Int. J. Heat Mass Transfer. Vol. 30, No. 12, pp. 2603–2609, 1987
Printed in Great Britain

0017–9310/87 $3.00+0.00

Heat transfer enhancement by means of flag-type insert in tubes

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(*Received* 29 *July* 1986 *and in final form* 7 *May* 1987)

Abstract—A flexible 'start-up' insert in the form of a rigid flag hinged on a diametral rod was found to enhance heat transfer in turbulent flow in a tube. The insert was capable of sideways oscillation and produced a heat transfer performance superior to a twisted tape and coiled wire start-up inserts (e.g. inserts of 1–10 tube diameters in length placed at the pipe inlet). Heat transfer over a tube length equivalent to 5–15 diameters downstream of the flag, was enhanced up to three times relative to that for the empty tube.

INTRODUCTION

THE ENHANCEMENT of convective heat transfer in tubes is a basic step towards high performance heat exchangers, particularly where tube-side heat transfer coefficients are limiting due to the nature of the fluid. Both capital and operational costs can be reduced if overall heat transfer coefficients in exchangers are increased since, in general, size, weight and pumping power can be made significantly smaller.

In general, performance can be improved by the insertion of various kinds of devices in a tube, such as a twisted tape, coiled wire, axial core inserts, as well as by the provision of internal ribs, fins or surface roughness. An increase in pressure drop will generally accompany heat transfer enhancement, and the choice of an appropriate insert must be carefully made to suit the fluid being handled and the flow regime, in order to optimize the net improvement.

In recent years, the performance of large numbers of potentially commercial enhancement devices have been reported. A comprehensive study has been made by Bergles *et al.* [1–4], with emphasis in twisted tape generated swirl flow. Beneficial effects are obtained with internal fins [5, 6], wire coils [7–9], slat blockage [10], wire brush [11], internal ribs [12], and other swirling devices [13, 14], just to name a few. In general, the accompanying pressure drop increase is larger than the heat transfer rate increase, although this feature does not necessarily demerit the inserts.

The bulk of those studies refer to inserts that occupy the whole length of the heat exchanger tube, the logic being that the performance of the whole of the surface is to be enhanced. However, flow destabilization is evident downstream of any insert, and the so-called 'start-up' inserts can be used advantageously [15–18]. These short lengths of inserts, placed at the inlet of the heat transfer tube, can be effective over as much as 60 diameters of downstream pipe. Other objects at the pipe inlet, such as a tangential vane [19] and a slot [20], have also been studied. Heat transfer rates can thus be increased by as much as 80% [15]. Although pressure drop data are scarce, it seems yet again that pressure drop increases are larger than heat transfer enhancement in these cases.

It is apparent, however, that all inserts reported are rigidly fixed inside the tube, and no gross insert motion is permitted. Since our object is essentially to destabilize the thermal boundary layer within the tube, we felt a device capable of dynamically interacting with the flow, and taking its driving energy from the flow, might produce the desired flow destabilization. Since the characteristic dimensions of such a single 'start-up' device could be large, a well-ordered secondary instability might be generated. This instability might be less 'dissipating' than the high shear devices characterized by wire inserts. In order to test these ideas, a simple flapping plate, or flag, was tested.

It consisted of a small, 30 gauge copper sheet, approximately 0.6 tube diameters wide and from 0.5 to 2 diameters long. This plate is hinged at its upstream end about a 0.1 mm diameter stainless steel rod that traverses the heat transfer tube along a vertical diameter. The flag thus manufactured is now placed at the inlet of the thin-walled copper heat transfer tube, and is in continuous motion due to the action of the fluid flow, oscillating very much the way a rigid flag does in free flow.

