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Packing size and shape effects on forced convection in large rectangular packed ducts with asymmetric heating

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Abstract

An experimental and numerical investigation of fully developed forced convection in large rectangular packed ducts is presented. The horizontally oriented ducts have the length-to-separation distance ratio of $L/H = 16$ with two aspect ratios of $W/H = 8$ and 4. A constant heat flux is supplied to the top wall, while the bottom and side walls are insulated. Packing of hard polyvinyl chloride Raschig rings with outside diameters of 48 and 34 mm, and expanded polystyrene spheres with diameters of 48, 38 and 29 mm are used in the air flow passage. The experiments are carried out for $200 < (Re_p = ud_p/\nu_f) < 1450$, and $4.5 < d_e/d_p < 9.0$. The pressure drop, the heat flux, axial and transverse temperature profiles of air flow inside the ducts are measured at the steady state. Similar experiments have also been carried out with empty ducts. Numerical predictions of two-dimensional quasi-homogeneous model are found to be in agreement with the experimental results of the packed ducts. The correlation equations for the Nusselt number are obtained. It is found that the introduction of packing into the air flow passage yields about a three times increase in the wall-to-air heat-transfer rate compared with that of the empty duct. © 1999 Elsevier Science Ltd. All rights reserved.

1. Introduction

Fluid dynamics and heat transfer behavior of fluid flow through packed tube and channel are of special interest because of wide applications in engineering practices, such as wall cooled reactors, heat exchangers and cooling of electronic equipment. In an early study Colburn [1] found that forced convection heat-transfer rate for air flowing through a uniformly heated packed tube, with a low tube-to-particle diameter, is about eight times higher than that of an empty tube. In a more recent work Chyrsler and Simons [2] reported that forced convection heat transfer to liquid Fluorocarbon-77 in a packed channel with glass spheres is about 10 times higher than that of an empty one. Introduction of the packing into the flow passage causes the thermal dispersion effect due to mixing of fluid, and decreases the resistance of thin, slow moving fluid film near to the heated wall. The forced convection in a packed rectangular duct with asymmetric heating has also shown a considerable increase in the wall-to-fluid heat-transfer rate [3–6]. The packed channels are used in solar air heaters [7,8] and are considered for air cooling in micro electronics [2,9].

In one of the early works, Fahien and Smith [10] used the numerical procedure to obtain the effect of the ratio of tube-to-particle diameter on convection by using the experimental data of Schwartz and Smith [11]. This point was made apparent earlier by Colburn [1] who found a maximum in the enhancement of heat transfer at a certain ratio of packing-to-tube diameter.

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Nomenclature

a_p	specific surface
A	flow cross-sectional area
C	constant in Eq. (1)
d	diameter
h	heat transfer coefficient
H	separation distance between top and bottom walls
G	mass flow rate
k	thermal conductivity
K	constant in Eq. (4)
L	packed section length of the duct
Nu_H	Nusselt number based on the separation distance, Eq. (8)
Nu_p	Nusselt number based on the particle diameter, Eq. (9)
P	pressure
Pr	Prandtl number
Q	heat flux
Re_H	Reynolds number based on the separation distance, Eq. (10)
Re_p	Reynolds number based on the particle diameter, Eq. (11)
T	temperature
u	superficial velocity
W	width of the duct
x	axial direction
y	radial direction.

Greek symbols

α_e	effective thermal diffusivity
ε	average void fraction
ν	kinematic viscosity
ρ	density
Φ	shape factor for irregular packing.

Subscripts

b	bulk
dy	dynamic
e	equivalent/effective
f	fluid
o	inlet
p	packing
st	static
t	tube
w	wall.

The ratio of tube-to-particle diameter, d_t/d_p, is directly related to the properties of the wall region especially for the gas-phase system due to increase in porosity [10–14]. Dixon et al. [13], Dixon [14,15], and Borkink and Westerterp [16] investigated the influence of the tube and particle diameter and shape, as well as their ratio on the radial heat transfer in packed beds of spheres, full cylinders and Raschig rings. Dixon [14,15] studied the region $5 < d_t/d_p < 12$, and $d_t/d_p < 4$, and pointed out that the wall effects tend to become unimportant for $d_t/d_p > 15$. Borkink and Westerterp [16] concluded that the values of h_w are less influenced by the particle shape. Most importantly they pointed out that the mixing phenomena on a particle scale may play a more important role in the process of radial heat transfer than usually is assumed. Tsotsas and Schulunder [17] analyzed the theoretical arguments, numerical calculations and experimental data from the literature, and pointed out the importance of physical aspect and not the mathematical formalism in the analysis. The rectangular packed beds with asymmetric heating have been investigated numerically with a vari-

