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HEAT TRANSFER AND PRESSURE DROP PERFORMANCE OF ROD BUNDLES ARRANGED IN SQUARE ARRAYS

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Abstract—The report covers results of measurements of pressure drop and heat transfer performed on two rod bundles with 9 and 16 rods, respectively, in square arrays. The measurements were carried out with helium for Reynolds numbers between 10^4 and 3×10^5. The rod bundles having a P/D ratio of 1·283 were operated at heat fluxes of up to 160 W/cm and wall temperatures of up to 700°C. Pressure losses at the spacers and the circumferential temperature distribution were also measured.

The new measured results and the available data from the literature were used to elaborate relations for pressure drop and heat transfer in rod bundles with rods arranged in square arrays.

NOMENCLATURE

A, cross section area;
C, drag coefficient;
C_v, modified drag coefficient;
c_p, specific heat at constant pressure;
D, tube diameter;
G, mass flow rate;
K, loss coefficient;
k, thermal conductivity;
L, length;
N, number of tubes;
Nu, Nusselt number;
p, pressure;
Δp, pressure drop;
P, pitch of tubes;
Pr, Prandtl number;
q, heat flux;
Q, heating power;
R, electrical resistivity;
Re, Reynolds number;
T, temperature;
u, velocity,
W, wall distance (Fig. 1);
y, wall distance;
α, heat transfer coefficient;
β, temperature coefficient of electrical resistivity;
ε, relative plugging of cross section;
λ, friction factor;
μ, dynamic viscosity,
ρ, density;
$\mathcal{R}$, gas constant.

Subscripts

a, acceleration;
B, bulk;
e, entrance;
el, electrical;
ex, exit;
f, friction;
h, hydraulic;
m, mean;
R, tube;
s, spacer;
t, total;

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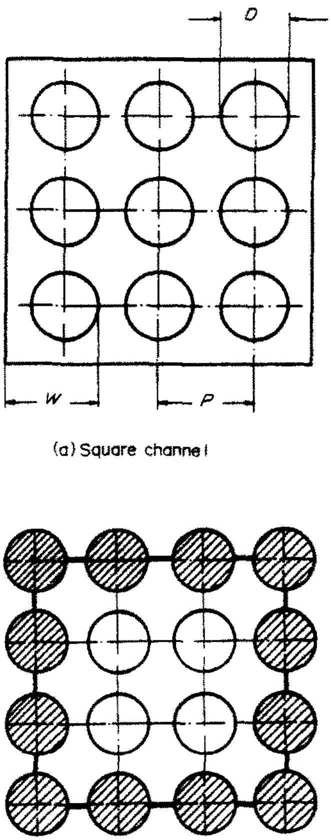
v, loss;
W, wall;
zu, input.

1. INTRODUCTION

KNOWLEDGE of the heat transfer and pressure drop behavior of rod bundles is very important in the design of nuclear reactors with rod-type fuel elements. Therefore, measurements were performed on two rod bundles with smooth surfaces in a square array within the framework of thermal-hydraulic studies for a gas cooled fast breeder reactor. A literature search revealed that so far only very few studies have been conducted on square arrays, less than on rod bundles in hexagonal arrays. At any rate, it is not possible to derive any general information from the data existing on rod bundles in square arrays. Hence, on the basis of data from the literature and new measured data, taking into account information possibly relevant to hexagonal arrays, safe and more accurate relations will be established in this report on heat transfer and pressure drop for design calculations.



(a) Square channel

(b) Quasi-infinite channel

FIG. 1. Rod bundles in square arrays.

In order to classify the many possible geometrical boundary conditions the scheme to be used in this paper will be briefly listed below. The following parameters can be distinguished:

(1) P/D ratio of the rods (Fig. 1).
(2) W/D ratio between the rods and the channel wall (Fig. 1).
(3) Number of rods in a rod bundle.
(4) Shape of the channel surrounding the rod bundle.
   (a) Square channel (Fig. 1a).
   (b) Quasi-infinite channel (Fig. 1b).
   (c) Other channels.

2. LITERATURE SURVEY

2.1 *Experimental investigations*

A number of experimental investigations of heat transfer and pressure drop in square rod bundle arrays have been performed. Table 1 is a compilation of the studies known to the authors; the main parameters of those studies are also listed.

Studies of the pressure drop in rod bundles of 9 rods were performed by Darling [1] and by Dingee and Chastain [2], who also investigated the heat transfer, for P/D ratios of 1·12–1·439. Parette and Grimble [3] studied the heat transfer with 9 rods for P/D = 1·14–1·2. 16 rods were used for measurements of the pressure drop for P/D = 1·073–2·00 by Galloway [4] and Galloway and Epstein, respectively [5], and by Grillo and Marinelli [6]. Grillo and Mazzone [7] recently studied the pressure drop with 36 rods for P/D = 1·3.

