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An experimental study of forced convection in a packed channel with asymmetric heating

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Abstract—An experiment has been carried out for forced convection of Freon-113 ($Pr = 8.06$) in a channel packed with small glass spheres (3, 5 and 6 mm in diameter) and with chrome steel spheres (6.35 mm in diameter), respectively. The experiments were carried out for $4000 < Re_h < 17\,000$ and $0.06 < \gamma < 0.13$ in a packed channel with $L/H = 9$. Both the heat flux to the wall and transverse temperature profiles inside the channel were measured. It is found that the Nusselt number increases as the particle diameter is decreased. The effects of the thermal conductivity ratio of the solid to fluid phase on the Nusselt number and temperature distribution are small for the range of the Reynolds number considered. Based on the 28 experimental runs, a correlation equation for the Nusselt number is obtained as $Nu_h = 0.31 Re_h^{0.814}\,e^{-1.63\gamma}$. It is also found that heat transfer in a packed channel is approximately three times higher than that in an empty channel for Re_h ranging from 5000 to 17 000.

1. INTRODUCTION

FORCED convection in packed tubes and channels has been the subject of intensive study in chemical engineering literature during the past six decades (see Cheng *et al.* [1] for a review of literature). As early as 1931, Colburn [2] found that the heat transfer rate for the forced convection of air through a packed tube is about eight times higher than that of an empty tube with a length to tube diameter ratio ranging from 17.8 to 36.0. The substantial increase in the heat transfer rate has been attributed to the mixing of fluid owing to the presence of the solid matrix, which is known as the thermal dispersion effect. During the ensuing years, more than 30 experiments have been reported on the forced convection of air through cylindrical and annular packed columns in chemical engineering literature [1].

The purpose of the early experiments was to obtain the appropriate heat transfer parameters for the numerical simulation of the performance of wall-cooled exothermic catalytic reactors, which was usually based on a plug flow model. For this purpose, correlation equations of the average effective radial thermal conductivity over the cross-section and the average wall heat transfer coefficient as a function of the appropriate dimensionless parameters were needed. In chemical reactor literature, these heat transfer parameters were usually obtained by matching the temperature data at the exit of the packed column with a theoretical model based on a plug flow with the assumption of a temperature slip at the wall of the heated packed tube. Correlation equations were obtained for the effective radial thermal conductivity and the wall Nusselt number as a function of the Reynolds numbers. It was found that while the effective radial thermal conductivity is linearly proportional to the Reynolds number, the exponent of the Reynolds number in the correlation equation for the wall Nusselt number varies widely from 0.33 to 1.0 [3, 4].

Li and Finlayson [5] attributed the disparity in the correlation equations for the wall Nusselt number to the entrance length effect while Dixon and Cresswell [6] attributed it to the failure of recognizing the additional parameters such as the dimensionless particle diameter (i.e. particle/tube diameter ratio) and fluid/particle thermal conductivity ratio. In a recent paper, Cheng *et al.* [1] have attributed the disparity of the wall Nusselt number correlation equations to the fact that the heat transfer rate was not measured directly, and to the method used for the determination of the wall Nusselt number in chemical reactor engineering literature.

In recent years, a number of experiments has been performed for forced convection in packed channels with different thermal boundary conditions. For example, Vafai *et al.* [7] performed an experiment on the forced convection of water in a channel filled with glass spheres; the channel was heated on one wall while the other walls were insulated. The average Nusselt numbers were measured at selected Reynolds numbers. Renken and Poulikakos [8, 9] carried out a similar experiment and reported the local Nusselt number at selected Reynolds numbers. Most recently, a similar experiment was conducted by Chrysler and Simons [10] for forced convection of Fluorocarbon-77 (with a Prandtl number of 22) in a packed channel filled with four sizes of glass spheres and with a channel length-to-plate separation distance (L/H) of 4.62.

NOMENCLATURE

B	constant in equation (8)
d_p	particle diameter
H	separation distance between the heated and cooled plates
k_d	stagnant thermal conductivity of the packed channel
k_f	thermal conductivity of the fluid phase
k_s	thermal conductivity of the solid phase
L	length of the packed channel
Nu_d	average Nusselt number based on the particle diameter and thermal conductivity of the fluid
Nu_{dd}	average Nusselt number based on the particle diameter and the stagnant thermal conductivity of the saturated packed bed
Nu_h	average Nusselt number based on the separation distance and the thermal conductivity of the fluid
q_w	average heat flux
Re_d	Reynolds number based on particle diameter and mean velocity in the packed channel
T_c	temperature of the cooled wall
T_h	temperature of the heated wall
u_m	mean velocity in the packed channel.

Greek symbols

γ	dimensionless particle diameter
Λ	thermal conductivity ratio of the fluid to the solid phase
μ_f	dynamic viscosity of the fluid
ρ_f	density of the fluid
ϕ	porosity of the packed channel.

A correlation equation for the average Nusselt number in terms of the Reynolds number was obtained. They reported that the heat transfer rate in a packed channel is about 10 times higher than that of an empty one.

As far as the authors are aware, the only published experimental work on forced convection in a packed channel heated asymmetrically was performed by Schroeder *et al.* [11]. The experiments were performed for the forced convection of water in a channel filled with glass spheres. At selected Reynolds numbers, transverse temperature profiles at the exit of the channel were measured but the heat flux was not measured. Correlations of the Nusselt number as a function of the Reynolds number were obtained indirectly based on the conventional method used in chemical reactor engineering literature by a comparison of transverse temperature data with a plug flow model as discussed earlier.