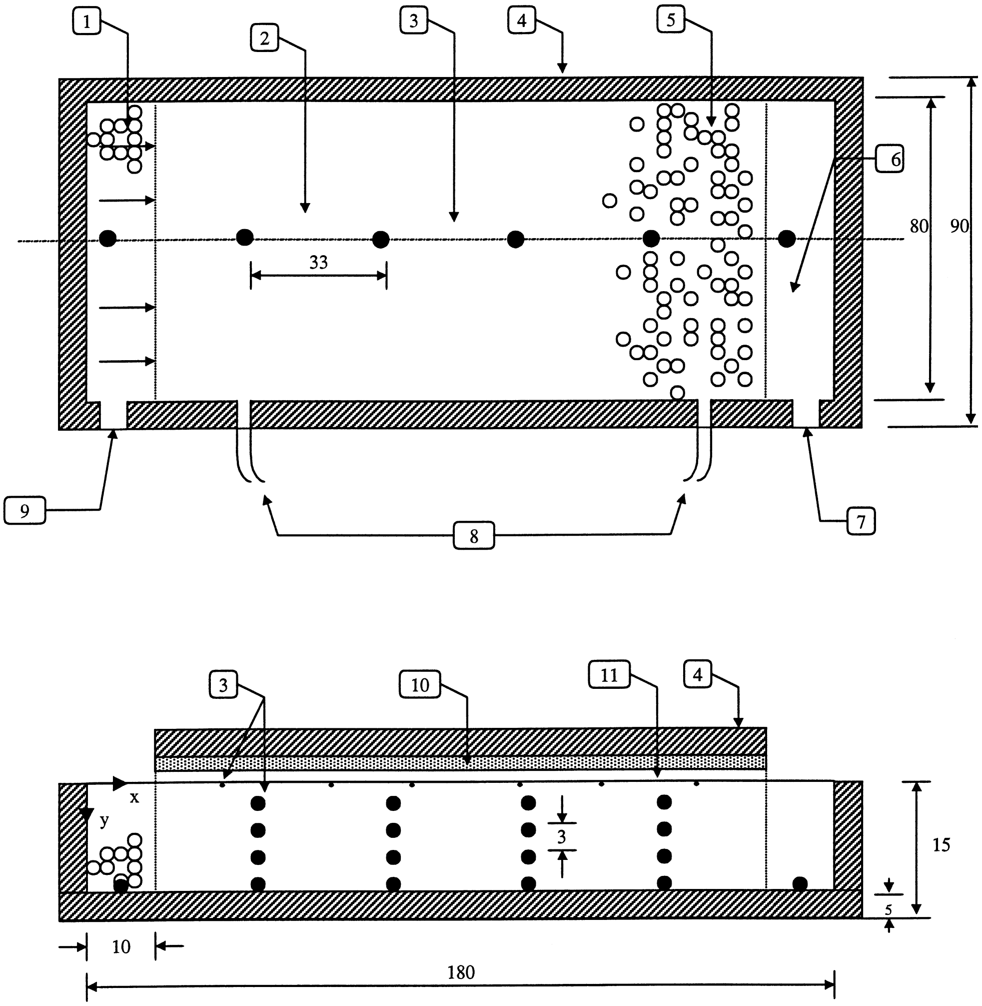

Fig. 1. Experimental apparatus.

able-porosity by Cheng and Vortmeyer [18], and experimentally by Demirel [5], and by Hwang et al. [6].

There has been no general agreement on the form of the correlation for the packed beds [17,19,20]. Various Nu_p versus Re_p correlations show considerable scatter and discrepancies, and most of the data used in these correlations are obtained from the radial heat transfer in exothermic catalytic reactors. However, small and large beds may behave differently which is particularly true for chemical reactors [21], and for beds of irregular packing due to more than one orientation [13,22]. This study presents experimental and numerical investigation of forced convection heat transfer in the 180 cm long rectangular packed ducts with the width-to-separation distance ratios of $W/H=8$ and 4. The horizontal flow passage is filled with Raschig rings and spheres of low thermal conductivity, and heated uniformly at the top wall only. The correlations for the Nusselt number are obtained for Raschig ring and for spheres. The correlation equations are valid for forced convection of air in the packed ducts for $200<Re_p<1450$, $4.5<d_e/d_p<9.0$ and the duct length-to-separation distance ratio of $L/H=16$. Packed duct measurements of the Nusselt numbers based on the separation distance, H, are also compared with those obtained for the empty ducts.

2. Experimental apparatus and procedure

The experimental apparatus is shown in Fig. 1. The horizontally oriented flow passage was made of 1 cm thick acrylic plate, while stainless steel of 0.1 cm thickness was used as the top heated wall. The overall dimensions of the ducts and the experiments are described as Cases I and II in Table 1. Two perforated

Table 1
Description of experimental setups

Property	Case I	Case II
Dimension of the ducts, cm	$10 \times 80 \times 180$	$10 \times 40 \times 180$
Total length, cm	180	180
Heated length, L, cm	160	160
Width, W, cm	80	40
Height, H, cm	10	10
Aspect ratio	8	4
Equivalent diameters of test sections, d_e, cm	29.4	21.7
Materials of top wall	Steel	Steel
Materials of side and bottom walls	Acrylic	Acrylic

plates were placed at the inlet and exit sections of the heated packed section for a better distribution of the incoming air flow and to support the packing. Each plate has small holes (5 mm diameter) of two rows in the middle surrounded by larger holes (10 mm diameter) of two single rows near to top and bottom plates. The heated packed section is preceded by a calming section of 10 cm in length, and followed by an exit section of 10 cm in length. All the side and bottom walls are insulated with a 5 cm thick insulating material with thermal conductivity of 0.14 W/m °C. The top wall is removable so that the packing can be filled in a random manner easily.

