The experimental work is performed for laminar flow of water inside the cylinder. As steady and pulsating flow with different frequencies. Carbon steel balls with 6.35 mm diameter are used as particles, which filling the tested cylinder.

An experimental set-up is designed and constructed to perform this aim for investigating the effect of pulsation frequencies on the amount of heat transferred, compared with steady flow for different water flow rates at different values of filling ratio with porous medium for the tested cylinder. The required experimental measurements of temperature, pressure, mass flow rate, frequency and pressure drop are collected for further data analysis. The operating parameters range are considered as; for Reynolds number from 400 to 2000, heat flux from 10 kW/m^2 to 60 kW/m^2 and pulsation frequencies from zero up to 5 Hz for different filling ratios from zero to unity.

Accepted March 31, 2004.

The obtained experimental results show that, for the considered range of the operating parameters Nusselt number and in turn heat transfer coefficient increase with increasing Reynolds number for steady flow and pulsating flow. Pressure drop also increases with increasing filling ratio with porous medium. Also, Nusselt number increase with increasing filling ratio with porous medium, for steady flow but for pulsating flow the variation of Nusselt number versus filling ratio with porous medium is monotonically. Also, it is found that, for filling ratio with porous medium equal to 0.35, Nusselt number for pulsating flow is bigger than steady flow and the value of pressure drop takes appropriate value as compared with other filling ratios. For higher values of filling ratios than 0.35, the value of Nusselt number for pulsating flow is lower than steady flow and the pressure drop takes higher values. Therefore, R_p =0.35 was considered the optimum value of filling ratios for pulsating flow in the studied operating range. It is found that, the optimum value for strouhal number was equal to 4 (which corresponding to f = 2 Hz) to gave higher values of Nusselt number at optimum value of filling ratio (R_p =0.35).

Good agreement was obtained when comparing the present experimental results with the previous results. Also, an empirical formula was derived for Nusselt number as a function of Reynolds number, and pulsation frequencies in the studied operating ranges.

Key words: Heat transfer, Pulsating flow, partially, porous medium and hot cylinder.

NOMENCLATURE

A : Surface area, m^2
D : Hot cylinder diameter, m
h : Heat transfer coefficient, $W/m^2.\ ^oC$
k : Thermal conductivity, $W/m.\ ^oC$
m : Mass flow rate, kg/s
Pr : Prandtl number ($Pr = Cp\ \mu / k$), -
q" : Heat flux, W/m^2
Rp : Filling Ratio ($Rp=(S/(0.5*D))$), -
St : Strouhal number ($St = f D / u$), -
u : Water velocity, m/s

Cp : Specific heat for fluid, J/kg. °C
f : Frequency, Hz
i : Specific enthalpy, J/kg
L : Hot cylinder length, m
Nu : Nusselt number ($Nu = h D / k$), -
Q : Heat transfer rate, W
Re : Reynolds number ($Re = \rho u D / \mu$), -
S : Thickness of porous medium, m
T : Temperature, °C

Greek symbols

μ : Dynamic viscosity of the fluid, kg/m.s
ρ : Density of the fluid, kg/m^3

Subscripts:

av : average
s : surface
t : total
w : water
i : inlet

o : outlet
st : steam
us : useful
g : gas
loss : loss

1. INTRODUCTION

Convection heat transfer in fluid saturated porous media motivated by a wide range of thermal engineering applications, such as geothermal systems, oil extraction, solid matrix heat exchangers, ground water pollution, thermal insulation, heat pipes, electronic cooling, filtration, chemical reactors, and the storage of nuclear wastes. Indeed, to fill the entire channel with a high-conductivity solid matrix can significantly enhance the heat transfer rate but at the expense of a considerable increase of the pressure drop. For this reason, forced convection in a composite system, in which a fluid-saturated porous material occupies only a part of the passage has been the topic of several investigations published in the literature [10] On the other hand, much attention has been given to both convective and conductive heat transferred by superimposing pulsation on the mean flow in a confined passageway. However, in the view of pressure drop and axial thermal diffusion, it is expected that a more effective enhanced heat transfer may be achieved by pulsating flow through a pipe partially filled with a porous medium. Advanced heat exchangers, regenerators and Stirling engines are some promising thermal engineering applications pertaining to the present investigation. In addition, a literature survey reveals that studies, which involve the present aspect, are relatively scarce and often incomplete.