Le Tourneau, Grimble and Zerbe [8] studied the pressure drop with 64 rods for P/D = 1·12 and 1·2, and Wantland [9] investigated pressure drop and heat transfer for P/D = 1·106 with

Table 1. List of investigations of rod bundles in square arrangements

Author	Lit.	Year	Z	P/D	W/D	D [mm]	Channel	Re-range ($\times 10^{-3}$)	Medium	Remarks
Ushakov *et al.*	[11]	1962	∞	1·0	—	17·6	—	2–80	water	pressure drop
Gunn, Darling	[12]	1963	∞	1·0	—	24·1	—	0·1–100	water	pressure drop
			4	1·31	1·157	26·7	square	0·3–40	and	pressure drop
Darling	[1]	1961	9	1·439	1·495	14·3	square	0·3–40	oil	pressure drop
Galloway	[4]	1964	16	1·073	1·041	11·87	square	$2\cdot10^{-3}$–14	water	
Galloway, Epstein	[5]	1965	16	1·23	1·12	10·36	square	$3\cdot10^{-3}$–32	+	
			16	1·47	1·24	8·68	square	0·1–38	polyethyl.	pressure drop
			16	2·00	1·51	6·38	square	0·1–46	glycol	
Grillo, Marinelli	[6]	1970	16	1·2833	1·238	15·1	square	10–300	water	pressure drop
Le Tourneau *et al.*	[8]	1957	64	1·12	1·06	8·46	square	3–60	water	pressure drop
			64	1·2	1·1	7·92	square	4–90	water	
Presser	[10]	1967	1/9	1·05	1·05	74	quasi-	10–240	air	pressure drop
			16/36	1·67	1·67	15	infinite	13–220	air	and heat transf.
Digee, Chastain	[2]	1957	9	1·12	1·12	12·7	square	30–600	water	pressure drop
			9	1·2	1·2	12·7	square	40–500	water	and
			9	1·27	1·27	12·7	square	40–700	water	heat transfer
Wantland	[9]	1956	100	1·106	1·28	4·8	square	1–10	water	press. d. + ht. tr.
Parette, Grimble	[3]	1956	9	1·2	1·1	63·5	square	34–160	air	heat transfer
			9	1·136	1·28	63·5	square	30–105	air	heat transfer

100 rods. These rod bundles were studied in square channels.

Presser [10] used quasi-infinite channels for his investigations for P/D ratios between 1·05 and 1·67; such channels are generated if rod bundles are interconnected at the positions of the closest distance between the rods of the outer row of rods (Fig. 1b). This shape of channel is said to very closely approximate a rod bundle of infinite extension.

The friction factor of rod bundles of an infinite extension can be measured only in the special case of rods contacting each other, i.e. $P/D = 1{\cdot}0$. Results of measurements were obtained for this case by Ushakov, Subbotin *et al.* [11] and by Gunn and Darling [1, 12]. Some of the publications mentioned were compiled by Sheinina [13].

2.2 *Theoretical studies*

The theoretical studies refer to rod bundles of an infinite extension only, i.e. they exclude the influence of the limiting channel wall on pressure drop and heat transfer.

The fundamental work was performed by Deissler and Taylor [14] and is a calculation of the velocity and temperature fields in the flow. Gunn and Darling [12] only mention a method of calculating the pressure drop. Further theoretical work was done by Osmachkin [15], Buleev, Polusukhina and Pyshin [16], Kokorev, Korsun and Petrovichev [17], Ibragimov *et al.* [18], Presser [10], Eifler and Nijsing [19], and by Subbotin and Ushakov [20]. In the work by Osmachkin [15] and Presser [10], the heat transfer is considered besides the pressure drop. Pressure drop and heat transfer are the topics also of the work by Aranovich [21] and recently by Ramm and Johannsen [22]; heat transfer only is considered by Borishanskiy, Gotovskiy and Firsova [23], by Kokorev, Korsun and Petrovichev [24] and by Markoczy [25].

Both the experimental data and the results of the theoretical studies by the various authors vary greatly, thus allowing no safe statement to be made about the dependence on geometrical parameters of pressure drop and heat transfer, respectively.

3. EXPERIMENTS

Measurements were performed in two test sections to provide more accurate relations for pressure drop and heat transfer of rod bundles arranged in square arrays.

3.1 *Test sections*

The test sections consist of 9 and 16, respectively, smooth tubes in a square array installed in a square channel. Both the circular tubes and the square channels were made of heat resistant steel (No. 4841). The main dimensions are shown in Table 2.

Table 2. Main dimensions of the test section

Number of tubes	9	16
Tube diameter (mm)	19	17·4
Tube wall thickness (mm)	1·2	1·2
P/D ratio	1·283	1·283
W/D ratio	1·23	1·27
Width across the flats of the square channel (mm)	76·5	93·8
Overall length (mm)	3660	3660
Heated length (mm)	2010	1740

A short unheated startup section of L_A = 500 mm made of copper tube of 3 mm wall thickness precedes each of the heated sections. Three and four heater tubes, respectively, are connected in parallel and in series for electrical heating with d.c. current. The tubes are centered by ceramic plates at the upper and lower ends and supported at four levels over their entire length by insulating ceramic pins guided in steel nipples. In this way, undue bowing of the tubes during operation is to be prevented. Suspension of the test sections in a carrier tube allows free expansion at the high operating temperatures encountered. The current is supplied from the top through the unheated startup section. At the bottom end, the tubes are connected by a copper bridge, the differential elongations between the heater tubes being accommodated by flexible copper strips.

