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مقایسه آزمونهای انتقال حرارت و پایین افتادن فشار مرطوب و خشک دربارهٔ بسته‌بندی‌های PVC چین‌دار صاف و زبر در برجهای خنک‌کننده

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چکیده

در این مقاله نتایج یک بررسی تجربی دربارهٔ عملکرد برج خنک‌کنندهٔ محتوی بسته‌بندی PVC نشان داده شده است. موضوعهای زیر مورد آزمایش قرار گرفته اند:

اثر سطح چین‌دار، اثر زاویهٔ ناهمواری و اثر فاصله‌دار بودن بسته‌بندی.

بررسی به دو قسمت تقسیم شده است: مقایسهٔ انتقال حرارت پوسته‌ای (فیلمی) با پایین افتادن فشار هوا، بدون جریان آب و مقایسهٔ تغییر انتالپی و پایین افتادن فشار در نمونهٔ برج خنک‌کننده با جریان آب.

هفت بسته‌بندی تجارتی مورد بررسی قرار گرفته است، که اندازهٔ گسترهٔ $۱/۷۰ > \frac{P}{D} > ۱/۱$ و $۵ \ge \frac{P}{e} \ge ۱$ را می‌پوشاند و نتیجه گیری یک ارتباط بدون دیمانسیون به بحث گذاشته شده است.

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G'	Mass flux (air),	kg/m^2s
E	Height of corrugation,	mm
L'	Mass flux (water),	kg/m^2s
k	Mass transfer coefficient,	kg/m^2s
p	Distance between repeated ribs,	mm
P	Pitch of packing, (see figures 5 & 6)	dimensionless
Pr	Air Prandtl number at reference temperature	dimensionless
S	Surface area of the packing per unit packing volume	m^2/m^3
Sc	Air Schmidt number at reference temperature	dimensionless
St	Stanton number	dimensionless
t_s	Packing surface temperature	°C
j_i	$StPr^{2/3} = \frac{D}{2Z} x \frac{t_{G2}-t_{G1}}{t_s}$	dimensionless
j_m	$StSc^{2/3} = \frac{K\alpha\nu}{S} x \frac{D}{D+\delta}$	dimensionless
u_a	Air velocity in packing	m/s
u_w	Water velocity in packing	m/s
u^*	Total velocity	m/s
Re_a	Air Reynolds Number=$G'D/\mu_a$	dimensionless
Re_w	Water Reynolds Number=$2L'D/\mu_w$	dimensionless

Greek Letters

α	Average value of thermal conductance in the packing	W/mK
δ	Packing thickness	mm
μ	Air viscosity	kg/ms
μ	Water viscosity	kg/ms
ν	Kinematic viscosity=μ/ρ	m^2/s
ρ	Density	kg/m^3

Subscripts

R	Rough surface
S	Smooth surface
a	Air

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Conclusion

Experiments were conducted to investigate the j factors and Fanning friction factor relations on the PVC packing for which no comprehensive investigations had previously been reported and lead us to choose the best cooling tower PVC packing by this method. The experiments were carried out for comparative types of packing in a counterflow cooling tower. From the experimental results and discussion on the performance characteristic of seven vertical parallel packings arrangement in forced draft counterflow cooling tower the following conclusions are reached;

1) All of the corrugated packings with the surface roughness showed an improvement in the heat transfer coefficient when compared to the smooth packing. This improvement ranged from 42 to 240 percent. These increase in heat transfer are accompanied by even greater increase in Fanning friction factor in case of increasing by either changes in Reynolds number or change in roughness geometry (p/e).

2) The smooth packing heat transfer and Fanning friction factor results were found to confirm the predictions of Johnson and of Fujita and agree to with the experiments measurements of Tezuka (1980).

3) It was found that the shape and configuration of the roughness projections are as important as the height of those projections in determining their effect on Fanning friction factor and mass transfer coefficient. It was found that a packing of particular interest was the packing C6, which had a horizontal main corrugation with the cross ribbing making an angle of 45°.

4) A rule for heat transfer in rough PVC packing is presented. By this rule the general problem of determining the four variable dependence of j, Re, p/e is reduced to a two variable system. This is analogous to the related development by Trass (1971) in which the three variable f, Re, p/e relationship for friction in rough pipes was reduced to a two variable system. The results of the present work as well as those from other experiments which has been conducted on smooth packing, were shown to support the heat transfer similarity rule.

(a) The following equation was used to model the sensible heat transfer over dry portion of seven packings.

$$j = 0.0323\ Re^{-0.466}(P/E)^{0.7}(p/e)^{0.5} \qquad (35)$$

(b) The combined heat and mass transfer occurring over the wet surface of the cooling tower was calculated by

$$j_m = 0.0425\ Re^{-0.55}(P/E)^{0.7}(p/e)^{0.5} \qquad (36)$$

(c) The Fanning friction factors for wet and dry conditions may be related by

$$f_{wet} = f_{dry}[0.115(RE)] + 0.145 \qquad (37)$$

over the Reynolds number range of 1500 to 22000. This equation applies only to the parallel cooling tower packing when $1.1 < P/D < 1.70$ and $1 \leq p/e \leq 5$.

Notation

D	Distance between the cooling tower packing,	mm
e	Height of roughness element (rib)	mm
e^+	eu^*/ν	m^{-1}
f	Friction factor $= \frac{D}{4Z} \times \frac{\Delta P}{\frac{\rho_a(u_a+u_w)}{2}}$	Dimensionless

These change in the turbulence level near the rough surface would have an effect on both the momentum and mass transfer rates. Disruption of the viscous sublayer and penetration of turbulence into the valley regions would result in rapid increases in the rates of both momentum and mass transfer. A greater increase in the latter would be expected, as proportionately more of the resistance to mass transfer occurs in the viscous region.

Discussion

When a fluid flow through a channel containing a solid system, various resistances to fluid flow (or friction factors)occur, according to the shape of the solid system. The conditions which result in high friction factors, produce strong eddies in the fluid and thereby increase the rate at which heat may be transmitted from the fluid to the solid system. In other words, the friction and heat transfer properties of the system are correlated, so that it is generally impossible to achieve high heat transfer properties with low friction.

In relatively wide packings, the fluid stream is almost completely separated from the walls, and a large proportion of the packing is filled with recirculating fluid. (e.g. Packing C1).

In relatively narrow, corrugated packings flow separation takes place near the ridge of every corrugation, and flow re-attachmant takes place upstream of the next ridge in the flow directions. The troughs of the corrugations are partly filled with re-circulating fluid. It was found that a packing of paticular interest was the packing C6, which has a horizontal main corrugation with the cross ribbing making an angle of 45°. Flow separation enhances the turbulence of the flow (compared with corresponding flow between smooth, straight wall) and thereby increases heat transfer rate and pressure drop. (e.g. Packing C6).