A numerical study by A. V. Kuznetsovc et al (2004) was made of turbulent flow inside partially filled with porous medium. The problem of modeling a turbulent flow in the porous/ fluid domain was thus reduced to thus problem of matching a laminar flow solution in the porous region and turbulent flow solution in the clear fluid region at the porous/fluid interface.

A numerical study by Sung, S.Y et al (2002) was made of flow and heat transfer characteristics of forced convection in a channel that is partially filled with a porous medium. The flow geometry models convective cooling process in a printed circuit board system with a porous insert. The channel walls are assumed to be adiabatic. Comprehensive numerical solutions are acquired to the governing Navier - Stokes equations, using the Brinkman-Forchheimer-extended Darcy model for the regions of porous media. Details of flow and thermal fields are examined over ranges of the operating parameters, the Reynolds number , the Darcy number the thickness of the porous substrate and the ratio of thermal conductivities Two types of the location of the porous block are considered. The maximum temperature at the heat source and the associated pressure drop are presented for operating parameter. Also, as the ratio of thermal conductivities increases for fixed Darcy number, heat transfer rates are augmented. Explicit influences of Reynolds number on the flow and heat transport characteristics are also scrutinized. Assessment is made of the utility of using a porous insert by cross comparing the gain in heat transport against the increase in pressure drop.

A numerical study by Guo et al (2001) was made of pulsating flow and heat transfer characteristics in a circular pipe partially filled with porous medium. The Brinkman-Forchheimer-extended Darcy model was adopted for the porous matrix region, which was attached to the pipe wall. The impacts of the Darcy number, the thickness of porous layer, the ratio of effective thermal conductivity of porous material to fluid, as well as the pulsating frequency, and the amplitude, were investigated. The enhanced longitudinal heat conduction due to pulsating flow and the enhancement convective heat transfer flow high conducting porous material were examined. The maximum effective thermal diffusivity was found at a critical thickness of porous layer. The effects of pulsating amplitude and frequency on heat transfer are also scrutinized.

A numerical study by Guo et al. (1997) the pulsating flow and heat transfer characteristics in a circular pipe partially filled with a porous medium. The Brinkman-Forchheimer-extended Darcy model is adopted for the porous matrix region, which is attached to the pipe wall. The enhanced longitudinal heat conduction due to pulsating flow and enhanced convective heat transfer from high conducting porous material are examined for different operating parameters. An optimal porous layer thickness was obtained

A experimental study by Chikh et al.(1995) was observed an enhanced heat transfer in an annular duct partially filled with a porous medium with high permeability and conductivity. The obtained results from the analytical solution show that increasing either the permeability or the thermal conductivity improves the heat transfer. Further, for highly permeable and conducting porous media, it may not be necessary to fill the gap completely to attain the maximum heat transfer

A numerical study by Kim et al. (1994) for heat transfer characteristics from forced pulsating flow in a channel filled with fluid-saturated porous media. The channel walls are assumed to be at uniform temperature. In comparison with the case of non-pulsating flow. the presence of flow pulsation brings forth a reduction in heat transfer in the entrance region and an enhancement of heat transfer at moderate downstream regions. Farther downstream, the influence of pulsation was neglected.

Poulikakos et al (1987) performed a theoretical study for fully developed convection heat transfer in a channel partially filled with a porous matrix. Two channel configurations are investigated, namely, circular pipe and parallel plates. A surprising finding was that the value of Nusselt number dependence on the thickness of the porous region is not monotonic. A critical value of the porous region thickness exists at which the value of Nusselt number reaches a minimum.

Poulikakos and Renken (1987) simulated numerically the problem of forced convection in a channel filled with a fluid-saturated porous medium. The temperature at the channel walls was assumed to be constant. Two channel configurations are investigated: parallel plates and circular pipe. The channeling phenomenon near the walls of both duct configurations enhanced the thermal communication between the fluid/solid matrix composite and the walls. This fact yielded an overall 22% increase in the value of Nusselt number in the fully developed region for the circular channel, compared to the value predicted when Darcy model was used.