In this paper, an experiment was carried out to investigate forced convection of Freon-113 (with a Prandtl number of 8.06) in a channel (with $L/H = 9$) heated asymmetrically and packed with glass spheres and chrome steel spheres. Both transverse temperature profiles and heat flux to the wall were measured. It was found that effects of thermal conductivity of particles on heat transfer characteristics are small for $Re_h \geqslant 4000$. Based on 28 experimental runs, a correlation equation for the Nusselt number was found to be given by $Nu_h = 0.31 Re_h^{0.814}\, e^{-1.63\gamma}$, where γ is the ratio of particle diameter to the separation distance between the heated and cooled walls. This correlation equation is valid for forced convection of Freon-113 in a packed channel for $4000 \leqslant Re_h \leqslant 17\,000$, $0.059 \leqslant \gamma \leqslant 0.125$, and the channel length-to-separation distance ratio of $L/H = 9$.

2. EXPERIMENTAL APPARATUS AND PROCEDURE

A forced convection Freon-113 loop was used for the experiments in this study. Details of the loop are described elsewhere [12]. A sketch of the test section is shown in Fig. 1. The test chamber is vertically oriented with Freon-113 flowing against gravity. The test section consists of four vertical walls: two aluminum plates serving as the heat source and the heat sink, and two acrylic plates serving as adiabatic side walls. The overall dimensions of the test chamber are 660.4 mm in length, 203.2 mm in width and 304.8 mm in depth. The cross-sectional flow area of the channel is 50.8×152.4 mm. The heated and cooled plates, 457.2 mm in length (L) and facing opposite from each other, are separated by a distance (H) of 50.8 mm. This heat transfer section is preceded by a calming section of 101.6 mm in length, and followed by an exit section of 50.8 mm in length. Both the calming and exit sections are made of acrylic plates. A perforated plate was installed on the bottom of the test chamber to promote a more uniform velocity profile entering the calming section while a top perforated plate was used to hold the spheres in place during the experiments. Both the walls and the perforated plates are removable so that the spheres can be packed easily into the test chamber.

An aluminum plate (203.2×457.2 mm), 31.75 mm thick, served as the heated plate. The back of the aluminum plate is mounted with five strip heaters.

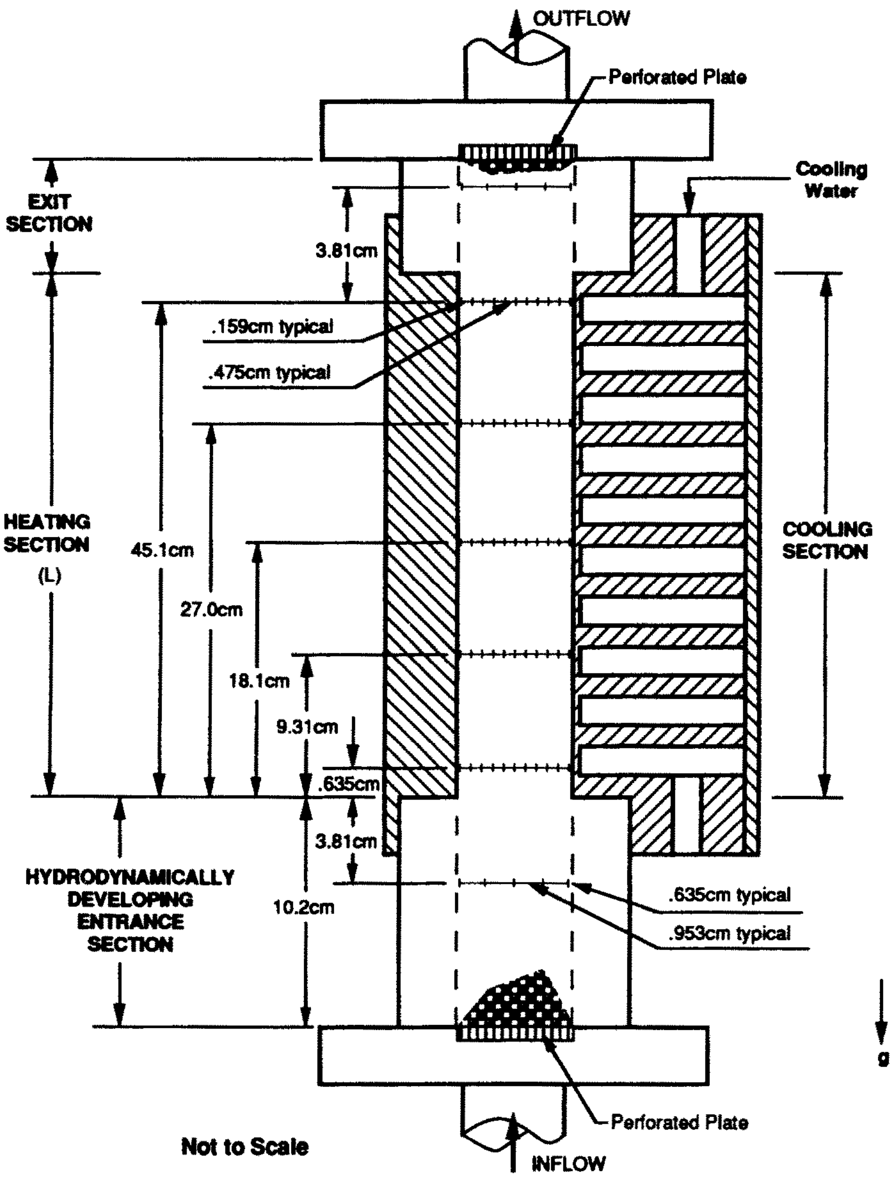


FIG. 1. Schematic of test section.

