Int. J. Heat Mass Transfer. Vol. 34, No. 1, pp. 199–207, 1991
Printed in Great Britain

0017-9310/91 \$3.00+0.00

Heat transfer from smooth and rough in-line tube banks at high Reynolds number

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(*Received* 16 *August* 1989)

Abstract—Pressure drop and heat transfer from smooth and rough in-line tube banks in cross-flow have been measured in the Reynolds number range $5\times10^4 \leqslant Re \leqslant 6\times10^6$. The coolant is pressurized air up to 40 bar. The roughness parameter is varied from $k/d = 0$ to 9×10^{-3}. The results from the local heat transfer measurements contribute to the understanding of the complicated flow around the tubes. Also for in-line arrangements a critical Reynolds number exists beyond which the heat transfer is improved while the pressure drop decreases. Finally, the entrance effect on heat transfer is considered.

1. INTRODUCTION

HEAT EXCHANGERS in cross-flow are applied when high heat transfer coefficients are required and the restrictions concerning the pressure drop are not too severe. This is, however, not the only reason, since other operation conditions may affect the decision on the heat exchanger type. The question whether in-line or staggered arrangements are favoured is not easy to answer. Generally staggered arrangements cause a higher heat transfer coefficient, but also a higher pressure drop compared to an in-line arrangement. This is, however, dependent on geometrical details (pitches), Reynolds number range and surface roughness. In the present work, for instance, a smooth in-line heat exchanger is treated which has the same heat transfer as the corresponding staggered version.

Various additional criteria are relevant for the selection of the heat exchanger type. An apparatus operating in a dusty surroundings must periodically be cleaned by a cleaning device which can easiest be applied in an in-line arrangement. Furthermore, helical type heat exchangers are very compact and not too sensitive to thermal stresses. They represent an in-line configuration with variable longitudinal pitches.

Finally staggered and in-line banks exhibit a different behaviour with respect to vibrations. Generally, staggered arrangements have a strong periodical contribution in the tube wake fluctuations and they are therefore prone to acoustic resonance effects. This periodicity, however, is easily predicted. In-line arrangements show a broad-banded wake spectrum and may therefore cause acoustic resonance only at the higher dynamic pressure of the flow. Those bundles, however, are sensitive to fluid elastic tube vibrations due to galloping, particularly when small pitches are applied.

The present paper deals with pressure drop and heat transfer from smooth and rough in-line tube arrangements. Very high Reynolds numbers must be verified to point out the effect of roughness clearly enough. It is the aim of this work to contribute to the understanding of flow and heat transfer phenomena observed in in-line arrangements. It is shown that the increase of heat transfer does not necessarily mean an increase of pressure drop. So this paper should not mainly complete the data stock of heat exchangers, but contribute to fundamental understanding in this field.

2. RESEARCH SITUATION

There are only a few papers dealing with the thermal hydraulics of cross-flow heat exchangers at high Reynolds numbers. Nearly at the same time a Russian group in Kaunas and a German group in Jülich started their research in this field on the background of nuclear technology. In 1967 a paper by Hammeke *et al.* [1] appeared on heat transfer and pressure loss of staggered, in-line and crossed bundles up to $Re = 2\times10^6$. In that research untreated rolled tubes were used having a roughness parameter of $k/d = 5\times10^{-4}$ which has already an effect on heat transfer and pressure drop. The data, valid for the in-line bundle, range up to $Re = 9\times10^5$. The curve representing the pressure drop coefficient has been averaged as independent of Re. It is, however, evident in spite of the large scale used in the figure that a certain dependence on Re exists as found also in the present study.

The Russian work is summarized in two books (1968 and 1982) [2, 3], giving empirical equations for the heat transfer and pressure drop results. While ref. [1] applied pressurized air or carbon dioxide to get high Reynolds numbers, refs. [2, 3] used a water channel. Here Reynolds numbers up to $Re = 10^6$ could be achieved. In 1968 a paper by Scholz [4] was published reporting the entrance effect of staggered and in-line tube banks. The experimental device was the same as

NOMENCLATURE

c_p	heat capacity	ρ	fluid density
d	tube diameter	τ_0	skin friction
h	roughness height	φ	angle of circumference measured from stagnation point
n	pointer	φ_s	angle of boundary layer separation
p	static pressure	φ_t	angle of boundary layer transition.
Δp	pressure drop		
s_t	transverse pitch of the tubes		
s_l	longitudinal pitch of the tubes		
u_s	velocity in the narrowest cross-section		Characteristic quantities
z	number of tube rows in the streamwise direction.	a	transverse pitch ratio, s_t/d
		b	longitudinal pitch ratio, s_l/d
		C_p	pressure coefficient, $1-(p(\varphi)-p_0)/((\rho/2)u_s^2)$
	Greek symbols	Nu	Nusselt number, $\alpha d/\lambda$
α	heat transfer coefficient	Pr	Prandtl number, $\eta c_p/\lambda$
η	fluid dynamic viscosity	Re	Reynolds number, $u_s d\rho/\eta$
λ	fluid thermal conductivity	ζ	pressure drop coefficient, $\Delta p/(z(\rho/2)u_s^2)$.

used by ref. [1], and the maximum Reynolds number was $Re = 10^6$. Niggeschmidt [5] tested a series of straight staggered and in-line tube banks with smooth surfaces reaching Reynolds numbers up to $Re = 8\times10^5$. In this investigation the critical Reynolds number was just exceeded. Groehn and Scholz [6] investigated four rough-surfaced in-line heat exchangers, exhibiting the same type of roughness elements as used in the present work, i.e. pyramids produced by a knurling process. Those authors applied very high roughness parameters compared to the present research, i.e. $k/d = 1.7\times10^{-2}$ and 3×10^{-2}. They investigated two different tube arrangements one being close to that of this work. The Reynolds number range covered was $7\times10^3 < Re < 8\times10^5$ using pressurized helium or air as a coolant.

