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AMPLIFICATION OF HEAT TRANSFER BY LOCAL INJECTION OF FLUID INTO A TURBULENT TUBE FLOW

W. A. TAUSCHER, E. M. SPARROW and J. R. LLOYD

Department of Mechanical Engineering, University of Minnesota, Minneapolis, Minesota, U.S.A.

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Abstract—It is demonstrated experimentally that local fluid injection into a turbulent tube flow gives rise to substantial increases in the heat transfer coefficients in the region downstream of the injection station. In the experiments, fluid was injected into a mainstream flow through a ring of discrete holes situated just upstream of an electrically heated test section. Water was the working fluid. The augmentation of the heat transfer coefficients due to injection is found to increase as the ratio of the injected flow to the test section flow increases. The extent of the augmentation is greater for low Reynolds number turbulent flows than for high Reynolds number turbulent flows.

NOMENCLATURE

c_p, specific heat at constant pressure;
d, tube diameter;
d_0, orifice bore diameter;
h, local heat transfer coefficient, $q/(T_w - T_b)$;
k, thermal conductivity;
$\dot{m}$, rate of flow in test section;
$\dot{m}_1$, rate of flow in hydrodynamic development section;
$\dot{m}_2$, rate of injected flow;
Nu, Nusselt number, hd/k;
Pr, Prandtl number, $c_p\mu/k$;
q, local heat transfer rate/area;
Re, Reynolds number, $4\dot{m}/\mu\pi d$;
T_b, fluid bulk temperature;
T_w, inside tube wall temperature;
x, axial coordinate;
μ, absolute viscosity.

Subscripts

fd, fully developed;
max, maximum Nusselt number.

INTRODUCTION

IN VARIOUS advanced technological applications, the injection of fluid through a wall into a flowing stream is a well established concept for decreasing the heat transfer between the stream and the wall. In the present research, an altogether different role of fluid injection is explored—that of *increasing* the heat transfer coefficient between a flowing stream and a surface. Experiments are performed wherein fluid is locally injected from the wall of a tube into a turbulent flow passing through the bore. Measurements in the region downstream of the injection station reveal large augmentations of the local heat transfer coefficient relative to that for a tube flow without injection. The experiments were carried out with water as the working fluid for both the mainstream and the injected flows. The injection station was situated just upstream of an electrically heated test section, the injection being accomplished by means of holes uniformly arranged in a ring around the circumference of the tube.

To the best knowledge of the present authors, the technique of injecting the same fluid to augment heat transfer coefficients has not heretofore been described in the heat transfer literature. Therefore, the range of potential applications remains unexplored. The fact that the fluid injection technique can provide a range of heat transfer coefficients for a corresponding range of injection rates is suggestive of its application to situations involving time-dependent heat loads and/or mainstream flows; for example, in conjunction with a control system actuated by the measurement of the tube wall temperature.

EXPERIMENTAL APPARATUS

The experiments were carried out in a closed loop fluid flow facility which is pictured schematically in Fig. 1. As shown therein, the system contains a primary branch whose main components are a pump, an adiabatic hydrodynamic development length, the electrically heated test section, and a heat exchanger. There is, in addition, a secondary branch which facilitates the injection of fluid into the primary branch at a point just upstream of the test section. The injection arrangement is indicated in the inset at the upper left of the figure.

The unheated starting section and the test section were both fabricated from type 304 stainless steel tubing having an i.d. of 0·752 in. and a wall thickness of $\frac{1}{16}$ in. The respective length to diameter ratios of these sections were 96 and 30. Heating of the test section was accomplished by passing an a.c. electric current longitudinally through the tube wall.

The injection unit was designed to provide radial inflow into the mainstream from a ring of discrete holes uniformly spaced around the circumference of the bore, the basic objective being to achieve the same flow through all holes. In overall configuration, the injection unit is an annular disk. Fluid was introduced into the unit