Wind tunnel experiments were performed with various degrees of packing surface wetting, ranging between completely dry and completely wet. However the transfer coefficient employed in modelling dry and wet condition was obtained from our experiments. From figure 8 the combined heat and mass transfer occurring over the wet surface of the cooling tower was modelled by

$$j_{wet} = 0.0425\ Re^{-0.55}(P/E)^{0.8}(p/e)^{0.6} \qquad (31)$$

From figure 9 the following j factor relation was used to model the sensible heat transfer over dry portion of the packing.

$$j_{dry} = 0.0323\ Re^{-0.466}(P/E)^{0.7}(p/e)^{0.5} \qquad (32)$$

For parallel cooling tower packing the j factor for wet and dry conditions can be related by

$$j_{wet} = j_{dry}[0.155\ \ln(Re)]-0.04 \qquad (33)$$

over the Reynolds Number range of 500 to 3000. This eqation applies only to the parallel cooling tower packing when $1.1<P/D<1.70$ and $1 \leq p/e \leq 5$. The Fanning friction factors for wet and dry conditions may be related by

$$f_{wet} = f_{dry}[0.115(Re)]+0.145 \qquad (34)$$

over the Reynolds number range of 1500 to 3000. This equation applies only to the parallel cooling tower packing when $1.1<P/D<1.70$ and $1 \leq p/e \leq 5$.

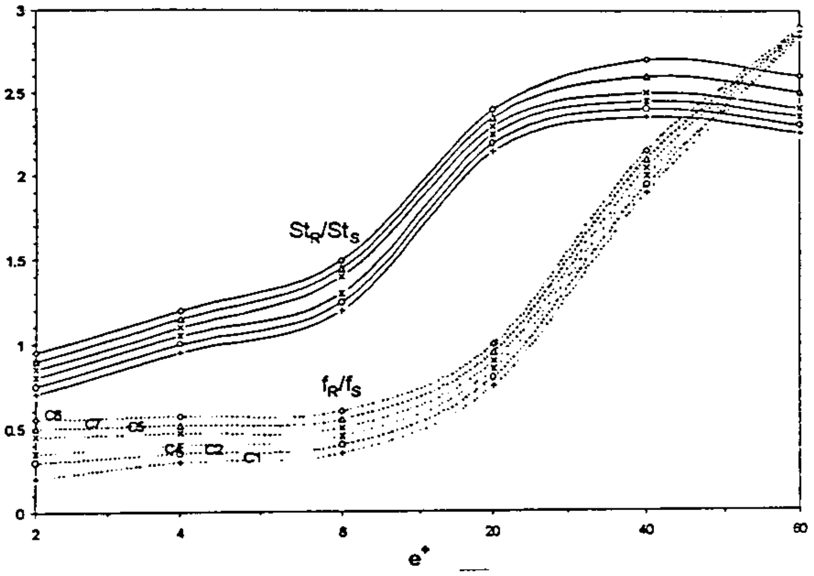


Fig. 10 Stanton number ratio verses e^+ for the packing with the rough surface.

The transition region is characterised by a marked increase is the transfer coefficient over the approximate range $20<e^+<45$. By comparing the data with the friction curve on the same graph, it can be seen that the increase in the mass transfer is much greater than in the friction factor over this range. Transition begins at about the same e^+ for both St and f, above $e^+ \sim 8$, however, the Stanton number ratio levels off, while the friction factor ratio continue to rise.

The transition region can be considered as the region in which the roughness emerges from a previously unaffected viscous sublayer. It is not necessary, however, to assume that the sublayer has not been changed by the presence of a submerged roughness. Perhaps a more acceptable description of the flow near the roughness elements is shown in Fig. 11.

The viscous sublayer may be considered to grow on the roughness element, the sublayer will be thicker between the ribs, and quite thin at the peaks. As Re_a increases, the sublayer will decrease in thickness. Eventually the thin layer at the peak will be disrupted, and the turbulence generated will be transmitted into the valleys between the roughness elements. As Re_a further increases, the valley regions will become more turbulent until, in the fully rough region, only a thin sublayer will remains.

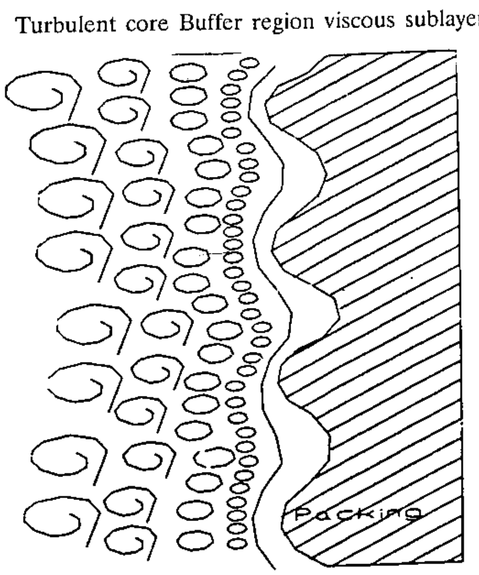


Fig. 11 Flow pattern near the rough wall

The fiction factor for the dry tests are 60% higher than the corresponding smooth packing friction factor. This differences in friction factor increase with increase of surface roughness up to 2.7 times for packing C6. In general, the friction factors of the cooling tower packings increase with increase Re_a and it decrease of Re_w. (Villar 1984).

The heat transfer data of the model cooling tower tests can be seen to be in fair agreement.In general $StSc^{2/3}$ increase with Re_w and the same way as $StPt^{2/3}$ with Re_a. The heat transfer data for the dry packing are much lower than the wet test results.

As it can be seen from Fig. 9a the smooth data follow an approximately linear plot, with a small negative slope, and except at high Reynolds number are in good agreement with the well known Blasius equation for smooth round tubes.

$$f= 0.0791\ Re^{-0.25} \quad (16)$$

A gradual positive deviation from the Blasius line can be observed as the Reynolds number is increase.

A linear least square curve fitting routine was use to obtain the following j factor correlations derived from the data presented in Fig. 8 and Fig 9.

i) Packing with the spacing of 40 mm and smooth surface

For dry condition $j_i=StPr^{2/3}=0.0323\ Re^{-0.673}$ (17)

For wet condition $j_m=StSc^{2/3}=0.04256\ Re^{-0.570}$ (18)

ii) Packing with the spacing of 50 mm and roughness of p/e=1

For dry condition $j_i=StPr^{2/3}=0.597\ Re^{-0.373}$ (19)

For wet condition $j_m=StSc^{2/3}=0.0306\ Re^{-0.470}$ (20)

iii) Packing with the spacing of 40 mm and roughness of p/e=3

For dry condition $j_i=StPr^{2/3}=0.595\ Re^{-0.320}$ (21)

For wet condition $j_m=StSc^{2/3}=0.0306\ Re^{-0.430}$ (22)

iv) Packing with the spacing of 40 mm and roughness of p/e=4

For dry condition $j_i=StPr^{2/3}=0.595\ Re^{-0.315}$ (23)