An earlier paper by the author [7] deals only with the flow through smooth and rough in-line tube banks. Here the roughness was produced by emery paper. The effects of the boundary layer flow were studied in the Reynolds number range of $4\times10^4 < Re < 10^7$. Some of those results will be used to explain the heat transfer and pressure drop in the present paper.

3. EXPERIMENTAL TECHNIQUES

To reach such high Reynolds numbers as in the present tests a high pressure wind tunnel was applied operating with air up to 40 bar. Details of this apparatus are found in ref. [8]. The test bundle itself consisted of seven rows and three tubes per row. The tube diameter was 0.15 m, the transversal pitch $a = 2.0$, the longitudinal pitch $b = 1.4$ and the cross-section of the rectangular channel 0.5×0.9 m^2. The heat transfer was measured by electrically heating a separate tube providing guard sections to account for wall effects. The local probe subtended a circumferential angle of $\Delta\varphi = 2.3$ which leads to a reasonable spatial resolution of the local data. The techniques of determining the local and integral quantities and the problems arising from the method used are discussed in an earlier paper [9].

The rough surface of the test cylinder was produced by knurling. Thus a regular arrangement of pyramidal roughness elements was obtained. As the expensive knurling process could not be applied to all tubes of the test bundle, the dummy tubes were covered by emery paper. In previous investigations [10] an attempt was made to find out the appropriate emery paper corresponding to each particular pyramidal roughness.

4. RESULTS

4.1. *Pressure drop*

In a previous paper dealing with pressure drop and heat transfer from a staggered tube bundle in cross-flow at high Reynolds numbers [11] it has been demonstrated that the flow mechanism is rather similar to that of a single circular cylinder in cross-flow. This became evident even for integral quantities such as the pressure drop coefficient. Its dependence on the Reynolds number and roughness parameter could be explained on the basis of the knowledge about the single cylinder.

For in-line arrangements this similarity is not obvious on a first view, but we will see that effects like the critical flow conditions also occur. Due to the geometrical situation, however, the effect on the pressure drop can be quite different from that obtained for staggered arrangements. In this context it has been shown earlier [7] that for transcritical flow conditions the pressure drop coefficient of an in-line tube bundle

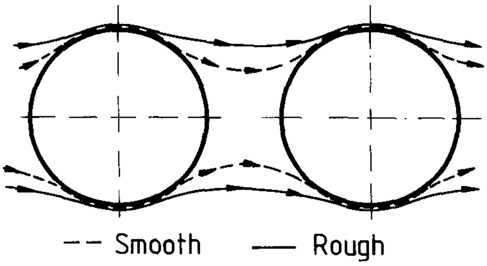

FIG. 1. Flow pattern of smooth or rough in-line tube bundles: ---, smooth; ——, rough.

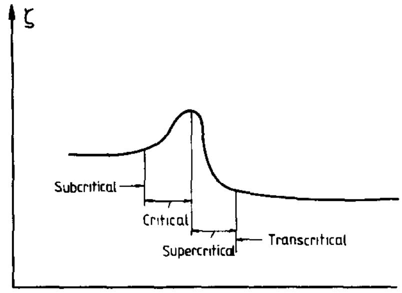

FIG. 3. Definition of the four flow regimes.

having the same geometry as used in the present tests decreases with increasing roughness parameter. This effect which is of great importance with respect to increasing the efficiency of heat exchangers could be explained with a view to the boundary layer parameter, particularly to the point of flow separation. Its variation with Reynolds number and surface roughness and the interaction with the point of impact in the succeeding row influence the flow pattern as shown qualitatively in Fig. 1. With increasing roughness parameter the transcritical flow separation occurs more upstream provoking a broader wake and leading to a larger value of angle of impact in the succeeding row. Thus the stream lines become 'smoother' with increasing roughness which results in a lower pressure drop compared to smooth surfaced tubes.

Figure 2 confirms the results of the pressure drop coefficient reported in ref. [7]. The just mentioned effect of decreasing pressure drop coefficient with increasing roughness parameter becomes obvious for high Reynolds numbers. The question may arise how the pressure drop of a tube bank can decrease with increasing roughness parameter. Of course, the friction stresses grow up with increasing roughness and cause also higher heat transfer coefficients. Their contribution to the total drag, however, is only a few per cent [7] whereas the rest is shape resistance. Thus the wall shear stresses control the pressure distribution which may in total result in a decrease of the pressure drop.

The particular curves have a shape as shown schematically in Fig. 3. As mentioned in a previous paper [7] four flow regimes can be distinguished: subcritical, critical, supercritical, and transcritical. The effect, however, is only of the order of $\pm 15\%$ and could be misunderstood as scattering of the results in a case where the experimental points are not as close to each other. In the present investigation, however, most of the experimental points are multiple measurements producing the same Reynolds number by increasing the velocity or the system pressure. Therefore, the effects observed can only be due to effects of the Reynolds number.