Kurzweg (1985) showed that pulsation produces an enhanced axial diffusion in the presence of an axial temperature gradient. The enhanced thermal diffusion can be thousands of times larger than the transport by axial molecular conduction.

Therefore, in the present work, the effect of pulsating frequency and different operating parameters on the convection heat transfer rate for pulsating flow inside a horizontal hot cylinder partially filled with a porous medium was studied experimentally.

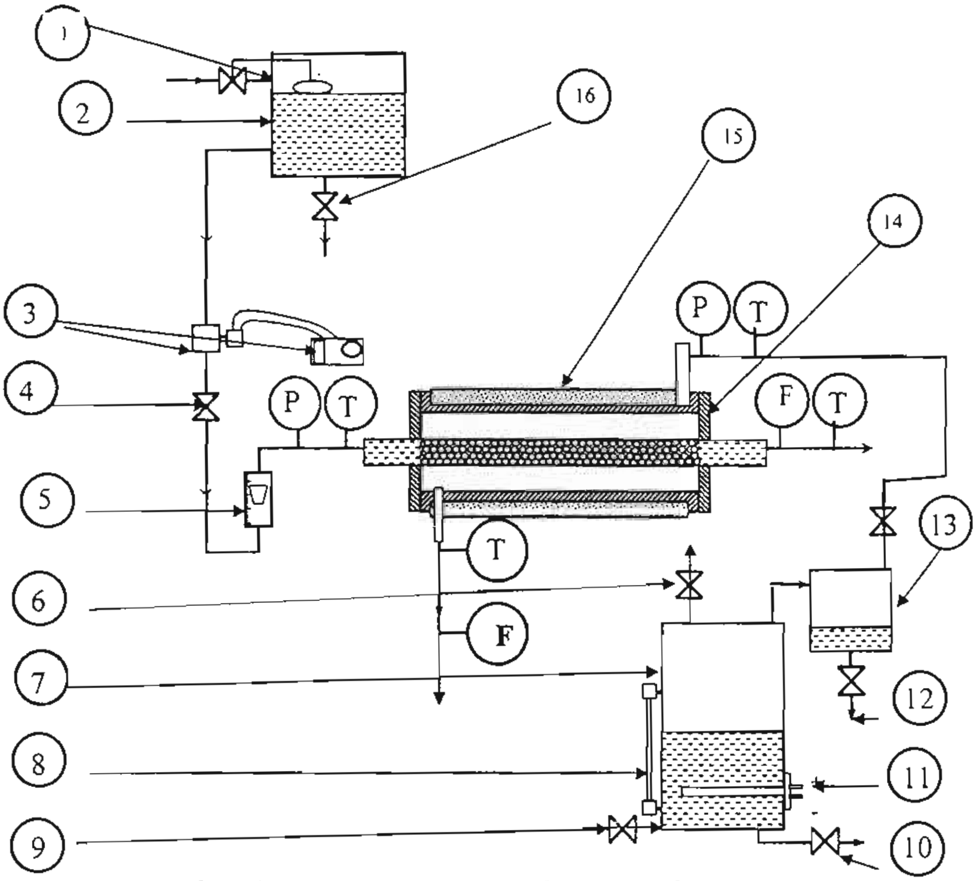
2. EXPERIMENTAL SET-UP

Experimental set-up is designed and constructed to evaluate the convection heat transfer rate for pulsating flow inside a hot cylinder partially filled with a porous medium. Figure (1) shows the schematic diagram for the experimental set-up, which performed to achieve this aim. The experimental set-up consists mainly of a horizontal test section, cooling water circuit and heating steam circuit. The details of the test section are illustrated in Fig. (2). It consists of a horizontal hot cylinder, which is filled by a porous medium and the outer surface is maintained at constant temperature by using a saturated steam in the annulus. The horizontal test cylinder is made of copper with 38 mm in diameter and 1 m long. The temperature of the outer surface of the tested cylinder was measured at different positions by using thermocouple wires, as shown in Fig. (2). The outer surface of the test section is insulated by using a 40 mm thickness of glass wool to minimize heat loss. Porous matrix consists of carbon steel balls having a nominal diameter of 6.35 mm. The porosity, ε of the porous medium was determined experimentally and found to be 0.4.