For wet condition $j_m=StSc^{2/3}=0.0306\ Re^{-0.392}$ (24)

v) Packing with the spacing of 35 mm and roughness of p/e=5

For dry condition $j_i=StPr^{2/3}=0.595\ Re^{-0.295}$ (25)

For wet condition $j_m=StSc^{2/3}=0.0306\ Re^{-0.375}$ (26)

vi) Packing with the spacing of 25 mm and roughness of p/e=4

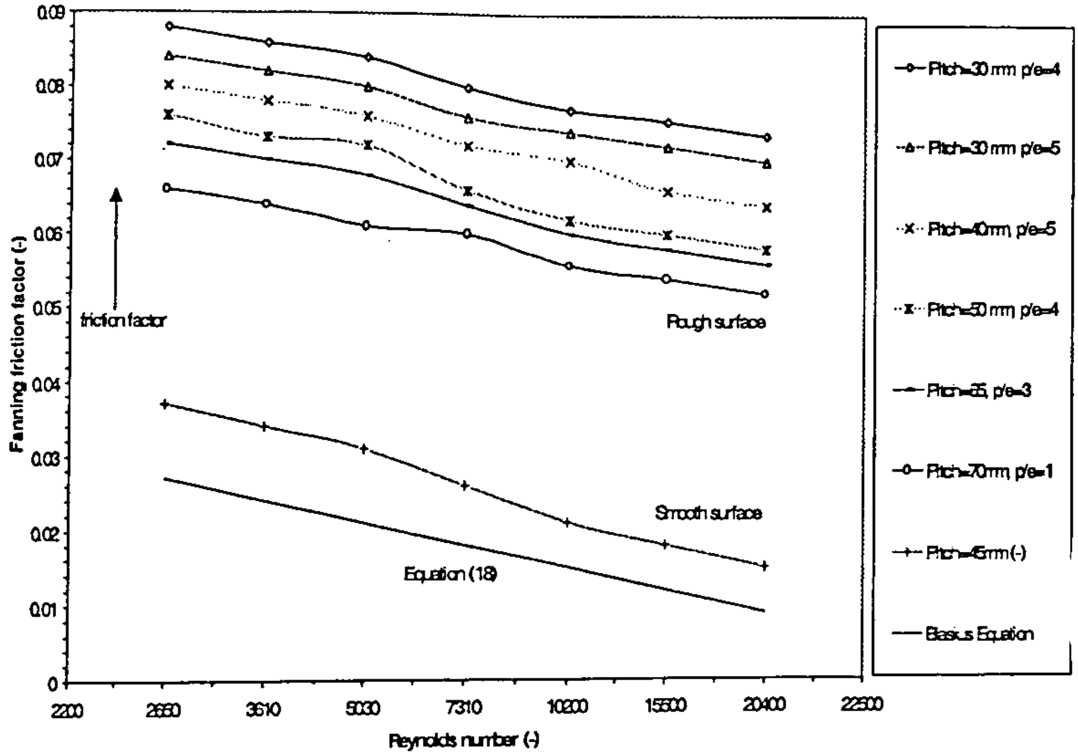
For dry condition $j_i=StPr^{2/3}=0.595\ Re^{-0.255}$ (27)

For wet condition $j_m=StSc^{2/3}=0.0306\ Re^{-0.326}$ (28)

vii) Packing with the spacing of 20 mm and roughness of p/e=5

For dry condition $j_i=StPr^{2/3}=0.595\ Re^{-0.215}$ (29)

For wet condition $j_m=StSc^{2/3}=0.0306\ Re^{-0.254}$ (30)



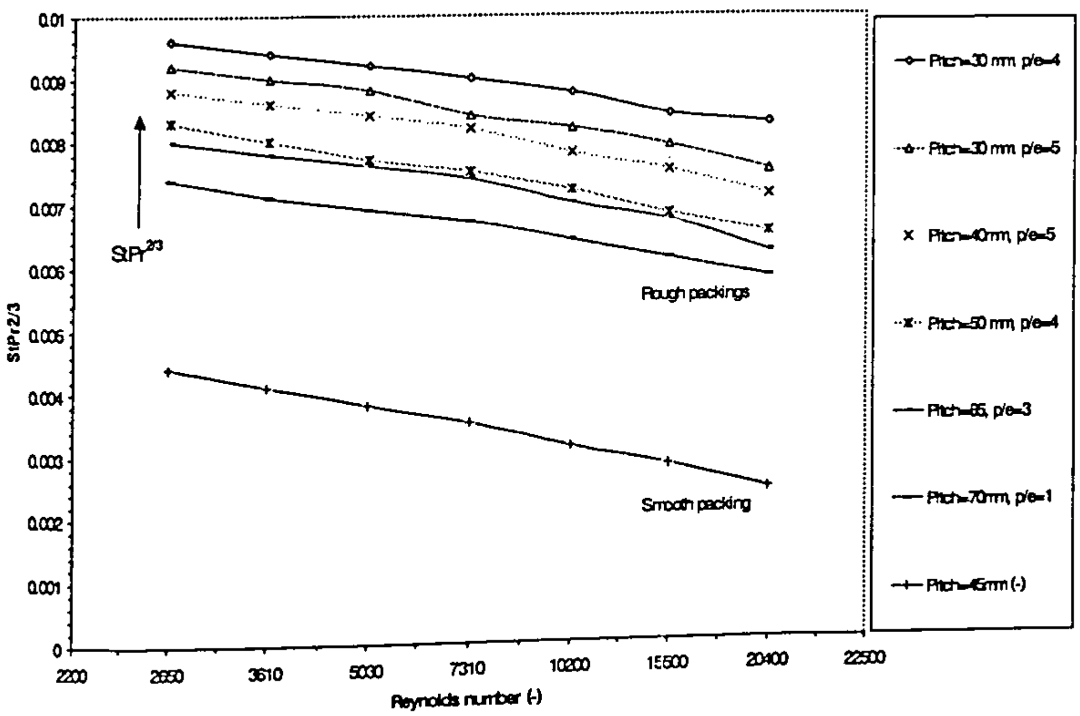


Fig. 9 Dry tests a) friction factor b) heat transfer test results for seven commercial packings.