Each heater has a maximum heating capacity of 350 W which equates a heat flux of 31 kW m^{-2}. The energy output of each strip heater was controlled by a rheostat. The entire heated test section is insulated with fiber glass. The accuracy of the power input to the heated plate is estimated to be $\pm 3\%$. The cooling jacket was made from a 76.2 mm thick aluminum plate. A 68.3 mm deep zigzag track, which was machined into the rearside of the cooled plate, provided the cooling water passage. Twelve J-type thermocouple probes (1.59 mm in diameter) in each plate were positioned at a distance of 1.59 mm from the inner walls to monitor the temperatures of the heated and cooled plates.

Two acrylic plates ($25.4 \times 101.6 \times 610$ mm), separating the heated and cooled plates, served as thermal insulation to ensure that heat transfer characteristics are two-dimensional in the test section. These clear acrylic plates allowed flow visualization and ensured gas bubbles did not exist in the test section. Five levels of J-type thermocouple probes (0.81 mm in diameter), located at 6.35, 9.21, 181, 270 and 451 mm from the inlet, were mounted on the acrylic plates. Each level has 11 thermocouples in the transverse direction with an even spacing of 4.76 mm except those near the walls. The thermocouples near the walls are located 1.59 mm from the walls. The thermocouple probes, installed in parallel with the heated and cooled plates, were used to measure the transverse fluid temperature in the packed channel. The heat loss through these two acrylic plates was estimated to be less than 1% of the power input to the heated plate.

A data acquisition system consisting of an IBM PS/2 computer, a MicroChannel architecture analog input board, six universal analog input multiplexers, and a screw terminal accessory board, was employed to record and display the flow rate of Freon-113 and temperatures from the thermocouples that were embedded in the porous medium and positioned in the heated and cooled plates. The accuracy of the flow rate measurement is $\pm 0.5\%$. The experimental accuracy of the temperature is $\pm 0.2°C$. Experimental uncertainties in the three Nusselt numbers Nu_h, Nu_d and Nu_{dd} (defined in equations (4), (6) and (7) below) are estimated to be ± 3.7, ± 4.3 and $\pm 4.8\%$, respectively.

The porous media used for the experiments were soda lime glass spheres with uniform diameters of 3, 5 and 6 mm as well as chrome steel spheres with a diameter of 6.35 mm. The heated wall was maintained at 38.5°C while the cooled wall was maintained at 22.5°C. Freon-113 entering the test section was maintained at 5 psig and 30.5°C (a subcooled liquid), which was the average temperature of the heated and cooled walls. These thermal boundary conditions ensured that the fluid was heated and cooled asymmetrically in the test section.

Experiments were first conducted for forced convection of Freon-113 in an empty channel which was maintained at the same thermal conditions mentioned above. By increasing the flow rate, the Reynolds number of the flow was increased from 5400 to 23 000.