EXPERIMENTAL APPARATUS AND PROCEDURE

A laboratory heat transfer apparatus was built to heat water flowing in a copper pipe with saturated steam as the heating medium condensing on its outer surface. Figure 1 shows a schematic diagram of the

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NOMENCLATURE

A	tube oscillation amplitude
c	specific heat of fluid
D	tube internal diameter
f	d'Arcy–Weisbach friction factor
h	in-tube heat transfer coefficient
k	fluid thermal conductivity
L	tube length
m	mass flow rate
Nu	Nusselt number, hD/k
Δp	pressure drop
Pr	Prandtl number, $c\mu/k$
Q	total heat flow
Re	Reynolds number, $uD\rho/\mu$
Re_v	vibration Reynolds number, $vA\rho/\mu$
T	temperature
U	overall heat transfer coefficient
u	bulk mean velocity
v	mean velocity of tube oscillation.

Greek symbols

μ	dynamic viscosity
ρ	mass density.

experimental facility, and Fig. 2 depicts the flag-type insert.

Two copper test tubes were employed: one of length $L = 1.6$ m, internal diameter $D = 14.4$ mm, and the other of $L = 1.78$ m, $D = 32$ mm. The settling length at the inlet was 1.2 m for the first and 2 m for the second test tube. Water was pumped at rates from 0.15 to 0.45 kg s^{-1}, corresponding to tube-side Reynolds numbers from 20 000 to 50 000 in the first test section. Rates were 0.2–2.5 kg s^{-1} yielding Reynolds numbers from 12 000 to 135 000 in the second tube. Water temperature was 15–60°C, corresponding to Prandtl numbers from about 8 to 3. Thermocouples were employed for tube surface and water inlet and outlet temperature measurement. A high-speed Hewlett–Packard series 9000 scanner coupled to a microprocessor allowed the measurement of mean temperatures over a period of several minutes, during which flow conditions were kept constant. Water mass throughput was measured by means of an orifice meter, which in turn was calibrated in separate tests.

The amount of total condensate was also metered. The condensate could be measured at eight fixed stations along the length of the second test section, each station corresponding to 210 mm of tube length, or about 7 diameters.

The pressure drop along the test section was measured by means of static manometers. These were connected to pressure taps at the inlet and outlet of the test section. The pressure taps were perpendicular

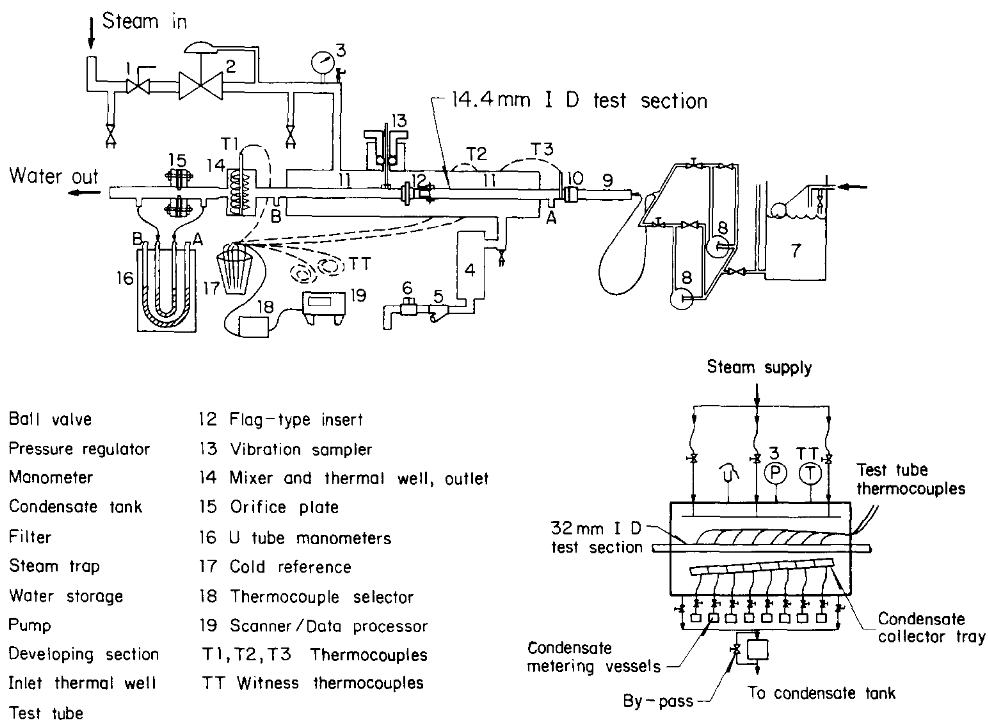

FIG. 1. Layout of experimental apparatus. Insert: detail of eight-station condensate collector.