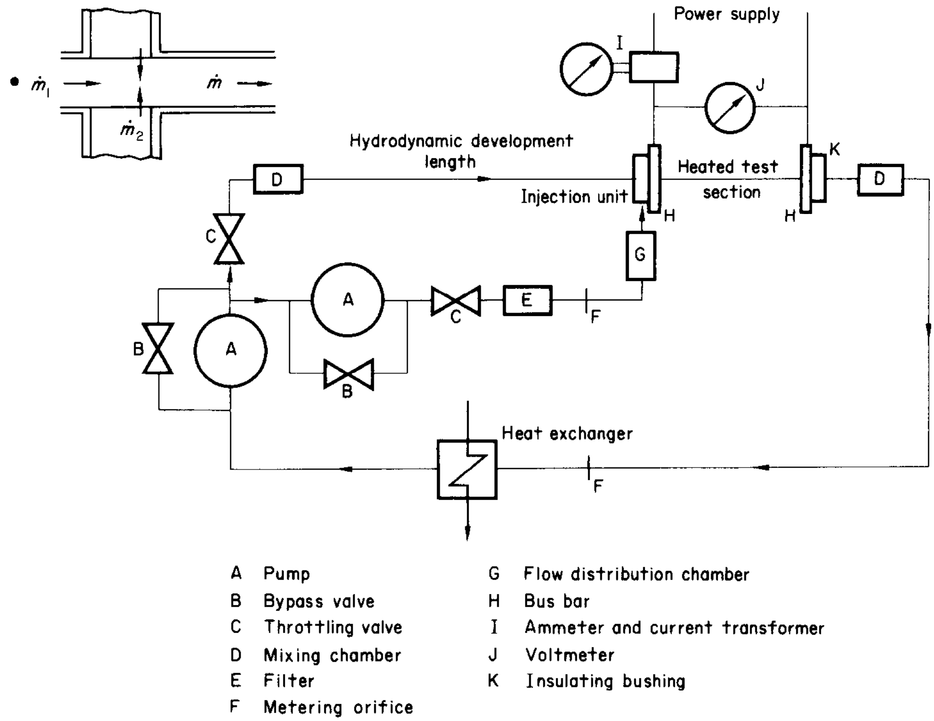

FIG. 1. Schematic diagram of the experimental apparatus.

at four positions spaced 90° apart on the outside surface. The fluid then passed into an annular chamber within the injection unit, from which it flowed radially inward into a second annular chamber, the two chambers being connected by eight uniformly spaced circular passages. This second chamber was, in turn, connected to a third chamber by a continuous slot. The third chamber served as the plenum from which the flow passed, via radial holes, into the bore. In the present experiments, there was a ring of thirty-two injection holes, each hole being 0·025 in. dia. The uniformity of the injected flow was verified by visual observations made with the injection unit removed from the test loop and further confirmed by the absence of marked circumferential variations in the measured wall temperatures of the test section.

The injection unit was fabricated, in the main, from a free machining plastic called delrin. Even with its several internal chambers, it was quite compact, being 3 in. o.d. and 1 in. thickness.

Temperature measurements were made with individually calibrated copper–constantan and iron–constantan thermocouples (30 gage). Twenty-seven such thermocouples were employed to measure temperatures on the outside surface of the test section tube, with some staggering in angular position and use of multiple thermocouples at selected axial stations to check on possible circumferential asymmetries. The thermocouples were affixed to the tube wall with copper oxide cement,* the lead wires being looped circumferentially around the tube to minimize conduction errors. Fluid bulk temperatures were measured by thermocouples installed in well-insulated mixing chambers situated at the inlet of the starting section and at the exit of the test section, and in the flow distribution chamber which supplies the injection unit.

The entire length of pipeline extending from the upstream mixing chamber to the downstream mixing chamber was submerged in silica aerogel insulation ($k = 0{\cdot}014$ Btu/hft°F). To further insure against extraneous heat losses, the experiments were performed such that the bulk temperature of the working fluid (approximately 80°F) was only slightly different from the temperature of the laboratory room. Water was the working fluid for all of the tests.

* It has been verified [1, 2] that this technique does not introduce detectable errors.

EXPERIMENTAL PROCEDURE AND DATA ANALYSIS

Preliminary experiments were performed to determine the maximum injection rate which the system was able to provide at the preselected operating temperature ($\sim$ 80°F). To find the maximum, the hydrodynamic entrance section was closed off, so that the test section was fed solely through the injection unit. For this mode of operation, the maximum test section Reynolds number was approximately 10 000.