Cooling water circuit consists mainly of constant head tank, pulsating generator and its control unit. Cooling water flows inside a horizontal tested cylinder as a pulsating flow or a steady flow. The pulsation generator comprises of a solenoid valve, which is equipped by an electronic control unit to vary and control the frequency of pulsating flow.

The heating steam circuit was provided the test section by the required heating steam. An electric boiler with 9 kW rated power with the basic dimensions of 0.6 m in diameter and

1.2 m height is used to generate the heating steam at the required conditions. A steam trap is installed before the test section directly to insure that the heating steam enters at dry saturation condition. The heating steam is flowing in the annulus of the test section, over the outer surface of the horizontal tested cylinder, and then it condensed and returns back to the boiler



1	Float valve	10,12,16	Drain valves
2	Constant head tank	11	Heating element
3	Pulsating generator	13	Steam trap
4	Control valve	14	Test section
5	Rotameter	15	Insulation
6	Safety valve		
7	Electric boiler	F	Flow measurement
8	Sight glass	P	Pressure measurement
9	Gate valve	T	Temperature measurement

Fig. (1) Schematic diagram for the experimental set-up.

3. EXPERIMENTAL MEASUREMENTS TECHNIQUE

To start any experiment, the experimental set-up was allowed to equilibrate for approximately one hour until steady state condition had been reached. Mass flow rate for the heating steam inlet to the steam jacket (annulus of the test section) could be controlled to obtain the required heat flux, which applied on the outer surface of the tested cylinder. Also, mass flow rate of cooling water was controlled. Pulsation frequency was adjusted to a certain value. Once the desired steady state was reached, the required measurements were taken. These measurements are temperature of cooling water at inlet and outlet of the tested cylinder; mass flow rate of cooling water and pressure drop through the tested cylinder. Pulsation frequency is also measured. Temperature and pressure at the inlet for the heating steam are measured. For condensate, temperature and mass flow rate are also measured. Outer surface temperature for the tested cylinder is measured at different positions along its length, and then inner surface temperature can be calculated. Temperatures were measured by using copper-constantan thermocouple wires type K, which are connected to a temperature recorder having minimum readable value of ± 0.1 °C. Inlet steam pressure was measured by Bourdon pressure gauge with minimum readable value of ± 0.05 bar. Pressure difference was measured by using an inclined U- tube manometer, which using mercury as measuring fluid. Water flow rate was measured by using flow meter. The amount of condensate was small then it measured by using a calibrated tank and stop watch.

In order to obtain a measure of the reliability of the experimental data an uncertainty analysis was performed for the principle parameters of interest. The root-mean-square random error propagation analysis was carried out in the standard fashion using uncertainties of the basic independent variables. These are included test cylinder dimensions, pressure, temperatures and mass flow rates, which are used to calculate the uncertainty in Nusselt number. The largest calculated uncertainties in the current investigation are less than 8.5 % for Nusselt number.