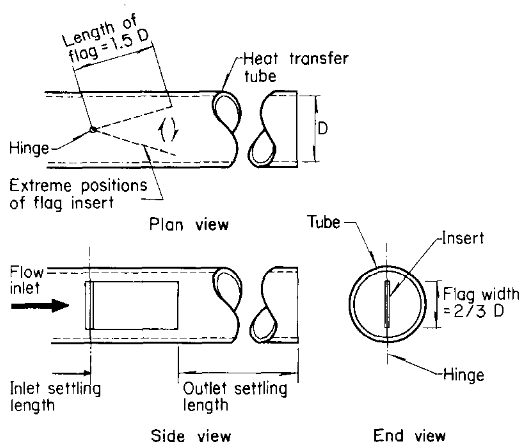

FIG. 2. Schematic view of flag-type promoter.

to the tube surface and formed an angle of 45° with the vertical.

Measurements for inlet and outlet water temperature and pressure were averaged in groups of three test runs under the same operating conditions. Where one of the three values disagreed by more than 3%, the three were discarded. The steam temperature was monitored to ensure steady state, and was kept between 102 and 102.5°C. Calibration of the thermocouples and the test procedure result in an inlet and outlet temperature uncertainty of 0.3°C typical and 1°C maximum.

The overall heat transfer coefficient was calculated by dividing the total heat flow (found by means of a heat balance on the water side) by the logarithmic mean temperature difference between water and steam and by the total heat transfer area. A 'local' heat transfer coefficient was estimated for each station from the condensate that was collected locally. This coefficient was found by dividing the local heat flow by the local temperature difference between the tube wall and the water.

The waterside local heat flux was compared to condensate-side heat flux and only sets of data that agreed within 10% were retained. Uncertainty in the evaluation of the overall heat transfer coefficient U and in the total heat flow Q are 7% typical and 12% maximum. Uncertainty in the estimate of the local (station) heat transfer coefficient is estimated at 15%.

The internal flow Reynolds number was calculated, for both un-enhanced and enhanced heat transfer, as for an empty tube, i.e. $Re = uD\rho/\mu$, where ρ and μ are respectively density and dynamic viscosity at mean bulk temperatures, D is internal tube diameter and u is mean fluid bulk velocity.

Finally, in order to check if the flag induced tube oscillation, surface vibration was monitored by means of a 25–30 000 Hz piezoelectric probe at the tube surface. The amplitude and frequency of oscillations were recorded with an uncertainty of about 5%.

The rig was commissioned and its plain-tube performance assessed as acceptable against standard correlations. Three flag sizes were then tested in the first test section. Each flag was located at the same point at the tube inlet, and each flag was of the same width and thickness. Flag lengths were 0.5, 1.5 and 2.5 cm. Their performance was compared with three other start-up inserts: (I) a flat, straight, ribbon, 18 cm long and 1.44 cm wide, 0.3 mm thick, made of copper, tightly fitted (but not welded) to the tube wall; (II) a twisted tape, 0.3 mm thick, 18.5 cm long, 1.44 cm wide, with a longitudinal pitch of 9 cm, made of copper, also tightly fitted, and (III) a 21 cm long wire coil insert, made of 1.2 mm diameter copper wire, pitch of 2.4 cm, again tightly fitted inside the pipe. The last one, being the longest of the set, occupied only the initial 13% of the total heating length of the test tube. The settling length available after the 'start-up' devices was about 1.39 m, or 95 internal tube diameters, in excess of the 60 or so diameters at which it was expected that all induced disturbances would have disappeared.