Two heating elements with twelve and seven loops of electric resistance heating wires are employed to supply uniform heat flux along the packed air flow passages for Cases I and II, respectively. The resistance wire distributed evenly and 2.5 cm above the top wall, and was supplied by an adjustable power input through a rheostat to control the heat flux. The heating elements covering the top wall are air tight and sufficiently insulated, so that the heat loss through the top and bottom walls was estimated to be less than 10% of the power input to the heated plate. Uniformity of the heat supply was tested for all the flow rates by the linear variation of top wall temperatures in the fully developed flow.

Air flow temperatures at three levels in the transverse direction and at four stations in the axial direction are measured by 12 K-type thermocouples of accuracy ± 0.2°C. The thermocouple probes are inserted from the side wall into the middle of the flow passage, and are located at 30, 63, 96 and 129 cm away from the calming section. At each station, temperatures of air flow are measured at 2, 5 and 8 cm in the transverse direction. Two thermocouples were placed at the inlet and exit sections of the flow passage. Four ‘cement on’ thermocouples with fast response time (10–20 ms) were placed on the bottom wall at 30, 63, 96 and 129 cm away from the inlet sec-

Table 2
The properties of the packing

Properties	Spherical EPS[a]	Spherical EPS[a]	Spherical EPS[a]	Raschig ring PVC[b]	Raschig ring PVC[b]
Inside diameter, mm	—	—	—	40	26
Outside diameter, mm	48	38	29	48	34
Height, mm	—	—	—	48	34
Equivalent diameter, mm	48	38	29	38.5	32.6
Void fraction, Case I	0.578	0.418	—	0.810	0.700
Void fraction, Case II	0.564	0.392	0.353	0.806	0.702
d_e/d_p, Case I	6.125	7.736	—	7.636	9.018
d_e/d_p, Case II	4.520	5.710	7.482	5.636	6.656
Number of units, Case I	931	2595	—	910	2959
Number of units, Case II	456	1261	3065	467	1490
Thermal conductivity, W/m K	0.037	0.037	0.037	0.14	0.14
Specific heat, kJ/kg °C	1.25	1.25	1.25	825	825
Density, kg/m^3	20	20	20	1320	1320

[a] EPS: Expanded polystyrene.
[b] PVC: Polyvinyl chloride.

tion for both Cases of I and II. Six 'cement on' thermocouples were mounted on the heated wall at 15, 41, 67, 93, 119 and 145 cm away from the inlet section for the Case I, and at 25, 47, 69, 91, 113 and 135 cm away from the inlet section for the Case II. For measuring the effectiveness of insulation and for calculation of heat loss, six thermocouples were distributed on both surfaces of insulating layers. All the thermocouples were shielded to reduce the radiation effects from surrounding. A data acquisition and control system of an *IBM PC* and *OMEGA WB-AAI16* board together with graphical interface application software is used to record and display the thermocouples.

Air flow rate from the main air supply line was measured by a Hygro-termo anemometer of type *OMEGA HHF710* together with the probe *HHF7-P1*. Air filter with a pressure regulator was placed in the air flow line. The anemometer has an accuracy of $\pm 1\%$ of reading (± 1 digit), and a resolution of 0.01 m/s. A sensitive inclined manometer filled with red Merium liquid (specific gravity of 0.8) was used for measuring the inlet pressure and the pressure drop in the packed section.

Raschig rings made of hard polyvinyl chloride in two different sizes, and spheres made of expanded polystyrene in three different sizes were used as the packing that was filled randomly through the top movable steel plate into the flow passage of the duct. The properties of the packing are given in Table 2.

Experiments were conducted for forced convection air flow in the packed duct under different mass flow rates and different heat fluxes to determine the thermal performance of the system. Heating started at the end of the calming section, giving a heated packed section of $L/H = 16$ for both Cases of I and II. Pressure drop, axial and transverse temperature profiles and heat fluxes were measured at steady state and recorded when two complete sets of temperature data showed no differences to within the experimental accuracy of ± 0.2°C for 30 min. The maximum temperature difference between the top plate and air flow at the exit was approximately 20°C. This was achieved by adjusting the power input to the resistance heater according to the air flow rate specified. The net increase in the air flow temperature was not larger than approximately 15°C. The experiments were also conducted with same ducts with empty flow passages to determine the effect of packing on the thermal performance. A total of 101 experimental runs, including the repeated ones, for packed flow passage, and 28 experiments for the empty duct were conducted. The experimental uncertainty in the Nusselt numbers mainly due to errors in heat balance and temperature readings are estimated to be in the range from 8 to 17%. The larger uncertainties are due to heat loss at relatively high heat fluxes and are due to smaller temperature differences between the wall and air flow temperatures.