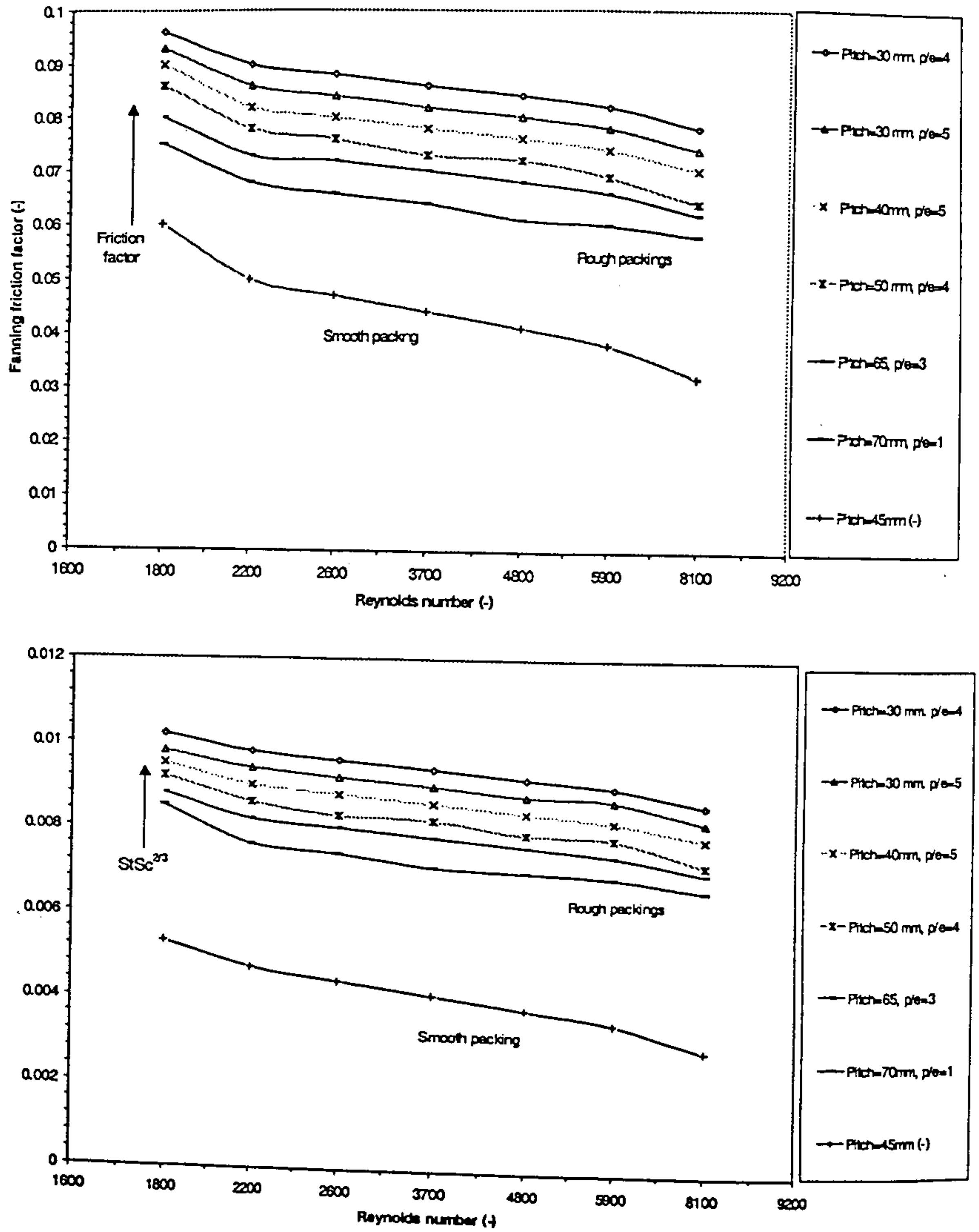


Fig. 8 Model cooling tower a) friction factor and b) heat transfer test results for seven commercial packings.

Test Group 7: Pitch = 30, P/D = 1.50, Rough surface p/e = 1, θ = 0 deg

Wet condition

Re	f/2	$StSc^{2/3}$
1800	0.093	0.0098
2200	0.086	0.0094
2600	0.084	0.0092
3700	0.082	0.009
4800	0.080	0.0088
5900	0.078	0.0087
8100	0.074	0.0082

Dry condition

Re	f/2	$StPr^{2/3}$
2650	0.084	0.0092
3610	0.082	0.009
5030	0.080	0.0088
7310	0.076	0.0084
10200	0.074	0.0082
15500	0.072	0.0079
20400	0.070	0.0075

omparison of the model cooling tower data with ie dry tests

Figs. 8 and 9 show the model cooling tower st results in dimensionless form with the iterpolated dry tests results for smooth and ugh packings. In these figures the model cooling wer data are presented as dimensionless groups f which $StSc^{2/3}$ the heat transfer factor rresponds to $StPr^{2/3}$ in the dry tests.

Fig.8a shows typical plots of friction factor f gainst Reynolds number. As has been shown, the iction factor of corrugated packing was affected y pitch, depth and also surface roughness Goshayshi & Missenden 1998). The friction ictor curve for packing C6 investigated in the resent test, lies above the curves for packing C7 y about 12%. The main reason is because the eight of the corrugation and the condition of the surface are dominant, the mean overall height of corrugations for packing C6 is 2.0 cm.

Fig. 8b shows valus of measured heat transfer factor, ($StSc^{2/3}$) plotted against the ratio of the Reynolds number for existing packings. The values for corrugated packing were up to 1.5 to 2.7 times higher than comparable smooth packing values. The $StSc^{2/3}$ values for rough and smooth corrugated packings decreased with the increas in pitch, and had a maximum value at P/D=1.5 and the ratio 4 for the ratio of distance between repeated ribs to height of rib, and the angle, θ, . 45°. The heat transfer factor for curve for packing C6 investigated in the present test, lies above the curves for packing C7 by about 15%. This cab be due to the better water distribution in the packing surface resulting of the angle of the cross ribbing.

Test Group 4: Pitch = 50, P/D = 1.43, Rough surface p/e = 4, θ = 0 deg

Wet model

Re	f/2	$StSc^{2/3}$
1800	0.086	0.0092
2200	0.078	0.0086
2600	0.076	0.0083
3700	0.073	0.0082
4800	0.072	0.0079
5900	0.069	0.0078
8100	0.064	0.0072

Dry model

Re	f/2	$StPr^{2/3}$
2650	0.076	0.0083
3610	0.073	0.008
5030	0.072	0.0077
7310	0.066	0.0075
10200	0.062	0.0072
15500	0.060	0.0068
20400	0.058	0.0065

Test Group 5: Pitch = 40, P/D = 1.32, Rough surface p/e = 4, θ = 0 deg

Wet test

Re	f/2	$StSc^{2/3}$
1800	0.09	0.0095
2200	0.082	0.009
2600	0.080	0.0088
3700	0.078	0.0086
4800	0.076	0.0084
5900	0.074	0.0082
8100	0.070	0.0078

Dry model

Re	f/2	$StPr^{2/3}$
2650	0.080	0.0088
3610	0.078	0.0086
5030	0.076	0.0084
7310	0.072	0.0082
10200	0.070	0.0078
15500	0.066	0.0075
20400	0.064	0.0071

Test Group 6: Pitch = 30, P/D = 1.50, Rough surface p/e = 4, θ = 45 deg

Wet test

Re	f/2	$StSc^{2/3}$
1800	0.096	0.0102
2200	0.090	0.0098
2600	0.088	0.0096
3700	0.086	0.0094
4800	0.084	0.0092
5900	0.082	0.009
8100	0.078	0.0086