3.2 *Test rig*

The investigations were performed in a helium test rig in which a maximum flow of 1·2 kg/s of helium at a maximum pressure of 50 bar is circulated in a closed circuit. The maximum heater capacity available is 500 kW.

The pressure drop along the entire test section was measured through pressure taps of 0·5 mm dia in the channel wall. In the 9-tube bundle, the absolute pressure was measured by means of sensitive pressure gauges of the 0·6 class and the pressure drop over the test section was determined by means of mercury, bromoform and water, respectively, in ∪-tube manometers. However, in a 16-tube bundle these measurements were performed by means of pressure transducers. The wall temperatures were measured by means of Ni–NiCr thermocouples with an insulated measuring junction. The thermocouples were soldered into the tube wall with vacuum gold solder. The inner wall temperature was measured by means of measuring devices inside of two central tubes of the 16-tube bundle. These measuring devices were equipped with two miniature thermocouples the diameter of which had been reduced to 0·35 mm at the tip. In addition to a short response time, this measuring device ensures a reproducibility of $<0{\cdot}5°C$. Gas inlet and outlet temperatures were measured by means of thermocouples with an insulated measuring junction.

The thermovoltages were recorded by potentiometer point recorders of the 0·25 class. The helium flow was measured by means of Venturi tubes and standard orifices, respectively. The electrical heating power was determined by the separate measurement of current and voltage. The coefficient of electrical resistivity was also determined from current and voltage measurements at various wall temperatures. For the range under investigation, a linear dependence of the resistivity was found to be $R = R_0\ (1 + \beta t)$, where R_0 is the resistivity at 0°C. The temperature coefficient of this material was

$$\beta = 0{\cdot}322\,.\,10^{-3}\ [1/°\mathrm{C}].$$

Further details about the test rig and the test sections can be found in [26, 27].

4. EVALUATION AND RESULTS

4.1 *Measurements of the pressure drop*

The overall pressure drop Δp_t of a tube bundle can be written as

$$\Delta p_t = \Delta p_e + \Delta p_f + \Delta p_a + \Delta p_{ex} + n\Delta p_s \quad (1)$$

with the inlet pressure drop

$$\Delta p_e = K_e \frac{\rho_e}{2} u_e^2 \quad (2)$$

where ρ_e is the density of the fluid at the inlet, u_e the inlet flow velocity and K_E a loss coefficient. Δp_f is the friction pressure drop which, according to Diamond and Hall [28], results from the integration of the momentum equation in gases with heat addition as

$$\Delta p_f = \lambda \frac{L}{D_h} \frac{1}{2\rho_m} \left(\frac{G}{A}\right)^2 \quad (3)$$

with the flow rate G and the flow cross section A, the length of the bundle L and the friction factor λ.

The mean density is defined as

$$\rho_m = \frac{p_e + p_{ex}}{T_e + T_{ex}} \frac{1}{\mathcal{R}} \quad (4)$$

where $\mathcal{R}$ is the gas constant and p_e and p_{ex} and T_e and T_{ex}, respectively, are the pressures p and temperatures T, respectively, at the inlet (e) and outlet (ex), respectively.

The hydraulic diameter D_h is defined as

$$D_h = \frac{4A}{P_t} \quad (5)$$

with the total wetted perimeter P_t.

Moreover, the acceleration pressure loss as obtained from the momentum equation is

$$\Delta p_a = \frac{1}{\rho_m}\left(\frac{G}{A}\right)^2 \frac{T_{ex} - T_e}{T_m}. \quad (6)$$

The pressure drop at the n spacers can be written as

$$\Delta p_s = C_s \frac{\rho}{2} u^2 \quad (7)$$

where C_s is the drag coefficient of the spacer.

Isothermal pressure drop measurements were conducted to determine the inlet loss coefficient K_e, the drag coefficient of the spacer C_s, and the friction factor λ as a function of the Reynolds number. The Reynolds number is defined as

$$Re = \frac{4G}{P_t \mu}. \quad (8)$$

The isothermally measured K_E and C_s were used to determine the friction factor λ_B (fluid properties referred to T_B) from the experiments with heat addition. Since the temperature varies greatly over the entire length of the bundle in the non-isothermal runs, it is necessary to determine the quantities of state and λ_B stepwise for accurate determination of λ_B. In [26, 27] it has been proved that the error caused by a simplified calculation with a density ρ_m averaged over the length of the test section, even in the most adverse case, is below 1·5 per cent. Therefore, the simplified method of calculation was applied to all the measured values.

In accordance with a suggestion by Taylor [29], the friction factors with heating are represented as

$$\lambda_B \sqrt{\left(\frac{T_W}{T_B}\right)} = F(Re_W) \quad (9)$$

over the Reynolds number Re_W at the wall.