With this information at hand, the following program of test runs was formulated and carried out. For a fixed test section Reynolds number of 10 000, runs were made for injection ratios $\dot{m}_2/\dot{m}$ of 0, $\frac{1}{8}$, $\frac{1}{4}$, $\frac{1}{2}$ and 1, where the quantities $\dot{m}_2$ and $\dot{m}$ (illustrated at the upper left of Fig. 1) respectively denote the rate of fluid injection and the flow rate in the test section. Next, with the test section Reynolds number fixed at 20 000, experiments were performed for $\dot{m}_2/\dot{m} = 0$, $\frac{1}{8}$, $\frac{1}{4}$ and $\frac{1}{2}$, the upper limit on the injection ratio being imposed by the pump capacity. Subsequent tests were made at system Reynolds numbers of 40 000 and 80 000, with corresponding maximum values of $\dot{m}_2/\dot{m}$ of $\frac{1}{4}$ and $\frac{1}{8}$.

All tests were performed at a bulk Prandtl number of approximately 6. Fully developed wall-to-bulk temperature differences ranged from 15 to 18°F, depending on the run. These temperature differences were sufficiently large to permit accurate measurements, but small enough to preclude significant variable property effects [1, 2]. Bulk temperature rises from the inlet to the exit of the test section were generally between 2–3°F, with correspondingly small deviations of the bulk fluid properties from those evaluated at the mean bulk temperature.

Local heat transfer coefficients were computed in accordance with the defining equation $h = q/(T_w - T_b)$. The heat flux q was determined directly from the ohmic heating in the tube wall (axial conduction and losses through the insulation were examined and found to be negligible). The local bulk temperature at any axial station was evaluated from a straight line passed through the calculated inlet and measured outlet bulk temperatures. The inside tube wall temperatures were calculated from the measured outside wall temperatures by applying steady-state heat conduction theory.

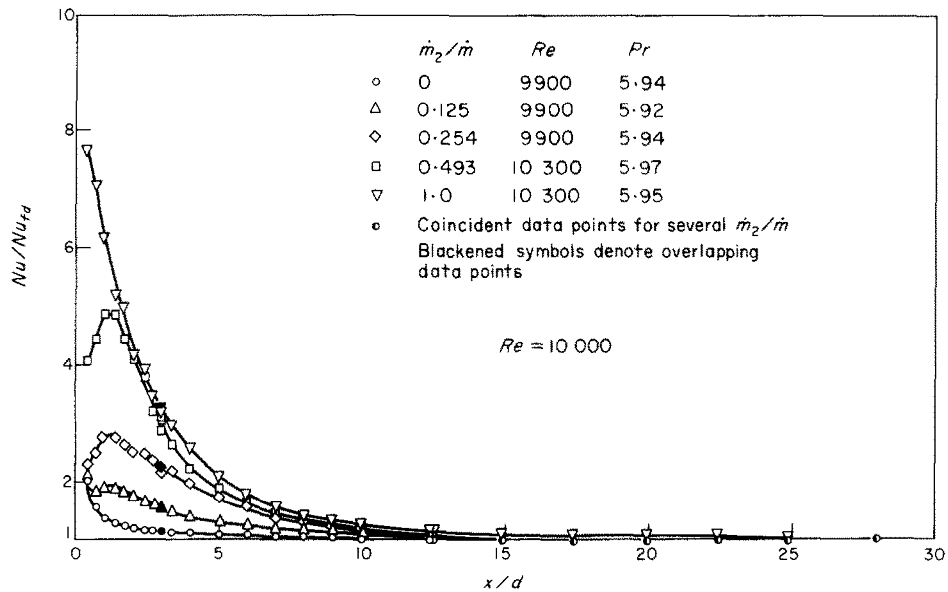

FIG. 2. Nusselt number distributions, $Re = 10\,000$.

The local heat transfer coefficents were employed to evaluate local Nusselt numbers. The fluid properties appearing in Nu, Re and Pr were introduced at the mean bulk temperature.

RESULTS AND DISCUSSION

Experimentally determined Nusselt number distributions are presented in Figs. 2–5, the successive figures corresponding to fixed Reynolds numbers of 10 000, 20 000, 40 000 and 80 000. On the ordinate, Nu is the local Nusselt number, while Nu_{fd} is the corresponding fully developed Nusselt number. The coordinate x measures distances along the length of the tube from the upstream extremity of the heated test section.* The data are parameterized by the ratio of the injection flow $\dot{m}_2$ to the total test section flow $\dot{m}$. A common ordinate scale is used for all figures to facilitate comparisons of results for different Reynolds numbers.

Attention may first be turned to Fig. 2, which contains results for the entire range of injection ratios $\dot{m}_2/\dot{m}$ between 0 and 1. Inspection of the figure reveals that fluid injection plays a decisive role in increasing the Nusselt number. The greatest augmentation occurs in the length of tube between 0 and 5 diameters of the point of injection, with significant effects also in evidence in the range from $5 \leqslant x/d \leqslant 10$. Furthermore, the greater the value of the injection ratio, the

* The injection holes were situated at $x/d = -1/3$.