3. Numerical procedures

A quasi-homogeneous, two-dimensional model, that is usually recommended for forced convection in packed beds [17], with an average void fraction was employed. It was assumed that the system is at steady state and fully developed, air and solid are in local thermal equilbrium, the axial dispersion of heat and the viscous effect in the flow are negligible [23,24], and the physical properties are constant. The governing equations are

$$\frac{dP}{dx} = \mu \frac{d^2u}{dy^2} - C\frac{1-\varepsilon}{d_p\varepsilon^3}\rho u^2 \tag{1}$$

$$u\frac{dT}{dx} = \frac{1}{\alpha_e}\frac{\partial^2 T}{\partial y^2} \tag{2}$$

where C is a constant and determined as 4.27 [25] for Raschig ring and taken as 1.75 for the spherical packing. Due to geometry of Raschig ring and the rectangular duct, a modification to Ergun equation [26] is consistent with the fact that Ergun equation predicts pressure drop lower than actual data for cylindrical packing [27]. By neglecting the viscous effect [23], the pressure drop for Raschig ring is obtained from

$$-\frac{dP}{dx} = 4.27\frac{G^2(1-\varepsilon)}{d_p\rho\varepsilon^3} \tag{3}$$

The maximum deviation between experimental and calculated pressure drop data is 9%.

The inlet and boundary conditions of Eqs. (1) and (2) are

$$u = 0, \quad \text{at} \quad y = 0 \tag{1a}$$

$$\frac{du}{dy} = 0, \quad \text{at} \quad y = H/2 \tag{1b}$$

$$T = T_0, \quad \text{at} \quad x = 0 \tag{2a}$$

$$\frac{\partial T}{\partial y} = -\frac{Q}{k_e}, \quad \text{at} \quad y = 0 \tag{2b}$$

$$\frac{\partial T}{\partial y} = 0, \quad \text{at} \quad y = H \tag{2c}$$

The energy equation coupled with the velocity field was solved using the Crank–Nicolson method with fifty grids in radial direction to determine the temperature field using an average void fraction, ε. The solution procedure is given in detail elsewhere [28]. The equivalent diameters of the duct, d_e and Raschig ring, d_p are given by [29]

$$d_e = \left[2.55K\frac{(WH)^2}{W+H}\right]^{1/3}, \quad d_p = \frac{6(1-\varepsilon)}{\Phi a_p} \tag{4}$$

where K is a constant that depends on W/H for a rectangular duct, and Φ is the shape factor for an irregular packing [29]. The effective thermal conductivity, k_e, in the effective thermal diffusivity, α_e, is expressed in terms of static, k_{st} and dynamic, k_{dy} contributions due to stagnant fluid and fluid flow, respectively

$$k_e = k_{st} + k_{dy} \tag{5}$$

The stagnant and dynamic thermal conductivities are given by [30]

$$k_{st} = \left[\varepsilon + (1-\varepsilon)\left(0.18 + \frac{2}{3}\frac{k_f}{k_p}\right)^{-1}\right] \tag{6}$$

$$k_{dy} = \frac{0.0025 Re_p}{1 + 46(d_p/d_e)^2} \tag{7}$$

4. Results and discussion

Experiments were conducted with the air flow superficial velocity range of 0.025–0.60 m/s, heat flux range of 50–350 W/m^2, and top wall temperature range of 35–60°C. A steady state is usually reached within an average time ranging from 4 to 6 h. All the heat transfer data is presented in terms of the Nusselt and the Reynolds numbers based on the separation distance of the heated and adiabatic walls, H, or the diameter of the particle, d_p. The Prandtl number is assumed to be constant. The Nusselt numbers based on H and d_p are

$$Nu_H = \frac{hH}{k_f} \tag{8}$$

$$Nu_p = \frac{hd_p}{k_f} \tag{9}$$

The Reynolds numbers based on H and d_p are

$$Re_H = \frac{uH}{\nu_f} \tag{10}$$

$$Re_p = \frac{ud_p}{\nu_f} \tag{11}$$

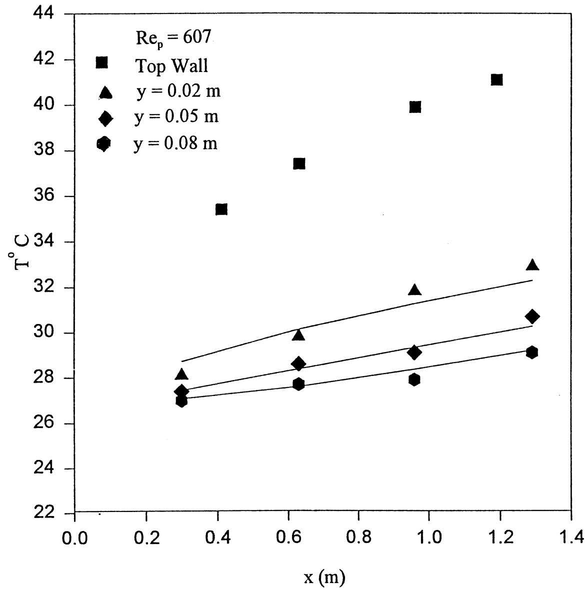

Fig. 2. Typical axial temperature profiles of air flow in the Case II duct packed with spheres of d_p = 2.9 cm at Re_p = 607.

where u is the superficial velocity, ν_f is the kinematic viscosity and k_f is the thermal conductivity of the fluid.