Dry model

Re	f/2	$StPr^{2/3}$
2650	0.088	0.0096
3610	0.086	0.0094
5030	0.084	0.0092
7310	0.080	0.009
10200	0.077	0.00875
15500	0.0755	0.0084
20400	0.0735	0.00825

Experimental Results

Heat transfer and friction data for corrugated PVC packing

Test Group 1: Pitch = 70, P/D = 1.40, Rough surface p/e = 1, θ = 45 deg

Wet test

Re	f/2	$StSc^{2/3}$
1800	0.075	0.0085
2200	0.068	0.0076
2600	0.066	0.0074
3700	0.064	0.0071
4800	0.061	0.0070
5900	0.060	0.0069
8100	0.058	0.0066

Dry model

Re	f/2	$StPr^{2/3}$
1500	0.066	0.0074
2600	0.064	0.0071
3700	0.061	0.0069
4800	0.060	0.0067
5900	0.0557	0.0064
7000	0.054	0.0061
8100	0.052	0.0058

Test Group 2: Pitch = 65, P/D = 1.65, Rough surface p/e = 3, θ = 0 deg

Wet test

Re	/2	$StSc^{2/3}$
1800	0.080	0.0088
2200	0.073	0.0082
2600	0.072	0.008
3700	0.070	0.0078
4800	0.068	0.0076
5900	0.066	0.0074
8100	0.062	0.007

Dry model

Re	f/2	$StPr^{2/3}$
2650	0.072	0.008
3610	0.070	0.0078
5030	0.068	0.0076
7310	0.064	0.0074
10200	0.060	0.007
15500	0.058	0.0067
20400	0.056	0.0062

Test Group 3: Pitch = 45, P/D = 1.13, smooth surface

Wet test

Re	f/2	$StSc^{2/3}$
1800	0.060	0.0053
2200	0.050	0.0047
2600	0.047	0.0044
3700	0.044	0.0041
4800	0.041	0.0038
5900	0.038	0.0035
8100	0.032	0.0029

Dry model

Re	f/2	$StPr^{2/3}$
2650	0.037	0.0044
3610	0.034	0.0041
5030	0.031	0.0038
7310	0.026	0.0035
10200	0.021	0.0031
15500	0.018	0.0028
20400	0.015	0.0024