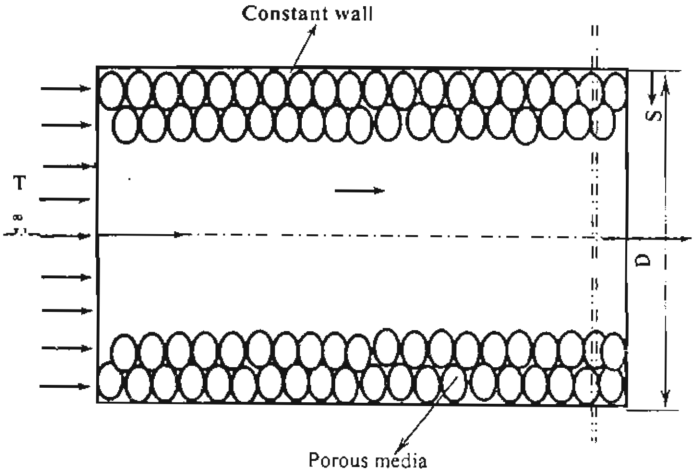


Fig.(2) Schematic diagram of Test Section

During the experimental work in the case of fully filled with porous medium the tested cylinder was emptied and refilled with the same amount of spheres several times. This was done to ascertain whether the experimental measurements would change significantly if the packing (which alter the microstructure of the porous medium in the vicinity of the cylinder wall) were changed. At steady state, the total input heat from the heating steam (Q_t) can be divided into useful heat to the water flow inside the tested cylinder (Q_{us}) and the remaining amount of heat can be transferred to the surroundings as heat loss (Q_{loss}). Then, useful heat can be determined as the difference between input heat and heat loss and calculated from measuring water flow rate and the temperature rise in water as;

$$Q_{us} = Q_t - Q_{loss} = m_w \, Cp_w \, (T_{w,o} - T_{w,i}) \qquad (1)$$

Where m_w , Cp_w , $T_{w,i}$ and $T_{w,o}$ are the amount of water flow rate, specific heat of water, inlet water temperature and outlet water temperature respectively. Water properties are calculated at average temperature ($T_{av,w} = (T_{w,i} + T_{w,o})/2$). The total input heat can be determined as;

$$Q_t = m_{st} (i_g - i_o) \qquad (2)$$

Where m_{st} , i_g and i_o are steam flow rate, specific enthalpy for dry saturated steam at inlet and specific enthalpy for condensate at outlet from the test section respectively.
Heat flux (q") can be calculated from the following equation as;

$$q'' = Q_{us} / A_s \qquad (3)$$

Where; $A_s = \pi D L$ (Inner cylinder heat transfer surface area).
D and L = Inner cylinder diameter and cylinder length respectively.
Convection heat transfer coefficient (h) can be calculated as;

$$h = q'' / (T_{s,i} - T_{w,av}) \qquad (4)$$

Where $T_{s,i}$ is the average value for the temperatures of the inner surface of the tested cylinder.

The average values for the dimensionless numbers like Nusselt number (Nu), Reynolds number (Re), Prandtl number (Pr) and Strouhal number (St) are defined according to the following equations as;

$$Nu = h D / k, \quad Re = \rho u D / \mu, \quad Pr = Cp \mu / k \ \& \ St = f D / u \qquad (5)$$

Where ρ, u, μ and f are water density, water velocity inside the cylinder, dynamic viscosity of water and pulsation frequency respectively.

5. RESULTS AND DISCUSSIONS

In designing heat exchangers it is important to enhance the amount of heat transferred to or from the working fluid and minimize the mechanical loss due to pressure drop. Then it is important to compromise between the pressure drop and heat transfer. In the present work, to reduce the pressure drop the tested cylinder was partially filled with a porous medium instead of full filling with porous medium. It is clear from Fig. (3) that, the pressure drop increases with increasing water velocity (or in turn Reynolds number). Also, it is observed that, decreasing values of filling ratio with porous medium cause a considerable decrease in the values of pressure drop.

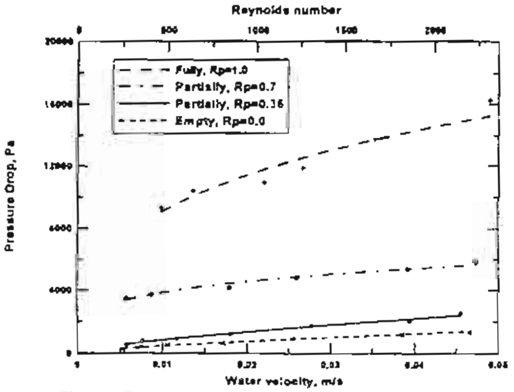


Fig. (3) Pressure drop for steady water flow inside the tested cylinder at different filling ratios.