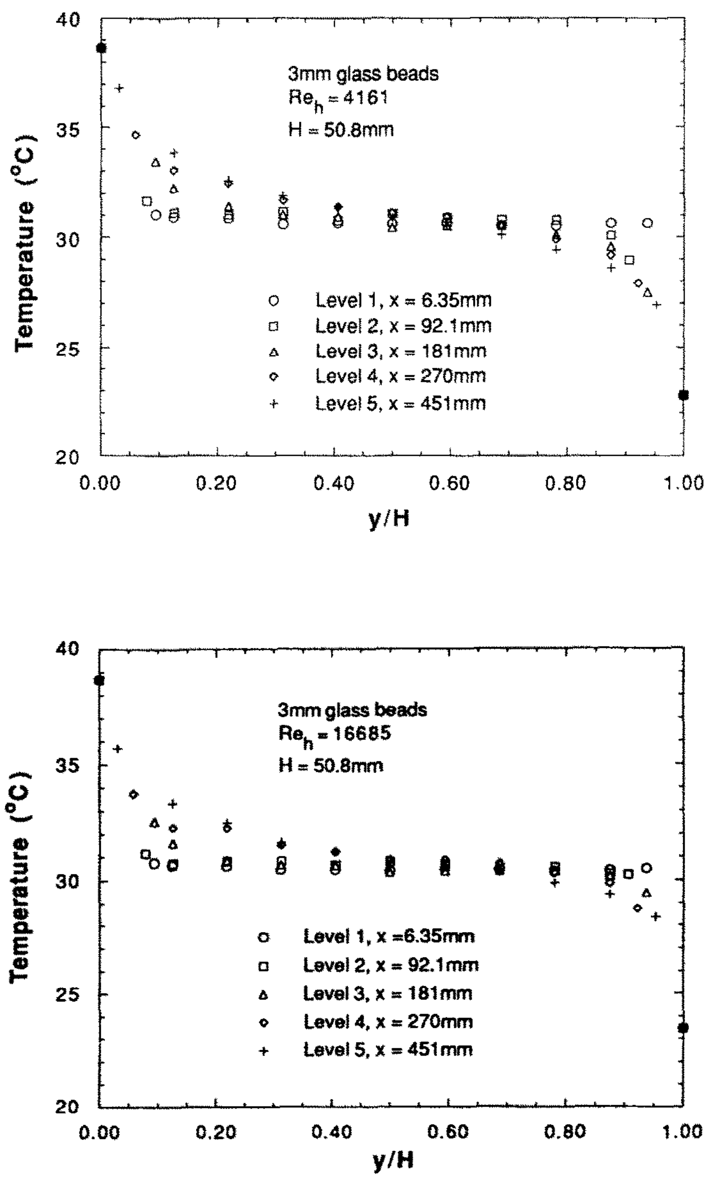


FIG. 2. Transverse temperature distributions in a packed channel filled with 3 mm glass spheres.

Both the transverse temperature profiles and heat flux to the wall were measured at a particular Reynolds number. Experiments were then conducted for forced convection of Freon-113 in the same channel filled with glass spheres having diameters of 3, 5 and 6 mm, respectively, and with chrome steel spheres having a diameter of 6.35 mm. The experiments were carried out for Reynolds number ranging from 2000 to 17 000.

3. DATA ANALYSIS AND RESULTS

All of the heat transfer data will be presented in terms of the Reynolds number. The Reynolds number in porous media flow can be defined based on two characteristic lengths: the separation distance of the heated and cooled walls, H, or the diameter of the solid spheres d_p. The Reynolds number Re_h based on the separation distance is

$$Re_h = \frac{\rho_f u_m H}{\mu_f}, \tag{1}$$

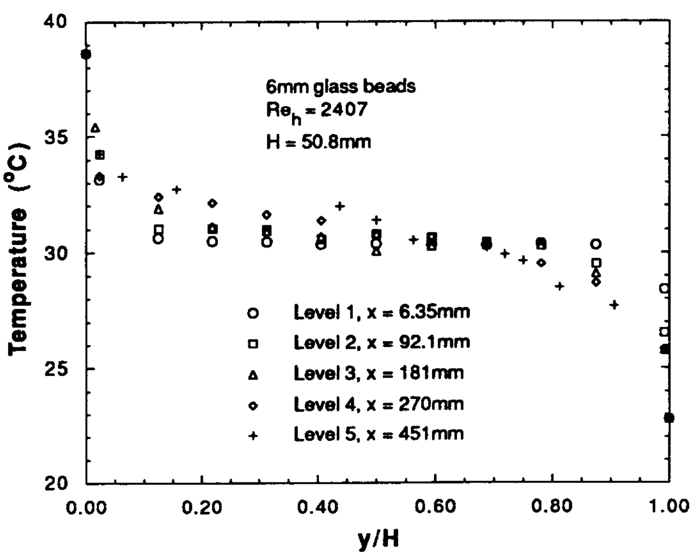
where ρ_f and μ_f are the density and the dynamic viscosity of the fluid, and u_m is the mean velocity in the packed channel. The Reynolds number Re_d based on the particle diameter is

$$Re_d = \frac{\rho_f u_m d_p}{\mu_f} \tag{2}$$

which is related to Re_h by

$$Re_d = \gamma\ Re_h, \tag{3}$$

where $\gamma = d_p/H$ is the dimensionless particle diameter.



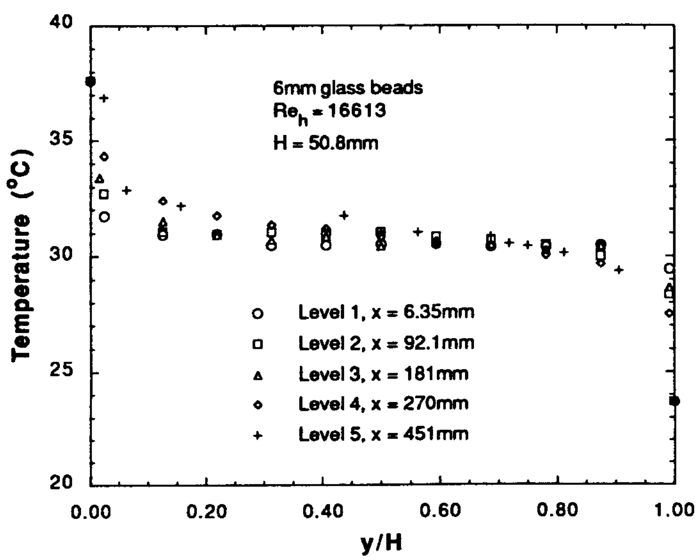


FIG. 3. Transverse temperature distributions in a packed channel filled with 6 mm glass spheres.