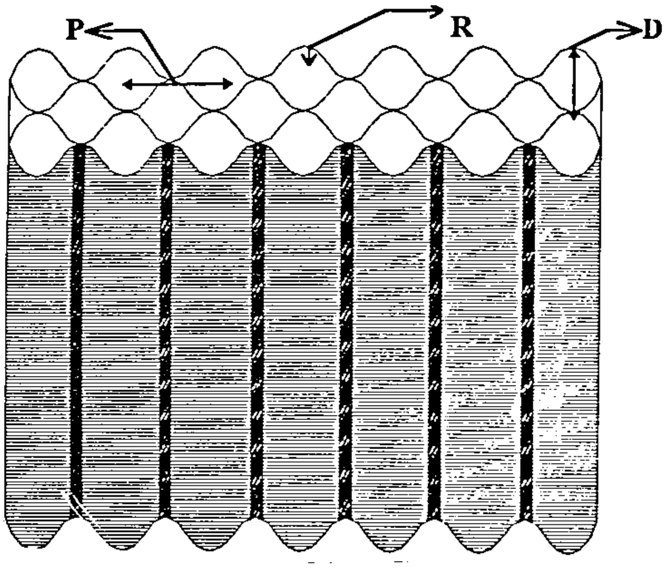


Fig. 5 Typical shape of smooth corrugated packing used in our experiment

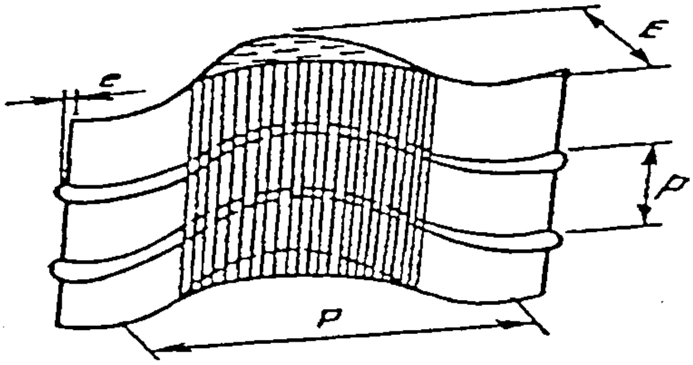


Fig. 6 Single cross ribbed sheet

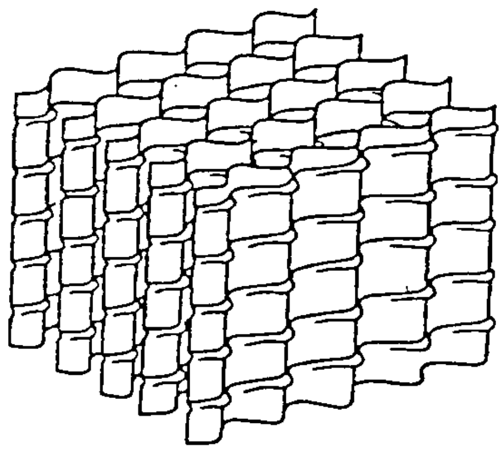


Fig. 7 Typical shape of rough corrugated packing used in our experiment

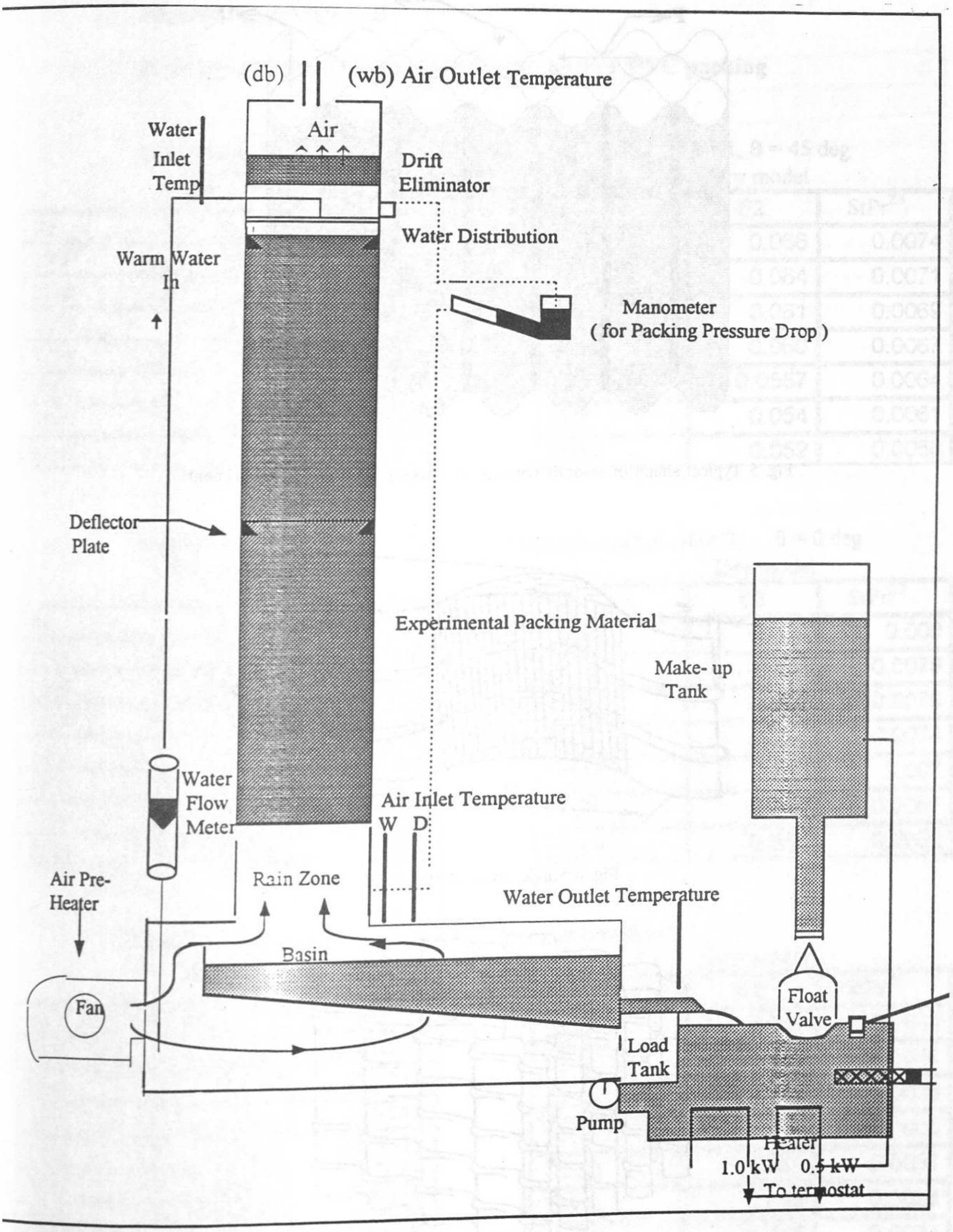


Figure 4 Schematic plan of our experimental counterflow forced draught cooling tower.

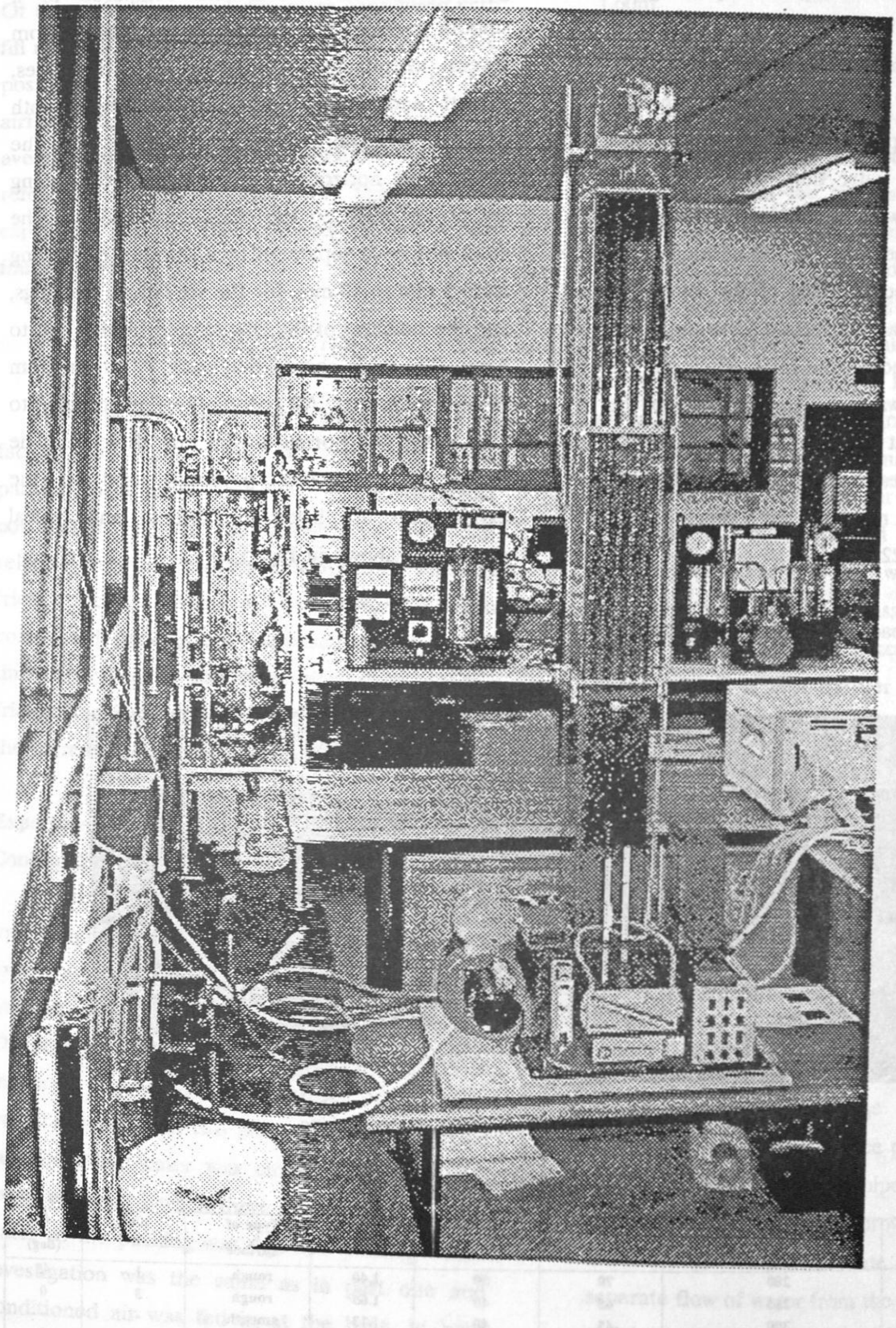


Fig. 3 Outside view of forced draft cooling tower in transport phenomena laboratory in chemical engineering department

ımp was 1.2 m below the top of the packings. A :ries of perimeter deflector plates as shown in ig. 4 was installed around the inner perimeter of ıe column. The column was made in the boratory of clear polycarbonate plastic to allow ɔservation of the water flow. These deflector ates removed the water film from the tower wall ıd redistributed the water in the packing zone. ; a result of deflection, most of the water was ansferred to the packing surface from the outer ıll, forming descending thin films, while air was ɔwn vertically upward, counter current to the ıter by the fan at the base. Measurement of at transfer and pressure drop was carried out in ɜ steady state and measured for a range of L/A ') from 0.45 to 2.22 kg/m^2s.

wer Packing

Six different packing, with six different geometries, were tested and compared to a smooth packing. The packing were obtained from several manufacturers and they were two types, smooth and ribbed, both of PVC. The smooth packing had horizontal corrugations and the ribbed had horizontal corrugations with ribbing set at an angle to the main corrugations. The cross ribs were separated by a distance p, ranging from 2 mm to 10 mm, for the six sample packings, and the height e of the ribs ranged from 1 mm to 3 mm. The main corrugation pitch, P, ranged from 30 mm to 70 mm. All corrugations are parallel to each other and normal to the flow direction. The forms of corrugated packings used in the experiments are listed in table 1, and typical shapes are shown in Fig. 5 to 7.