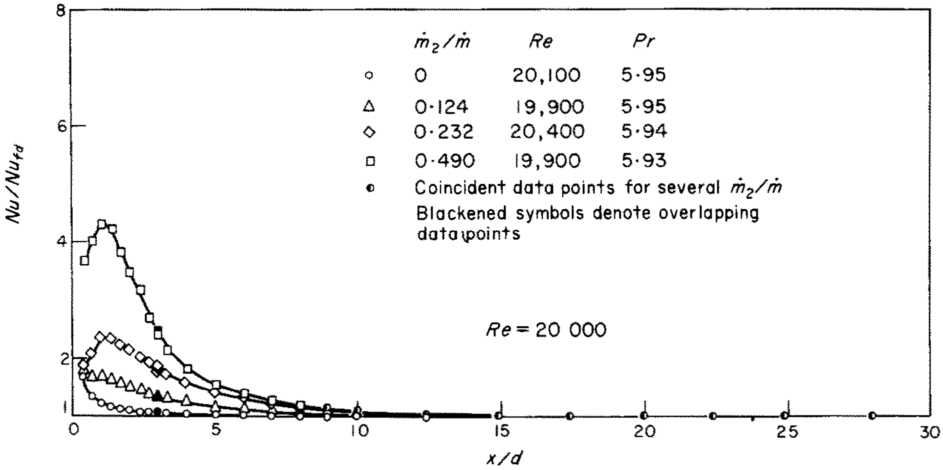

FIG. 3. Nusselt number distributions, $Re = 20\,000$.

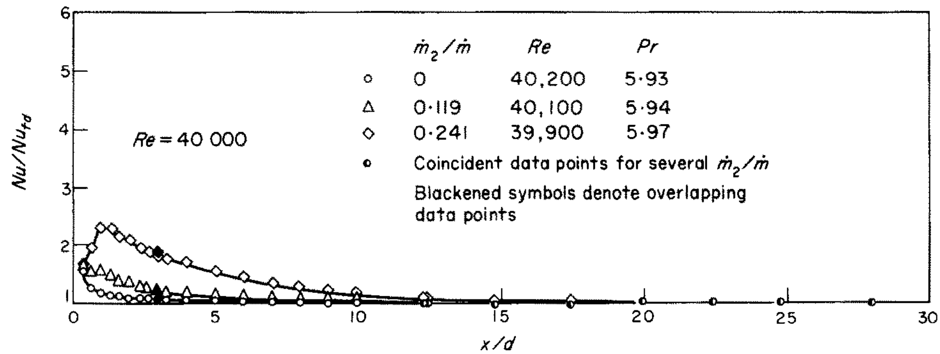

FIG. 4. Nusselt number distributions, $Re = 40\,000$.

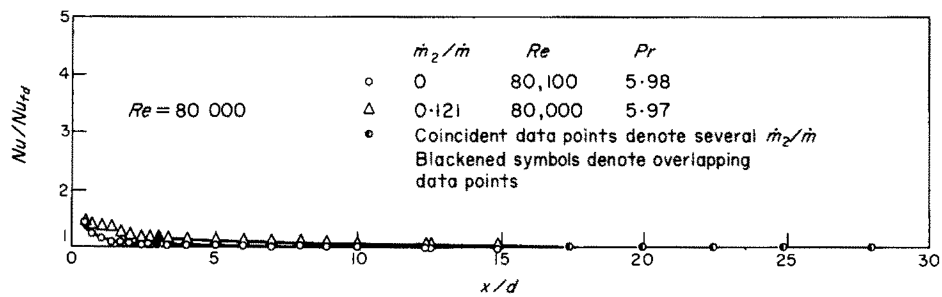

FIG. 5. Nusselt number distributions, $Re = 80\,000$.

more pronounced is the augmentation of the Nusselt number.

For injection ratios other than 0 and 1, the Nusselt number distributions display a maximum value at $x/d \sim 1$. In the absence of injection, that is, $\dot{m}_2 = 0$, the distribution curve is characterized by the usual monotonic decrease with increasing downstream distance. A similar behavior (but with dramatically higher Nu/Nu_{fd}) is in evidence for the case in which the test section flow is due only to injection, that is, when $\dot{m}_2 = \dot{m}$.