Figure 2 shows the measured friction factors of the 9-tube bundle, Fig. 3 the friction factors of the 16-tube bundle over the Reynolds number. For comparison, the curve of the friction factor of the smooth circular tube was also entered in accordance with Maubach [30]

$$\frac{1}{\sqrt{\lambda}} = 2{\cdot}035 \lg Re(\sqrt{\lambda}) - 0{\cdot}989. \quad (10)$$

Two important results can be derived from the measurements:

(1) The method of correlation of friction factors of smooth tubes as suggested by Taylor also very well correlates the measured results with and without addition of heat in the case of rod bundles. At any rate, no difference can

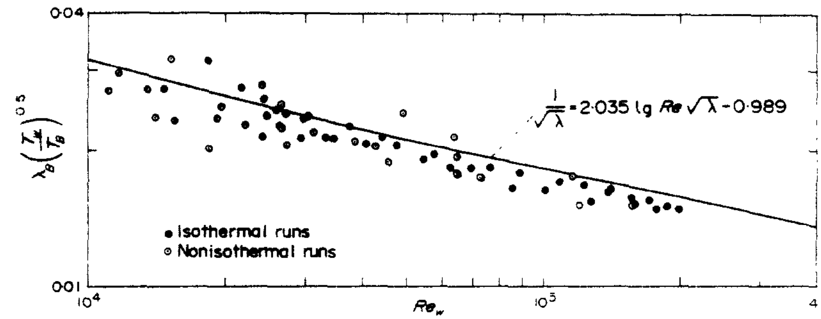


FIG. 2. Experimental friction factors 9-rod-bundle.

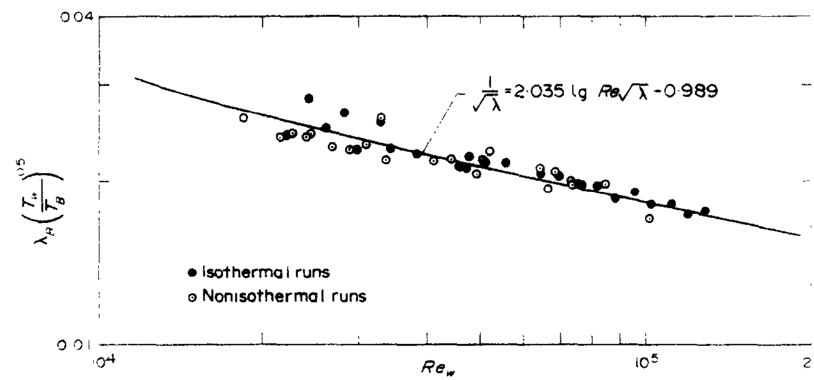


FIG. 3. Experimental friction factors 16-rod-bundle.

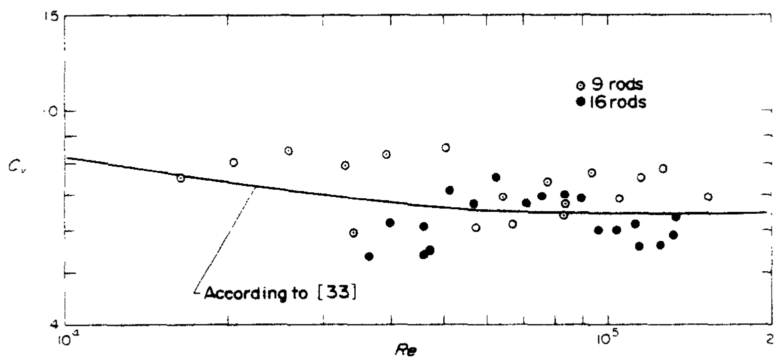


FIG. 4. Drag coefficients of the spacer

be found between isothermal and non-isothermal friction factors.

(2) The friction factors for the 9-tube bundle are on the average approximately 8 per cent below the values for circular tubes; the friction factors of the 16-tube bundle agree with those for the circular tubes. This is due to the fact that, given the same P/D ratio for both tube bundles, the W/D ratio of the 9-tube bundle is lower than in the 16-tube bundle. As outlined in [31, 32], there will be a uniform flow distribution through rod bundles in hexagonal arrangements approximately for $P/D = W/D$ at a maximum value of the friction factor. At larger or smaller wall distances the friction factor will always be smaller. The same effect of course can be found also in rod bundles in square arrays. In the rod bundles investigated, $P/D = 1{\cdot}283$; the W/D ratios are 1·23 in the 9-tube bundle and 1·27 in the 16-tube bundle.

The drag coefficients measured for the spacers are plotted over the Reynolds number in Fig. 4 as modified drag coefficients C_v which, according to [33, 34] turn out to be

$$C_v = C_B \frac{1}{\varepsilon^2} \tag{11}$$

with

$$\varepsilon = \frac{A_r}{A_t} \tag{12}$$

where A_r represents the flow cross section plugged by the spacers. Agreement with the curve determined from measurements of the loss of pressure at various grids is excellent. The relatively large scatter is due chiefly to the small pressure drop at the grids, because the relatively small amount of plugging of the cross section ε amounted to only 15·6 and 17·1 per cent, respectively, for the 9- and the 16-tube bundles.