The just mentioned paper by Scholz (see Fig. 3 of ref. [4]) reports on pressure drop measurements of in-line bundles of a different number of rows. Starting from one row and continuing up to ten rows the development of the curve shape is impressively demonstrated. For a large number of rows a certain waviness of the pressure drop curve remains which can be recognized as a remaining effect of the critical flow conditions.

Figure 2 contains additional experimental data by Groehn and Scholz [6] for a very rough surfaced bundle of nearly the same geometry as the present one. Their results fit very well into the present roughness parameter field. It is evident that for transcritical flow conditions the pressure drop coefficient of the rough bundle is considerably lower—about 30%—than for the smooth bundle. At the same time the heat transfer is up to 40% higher, which results in a higher efficiency.

The critical Reynolds number is indicated in Fig. 2 by a sudden increase followed by a gradual decrease of the pressure drop coefficient. These data are evaluated for the use in Fig. 4 exhibiting the dependence of the critical Reynolds number on the surface

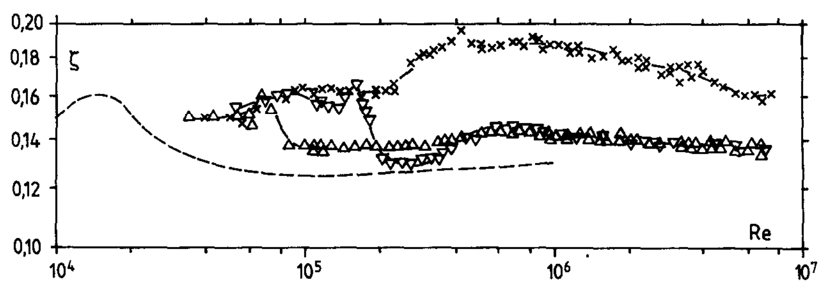

FIG. 2. Pressure drop coefficient of an in-line tube bank: ×, smooth; ▽, $k/d = 3\times10^{-3}$; △, $k/d = 9\times10^{-3}$; ---- ref. [6], $k/d = 1.7\times10^{-2}$.

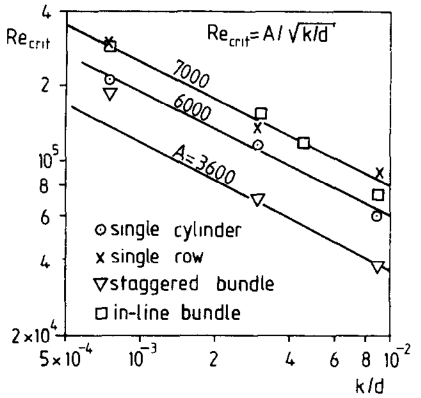

FIG. 4. Effect of surface roughness on the critical Reynolds number.

parameter. The figure contains also previous results [11] for the single cylinder, a staggered tube bundle and for one tube row. The present data for the in-line bundle collapse with those of the single row. The corresponding correlation is

$$Re_{crit} = 7000/\sqrt{(k/d)}.$$

4.2. *Local distribution of static pressure, skin friction and heat transfer*

To get more detailed information about the flow and heat transfer from the bundle local boundary layer quantities such as static pressure and skin friction are considered. Furthermore, the local heat transfer coefficient was measured. The techniques used for the determination of the flow parameters are reported in ref. [12] for the smooth surface and in ref. [13] for the rough case, respectively. Particularly, the application of the skin friction surface–fence is described together with details of the calibration procedure.

Figure 5 represents the experimental results of the local static pressure, the skin friction and the heat transfer of a smooth in-line tube bundle. The parameter chosen is the Reynolds number.

Figure 5(a) shows the static pressure distribution of a tube in the fifth row. The pressure maximum in the front part of the tube indicates the point of impact of the separated flow from the preceding tube. This location is characterized in Fig. 5(b) by the vanishing of the skin friction. From Fig. 5(c) it is evident that at this position the heat transfer also exhibits an intermediate maximum like a stagnation point heat transfer. This intermediate maximum can no longer be detected for the case representing the highest Reynolds number as the boundary layer becomes turbulent immediately downstream of the point of impact.

The point of impact varies with the Reynolds number. For subcritical conditions, i.e. $Re < 4 \times 10^5$, the stagnation line can be identified near $\varphi = 30°$. At supercritical and transcritical Reynolds numbers this location shifts to the vicinity of $\varphi = 40°–45°$. Upstream of the point of impact a recirculating zone exists indicated by the negative skin friction. Down-