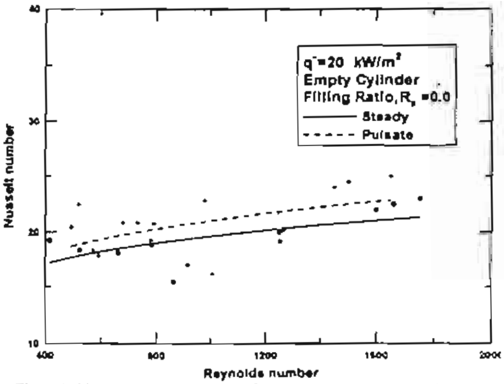


Fig. (4) Nusselt number versus Reynolds number for pulsating flow compared with steady flow for empty cylinder (R_p=0.0).

On the other hand, the influence of pulsation on the enhancement of convection heat transferred was examined for cylinder partially filled with porous medium and compared with empty and fully filled cylinder with porous medium. The experiments were performed for laminar flow with 400 < Re < 2000. Figure (4) shows that, for empty cylinder Nusselt number increases with increasing Reynolds number, as expected, for both steady and pulsating flow. Also, it is noticed that, for empty cylinder pulsating flow gave higher values for Nusselt number than steady flow at the same value of heat flux. This enhancement was pertained as, pulsation produced an enhanced in axial diffusion in the presence of an axial gradient in temperature. The enhanced in thermal diffusion was larger than the transport by axial molecular conduction.

As shown in Fig. (5), Nusselt number for pulsating flow takes higher values than steady flow for partially filling with porous medium at filling ratio equal to 0.35. This means that, an enhancement in the amount of heat transfer is obtained. But for higher values of filling ratio (R_p =0.7 and R_p =1), Nusselt number for pulsating flow takes lower values than steady flow, as shown in Fig. (6) and Fig. (7) respectively. It is important to collect the behavior of Nusselt number, which appears in figures (4-7) in a common graph to illustrate the effect of pulsation on the water flow inside the tested cylinder at different filling ratios with porous medium . Figure (8) shows the variation of Nusselt number against filling ratio with porous medium for pulsating flow compared with steady flow at certain value of Reynolds number. It is observed from Fig. (8) that, for steady flow Nusselt number increases with increasing filling ratio due to turbulence compared with the empty cylinder. This increase in

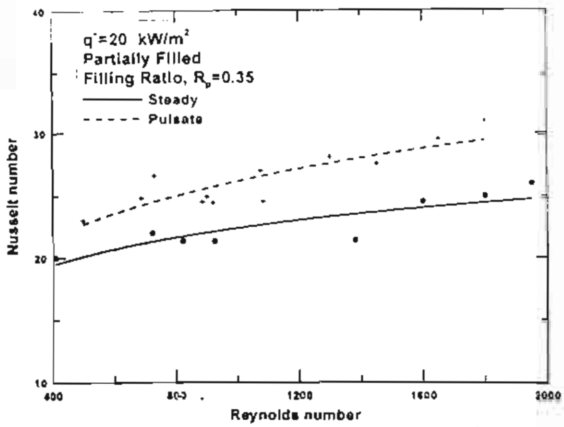


Fig. (5) Nusselt number versus Reynolds number for pulsating flow compared with steady flow for partially filled cylinder with porous medium (R_p =0.35).

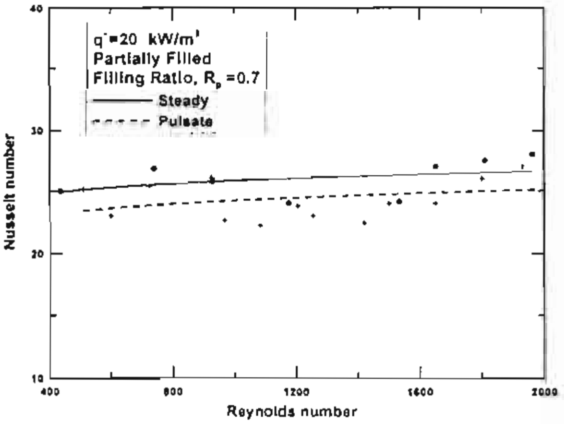


Fig. (6) Nusselt number versus Reynolds number for pulsating flow compared with steady flow for partially filled cylinder with porous medium (R_p =0.7).