4.2 *Heat transfer measurements*

In the 9-tube bundle, the thermocouples were arranged so as to measure the longitudinal distribution of the wall temperature of the central rod and, in a specific cross section, also the temperatures of a corner rod and a central rod of the outer row. In the 16-tube bundle, a movable measuring device was used in addition to the firmly installed thermocouples to determine the longitudinal and the circumferential temperature distributions.

The values measured for the wall temperatures were corrected because of the temperature gradient in the wall. Since the development of the flow and the temperature distribution was repeatedly disturbed by the spacers installed, a level was chosen for each of the two tube bundles for evaluation of the results in which approximately fully developed flow conditions existed.

In the tube bundle with 9 tubes, this level was at $L/D_h = 80$, and in the 16-tube bundle it was at $L/D_h = 65$. For this position, the Nusselt number was determined as

$$Nu = \frac{\alpha \, . \, D_h}{k} \tag{13}$$

with k as the thermal conductivity, and the heat transfer coefficient α calculated as follows:

$$\alpha = \frac{q}{T_W - T_B}. \tag{14}$$

From the heating power supplied Q_{zu}, the average power per unit area q is determined by division by the surface:

$$g = \frac{Q_{zu}}{N\pi D L_h} \tag{15}$$

where N is the number of heater rods.

The local power per unit area at the measuring level, the dependence of electrical resistivity on temperature taken into account, then is

$$q_M = q \frac{1 + \beta t_{WM}}{1 + \beta t_{W_{av}}} \tag{16}$$

where t_{WM} is the measured wall temperature at the measuring level, $t_{W_{av}}$ is the mean wall temperature at the test section.

The power input Q_{zu} was determined from the electric power Q_{el} and the heat losses Q_v as

$$Q_{zu} = Q_{el} - Q_v \tag{17}$$

with the heat losses being calculated.

Due to the excellent insulation [26, 27], heat losses were small.

The calculation of the bulk temperature T_B was performed on the basis of the assumption of a constant gas temperature over the cross section:

$$T_B = T_e + \frac{Q_{zu}(x/L)}{Gc_p}. \quad (18)$$

In the bundle with 9 tubes, this assumption was fulfilled by the choice of $W/D = 1{\cdot}23$ for the outer tubes. In this way, a bulk flow was established in the subchannels which corresponded to the fraction of the heating area and amounted to $\frac{1}{9}$ of the total flow in the central channels. Under this assumption, the Nusselt and Reynolds numbers were calculated for the central cell only. When using the hydraulic diameter of the central cell:

$$D_\infty = \frac{4[(P/D)^2 - (\pi D^2/4)]}{\pi D} \quad (19)$$

the result is

$$Nu_B = \frac{\alpha . D_\infty}{k} \quad (20)$$

and the Reynolds numbers for the 9-tube bundle are

$$Re_{B-9} \doteq \frac{(\rho w)_\infty D_\infty}{\mu} = \left(\frac{G}{9}\right)\frac{4}{\mu\pi D} \quad (21)$$

and those for the 16-rod bundle are

$$Re_{B-16} = \frac{(\rho w)_\infty D_\infty}{\mu} = \left(\frac{G}{16}\right)\frac{4}{\mu\pi D}. \quad (22)$$

However, for the 16-rod bundle, $W/D = 1{\cdot}27$, that is to say, the distance from the wall is greater than in the bundle of nine. In this way, a larger fraction of the bulk flow passes through the outer channels and correspondingly less will flow in the central channels. This gives rise to a more rapid increase in the gas temperature of the central channels relative to the calculated values. This effect makes the calculated temperature difference $T_W - T_B$ too large and makes for an underestimation of α and thus of the Nusselt number in the calculation. In addition, this influences the Reynolds number, underestimating it in the calculations of the central channels. In the evaluation, the fluid properties shown in [26] were used which differ only slightly from the values recommended by Pfriem [35] in the range investigated here.

Figure 5 shows the measured results. For

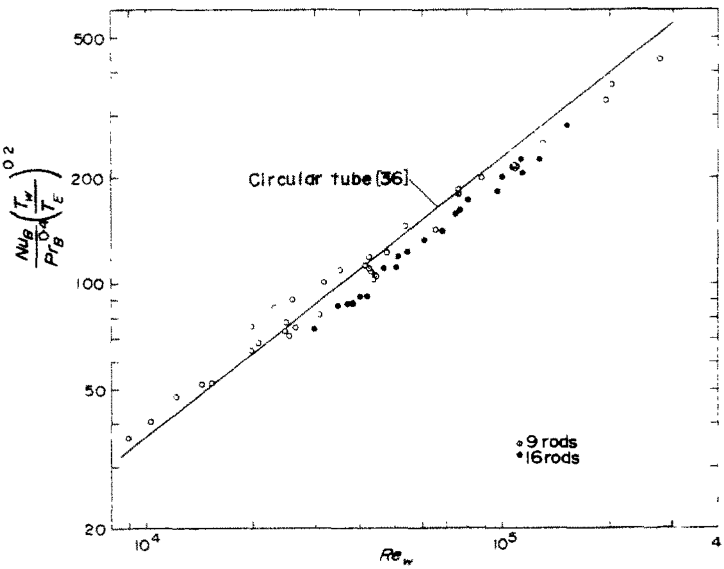


FIG. 5. Experimental Nusselt numbers.