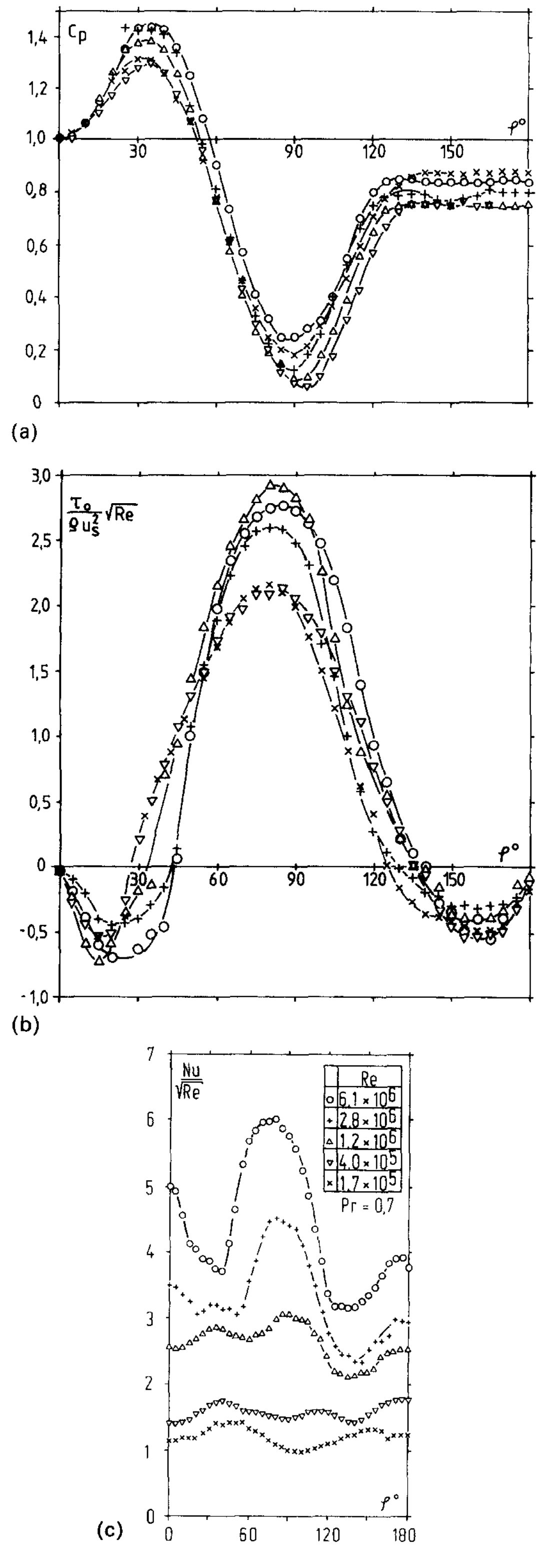

FIG. 5. Local distributions of: (a) static pressure; (b) skin friction; (c) heat transfer. Smooth in-line tube bundle.

stream of the point of impact a strong negative pressure gradient occurs causing an accelerated flow with increasing skin friction. Immediately upstream of the sign change of the pressure gradient near $\varphi = 90°$ the

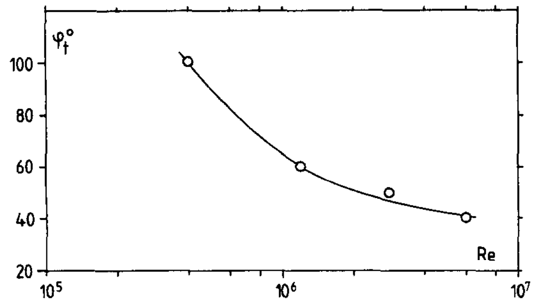

FIG. 6. Location of the transition point for a smooth in-line tube bundle.

skin friction exceeds its maximum, strongly diminishing under the effect of the following adverse pressure gradient. At the point of zero-skin friction the boundary layer separation is reached. It is obvious that the skin friction maximum for $Re > 4\times10^5$ no longer varies with $\sqrt{(Re)}$ as the boundary layer is turbulent. Thus the peak value increases with increasing Reynolds number.

The surface area covered by an established boundary layer ranges from about $\varphi = 30°$ to $140°$ showing a small variation with Reynolds number concerning the points of impact and separation. For subcritical flow conditions ($Re = 1.7\times10^5$) a laminar boundary layer exists in this part showing a decrease of local heat transfer with increasing Reynolds number (Fig. 5(c)). The same holds for $Re = 4\times10^5$, but this curve exhibits a maximum indicating that the boundary layer becomes turbulent at about $\varphi = 90°$. With increasing Reynolds number the transition point shifts upstream. This is characteristic for the supercritical flow regime. Finally, the transition occurs immediately near the point of impact ($Re = 6.1\times10^6$) which means that transcritical flow conditions are reached.

Beyond the separation point the rearward recirculation region begins. The base pressure here corresponds with the value of the pressure drop coefficient, i.e. high base pressure causes a low pressure drop and vice versa.

Figure 6 representing the transition point from laminar to turbulent boundary layer results from the evaluation of Fig. 5(c). It is seen that the location of transition shifts upstream with increasing Reynolds number.

Figures 7(a)–(c) illustrate the results of static pressure, skin friction and local heat transfer for a rough in-line tube bundle. Due to the high roughness parameter, $k/d = 9\times10^{-3}$, the flow is already transcritical for the three Reynolds numbers, i.e. the boundary layer is turbulent throughout the total length.