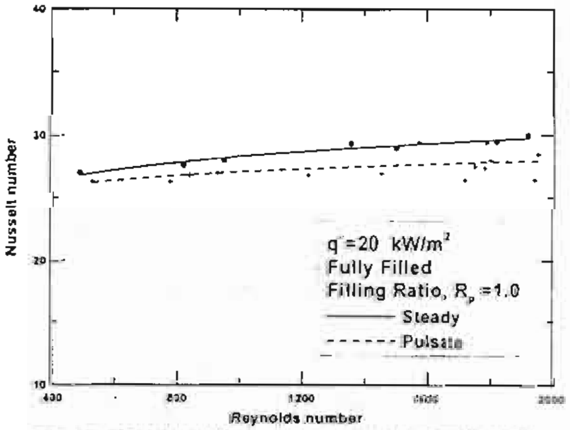


Fig. (7) Nusselt number versus Reynolds number for pulsating flow compared with steady flow for partially filled cylinder with porous medium (R_p =1.0).

the amount of heat transferred was attributed to the increase of the channeling velocity in the void region of the porous media, which is in contact with the tube surface. The impact of pulsation on the heat transfer enhancement for empty cylinder was expected due to the enhanced axial heat diffusion. This is based on the fact that large oscillating temperature gradients in the direction normal to the wall are produced and an axial temperature gradient is present. Figure (8) shows that, for pulsating flow the variation of Nusselt number with filling ratio with porous medium is monotonically. For filling ratio equal to 0.35 the increase in the value of Nusselt number or in turn the amount of heat transferred due to pulsation for partially filled was bigger than due to partially filled for steady flow. Then R_p =0.35 is considered the optimum value of filling ratio for pulsating flow in the range of operating condition. Also ,Fig(8) shows a comparison between the present results with the previous data, which obtained by Guo et al [2]. It is observed that the average values of Nusselt number for the present work for pulsating flow take the same trend with the previous data.

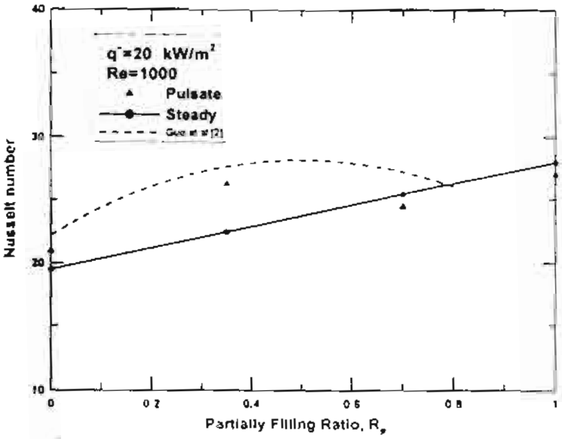


Fig. (8) Comparison between Nusselt number as a function of filling ratio with porous medium for pulsating flow and steady flow compared with Guo et al [2]

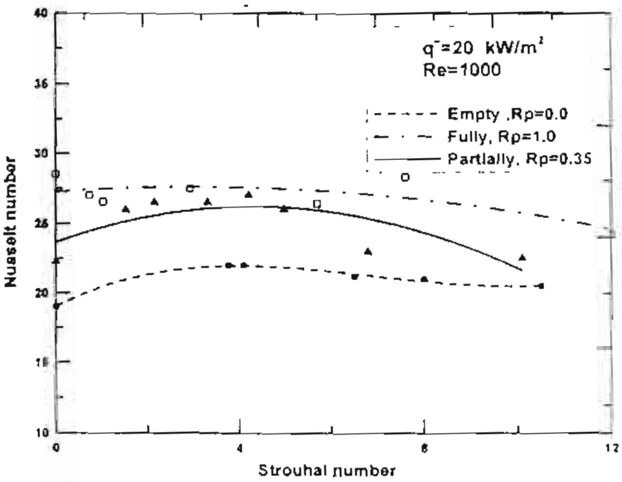


Fig. (9) Nusselt number versus Strouhal number for different values of filling ratio with porous medium

Figure (9) shows the variation of Nusselt number versus Strouhal number for different values of filling ratio with porous medium. Physically, the optimum frequency means that there is a sufficient time for heat to flow from the wall to the fluid core or before the temperature reverses itself within the core. It is found that, the optimum value for Strouhal number was equal to 4 (which corresponding to f = 2 Hz) to gave higher values of Nusselt number at optimum value of filling ratio (R_p =0.35), for the studied range of the operating parameters.