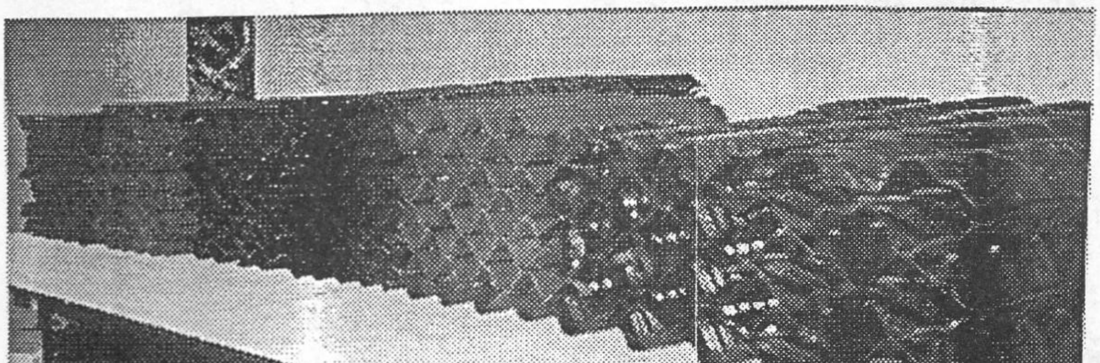


Fig.2 Photograph showing the geometries of the packing

Table 1: Shapes of corrugated packing used for experiments

:st ɔup	Type of Corrugation	Surface area per unit volum (m^2/m^3)	Pitch (mm)	Spacing (mm)	P/D	Type of surface	p/e	θ (deg)
1	sinusoidal	200	70	50	1.40	rough	1	45
2	sinusoidal	250	65	40	1.65	rough	3	0
3	triangular	300	45	40	1.13	smooth	-	-
4	triangular	350	50	35	1.43	rough	4	0
5	hexagonal	470	40	25	1.32	rough	5	0
6	sinusoidal	500	30	20	1.50	rough	4	45
7	triangular	500	30	20	1.50	rough	5	0

Of the eight fills tested, five were configured with fill material parallel to the air flow and three were positioned with the fill perpendicular to the airflow. The j factor was normalised by using the average value of ka for the crossflow fills at the reference values L' and G'. The resulting expression for the fills oriented to airflow was found to be;

$$j_m = 0.942\,(Re_a)^{-070}(D)^{0.73} \qquad (15)$$

In this present paper, the j_i factors and the j_m factors for many types of corrugated PVC packing, including smooth and rough surfaces and corrugated packing are investigated, and the relation between packing j factor and Fanning friction factor is discussed. The j factors for rough corrugated packing are increased by 1.5 to 2.7 times compared to smooth packing values, but the friction factor for the packings also increases with the increase in heat transfer performance.

Experimental Apparatus and Its Description

Cooling Tower

The cooling tower used for these experiments was designed for experimental investigation. The tower was operated at typical prevailing design weather conditions for cooling towers in the UK. The tower was a forced draft, counterflow type as shown in Fig. 4. The heat transfer experiments were carried out in two parts. Part 1 was carried out when the water was circulated through the tower and part 2 when there was no circulation of water so the packing was dry. The packing under investigation was the same as in part one and conditioned air was fed in at the base. In both parts, the cross sectional test area, A, was 0.15 m x 0.15 m. and the column packed height, Z, was 1.60m.

Air Distribution

A centrifugal, forward curved impeller fan was used to supply air to the cooling tower. It was accommodated at the bottom of the tower and was driven by a 2.25 kW ac motor . Two nozzles were installed at a distance of 0.08 m from the edge of the inlet air and 0.08 m from the outlet air and were connected to a micromanometer type APM 2000 of range 0 to 2000 Pa with an accuracy of $\pm 1\%$FSD± 1 digit(i.e. maximum of 1.2 pa error in our measurements). The micromanometer was calibrated to measure the air flow rate through the tower. Air inlet (t_{G2}) and outlet temperature (t_{G1}) dry bulb and wet bulb were measured with the help of mercury in glass thermometers of range 0-50°C with an accuracy of 0.1 K. Measurement of heat transfer and pressure drop was carried out in the steady state for a range of G/A (G') 0.20 to 1.50 kg/m^{2}s. A damper was provided in the air inlet to control the air flow rate through the tower. Air entered the tower, passed through the test section and drift eliminator and was ducted outside of the building.

Water Distribution System

The water distribution system for the cooling tower provided for entry at the top of the tower. Water was taken from the sump tank by a centrifugal pump, through the pipe line header to three smaller distribution pipes above the test section. Hand valves were provided in the water inlet line to control the rate of water flow. A separate flow of water from the tank to the heater and back through the pump allowed the desired temperature of water entering the tower (27 to 37°C) to be maintained. The water level in the hydrodynamic conditions in both wet and dry operation must be the same for the analogy to hold. (McQuiston 1978).

Colburn (1933) presented a general method for (dry) forced convection heat transfer data, involving a range of fluids, (air, water), and configurations, (flow inside tubes, etc.). He was able to show that the forced convection heat transfer in a region of fully developed flow, i.e. away from the inlet and the exit could be shown by;

$$j_i = j_{dry} = St\,Pr^{2/3} = \left(\frac{\alpha}{Gc_p}\right)Pr^{2/3} \qquad (6)$$

Another significant result of this work was to prove Reynolds' contention that heat transfer and friction data could be simply related. Colburn showed that this relation, in terms of j factor is

$$j_i = f/2 \qquad (7)$$

f/2 is the Faning friction factor, in this case referred to the mean air velocity in the dry passage. Further details of this study were published by Chilton and Colburn (1933), the result being that a mass transfer j_{wet} factor could be defined in the same manner as the sensible j factor.

$$j_m = j_{wet} = \left(\frac{K}{G}\right)Sc^{2/3} \qquad (8)$$

Smith and Norman (1957) showed that the above equation in the present instance of a fixed Prandtl number lead to an equation of the form,

$$j_i = \frac{f/2}{b + c(f/2)^{1/2}} \qquad (9)$$

where b is a constant which depends upon the Prandtl number and c is a constant which depends both on the Prandtl number and the surface condition.