It is evident from Fig. 7(b) that the point of impact is located near the position identified for the smooth tube at the transcritical flow condition, i.e. near $\varphi = 45°$. The point of boundary layer separation, however, has shifted upstream to $\varphi = 110°$ as the

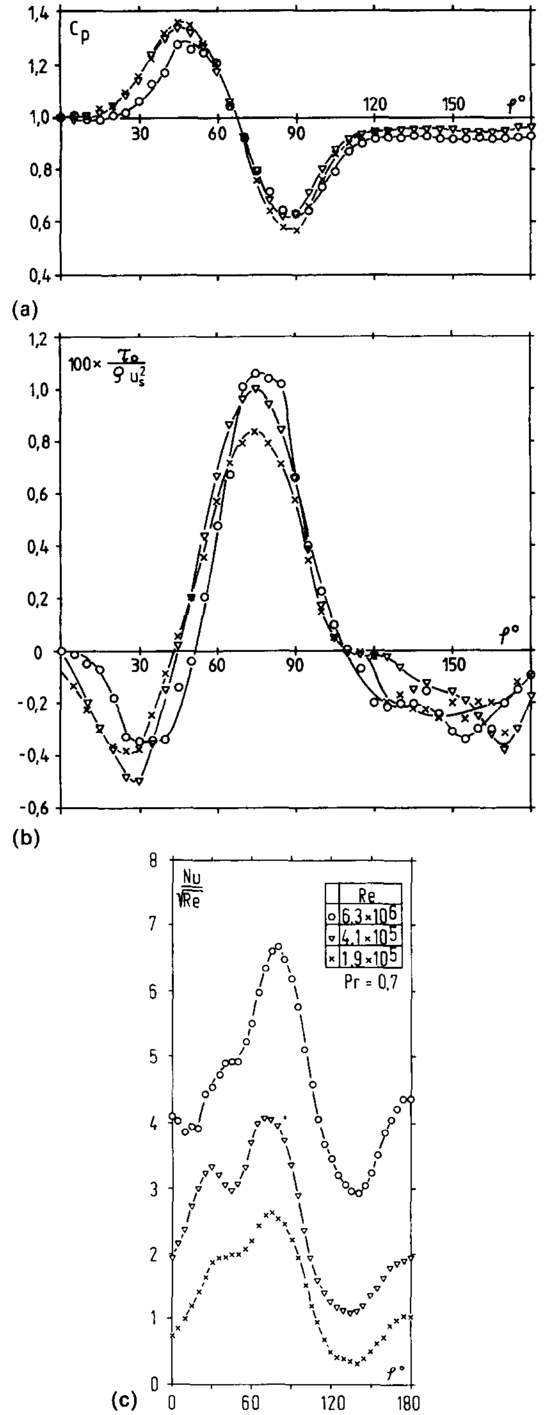

FIG. 7. Local distributions of: (a) static pressure; (b) skin friction; (c) heat transfer. Rough in-line tube bundle $k/d = 9\times10^{-3}$.

friction forces have considerably increased due to the rough surface. For example, the peak value of the skin friction at $Re = 5\times10^6$ is by one order of magnitude larger than for the smooth case. Note that the skin friction was made dimensionless in Fig. 7(b) without normalizing with $\sqrt{(Re)}$. This seemed to be useful as the skin friction is nearly independent of Re for the flow past rough surfaces at transcritical conditions.

Figure 7(c) shows that the heat transfer strongly increases immediately downstream of the point of impact which indicates the existence of turbulent boundary layers. It is surprising to see that if one integrates the local values the integral of the heat transfer for $Re = 6 \times 10^6$ is nearly the same for the rough and the smooth bundle. Gradual differences occurring in the boundary layer area are compensated for in the recirculation zones. This evidence is confirmed by the results of the integral heat transfer measurement (Fig. 8).

4.3. *Total heat transfer*

The total heat transfer from in-line bundles of different surface roughness is shown in Fig. 8. In the low and the high Reynolds number regimes, i.e. at subcritical and at transcritical collapse while an improvement of heat transfer is observed in the critical and supercritical flow range for the rough bundles. The maximum effect is about 40%. This improvement is, of course, due to the premature onset of turbulence generated by the surface roughness.

The curves for the two rough bundles collapse beyond the particular critical Reynolds numbers. In this context Fig. 2 also shows that for transcritical conditions the pressure drop curves nearly collapse. Similar results are known from earlier tests with staggered tube bundles. It may be concluded that already very small surface roughnesses cause a maximum effect on heat transfer and that it is useful to apply roughnesses greater than $k/d \approx 5 \times 10^{-3}$ only if the range of improved heat transfer is required at lower Reynolds numbers.

The comparison of the total heat transfer from a smooth in-line bundle with that of a corresponding staggered apparatus reveals that the experimental curves collapse over the whole range of investigation. At the same time the pressure drop is lower for the in-line arrangement in the subcritical and transcritical regime which means that the efficiency of the in-line bundle is higher.

If rough surfaces can be used for a technical application the present results show that rough in-line arrangements have some advantages: the pressure drop coefficient of the rough in-line bundle is lower for some Reynolds number range than that of the smooth bundle. As at the same time the heat transfer is improved the rough bundle will have a higher efficiency than the smooth one.

It may be of interest to know that the influence of surface roughness on heat transfer is more important for staggered arrangements than for in-line configurations. This may be due to the fact that the improvement of heat transfer is predominantly associated with boundary layer effects and that the percentage of the surface covered with an established boundary layer is higher for staggered than for in-line bundles.

Experimental results of other authors exist only for $Re < 10^6$. In this range the present data of the smooth bundle agree within 10% with those of Hammeke *et al.* [1], Niggeschmidt [5] and Žukauskas [3] (Fig. 9). Applying the general formulae, given in ref. [14], to a bundle of the present geometry good agreement is achieved in the subcritical regime. For transcritical conditions, however, an underprediction of about 30% compared to the present results occurs.