The numerical model with an average void fraction yields a plug flow type velocity distribution in the packed duct except near the wall region [28]. The typical measured and predicted axial temperature profiles of air flow and top wall temperatures in the packed duct of the Case II filled with spheres d_p = 2.9 cm are given in Fig. 2. The typical experimental and calculated values of transverse temperature distributions of

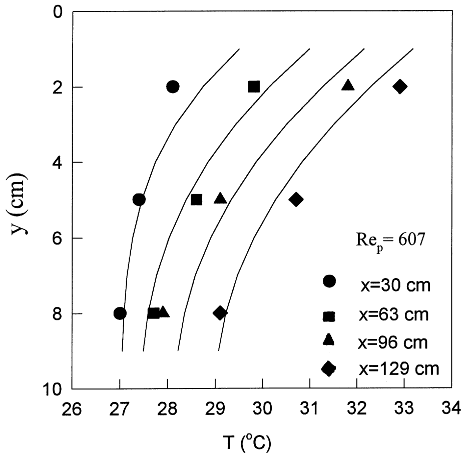

Fig. 3. Typical radial temperature profiles of air flow in the Case II duct packed with spheres of d_p = 2.9 cm at Re_p = 607.

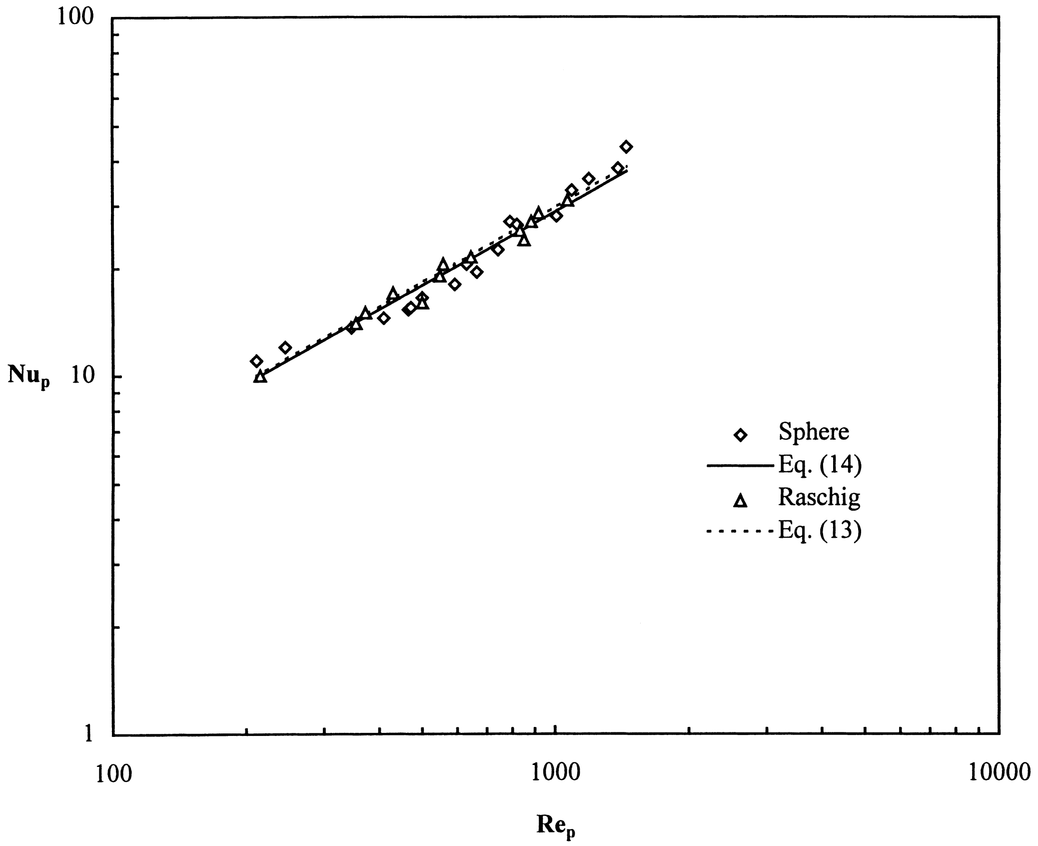

Fig. 4. Average Nusselt numbers based on the packing diameter, Nu_p for both Cases of I and II.

air flow for packed duct of the Case II filled with spheres of $d_p = 2.9$ cm are shown in Fig. 3. Numerical representations of temperature profiles are poor in the vicinity of heated wall mainly due to sharp drop in fluid temperature and assumption of constant void fraction in the model [12,15].

The direct measurements of heat flux and temperature of air flow have been used to calculate the values of Nu_p. The fluid temperature measurements at the end of the stations are fitted with a quadratic function and the bulk temperature of fluid flow, T_b, is obtained as

$$T_b = \frac{\int_A T\,dA}{\int_A dA} \tag{12}$$

A plot of Nu_p versus Re_p for the two types of packing and for Cases I and II is presented in Fig. 4. A steady increase of Nu_p with Re_p is observed, although the values of Nu_p for Raschig ring are slightly higher than that of spherical packing. This is line with the findings of Dixon [14], and Borkink and Westerterp [16] for spheres and Raschig rings. A correlation equation based on the results of the 20 experimental runs for the cylindrical packing is obtained as

$$Nu_p = 0.24 Re_p^{0.7},\ 5.6 < d_e/d_p < 9$$
$$(0.11 < d_p/d_e < 0.18),\ 200 < Re_p < 1450 \tag{13}$$