comparison, the related based on Dittus–Boelter [36] is plotted:

$$Nu_B = 0{\cdot}023\, Re_B^{0{\cdot}8}\, Pr_B^{0{\cdot}4} \left(\frac{T_W}{T_e}\right)^{-0{\cdot}2} \tag{23}$$

which had been expanded by the temperature factor T_W/T_e with the inlet temperature T_e. The exponent of the temperature factor is chosen as $n = -0{\cdot}2$ in accordance with experiments by Dalle Donne and Meerwald [37] in smooth annuli.

The measured results for the 9-tube bundle show a considerable amount of scatter; on the average, they are about 7 per cent below the tube values in accordance with equation (23). The Nusselt numbers of the 16-tubr bundle are parallel the tube values at a level approximately 15 per cent lower. The main reason for the values of the 16-tube bundle being clearly lower is the larger distance from the wall in the 16-tube bundle, as outlined above.

4.3 *Circumferential temperature distribution*

A temperature measuring device was installed in the central rod of the 16-rod bundle to determine the circumferential temperature distribution. At a temperature difference of 55°C between the gas and the wall the temperature variation on the circumference was within ±1°C. From the theoretical results by Deissler and Taylor [14], a maximum temperature difference on the circumference of $\Delta T = 12$°C would result with the thermal conduction in the wall taken into account. The results by Deissler and Taylor greatly overestimate the temperature variation on the circumference, as has already been stated by Dingee and Chastain [2], Hoffman, Wantland and Stelzman [38], Miller, Byrnes and Benforado [39], and Palmer and Swanson [40].

All the measured values are tabulated in [26, 27].

5. COMPARISON OF DATA FROM THE LITERATURE WITH THE MEASURED VALUES

5.1 *Friction factors*

Figure 6 is a compilation of all the measured results and theoretically determined values

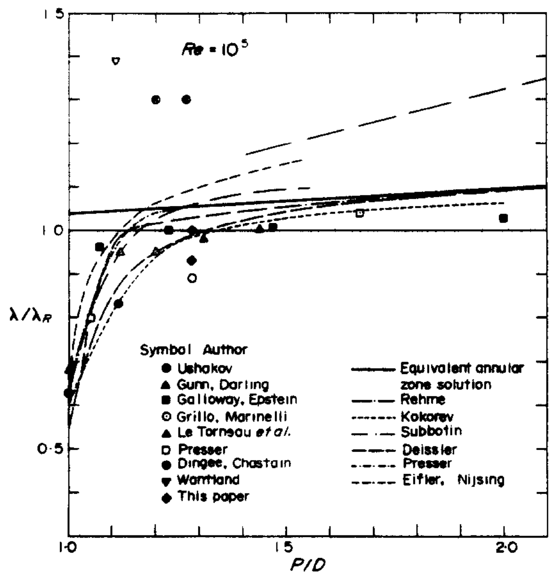


FIG. 6. Friction factors of rod bundles in square arrays.

known to the authors for a Reynolds number of $Re = 10^5$, as far as this is possible. All data were referred to the circular tube values in accordance with equation (10). Also included in the compilation, in analogy with the hexagonal array [32], was the annular zone solution generated under the assumption of Nikuradse's law of the wall [41] for the velocity profile

$$u^+ = 2{\cdot}5 \ln y^+ + 5{\cdot}5 \tag{24}$$

with the dimensionless flow velocity u^+ and the dimensionless distance from the wall y^+. In analogy with the relation for hexagonal arrays derived in [32], the following approximation equation can be indicated for the annular zone solution for square arrays:

$$\lambda/\lambda_R = 1{\cdot}04 + 0{\cdot}06\,(P/D - 1) \tag{25}$$

for $P/D < 2{\cdot}0$ and $Re = 10^5$.

The annular zone solution is a very good approximation of the maximum value of the friction factor in rod bundles at $P/D > 1{\cdot}2$. As is shown in Fig. 6, all the measured results except for the very high values by Wantland [9] (low Reynolds number !) and Dingee and Chastain [2] (inaccurate measurements) are below the annular zone solution which, in this way, is confirmed very well as the upper limit of the friction factor for square arrays. This maximum value is approximated most closely by rod bundles of an infinite extension. Rod bundles *with* channel walls always have lower friction factors. As has been measured by Presser [10], Courtaud, Ricque and Martinet [42] and Ibragimov, Isupov and Subbotin [43] and confirmed by theoretical investigations of laminar flow in rod bundles [44], a rod bundle with a specific P/D ratio for $W/D < P/D$ will first of all exhibit an increase in the friction factor, a flat maximum at $W/D \approx P/D$ which is close to the value for a rod bundle of infinite extension, depending upon the number of rods and, upon further increase in W/D, a decrease of the friction factor. Calculations by Eifler and Nijsing [45] confirm these statements.