3.1. *Temperature profiles*

Transverse temperature profiles at different axial positions in the packed channel filled with 3 and 6 mm glass spheres are presented in Figs. 2 and 3 at low and high Reynolds numbers. Since the inlet fluid temperature is the average temperature of the heated and cooled walls, the fluid is heated in the left half of the channel and is cooled in the right half of the channel (see Fig. 1). The transverse temperature data in the channel packed with chrome steel spheres 6.35 mm in diameter are presented in Fig. 4. It is observed that it takes longer for the fluid through the packed channel to develop thermally to a linear temperature profile at higher Reynolds numbers. The trend holds true for different sphere sizes. There is only a slight difference in the temperature distributions in the packed channel filled with glass spheres or chrome steel spheres in the Reynolds number range under consideration. When examining the temperature data, it should be noted that (i) the precise locations of the thermocouples are uncertain and (ii) it is uncertain as to whether fluid or solid temperatures were measured.

A comparison of the transverse temperature profiles at $x = 451$ mm (from the inlet) in the channel with and without porous media at $Re_h = 10\,000$ is presented in Fig. 5. It is shown that the temperature gradient at the wall decreases when porous media are packed in the channel.

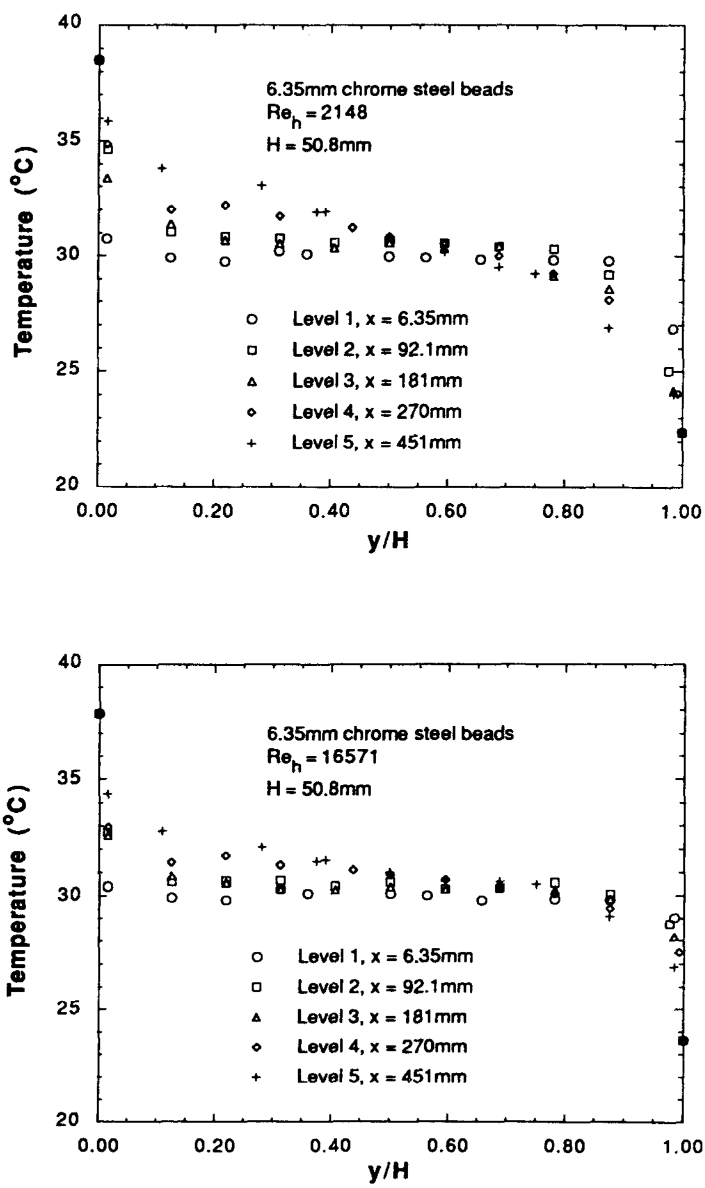


FIG. 4. Transverse temperature distributions in a packed channel filled with 6.35 mm chrome steel spheres.

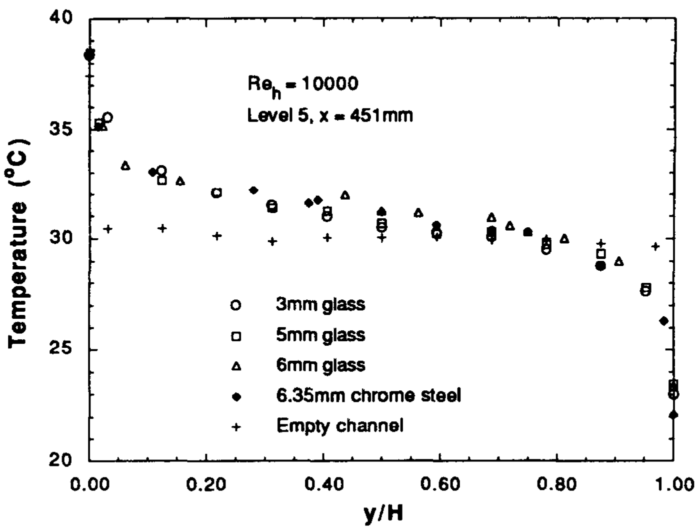


FIG. 5. Comparison of transverse temperature distributions in a packed channel.