RESULTS

A first set of values of heat flow for various flow rates was obtained for the smooth tube without an insert in the smaller heat transfer test section. Using a Wilson plot technique similar to that given in ref. [21], the combined value of the wall resistance and the condensing resistance was determined. The tubeside convection coefficient was then calculated, and the value of the Nusselt number was thus established. A comparison between results and a standard correlation [22] for the smooth tube was made and the performance of the apparatus and the measuring technique were deemed satisfactory. A similar procedure was then followed for tests with the flag-type insert, using the 1.5 cm long flag placed at the inlet of the test section, and with the helical wire, the twisted tape and the straight tape. A dimensionless representation of these results is included in Fig. 3. Since these results refer to the whole tube length, the enhancement effect of the inserts has been averaged along the test section.

Friction factors for the smooth tube and for the test section including the above-mentioned inserts were calculated from the definition, using pressure drop readings obtained during the heat transfer experiments for this purpose. Although a certain scatter of data points was apparent, as Fig. 4 shows, the general tendency was considered satisfactory. The results for the smooth tube seem to fall near the region indicated by work from other authors [22].

Heat transfer results due to the static start-up inserts I to III in Fig. 3 follow a characteristic parallel to that of smooth tube heat transfer, within the range of Reynolds numbers from 20 000 to 50 000. Pressure drop enhancement, on the other hand, increases more rapidly at high Reynolds numbers, following a divergent characteristic as shown in Fig. 4. Consequently, for the static inserts tested, the higher the flow velocity, the higher the pressure drop penalty. The trend of the

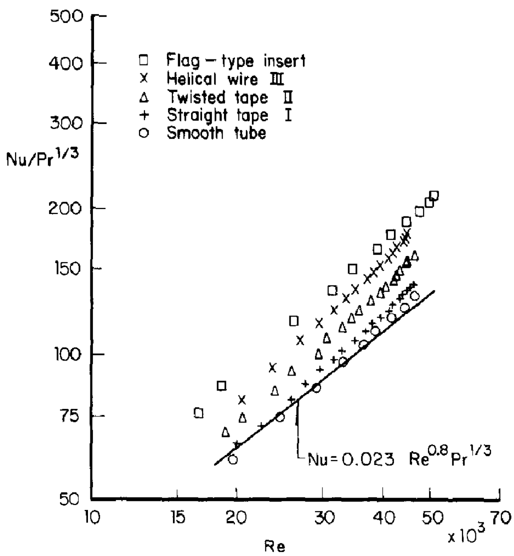

FIG. 3. Dimensionless heat transfer for smooth tube and for fixed inserts tested, compared with flag-type insert (experimental).

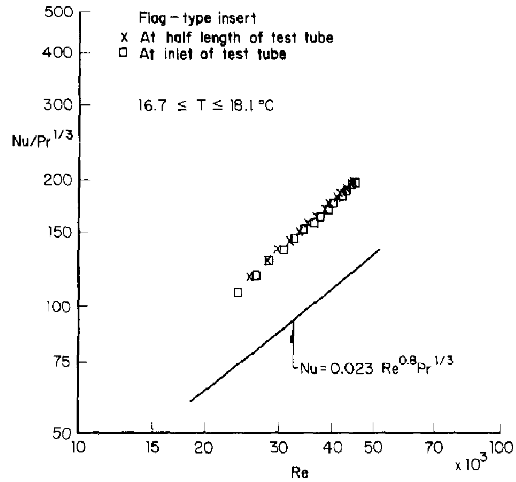

FIG. 5. Dimensionless heat transfer for flag-type insert at inlet and at half length of test section.

results for the flag-type promoter is quite different: heat transfer enhancement increases with flow rate, and pressure drop characteristic is very similar to that of the smooth tube. The trends of these results reveal that the mechanism of heat transfer enhancement in the presence of a flexible insert is likely to be very different from the enhancement mechanisms at work with rigid, static inserts.

A further set of tests was carried out with the flag placed halfway downstream. This arrangement would result in allowing only 48 tube diameters downstream of the insert for the enhancement to be effective, assuming that no upstream effects are at work. Results for this flag position and for the flag placed at the inlet of the test section are shown in Fig. 5, which indicates a similar enhancement in both cases. It is clear therefore that the enhancement effect has disappeared within the available downstream settling length.