Among the theoretical papers, the results by Subbotin and Ushakov [20] and by Eifler and Nijsing [19] are higher than the annular zone solution. This is due to the fact that velocity profiles were used in the calculations which were based on measurements performed by Brighton and Jones [46] on annular gaps with small diameter ratios. In our opinion, these results have been misinterpreted in view of the fact that Brighton and Jones in their evaluation assumed a coincidence of zero shear stress and maximum velocity. This assumption does not hold true for asymmetrical velocity profiles of the type found here [47]. Also the results of the calculations of Ramm and Johannsen [22] for rod bundles arranged in square arrays are 6 per cent higher than the annular zone solution for $P/D > 1{\cdot}2$. This is very surprising because the results reported in the same paper for rod bundles in hexagonal arrangements are a few percent lower than the annular zone solution in this case. The measured results confirm the theoretical results elaborated by Deissler and Taylor [14], Kokorev *et al.* [17] and Presser [10]. Also the friction factors for rod bundles of infinite extension in a square array determined by means of a recently suggested method of calculating friction factors in non-circular channels [48] based on the laminar solutions [44], are in excellent agreement with the measured data.

5.2 *Nusselt numbers*

Figure 7 is a compilation of the data found in the literature on Nusselt numbers in turbulent flows through rod bundles arranged in square arrays, i.e. both theoretical and experimental data. The Nusselt numbers were referred to the circular tube values according to equation (23). Measured results and theoretical data scatter greatly.

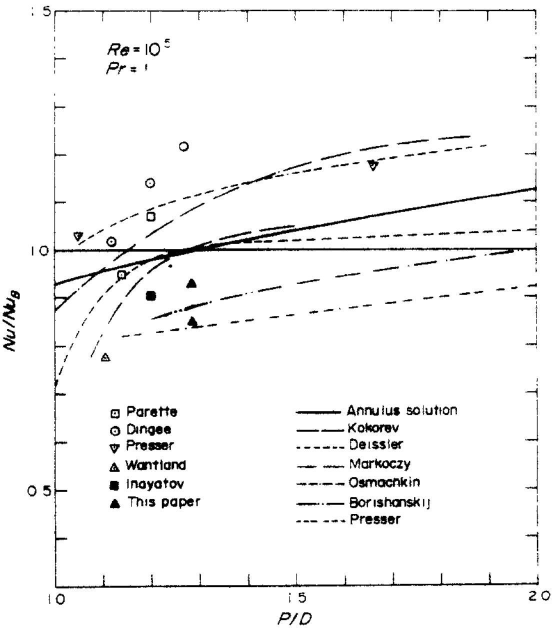


FIG. 7. Nusselt numbers of rod bundles in square arrays.

In analogy to the friction data also an annulus solution was included in the compilation, as suggested by Dalle Donne [49] for hexagonal rod bundles. For annuli with a heated inner tube, Dalle Donne and Meerwald [37] recommend the relationship

$$Nu_B = 0{\cdot}018\, Re_B^{0\cdot8}\, Pr_B^{0\cdot4} \left(\frac{D_2}{D_1}\right)^{0\cdot16} \left(\frac{T_W}{T_e}\right)^{-0\cdot2} \quad (26)$$

which dates back to Petukhov and Roizen [50].

Taking into account the hydraulic diameter in a rod bundle of infinite extension in a square array

$$D_\infty = D\left[\frac{4}{\pi}\left(\frac{P}{D}\right)^2 - 1\right] \tag{27}$$

and the changed boundary condition in rod bundles relative to annuli, i.e. no opposite wall, results in

$$\frac{Nu_B}{Re_B^{0\cdot 8}\,Pr_B^{0\cdot 4}}\left(\frac{D_h}{D_\infty}\right)^{0\cdot 2} = 0{\cdot}018\left(\frac{2}{\sqrt{\pi}}\,\frac{P}{D}\right)^{0\cdot 16} \times \left(\frac{T_W}{T_e}\right)^{-0\cdot 2} \tag{28}$$

or

$$\frac{Nu_B}{Re_B^{0\cdot 8}\,Pr_B^{0\cdot 4}} = 0{\cdot}018\left(\frac{2}{\pi}\frac{P}{D}\right)^{0\cdot 16} \times \left(1 + \frac{2}{\sqrt{\pi}}\,\frac{P}{D}\right)^{0\cdot 2}\left(\frac{T_W}{T_e}\right)^{-0\cdot 2}. \tag{29}$$

This relation should be an upper limit of the Nusselt number of rod bundles arranged in square arrays, like the annular zone solution in the case of the friction factor. The new measured values, which were evaluated for a tube bundle of infinite extension, are clearly below equation (29).