Guillory and McQuiston (1976) performed experiments on parallel plate heat exchangers and observed that the j_m factor was greater than the j_i factor, on a wet surface.

Tezoka and Fujita(1980) plotted the heat transfer factor, α, against pressure drop for film type packings made of aluminium. Their results may be expressed as;

$$1.3\,j_m = j_i = (0.7-1.4)f/2 \qquad (10)$$

They also showed that the j factors for the cooling tower packing could be calculated from the correlations below;

$$j_i = \frac{D}{2Z} \times \frac{t_{G2} - t_{G1}}{t_s} \times Sc \qquad (11)$$

$$j_m = \frac{Kav/L}{S} \times \frac{D}{D+\delta} \qquad (12)$$

Chen (1985) showed that the sensible Colburn j factor and the Reynolds number could be related by;

$$j = CRe^{m-1} \qquad (13)$$

Upon tacking the logarithm of both sides

$$\log j = \log C + (m-1)\log Re \qquad (14)$$

When (m-1) is the slope of the line and log C is the intercept of the line.

More recently Johnson and Bartz (1990) obtained heat and mass transfer, and pressure data for crossflow films with splash type packings.

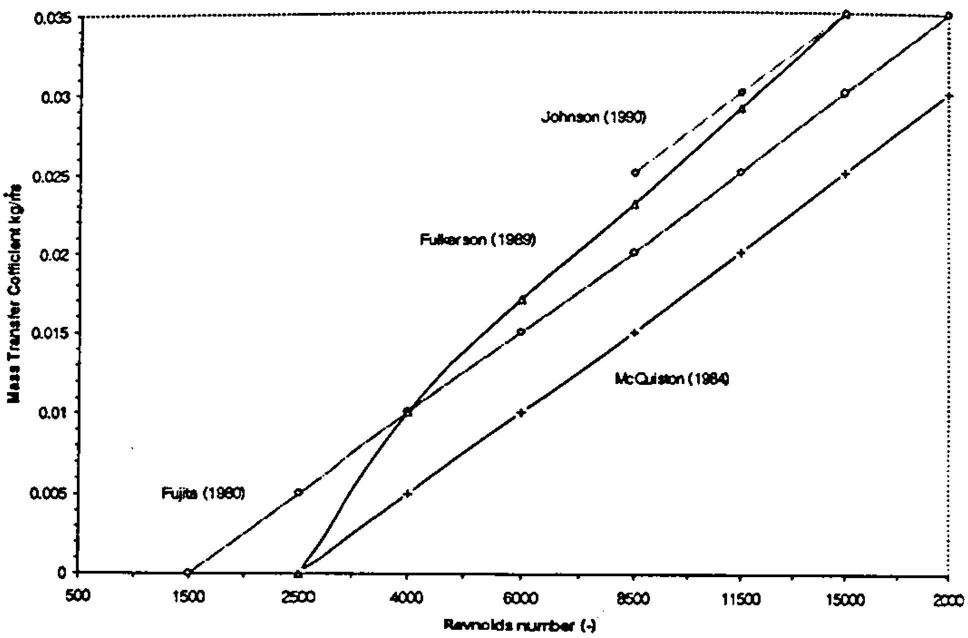


fig. 1 Some experimental results from the literature for mass transfer characteristics of cooling tower packing

One of the most common methods used in the design of heat transfer surfaces involves a direct analogy to dry operation. In the dry tests there is one variable less than for the wet cooling tower model (the water mass velocity or the water film Reynolds number Re_w) and also a range of geometries can be tested in less time than for the corresponding wet cooling tower tests.

In the design of heat transfer surfaces, the basic design equation for sensible calculations

$$Q_t = h_t A \Delta t_m \qquad (1)$$

is replaced by

$$Q_i = h_i A \Delta t_m \qquad (2)$$

using the surface coefficient h obtained in dry tests, where

$$h_i = h_t / c_p \qquad (3)$$

This method can be justified on theoretical grounds, since the differential energy conservation for sensible heat transfer is given by;

$$u\frac{\partial t}{\partial x} + v\frac{\partial t}{\partial y} - \alpha\frac{\partial^2 t}{\partial y^2} = 0 \qquad (4)$$

and the differential conservation equation for total energy transfer in a system involving mass diffusion

$$u\frac{\partial t}{\partial x} + v\frac{\partial t}{\partial y} - \alpha\frac{\partial^2 i}{\partial y^2} = 0 \qquad (5)$$

are mathematically identical. Physically, however, the independent variable for the dry case (thermodynamic temperature) is replaced by enthalpy in the wet case. Equation (5) strictly applies only when the Prandtl and Schmidt numbers for the mixtures are equal. Also, the

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COMPARISON OF WET AND DRY HEAT TRANSFER AND PRESSURE DROP TESTS OF SMOOTH AND ROUGH CORRUGATED PVC PACKINGS IN COOLING TOWERS

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ımmary

This paper presents the results of an experimental investigation of the performance of a cooling tower with PVC packing. The following were examined; the effect of surface roughness, the effect of the angle of roughness and the effect of packing spacing. The investigation was divided into two parts: comparison of film heat transfer with air pressure drop, without water circulation and comparison of enthalpy change and pressure drop in the model cooling tower, with circulation of water.

Seven commercial packing were investigated, covering a size range of $1.1<P/D<1.70$ and $1\leq p/e\leq 5$ and a discussion of the dimentionless correlation resulting is given.

troduction

Heat and mass transfer between a falling ;uid film down a vertical wall and air flowing wards, contacting directly with the water film, is important and interesting phenomenon in dustrial apparatus such as cooling towers. A mber of investigations concerning this have en made from both theoretical and applied wpoints (Bukowski 1995 ,Nabhan 1994, Kranc 93, Johnson 1990, Egberongbe 1990, Johnson 90 Fulkerson 1989, Fujita 1980). When these perimental results are compared, however, nsiderable differences are found in both the pes and intercepts of the logarithmic plots, as en from Fig. 1, which compares some results m the literature for mass transfer under simple nditions of air and water film exchange, along a ooth packing surface.

These discrepancies are considered to arise m differences in experimental conditions, such the shape and dimensions of the packing, the hydrodynamic and thermodynamic conditions of the inlet air and water, the values introduced for physical properties of air and water, and also from the manner of treating the measured data.

The large number of experimental and theoretical studies mentioned above, all have been performed for heat and mass transfer on smooth surfaces. However, many current cooling tower packing designs involve surfaces that are rough rather than smooth. In fact, no packing surfaces are totally smooth and turbulence promoters are often introduced to improve heat and mass transfer rates. Relatively few papers has been published so far on the effect of surface roughness on the heat and mass transfer in cooling tower packings.

Most studies of heat and mass transfer in smooth packings have consisted of experimental measurements resulting in only a partial understanding of the effects of roughness.