A correlation equation based on the results of the 33 experimental runs for the spherical packing is obtained as

$$Nu_p = 0.25 Re_p^{0.69},\ 4.5 < d_e/d_p < 7.$$
$$(0.13 < d_p/d_e < 0.22),\ 200 < Re_p < 1450 \tag{14}$$

The correlation Eqs. (13) and (14) are shown in Fig. 5 along with the various other correlations reported in the literature. It can be observed that the present results indicate slightly lower values of Nu_p than those of the previous correlations. The present measurements are for the packing with very low thermal conductivity that may influence the radial thermal dispersion adversely [13], hence may cause low values of Nu_p. The

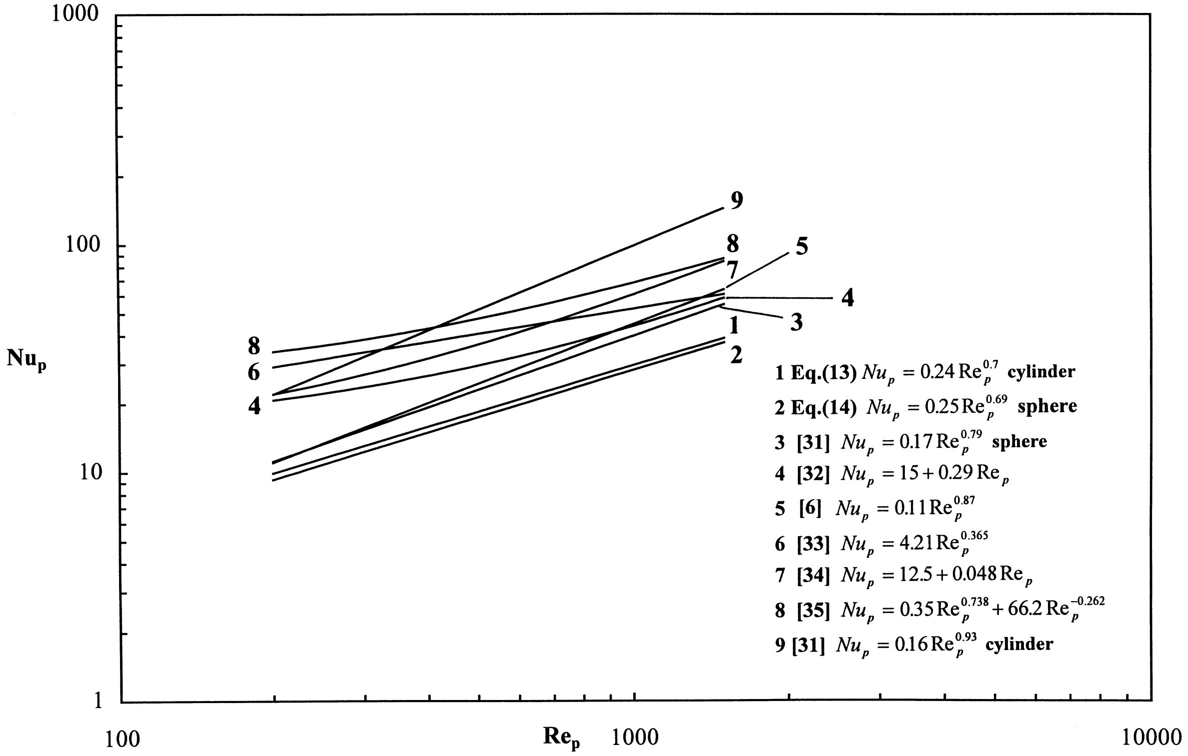

Fig. 5. Correlation equations for Nu_p. 1: $5 < d_e/d_p < 9$, cylinder, polyvinyl choride; 2: $4 < d_e/d_p < 8$, sphere, expanded polystyrene; 3: $d_t/d_p = 10$, cylinder, metal; 4: $3 < d_t/d_p < 5$, sphere, celite; 5: $9 < d_e/d_p < 20$, sphere, glass, chrome steel; 6: $14 < d_t/d_p < 28$, cylinder, celite, $8 < d_t/d_p < 16$, sphere, alundum; 7: $d_t/d_p = 17$, cylinder; 8: $5 < d_t/d_p < 50$, sphere, cylinder, ceramic; 9: $d_e/d_p > 10$, cylinder.