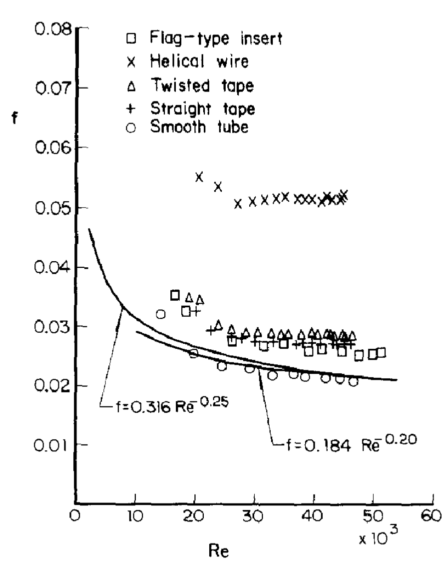

FIG. 4. Friction factor for smooth tube and for tests with various inserts, as in Fig. 3.

The observation that the flag-type insert produces enhancement confined to a few tube diameters downstream, is supported by local heat flux measurements. The flag insert was installed at the inlet of the first test station (each 'station' comprising a test tube length of about 7 diameters). In tests performed with the larger, 32 mm diameter test section, a very sudden fall of heat transfer was evident in the test section downstream of the insert. However, a very marked enhancement is at work in the test station where the flag insert is placed (i.e. for positions from 1 to 7 tube diameters downstream of the insert). Figure 6 presents the dimensionless tubeside heat transfer coefficient for each test station. The enhancement in the first station varied from a factor of more than 2 for Reynolds numbers of 20 000 to a factor of around 3 for Reynolds numbers of 100 000. In the second station, a small enhancement, up to 1.5 times the empty tube heat transfer, was measured. No traces of activity could be detected at points further removed from the insert.

Heat transfer results appeared not to be sensitive to flag length. However, pressure drop readings show a slight increase in friction when longer flags are used. Pressure drop test results without heat transfer, where water temperature was kept constant at 16°C, are shown in Fig. 7. Pressure drop data in the presence of heat transfer were not as clearly differentiated for the various flag lengths.

It is apparent that the mechanism responsible for enhancement in the presence of a flag insert is not of the same nature as that associated with the static inserts tested. In order to verify if tube surface vibration was producing heat transfer intensification, the oscillation at the tube surface was measured in the direction perpendicular to the tube axis, in the smaller test section. Surface vibration characteristics revealed

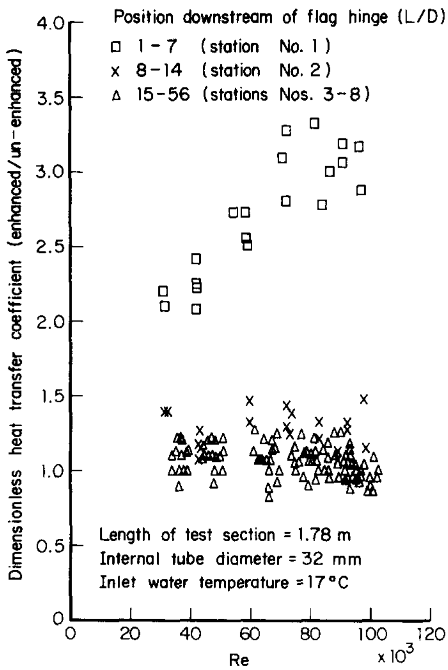

FIG. 6. Enhanced in-tube heat transfer coefficient for various positions downstream of the flag ('stations' numbered 1–8 downstream of the flag).