3.2. *Nusselt numbers*

The average Nusselt number for forced convection in a packed channel with asymmetric heating can be defined based on the plate separation distance H and the fluid thermal conductivity k_f ($k_f = 0.0744$ W m^{-1} K^{-1} for Freon-113 at 30°C) as

$$Nu_h = \frac{q_w H}{k_f(T_h - T_c)} \tag{4}$$

where q_w is the average heat flux on the heated wall to the packed channel while T_h and T_c are the temperatures of the heated and cooled walls. A plot of Nu_h vs Re_h for a channel with and without porous media is presented in Fig. 6. It is shown that as the value of Re_h is increased, the value of Nu_h varies gradually from a horizontal line (i.e. conduction predominant) to an inclined straight line where convection becomes predominant. The value of Nu_h decreases as the particle size is increased although its effect is small for $0.059 \leqslant \gamma \leqslant 0.125$. The effect of thermal conductivity of solid particles is shown to have a negligible effect on the value of Nu_h at high values of Re_h, which is consistent with previous observations by Yagi *et al.* [13]. This can be attributed to the fact that convection rather than conduction is predominant at high Reynolds numbers. Based on the results of the 28 experimental runs listed in Table 1, the following correlation for the value of Nu_h is

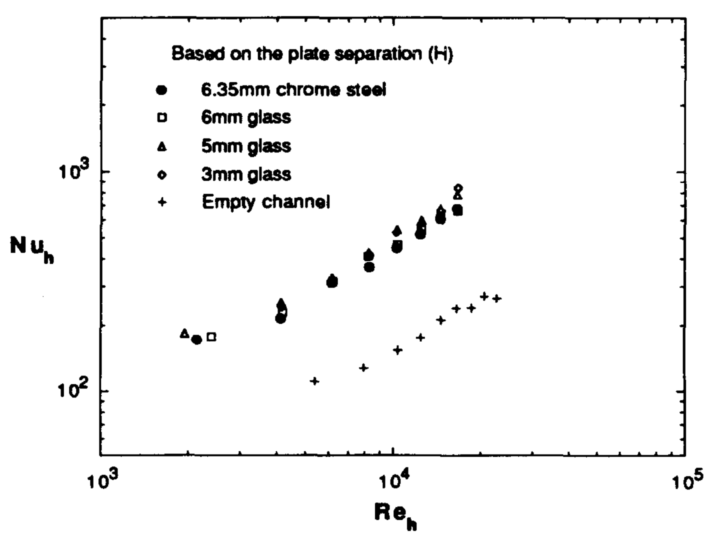


FIG. 6. Average Nusselt number Nu_h in a channel with and without porous media.

Table 1. Experimental data

Run No.	GPM	$T_h - T_c$ (°C)	q_w (W m^{-2})	Re_h	Nu_h
3 mm glass beads, $\phi = 0.3332$, $\gamma = 0.0591$, $k_d = 0.4296$ W m^{-1} K^{-1}					
r148	4.01	15.86	5629	4161	242
r156	5.99	15.67	7358	6215	321
r161	8.03	14.48	8802	8332	415
r166	9.96	15.37	11 890	10 335	528
r169	11.99	14.42	12 119	12 441	574
r172	14.06	14.51	14 056	14 589	662
r176	16.08	15.21	18 709	16 685	840
5 mm glass beads, $\phi = 0.3465$, $\gamma = 0.0984$, $k_d = 0.4192$ W m^{-1} K^{-1}					
r177	1.88	16.11	4357	1951	185
r188	3.99	14.95	5590	4140	255
r196	5.96	14.92	7196	6184	329
r201	7.98	14.73	9215	8280	427
r206	10.00	15.03	11 928	10 376	542
r209	12.05	13.88	12 149	12 504	598
r212	14.02	14.04	13 827	14 548	673
r215	15.99	13.91	15 903	16 592	781
6 mm glass beads, $\phi = 0.3427$, $\gamma = 0.118$, $k_d = 0.4191$ W m^{-1} K^{-1}					
r104	2.32	15.48	4145	2407	179
r115	4.03	15.79	5287	4182	229
r124	5.98	15.40	7152	6205	317
r131	7.93	15.19	9143	8228	411
r135	9.99	15.03	10 216	10 366	464
r139	12.04	14.62	11 727	12 493	548
r142	13.99	14.84	13 506	14 517	621
r145	16.01	13.94	13 463	16 613	659
6.35 mm chrome steel beads, $\phi = 0.3423$, $\gamma = 0.125$, $k_d = 1.324$ W m^{-1} K^{-1}					
r261	2.07	16.11	4110	2148	174
r272	3.98	16.23	5131	4130	216
r280	5.94	15.31	7039	6164	314
r286	8.00	15.40	8280	8301	367
r291	9.96	15.26	10 009	10 335	448
r295	12.00	14.80	11 238	12 452	519
r299	14.01	14.67	13 094	14 537	609
r302	15.97	14.23	14 110	16 571	677
Empty channel, $k_f = 0.0744$ W m^{-1} K^{-1}					
pr16	5.19	15.70	2541	5385	111
pr18	7.67	15.46	2930	7959	129
pr10	10.05	15.39	3489	10 428	155
pr20	12.02	14.38	3734	12 472	177
pr12	14.07	16.54	5142	14 600	212
pr22	15.90	14.62	5122	16 498	239
pr24	17.86	13.99	4913	18 532	240
pr27	19.80	13.85	5503	20 545	271
pr28	21.85	15.56	6050	22 672	265

obtained:

$$Nu_h = 0.31 Re_h^{0.814} e^{-1.63\gamma} \quad (5)$$

where $\gamma = d_p/H$. Equation (5) is valid for the forced convection of Freon-113 at $4000 \leqslant Re_h \leqslant 17\,000$, $0.06 \leqslant \gamma \leqslant 0.13$, and $L/H = 9$. As shown in Fig. 6, the value of Nu_h is about three times higher for a packed channel than that of an empty channel. The enhanced heat transfer rate is due to the transverse thermal dispersion effect.