Another effect of fluid injection, in addition to augmenting the magnitude of the Nusselt number, is to increase the length of the thermal development region. The lengthening of the development region is accentuated with increasing values of the injection ratio.

Figures 3–5, each corresponding to a successively higher Reynolds number, show trends that are similar to those already discussed for Fig. 2. However, for any fixed injection ratio $\dot{m}_2/\dot{m}$, the augmentation of the Nusselt number diminishes with increasing Reynolds number. For instance, for $\dot{m}_2/\dot{m} = \frac{1}{4}$, the maximum values of Nu/Nu_{fd} for $Re = 10\,000$ and $Re = 40\,000$ are 2·8 and 2·3 respectively. Thus, fluid injection is more effective in low Reynolds number turbulent flows than in high Reynolds number turbulent flows. This finding is plausible inasmuch as the intrinsic turbulence levels of the latter flows are higher than those of the former flows.

The maximum Nusselt numbers that are in evidence in the Nu/Nu_{fd} distributions of Figs. 2–5 have been correlated by the relation $Nu_{max} = 1{\cdot}08\ [(\dot{m}_2/\dot{m})Re]^{0{\cdot}7}$, which can be applied for estimating the Nu_{max}/Nu_{fd} ratio for operating conditions other than those investigated herein.

It is of interest to compare the heat transfer characteristics associated with fluid injection with those associated with flow separation induced by the presence of an obstacle. In [3], experiments were performed in which an orifice was situated at the inlet cross section of an electrically heated tube. Representative Nusselt number results from [3] and from the present investigation have been brought together in Fig. 6, where d_0/d denotes the ratio of the orifice bore diameter to the tube diameter. On the basis of evidence shown in Fig. 6 of the reference, the difference in the Prandtl numbers of the two sets of results should have no effect on the comparison of results.

Inasmuch as the numerical values of $\dot{m}_2/\dot{m}$ have no clear relationship to the numerical values of d_0/d, it is not relevant to compare any specific dashed curve with any specific solid curve. Rather, the information which can be drawn from Fig. 6 concerns the shapes of the

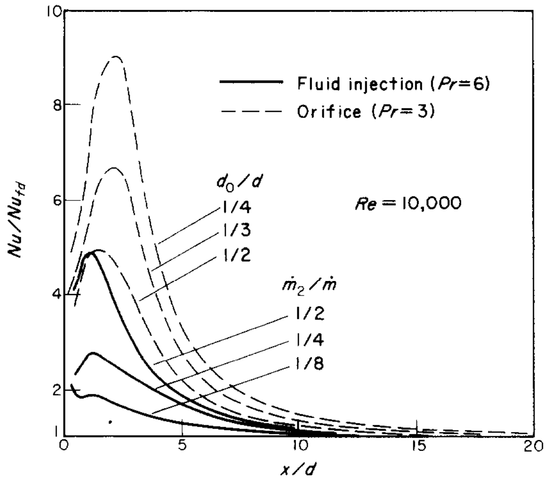

FIG. 6. Comparison of results for fluid injection and orifice-induced separation.

distribution curves. The fact that both sets of distribution curves have qualitatively similar shapes may, at first thought, suggest that similar phenomena are controlling the heat transfer processes in the two types of flows being compared. Although there may well be certain general phenomena that are common to both flows (for instance, separation), there are, undoubtedly, great differences in the flow patterns. Various aspects of the flow field will now be discussed.

Consider first the flow field in the presence of fluid injection. The unidirectional flow delivered by the hydrodynamic development section encounters a fluid screen, consisting of radial jets, when it reaches the injection unit. There are spaces between the jets* through which the mainstream flow can pass. The flow field just downstream of the injection station is highly complex. Experiments on a *single* jet injected normal to the wall of a wind tunnel and transverse to the main stream have shown that the flow downstream of the jet contains regions of separation and backflow, as well as a velocity component normal to and away from the wall [4]. Evidently, the downstream region is one characterized by strong mixing, even more so in the presence of a large number of adjacent, interacting jets.

Another factor which acts to augment the turbulence level is the confrontation and collision of the individual jet streams in the central region of the tube.

The main stream bends the jets and arches the jet columns in the streamwise direction. It is this downstream displacement of the jets that may be responsible, at least in part, for the maxima that are in evidence in Figs. 2–5. In addition, the interaction between the arched jets and the main stream sets up a secondary flow within the jets themselves [4].