If one assumes that equation (29) represents the maximum value of the Nusselt number in rod bundles arranged in square arrays, the calculations by Presser [10] and Markoczy [25] result in values which are too high and the calculations by Osmachkin [15] and Borishanskiy *et al.* [23] in values which are too low. The data by Deissler and Taylor [14] and those by Kokorev, Korsun and Petrovichev [24] can be recommended for the range of $P/D > 1{\cdot}3$, since the annulus solution of course is a good approximation for rod bundle geometries only for higher P/D ratios ($P/D > 1{\cdot}2$).

Because of the large amount of scatter in the measured values and the different boundary conditions of the experimental investigations, a safe statement about the heat transfer behavior of rod bundles in square arrays will not be possible until more experimental results, especially of more systematic studies, are available.

6. SUMMARY

In conclusion, it can be said that there is an upper limit, the annular zone solution, for the friction factor for an axial flow through rod bundles in square arrays. In a good approximation, this can be indicated to be

$$\lambda/\lambda_R = 1{\cdot}04 + 0{\cdot}06\,(P/D - 1) \tag{30}$$

for $P/D < 2{\cdot}0$ and $Re = 10^5$. This relation is confirmed excellently by the measured results. The friction factors under heated conditions are in good agreement with the isothermal values, if the friction factors are multiplied by $(T_W/T_B)^{0\cdot 5}$ and represented over the Reynolds number at the wall, as suggested by Taylor.

For the heat transfer, the upper limit of the Nusselt number recommended from results of measurements performed by the authors in analogy with the pressure drop is

$$\frac{Nu_B}{Re_B^{0\cdot 8}\,Pr_B^{0\cdot 4}} = 0{\cdot}018\left(\frac{2}{\pi}\,\frac{P}{D}\right)^{0\cdot 16} \times \left(1 + \frac{2}{\sqrt{\pi}}\,\frac{P}{D}\right)^{0\cdot 2}\left(\frac{T_W}{T_e}\right)^{-0\cdot 2}. \tag{31}$$

Safe statements about the Nusselt numbers require other experimental studies, especially systematic ones, to be performed.

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TRANSFERT THERMIQUE ET PERTE DE CHARGE DANS DES FAISCEAUX DE BARRES DISPOSEES EN RESEAUX CARRES

Résumé—L'article rapporte des résultats de mesures de perte de pression et de transfert thermique obtenus sur deux faisceaux à 9 et 16 barres en réseaux carrés. Les mesures sont faites avec l'hélium pour des nombres de Reynolds entre 10^4 et 3.10^5. Les faisceaux ayant un rapport $P/D = 1{,}283$ fonctionnent pour des flux de chaleur atteignant 160 W/cm et des températures allant jusqu'à 700°C. On mesure aussi les chutes de pression aux entretoises et la distribution circonférentielle de la température. Ces nouveaux résultats des mesures et les données disponibles dans la bibliographie sont utilisés pour établir des relations concernant la chute de pression et le transfert thermique dans des faisceaux de barres en réseaux carrés.

WÄRMEÜBERGANG UND DRUCKVERLUST AN STABBÜNDELN MIT QUADRATISCHER ANORDUNG DER STÄBE

Zusammenfassung—Es wird über Messergebnisse des Druckabfalles und des Wärmeüberganges an zwei Stabbündeln mit 9 beziehungsweise 16 Stäben in quadratischer Anordnung berichtet. Die Messungen wurden mit Helium für Reynolds-Zahlen zwischen 10^4 und 3×10^5 durchgeführt. Die Stabbündel hatten ein Teilungsverhältnis $P/D = 1{,}283$ und wurden bei Wärmestromdichte bis zu 160 W/cm² und Wandtemperaturen bis zu 700°C betrieben. Die Druckverluste an den Abstandshaltern und die Temperaturverteilung in Umfangsrichtung wurden ebenfalls gemessen.

Die neuen Messergebnisse und vorhandene Daten aus der Literatur dienten zur Ermittlung von Wärmeübergangs- und Druckverlustbeziehungen für Stabbündel mit quadratischer Anordnung der Brennstäbe.

ХАРАКТЕРИСТИКИ ТЕПЛООБМЕНА И ГИДРАВЛИЧЕСКОГО СОПРОТИВЛЕНИЯ СТЕРЖНЕЙ, СОБРАННЫХ В ПУЧКИ КВАДРАТНОГО СЕЧЕНИЯ

Аннотация—В статье изложены результаты измерения перепада давления и теплообмена в двух квадратных пучках стержней из 9 и 16 стержней соответственно. Измерения проводились в потоке гелия для чисел Рейнольдса от 10^4 до 3×10^5. Пучки стержней с относительным шагом $P/D = 1{,}283$ работали с тепловой нагрузкой до 160 вт/см при температуре стенки до 700°C. Измерялись также перепады давления в пучках и распределение температуры по окружности стержней.

Полученные результаты измерений и опубликованные в литературе данные использовались для уточнения зависимостей перепада давления и теплообмена в стержнях, собранных в пучки квадратного сечения.