In chemical reactor engineering literature, the Nusselt number is usually expressed in terms of the particle diameter d_p and the thermal conductivity of the fluid k_f. Thus, we may define the Nusselt number as

$$Nu_d = \frac{q_w d_p}{k_f(T_h - T_c)}. \quad (6)$$

The porous media data in Fig. 6 are replotted in Fig. 7 where the values of Nu_d vs Re_d are presented. It is seen that the values of Nu_d increase as the particle diameter is increased. The effect of particle size on Nu_d is pronounced at low Reynolds numbers. The effect of thermal conductivity of particles, however, appears to be small.

The average Nusselt number can also be defined based on the particle diameter d_p and the stagnant thermal conductivity (k_d) of the packed channel as follows:

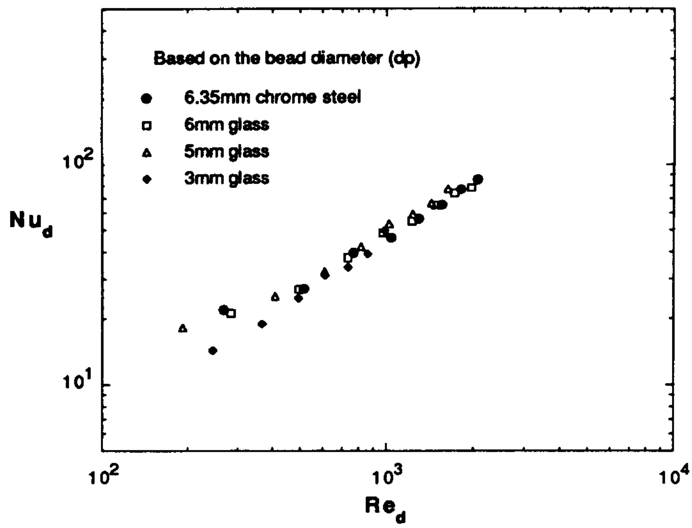


FIG. 7. Average Nusselt number Nu_d in a packed channel.

$$Nu_{dd} = \frac{q_w d_p}{k_d(T_h - T_c)} \tag{7}$$

where k_d is given by the semi-analytical formula [14]

$$\frac{k_d}{k_f} = 1-(1-\phi)^{1/2} - \frac{2(1-\phi)^{1/2}}{1-\Lambda B}\left\{\frac{(1-\Lambda)B}{(1-\Lambda B)^2}\ln(\Lambda B) + \frac{B+1}{2} + \frac{B-1}{1-\Lambda B}\right\} \tag{8}$$

with $\Lambda = k_f/k_s$ denoting the ratio of the thermal conductivity of the fluid phase to that of the solid phase, and $B = 1.25\{(1-\phi)/\phi\}^{10/9}$ for a packed-sphere bed where ϕ is the average porosity of the bed. It is relevant to note that $k_s = 1.4$ W m^{-1} K^{-1} for glass spheres and $k_s = 45$ W m^{-1} K^{-1} for chrome steel spheres.

A plot of Nu_{dd} vs Re_d for different sizes of the glass sphere and chrome steel spheres is presented in Fig. 8. It is shown that the shape of the curve for glass spheres in this graph is similar to those presented in Fig. 7. However, the Nusselt numbers Nu_{dd} for the chrome steel spheres are much lower than those of glass spheres. This is because the stagnant thermal conductivity of the F-113/chrome steel spheres system is much higher than that of the F-113/glass spheres system.

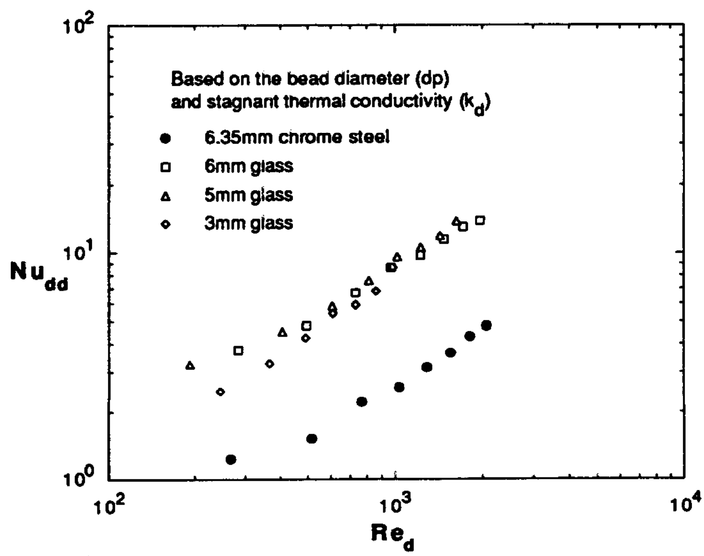


FIG. 8. Average Nusselt number Nu_{dd} in a packed channel.