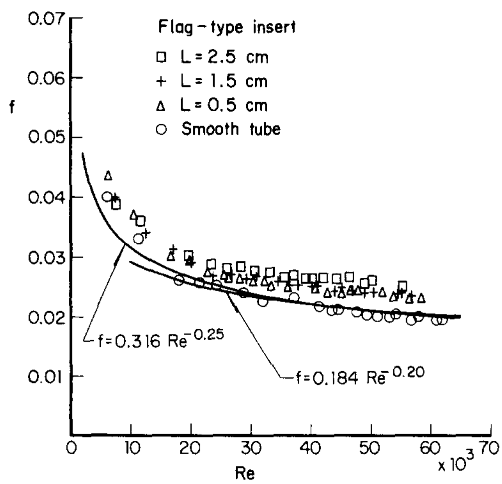

FIG. 7. Friction factor for flags of three different lengths.

that oscillations exhibit constant frequency, irrespective of flow rate, and amplitude growing with flow throughput, as shown in Fig. 8. These results refer to the 1.5 cm long flag (about 1 tube diameter), the twisted tape and the smooth tube in isothermal flow at 16°C.

A 'vibration Reynolds number' is defined as the product of the mean vibration velocity by the vibration amplitude divided by the liquid kinematic viscosity [23]. It is found to be about four orders of magnitude smaller than the axial flow Reynolds number. Furthermore, the vibration Reynolds number has similar values for the various inserts studied and does not exhibit a parallelism with the enhancement observed. Vibration measurement suggests that surface oscillation is unlikely to play a major part in enhancement by means of flag-type inserts [24].

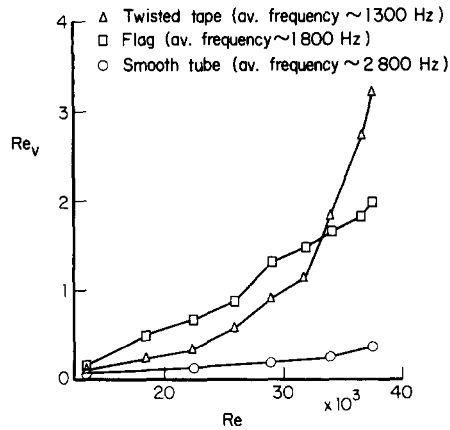

FIG. 8. Vibrational Reynolds number vs main-flow Reynolds number for smooth tube, short twisted tape and 1.5 cm long flag-type insert.

DISCUSSION

The results agree well with previous findings in start-up heat transfer enhancement in turbulent flow [15, 25]. As shown in Figs. 3 and 4, a short twisted tape can produce an increase of 30% in heat transfer at $Re = 40\,000$, although pressure drop increase is about 1.5 times the heat transfer increase. The coiled wire produces a 40% increase in heat transfer, at $Re = 45\,000$, but the pressure drop goes up by a factor of about three with respect to the smooth tube. Of the inserts tested, only the flag exhibits a larger enhancement in heat transfer than in pressure drop. The enhancement characteristics of the flag seem to be independent of flag length for flag lengths between 0.5 and 2 tube diameters.

It should be noted that, in the case of the short static (i.e. non-flapping) inserts tested, heat transfer enhancement, when expressed in dimensionless form with respect to the smooth tube heat transfer, diminishes as Reynolds number increases. However, pressure drop data reveal that friction losses will increase as Reynolds number increases. Thus, the beneficial effects of short static inserts decrease as flow rate is increased. In comparison, the flexible insert results in increased enhancement benefits (higher heat transfer enhancement, lower pressure drop enhancement) as Reynolds number is increased, becoming therefore much more attractive at higher velocities.

It might be thought that enhancement effects could arise from vibrations induced on the heat transfer surface by the flag. Direct vibration measurement seems to deny this possibility, since for vibration to be important in heat transfer enhancement in turbulent flow, the vibration Reynolds number would need to be some 10 000 times larger than it is at present [24]. The vibration characteristics of a tube with a flag-insert at work are very similar to those for empty tubes, which suggests that no vibration risk of tube mechanical failure is introduced by this enhancement device.

No further attempt has as yet been made in this work to visualize or to otherwise study the characteristics of the flow past a flag insert. It might seem that the unsteady character of the flow produces the interesting enhancement observed. The geometry of the flag appears to be better suited to induce an effective mixing motion in the core of the fluid than to generate highly dissipative 'eddies', or turbulence, and this could explain the low increase in pressure drop.