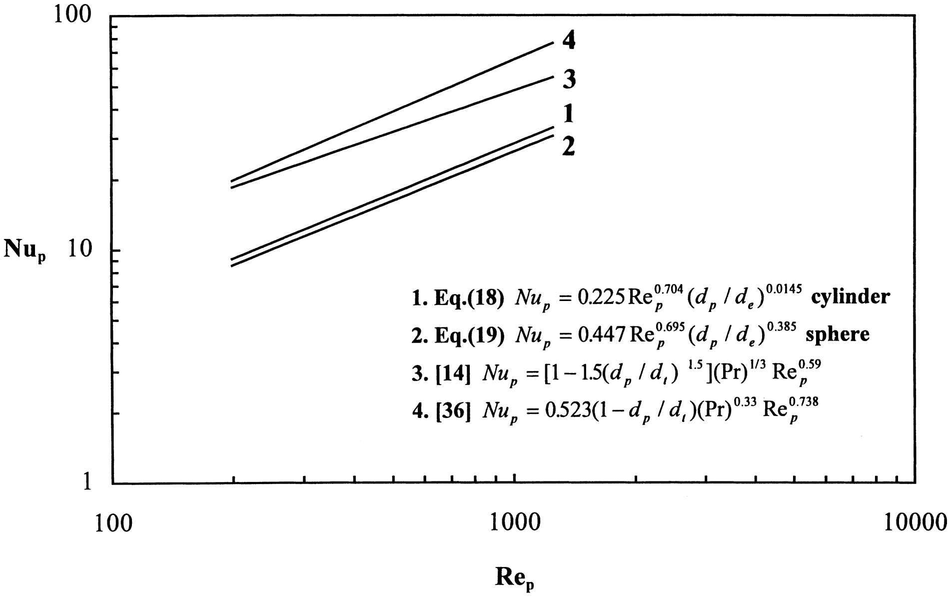

Fig. 6. Correlation equations for Nu_p in terms of Re_p and d_p/d_e or d_p/d_t.

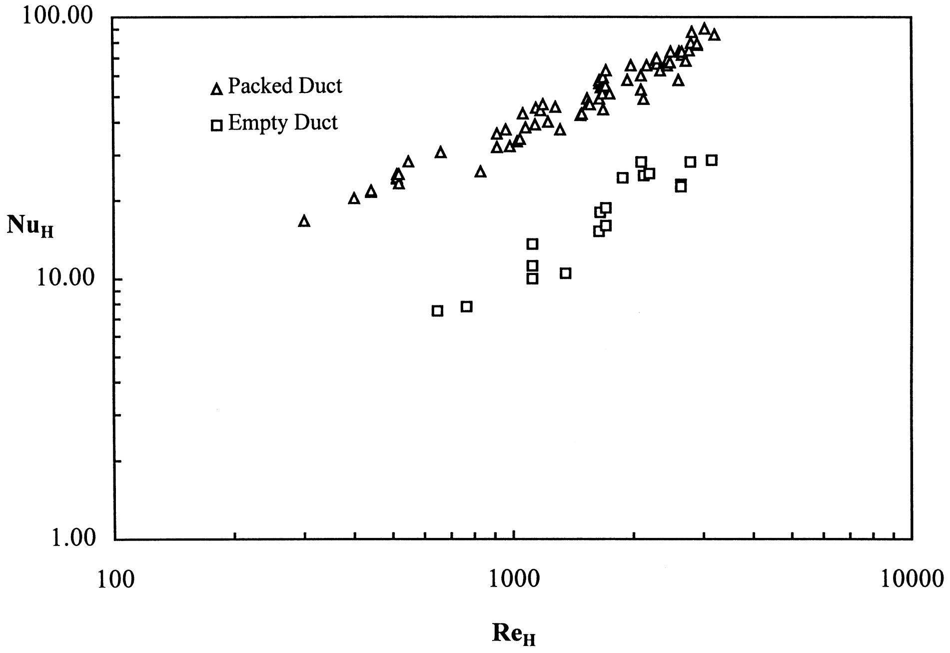

Fig. 7. Average Nusselt numbers based on the separation distance, Nu_H in the ducts of Cases I and II with and without packing.

equations shown in Fig. 5 are the results of the wide range of experimental conditions and show the discrepancies in the correlations. However the dependencies of Re_p, as indicated by the slopes of the curves, are more or less close to one another except the one given for cylindrical packing by Li and Finlayson [31].

The fluid phase transfer parameters can be obtained from mass transfer experiments by analogy. The most accurate estimates of the wall Sherwood group have been obtained from the electrochemical method for the determination of mass transfer rates. For water Dixon et al. [13] obtained

$$Nu_p = [1 - 1.5(d_t/d_p)^{-1.5}](Pr)^{1/3} Re_p^{0.59} \tag{15}$$

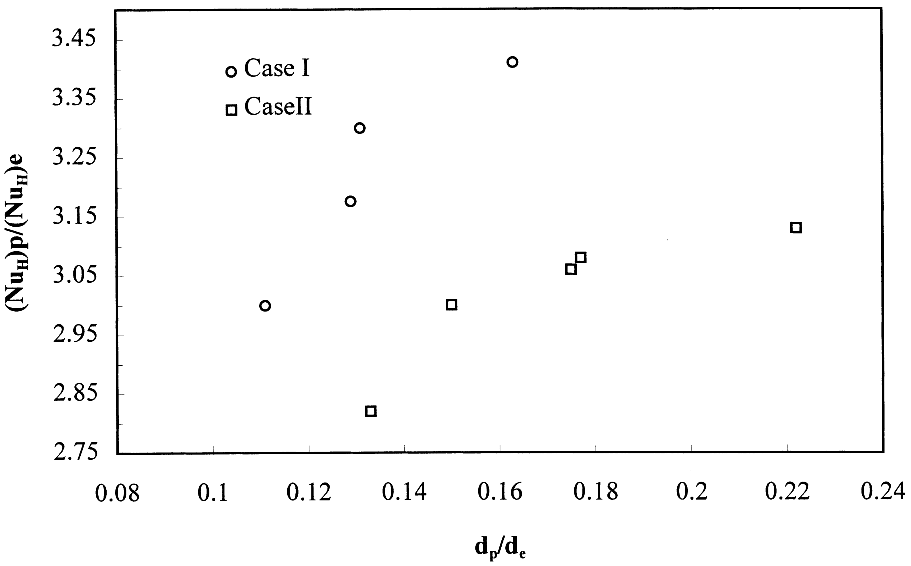

Fig. 8. Change of the ratio of Nu_H in packed-to-empty duct with d_p/d_e.