4. CONCLUSIONS

An experimental investigation has been performed for the forced convection of Freon-113 in a packed channel heated asymmetrically. The channel was filled with glass spheres or chrome spheres of uniform size having diameters ranging from 3 to 6.35 mm. The experiments were carried out for a Reynolds number Re_h (based on the separation distance between the heated and cooled walls as a characteristic length) from 2000 to 17 000. Both temperature distributions in the packed channel and the Nusselt number of the wall at various Reynolds numbers and dimensionless particle diameters were measured. It was found that the effect of the thermal conductivity of the solid particles has a negligible effect on the Nusselt number. Based on the results of 28 experimental runs for $4000 \leqslant Re_h \leqslant 17\,000$ and $0.06 \leqslant \gamma \leqslant 0.13$, a correlation equation for the Nusselt number is obtained as $Nu_h = 0.31 Re_h^{0.814} e^{-1.63\gamma}$, where γ is the dimensionless particle diameter. It was found that the forced convective heat transfer of Freon-113 in a packed channel (with $L/H = 9$) is about three times higher than that in an empty channel at the same Reynolds number.

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ETUDE EXPERIMENTALE DE LA CONVECTION FORCEE DANS UN CANAL A LIT FIXE AVEC CHAUFFAGE ASYMETRIQUE

Résumé—Une expérimentation a été conduite sur la convection forcée du Freon 113 ($Pr = 8{,}06$) dans un canal garni de petites sphères de verre (3,5 et 6 mm de diamètre) ou de sphères d'acier au chrome (6,35 mm de diamètre). Les expériences correspondent à $4000 < Re_h < 17{,}000$ et $0{,}06 < \gamma < 0{,}13$ dans le canal avec $L/H = 9$. On mesure à la fois le flux thermique à la paroi et les profils transverses de température dans le canal. On trouve que le nombre de Nusselt augmente quand le diamètre de la particule diminue. Les effets du rapport des conductivités thermiques du solide et du fluide sur le nombre de Nusselt et la distribution de température sont faibles pour le domaine de nombre de Reynolds considéré. Une formule est obtenue à partir des 28 essais expérimentaux et $Nu_h = 0{,}31\ Re_h^{0.814}\,e^{-1.66\gamma}$. On trouve aussi que le transfert thermique dans un canal à lit fixe est approximativement trois fois plus grand que dans un canal vide pour Re_h variant entre 5000 et 17 000.

EXPERIMENTELLE UNTERSUCHUNG DER ERZWUNGENEN KONVEKTION IN EINEM SCHÜTTUNGSKANAL MIT ASYMMETRISCHER BEHEIZUNG

Zusammenfassung—Die erzwungene Konvektion von Freon 113 ($Pr = 8{,}06$) in einem mit Glaskugeln (Durchmesser 3, 5 und 6 mm) bzw. Chromstahlkugeln (Durchmesser 6,35 mm) gefüllten Kanal wird experimentell untersucht. Die Versuche wurden im Bereich $4000 < Re_h < 17\,000$ und $0{,}06 < \gamma < 0{,}13$ in einem Kanal mit dem Abmessungsverhältnis $L/H = 9$ durchgeführt. Gemessen wurden sowohl die Wärmestromdichte an der Wand als auch Temperaturprofile innerhalb des Kanals. Es zeigt sich, daß die Nusselt-Zahl mit abnehmenden Partikeldurchmesser ansteigt. Im untersuchten Bereich ist der Einfluß des Wärmeleitfähigkeitsverhältnisses zwischen fester und flüssiger Phase auf die Nusselt-Zahl und die Temperaturverteilung gering. Aus 28 Versuchsreihen wurde für die Nusselt-Zahl die Korrelationsgleichung $Nu_h = 0{,}31\ Re_h^{0.814}\,e^{-1.63\gamma}$ ermittelt. Weiterhin ergab sich, daß für $5000 < Re < 17\,000$ die Wärmeübertragung im Schüttungskanal ungefähr dreifach größer ist als im leeren Kanal.

ЭКСПЕРИМЕНТАЛЬНОЕ ИССЛЕДОВАНИЕ ВЫНУЖДЕННОЙ КОНВЕКЦИИ В ЗАПОЛНЕННОМ УПАКОВКОЙ КАНАЛЕ С АСИММЕТРИЧНЫМ НАГРЕВОМ

Аннотация—Экспериментально исследовалась вынужденная конвекция фреона-113 ($Pr = 8{,}06$) в канале, упакованном соответственно небольшим стеклянными шриками (диаметром 3,5 и 6 мм) и шариками из хромистой стали (диаметром 6,35 мм). Эксперименты проводились при $4000 < Re_n < 17\,000$ и $0{,}06 < \gamma < 0{,}13$ в канале с $L/H = 9$. Измерялись тепловой поток к стенке и поперечные профили температур внутри канала. Найдено, что число Нуссельта возрастает с уменьшением диаметра частиц. Влияние отношения теплопроводностей твердой и жидкой фаз на число Нуссельта и распределение температур в исследуемом диапазоне исменения числа Рейнольдса несущественно. На основе 28 опытов получено соотношение для числа Нуссельта $Nu_h = 0{,}31 Re_h^{0{,}814}\, e^{-1{,}63\gamma}$. Найдено также, что теплоперенос в канале с упаковкой приблизительно в три раза выше, чем в незаполненном канале при изменении значения Re_h от 5000 до 17 000.