Groehn and Scholz [6] investigated in-line tube bundles of nearly the same geometry as in the present

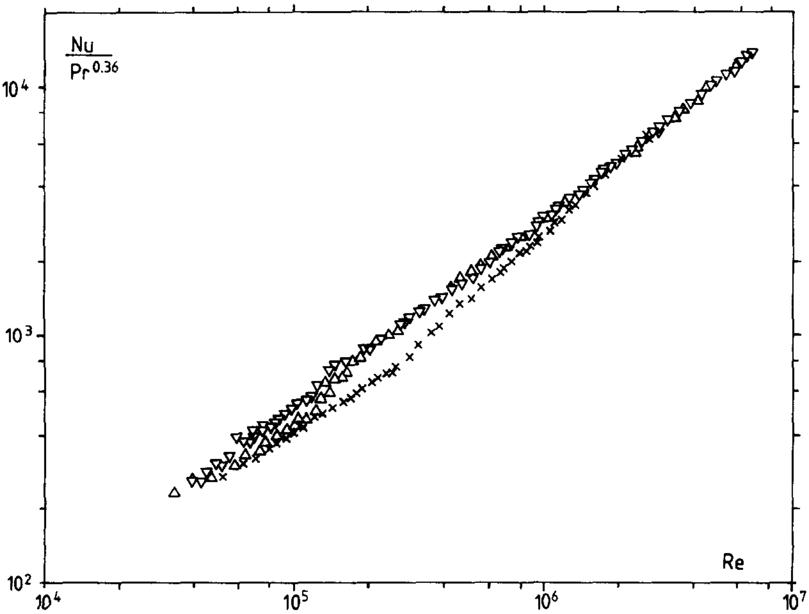

FIG. 8. Total heat transfer of an in-line tube bundle. Symbols as in Fig. 2.

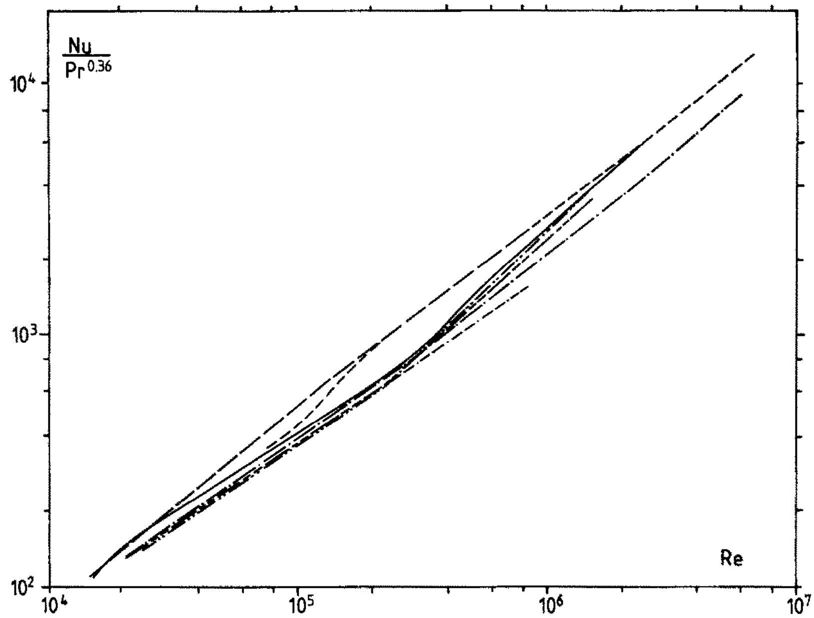

FIG. 9. Data comparison for in-line tube bundles. Smooth surface: — · · —, ref. [1]; ——·——, ref. [5]; ————, ref. [2]; — · —, ref. [14]; ——, present. Rough surface: ——, ref. [6], $k/d = 1.7\times10^{-2}$; – – – –, present, $k/d = 3\times10^{-3}$.

work. The roughness parameters, however, were $k/d = 1.7\times10^{-2}$ and 3×10^{-2}, i.e. more than a factor of three higher than the coarsest one used in these tests. The experimental curves for $k/d = 1.7\times10^{-2}$ collapse with the present one for rough surfaces over the whole range of investigation. The data for $k/d = 3\times10^{-2}$ are even somewhat lower. This confirms the aforementioned statement that the total heat transfer cannot exceed the threshold already reached for $k/d = 3\times10^{-3}$. If the roughness becomes much higher a secondary flow probably establishes at the base of the roughness elements which may act as a heat resistance.

4.4. *Entrance effects*

The flow conditions in the entrance section of a tube bundle are different from those in the interior. Therefore, an effect on heat transfer is expected. In the first row the oncoming velocity and the turbulence level are lower than inside the heat exchanger. Furthermore, the frontal separated region does not exist, but an established boundary layer, which is laminar up to very high Reynolds numbers. The second row stands already in the wake of the first one. The outer flow, however, has not yet reached the high turbulence level as occurring further downstream. Thus the lowest values of heat transfer are expected in the first row and the second row will come close to the average value. This is to some extent different from staggered configurations since here the second row is positioned downstream of the gaps between the tubes and not immediately in the wake.