In light of the foregoing discussion, the augmentation of the heat transfer due to fluid injection is entirely plausible.

The flow field downstream of an orifice is less complex than that accompanying fluid injection. A closed separated region is set up downstream of the orifice. Within the separated region, there is substantial eddying and mixing which augments the heat transfer. Viscous forces induced by the no-slip boundary condition at the downstream face of the orifice tend to inhibit mixing in the adjacent fluid, creating conditions leading to the maxima shown in Fig. 6 (dashed lines).

* For instance, at the tube wall, the jets occupy ($32 \times 0{\cdot}025$ in.) = 0·8 in. of the 2·36 in. circumference.

CONCLUDING REMARKS

It has been demonstrated that fluid injection into a turbulent tube flow can provide substantial augmentation of the heat transfer coefficients in the region downstream of the injection station. The findings of this investigation bring into focus several areas for fruitful additional study. From the standpoint of basic research, there is the exploration of the flow field induced by fluid injection. On the developmental side, the influence of the diameter and spacing of the injection holes remains to be explored.

Fluid injection may be regarded as an *active* technique for augmenting turbulent heat transfer coefficients inasmuch as the extent of the augmentation can be varied by varying the injection flow rate. This characteristic may have application in automatic control systems. In contrast, a fixed separation-inducing obstacle (for example, an orifice, ribs, twisted tape) provides a heat transfer augmentation which is more or less invariable.

ACKNOWLEDGEMENT

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AMPLIFICATION DU TRANSPORT DE CHALEUR PAR INJECTION LOCALE DE FLUIDE DANS UN ÉCOULEMENT TURBULENT DANS UN TUBE

Résumé—On démontre expérimentalement que l'injection locale de fluide dans l'écoulement turbulent dans un tube donne naissance à des augmentations substantielles des coefficients de transport de chaleur dans la région en aval du lieu de l'injection. Dans les expériences, le fluide était injecté dans un écoulement principal à travers un anneau de trous séparés placé juste en amont d'une section d'essai chauffée électriquement. Le fluide de travail était de l'eau. L'augmentation des coefficients de transport de chaleur due à l'injection augmentait avec le rapport de l'écoulement injecté à l'écoulement dans la section d'essai. La valeur de l'augmentation est plus grande pour les écoulements turbulents à faibles nombres de Reynolds que pour ceux à nombres de Reynolds élevés.

VERGRÖSSERUNG DES WÄRMEÜBERGANGS DURCH ÖRTLICHE FLÜSSIGKEITS EINSPRITZUNG IN EINE TURBULENTE ROHRSTRÖMUNG

Zusammenfassung— Es wird experimentell gezeigt, dass örtliche Flüssigkeitseinspritzung in eine turbulente Rohrströmung eine wesentliche Erhöhung des Wärmeübergangskoeffizienten in der Zone stromabwärts von der Einspritzstell ergibt. In den Versuchen wurde die Flüssigkeit in die Hauptströmung eingespritzt durch einem Ring einzelner Bohrungen, die direkt vor einer elektrisch beheizten Messstrecke angebracht waren.

Hierbei wurde mit Wasser als Flüssigkeit gearbeitet. Der Anstieg des Wärmeübergangskoeffizienten infolge der Einspritzung wächst mit dem Verhältnis der eingespritzten Flüssigkeitsmenge zum Gesamtdurchsatz. Der Betrag der Vergrösserung ist für die betrachtete turbulente Strömung bei niedrigen Reynoldszahlen grösser als bei hohen Reynoldszahlen.

ИНТЕНСИФИКАЦИЯ ТЕПЛООБМЕНА ПУТЕМ ЛОКАЛЬНОГО ВДУВА ЖИДКОСТИ В ТУРБУЛЕНТНЫЙ ПОТОК

Аннотация—Экспериментально показано, что локальный вдув жидкости в турбулентный поток приводит к увеличению коэффициентов теплообмена в области вниз по потоку от места вдува. В экспериментах жидкость вдувалась в основной поток через кольцо дискретных отверстий, расположенных вверх по потоку от электрически нагреваемого экспериментального участка. Вода служила рабочей жидкостью. Найдено, что увеличение коэффициентов теплообмена за счет вдува возрастает с увеличением отношения вдуваемого потока к потоку на экспериментальном участке; это увеличение гораздо больше при низких числах Рейнольдса для турбулентных потоков, чем при больших числах Рейнольдса в этом же случае.