For gaseous systems Colledge and Paterson [36] obtained

$$Nu_{\rm p} = 0.523(1 - d_{\rm p}/d_{\rm t})(Pr)^{0.33} Re_{\rm p}^{0.738} \tag{16}$$

From the present measurement a correlation equation in terms of $Re_{\rm p}$ and $d_{\rm p}/d_{\rm e}$ for Raschig ring is obtained as

$$Nu_{\rm p} = 0.225 Re_{\rm p}^{0.704}(d_{\rm p}/d_{\rm e})^{0.0145} \tag{17}$$

Eq. (17) is valid for $0.11<d_{\rm p}/d_{\rm e}<0.18$ or $5.6<d_{\rm e}/d_{\rm p}<9.0$ and for $200<Re_{\rm p}<1450$. A correlation equation for the spherical packing is also obtained as:

$$Nu_{\rm p} = 0.447 Re_{\rm p}^{0.695}(d_{\rm p}/d_{\rm e})^{0.385} \tag{18}$$

Eq. (18) is valid for $0.13<d_{\rm p}/d_{\rm e}<0.22$ or $4.5<d_{\rm e}/d_{\rm p}<7.7$ and for $200<Re_{\rm p}<1450$. Eqs. (15)–(18) have been plotted at $d_{\rm p}/d_{\rm e}=0.15$ and shown in Fig. 6. The values of $Nu_{\rm p}$ obtained from Eqs. (17) and (18) are lower than those obtained from the analogy of mass transfer. A plot of Nu_H vs Re_H for the air flow passage with and without packing is presented in Fig. 7. It is shown that the values of Nu_H obtained from all packing are about three times higher for a packed duct than that of an empty duct. Similar results from steel and glass spheres are also reported by Hwang et al. [6]. As seen from Fig. 7, the transverse thermal dispersion effect yields the enhanced heat transfer rate. In Fig. 8, the ratio of the Nusselt numbers, based on the separation distance and obtained from all packing, for the packed duct to the empty duct, $(Nu_H)_{\rm p}/(Nu_H)_{\rm e}$, versus $d_{\rm p}/d_{\rm e}$ is plotted for the Case I at $Re_H=1700$ and for the Case II at $Re_H=2600$. It was shown that the ratio of $(Nu_H)_{\rm p}/(Nu_H)_{\rm e}$ steadily increases as $d_{\rm p}/d_{\rm e}$ increases, although the increase slows down at higher values of $d_{\rm p}/d_{\rm e}$.

5. Conclusions

Flow and heat transfer characteristics have been investigated experimentally and numerically for the large rectangular packed ducts with $L/H=16$, and with $W/H=8$ and 4. In the horizontally oriented ducts Raschig rings and spherical packing with low thermal conductivity and with the range $0.11 < d_{\rm p}/d_{\rm e} < 0.22$ are used in the air flow passage. The ducts are heated from the top wall by uniform heat flux, while the other walls are insulated. Pressure drop, heat flux, temperature distribution of air flow and the Nusselt number at various Reynolds number were measured. Flow and temperature fields are obtained numerically from the quasi-homogeneous two-dimensional model with an average void fraction. It was found that the plug flow prevailed in the core region of the packed duct. The influence of the shape of packing on the heat transfer is marginal, and the Nusselt numbers obtained for Raschig ring and spherical packing are close to each other. The Nusselt numbers based on the particle diameter have been correlated in terms of the Reynolds number as $Nu_{\rm p}=0.24Re_{\rm p}^{0.7}$; $0.11 < d_{\rm p}/d_{\rm e} < 0.18$; for Raschig ring, and $Nu_{\rm p}=0.25Re_{\rm p}^{0.69}$; $0.13 < d_{\rm p}/d_{\rm e} < 0.22$ for the sphere when the $Re_{\rm p}$ change in the range $200 < Re_{\rm p} < 1450$. The correlation equations in terms of $Re_{\rm p}$ and $d_{\rm p}/d_{\rm e}$ for Raschig ring are obtained as $Nu_{\rm p}=0.225Re_{\rm p}^{0.704}(d_{\rm p}/d_{\rm e})^{0.0145}$ that is valid for $0.11 < d_{\rm p}/d_{\rm e} < 0.18$, and for the spherical packing as $Nu_{\rm p}=0.447Re_{\rm p}^{0.695}(d_{\rm p}/d_{\rm e})^{0.385}$ that is valid for $0.13 < d_{\rm p}/d_{\rm e} < 0.22$. It was found that the forced convection heat transfer from wall-to-fluid in the packed ducts is about three times higher than that in an empty duct at the same Reynolds number.

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