The experimental data of the smooth tube are displayed in Fig. 10(a). For $Re = 10^5$ the flow is still subcritical throughout the bundle. The fraction of the first row heat transfer is 60% compared to the average value. The values for $Re = 5\times10^5$ and 10^6 would have been expected somewhat lower, but it is seen from the local heat transfer distributions (Fig. 4(c)) that the improvement of heat transfer due to the turbulence effects is not so important. The entrance effect becomes severest for the highest Reynolds number, $Re = 5\times10^6$. Whereas the boundary layers in the interior of the bundle are turbulent (Fig. 4(c)), the front part of the first row is covered up to about $\varphi = 60°$ with a laminar boundary layer. This is known from earlier experiments (Figs. 6 and 8 of ref. [7]). Therefore, the first row contributes only 40% to the average value.

Figure 10(b) shows the trend that for $k/d = 3\times10^{-3}$ the heat transfer of the first row increases with increasing Reynolds number for $Re > Re_{crit}$. This results from the fact that the boundary layer of the first row strongly turns out to undergo transition to turbulence with increasing Reynolds number under the effect of surface roughness. In Fig. 10(c) which represents the data for $k/d = 9\times10^{-3}$ the frontal boundary layer for $Re = 5\times10^6$ is already turbulent in the immediate vicinity of the stagnation point. Thus we find a contribution of 90% in the first row.

The entrance effect is, of course, also influenced by the geometrical conditions of a bundle which is not treated herein. The aim of this paper, however, is

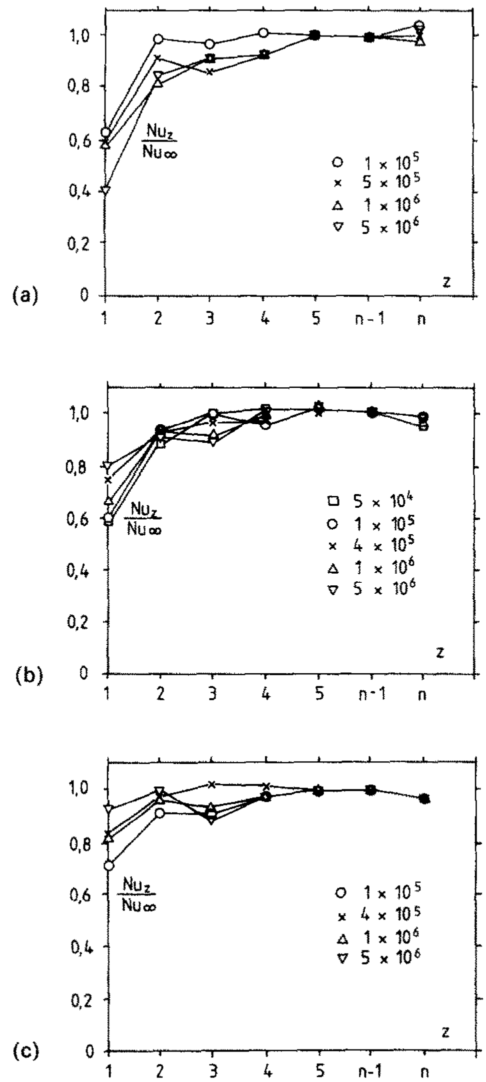

FIG. 10. Entrance effect on heat transfer of in-line tube bundles: (a) smooth; (b) $k/d = 3\times10^{-3}$; (c) $k/d = 9\times10^{-3}$.

to point out the fundamental aspects governing the entrance effect.

5. CONCLUSION

For an in-line tube bundle of transverse pitch $a = 2$ and of longitudinal pitch $b = 1.4$ the flow and heat transfer mechanisms around the critical Reynolds number and the effect of surface roughness are exemplarily studied. In-line arrangements inherently have the potential to decrease the shape resistance and thereby the pressure drop by applying rough surfaces. At the same time the heat transfer can be improved, at least in restricted ranges of the Reynolds number. Thus surface roughening of in-line arrangements is an appropriate measure to increase the efficiency of a heat exchanger. The roughness parameter determines the value of the critical Reynolds number from which the improvement of heat transfer starts. The maximum enhancement to be obtained already occurs for $k/d \approx 3\times10^{-3}$. A further increase of the roughness parameter does not result in a higher heat transfer.

Compared to a staggered tube bundle the entrance effect of an in-line arrangement does not touch so many rows, as the tubes of the second row are already exposed to the wake of the preceding tubes. Particularly the first row shows a considerable departure from the average value which can range from 40 to 90%. Low values occur when the boundary layer in the front part of the tube belonging to the first row is still laminar, whereas turbulent boundary layers prevail in the interior of the heat exchanger. This is, for instance, the case for a smooth bundle at high Reynolds number.

Acknowledgement—This work was done with the valuable help of my co-workers H. Gillessen, F. Hoffmanns, H. Reger and R. Rommerskirchen. I would like to thank them for their engagement in preparing, performing and evaluating the numerous tests.