It should be noted at this point that whilst 'start-up' promoters may produce very large local increases in film coefficient, the increase, averaged over what may be a very long tube, would be clearly less impressive. Obviously, a system of multiple 'start-up' promoters can be considered, and flag-type inserts can be staggered down the flow tube in such a way as to maximize their overall effect on the tube as a whole [23]. A 'full-tube' promoter can thus be achieved, and comparisons with alternative enhancement devices can be made on a more rational basis, where clearly the flag is capable of superior performance. The choice of axial flag spacing results from an optimization exercise. The 200% typical heat transfer enhancement encountered in test station 1, Fig. 6, for a single flag, could conceivably be applied to the tube as a whole if flags were installed about 7 tube diameters apart.

CONCLUSIONS

A simple and effective tubular turbulent flow heat transfer enhancement device is obtained if a small rectangle of sheet metal is allowed to flap inside a tube in a flag-type fashion. The device produces a heat transfer enhancement that increases with fluid velocity in turbulent flow. Pressure drop increases are not as high as heat transfer increases, a feature not seen with other forms of static promoter tested.

Typically, a single flag in a tube of a length equal to 90 diameters can produce a 300% overall increase in heat transfer with minimal effect on pressure drop, when compared with other short promoters. Most of the enhancement is concentrated along a tube length of about 10 tube diameters downstream of the flag, where local coefficients are increased by about three fold. Multiple flag arrangements could produce heat transfer enhancement of up to 300% at Reynolds numbers of 20 000–120 000, if spaced axially along the tube to prevent the natural tendency of the flow to settle downstream of the flag.

Acknowledgements—Part of this work was done at the Institute of Engineering, UNAM, Mexico, and supported by a grant from Fundación Ricardo J. Zevada. Thanks are due to Norberto Chargoy for the construction of the experimental device and for gathering data.

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ACCROISSEMENT DU TRANSFERT THERMIQUE DANS DES TUBES PAR DES INSERTS DE TYPE FLOTTANT

Résumé—Un insert flexible sous la forme d'un drapeau rigide monté sur une tige diamétrale augmente le transfert thermique dans les écoulements turbulents dans un tube. L'insert est animé d'oscillations latérales et produit des performances de transfert thermiques supérieures à des rubans torsadés ou des fils spiralés. Le transfert thermique sur une longueur équivalente de tube de 5–15 diamètres en aval du drapeau est augmenté jusqu'à trois fois la valeur relative au tube entier.

ERHÖHUNG DES WÄRMEÜBERGANGS IN ROHREN DURCH VERWENDUNG EINES DREHBAR GELAGERTEN BLECHSTREIFENS

Zusammenfassung—Ein beweglich eingebauter, starrer Blechstreifen, der an einem diametral befestigten Stab drehbar aufgehängt ist, wurde zur Erhöhung des Wärmeübergangs im Einlauf bei turbulenter Rohrströmung eingesetzt. Durch die drehbare Lagerung waren Seitwärts-Oszillationen möglich. Dadurch wurde im Einlauf eine höhere Steigerung des Wärmeübergangs als durch Verwendung verdrehter Bänder oder gewundener Drähte erreicht (Länge beispielsweise 1–10 facher Rohrdurchmesser im Einlauf). Der Wärmeübergang hinter dem Blechstreifen wurde auf eine Länge des 5- bis 15-fachen Rohrdurchmessers bis zum 3-fachen Wert, bezogen auf ein Rohr ohne Einbau, verbessert.

ИНТЕНСИФИКАЦИЯ ТЕПЛООБМЕНА С ПОМОЩЬЮ ПРЯМОУГОЛЬНОЙ ВСТАВКИ В ТРУБАХ

Аннотация—Найдено, что с помощью гибкой вставки, выполненной в виде жесткого флажка, подвешанного на диаметральном стержне, можно интенсифицировать теплообмен при турбулентном течении в трубе. Такая вставка способна создавать колебания и обеспечивать более интенсивный теплообмен, чем шнек или спираль (например, трубчатые вставки длиной 1–10 калибров на входе в трубу). По сравнению с трубой без вставки получена интенсификация теплообмена в 3 раза на длине трубы, равной 5–15 калибрам от вставки вниз по потоку.