For the tested operating range the following empirical correlation is obtained as;

$$Nu=0.644\ Re^{0.661}(1+St\ e^{0.35-Rp}) \qquad (6)$$

CONCLUSIONS

Convection heat transfer for pulsating flow inside a horizontal hot cylinder partially filled with a porous medium is investigated experimentally. Water is used as a working fluid for pulsating and steady fluid flows. The tested cylinder is filled with saturated spherical beads porous media and the outer surface is exposed to saturated steam to maintain its surface at constant wall temperature. The studied operating parameters are Reynolds number; pulsation frequency and the effect of partially filled with porous material. The influence of pulsating flow on Nusselt number is investigated at different values of Reynolds numbers and filling ratios.

The obtained experimental results show that, for the considered range of the operating parameters Nusselt number increases with increasing Reynolds number for steady flow and pulsating flow. Pressure drop also increases with increasing filling ratio with porous medium. Also, Nusselt number increase with increasing filling ratio with porous medium, for steady flow but for pulsating flow the variation of Nusselt number versus filling ratio with porous medium is monotonically. Filling ratio equal to 0.35 was considered the optimum value of filling ratios for pulsating flow in the studied operating range. At this value, the optimum value for Strouhal number was equal to 4 (which corresponding to f = 2 Hz) to give higher values of Nusselt number, for the studied range of the operating parameters. An empirical formula was derived for Nusselt number as a function of Reynolds number, and pulsation frequencies in the studied operating ranges.

REFERENCES

[1] **A.V.Kuznetsov and S.M.Becker** (2004) "Effect of the interface roughness on turbulent convection heat transfer in a composite porous / fluid duct" Int. Comm. Heat Mass Transfer, vol.31, No.1, pp. 11-20

[2] **Sung, S.Y. Kim and J.M. Hyun**, 2002 "Forced Convection from an Isolated Heat Source in a Channel with Porous Medium," International Journal of Heat and Fluid Flow. Vol.21, No.6, pp.527-535.

[3] **Guo, Z., Kim, S. Y., and Sung, H. J.**, 2001 "Pulsating flow and heat transfer in a tube partially filled with a porous medium," Numerical Heat Transfer-Part A, Vol.35, No 5, pp.517-527

[4] **Guo, Z., Kim, S. Y., and Sung, H. J.**, 1997 "Pulsating flow and heat transfer in a pipe partially filled with a porous medium," Int. Journal Heat and Mass Transfer Vol. 40, No. 17, pp. 4209-4218.

[5] **Chikh, S., Boumedien, A., Bouhadef, K. and Lauriat, G.**, 1995 "Analytical solution of non-Darcian forced convection in an annular duct partially filled with a porous medium" Int. Journal of Heat and Mass Transfer Vol. 38, pp. 1543-1551.

[6] **Kim, S. Y., Kang, B. H. and Hyun, J. H.**, 1994 "Heat transfer from pulsating flow in a channel filled with porous media," Int. Journal Heat and Mass Transfer Vol. 37, No. 14, pp. 2025-2033.

[7] **Poulikakos, D., and Kazmierczak, M.**, 1987 "Forced convection in a duct partially filled with a porous material," ASME Journal of Heat Transfer Vol. 109, pp. 653-662.

[8] **Poulikakos, D., and Renken, K.**, 1987 "Forced convection in a channel filled with porous medium, including the effects of flow inertia, variable porosity, and Brinkman friction," ASME Journal of Heat Transfer Vol. 109, pp. 880-888.

[9] **Kurzweg** (1985) pulsation produces an enhanced axial diffusion in the presence of an axial temperature gradient" ASME Journal of Heat Transfer Vol. 107, pp. 380-388

[10] **Claus Borgnakkeand Richard E. sonntay** 1997 "Table of Thermodynamic and transport Properties "John Wiley & sons Inc.