REFERENCES

1. K. Hammeke, E. Heinecke und F. Scholz, Wärmeübergangs- und Druckverlustmessungen an querangeströmten Glattrohrbündeln, insbesondere bei hohen Reynolds-Zahlen, *Int. J. Heat Mass Transfer* **10**, 427–446 (1967).
2. A. Žukauskas, V. Makarevičius and A. Šlančiauskas, *Heat Transfer in Banks of Tubes in Cross Flow of Fluid, Teplofizika*. Mintis, Vilnius (1968) (in Russian).
3. A. Žukauskas, *Convective Heat Transfer*. Nauka, Moscow (1982).
4. F. Scholz, Einfluß der Rohrreihenzahl auf den Druckverlust und Wärmeübergang von Rohrbündeln bei hohen Reynolds-Zahlen, *Chemie-Ingr-Tech.* **40**(20), 988–995 (1968).
5. W. Niggeschmidt, Druckverlust und Wärmeübergang bei fluchtenden, versetzten und teilversetzten querangeströmten Rohrbündeln, Diss. TH Darmstadt, D17 (1975).
6. H. G. Groehn und F. Scholz, Wärme- und strömungstechnische Untersuchungen an fluchtenden Rohrbündelwärmetauschern aus pyramidenförmig aufgerauhten Rohren, Jül-Rep. 1437, July (1977).
7. E. Achenbach, On the cross flow through in-line banks with regard to the effect of surface roughness, *Wärme- und Stoffübertr.* **4**, 152–155 (1971).
8. H. Grosse und F. Scholz, Der Hochdruck-Gaskanal, *Kerntechnik* **7**(4), 150–158 (1965).
9. E. Achenbach, Total and local heat transfer from a smooth circular cylinder in cross-flow at high Reynolds number, *Int. J. Heat Mass Transfer* **18**, 1387–1396 (1975).
10. E. Achenbach, The effect of surface roughness on the heat transfer from a circular cylinder to the cross flow of air, *Int. J. Heat Mass Transfer* **20**, 359–369 (1977).
11. E. Achenbach, Heat transfer from a staggered tube bundle in cross-flow at high Reynolds numbers, *Int. J. Heat Mass Transfer* **32**, 271–280 (1989).
12. E. Achenbach, Distribution of local pressure and skin friction around a circular cylinder in cross-flow up to $Re = 5\times10^6$, *J. Fluid Mech.* **34**, 625–639 (1968).
13. E. Achenbach, Influence of surface roughness on the cross-flow around a circular cylinder, *J. Fluid Mech.* **46**, 321–335 (1971).
14. *VDI-Wärmeatlas*. VDI, Düsseldorf (1984).

TRANSFERT THERMIQUE A GRAND NOMBRE DE REYNOLDS POUR DES TUBES LISSES OU RUGUEUX EN LIGNE

Résumé—La perte de charge et le transfert thermique pour des assemblages de tubes lisses ou rugueux en ligne, attaqués transversalement par l'écoulement, sont mesurés dans le domaine de nombre de Reynolds $5\times10^4 \leqslant Re \leqslant 6\times10^6$. Le fluide refroidissant est l'air comprimé jusqu'à 40 bar. Le paramètre de rugosité k/d varie de 0 à 9×10^{-3}. Les résultats des mesures locales de transfert thermique contribuent à la compréhension de l'écoulement autour des tubes. Aussi pour des arrangements en ligne, il existe un nombre de Reynolds critique pour lequel le transfert thermique est augmenté alors que la perte de charge est diminuée. On considère enfin l'effet d'entrée sur le transfert thermique.

WÄRMEÜBERGANG VON GLATTEN UND RAUHEN FLUCHTENDEN ROHRBÜNDELN BEI HOHEN REYNOLDS-ZAHLEN

Zusammenfassung—An fluchtenden querangeströmten Rohrbündeln glatter oder rauher Oberfläche wurden der Druckverlust und der Wärmeübergang im Bereich der Reynolds-Zahlen $5\times10^4 < Re < 6\times10^6$ gemessen. Als Strömungsmedium diente Luft unter einem Druck bis zu 40 bar. Der Rauhigkeitsparameter wurde in den Grenzen $0 < k/d < 9\times10^{-3}$ variiert. Die Ergebnisse der örtlichen Wärmeübergangsmessungen geben Aufschluß über die komplizierten Strömungsvorgänge an den Wärmetauscherrohren. Es zeigt sich, daß auch für fluchtende Rohranordnungen eine kritische Reynolds-Zahl existiert, bei deren Überschreiten eine Verbesserung des Wärmeüberganges und eine Verminderung des Druckverlustbeiwertes beobachtet wird. Im letzten Teil der Arbeit wird der Einlaufeffekt auf den Wärmeübergang behandelt.

ТЕПЛОПЕРЕНОС ОТ КОРИДОРНЫХ ПУЧКОВ ГЛАДКИХ И ШЕРОХОВАТЫХ ТРУБ ПРИ ВЫСОКОМ ЧИСЛЕ РЕЙНОЛЬДСА

Аннотация—Измерены перенады давления и теплоперенос в коридорных пучках гладких и шероховатых труб при их поперечном обтекании в диапазоне значений числа Рейнольдса $5\times10^4 \leqslant Re \leqslant 6\times10^6$. Охлаждающий воздух находился под давлением до 40 бар. Параметр шероховатости изменялся от $k/d = 0$ до 9×10^{-3}. Результаты измерений локального теплопереноса позволили выяснить сложный характер обтекания труб. Кроме того, для коридорных расположений обнаружено критическое число Рейнольса, выше которого теплоперенос улучшается, а перепады давления снижаются. Рассмотрено также влияние на теплоперенос входного участка.