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Development of turbulent heat transfer over the length of vertical tubes in the presence of mixed air convection

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Abstract—Experimental results are presented for heat transfer and turbulent transport in heated vertical circular tubes in the presence of mixed convection at $Re = 5100$ and 9000. For the first time the measurements of turbulent transports ($\overline{u'v'}$ and $\overline{u_i'T'}$) were made. It is shown that turbulent heat transport near the wall is suppressed to a greater degree than momentum transport, being thus responsible for a sharp reduction in heat transfer with a slight change in the friction coefficient. Buoyancy forces influence turbulence differently over the cross-section—the suppression of turbulence near the wall and its generation in the flow core. The anisotropy of longitudinal and radial turbulent heat transport is shown to undergo the reversal of sign in the flow core.

1. INTRODUCTION

IT WAS found experimentally [1] that there was a decrease in heat transfer during the upward flow in heated vertical channels. The heat transfer coefficient was in this case by a factor of 2–3 smaller than in forced convection, thus leading to the overheating of the channel walls at a fixed heat flux on them.

Recent years have seen the appearance of a number of experimental [2–4] and computational–theoretical [5–9] works dealing with the study of these flows, nevertheless the specific features of turbulent heat and momentum transport, which are responsible for the non-monotonous behaviour of the heat transfer coefficient, remain to be learned. This uncertainty is reflected in the failure of different computational models, which, while giving almost the same results on heat transfer, may lead not only to quantitative, but also to qualitative discrepancies in the results for other characteristics and even for the friction coefficient.

The acquisition of experimental data on turbulent transport was limited because the well-known hot-wire measurement techniques were intended for large measuring volumes and this precluded the possibility for the measurement of its parameters in the wall region where the flow velocity and temperature gradients were high. This gap in the experimental evidence has been filled by the application of a combination of the two-colour laser Doppler anemometer with a miniature resistance thermometer [10].

2. EXPERIMENT

Measurements were made on two experimental set-ups composed of stainless steel tubes with inner diameters (d) of 46 and 78.6 mm. The lengths of the tubes were $l_1' = 110d$ and $l_2' = 85.5d$; the lengths of the heated stretches were $l_1 = 87d$ and $l_2 = 68.3d$, respectively. Some measurements were also made with shorter heated lengths by shifting the lower current lead. The regimes investigated had Reynolds numbers $Re = u_b d/\nu = 5100$ and 9000 and Grashof numbers $Gr = g\beta q_w d^4/\lambda\nu^2 \leqslant 3\times10^7$. This allowed the examination of the influence of buoyancy forces on the turbulent transfer characteristics during the deterioration of heat transfer under air upflow through vertical heated pipes.

In experiments, use was made of the 55X and 56C/N DANTEC anemometers incorporated into an automatic measuring complex with a HP 9830 computer. To measure the averaged velocity $u(y)$, the intensities of fluctuations of the longitudinal and normal velocities ($\sigma_u = \sqrt{(\overline{u'^2})}$, $\sigma_v = \sqrt{(\overline{v'^2})}$) and the Reynolds stresses $\overline{u'v'}$, use was made of a two-colour LDA system involving counting-type processors.

The averaged temperature $T(y)$ and the intensity of temperature fluctuations $\sigma_T = \sqrt{(\overline{T'^2})}$ were measured by a thermoanemometer operating in the resistance thermometer mode. The temperature was measured by probes with wire diameters of 1 and 2.5 μm. The velocity–temperature correlations $\overline{u'T'}$ and $\overline{v'T'}$, that characterize the turbulent heat transport, were measured by an LDA system combined with a resistance thermometer. The shear stress τ_w and the wall heat flux density q_w were determined from the measured gradients of averaged velocity and temperature profiles near the wall. These values were used for finding the dynamic velocity $u_\tau = \sqrt{(\tau_w/\rho)}$ and temperature scale $T_* = q_w/\rho c_p u_\tau$, and also for determining the friction coefficient ξ and heat transfer.

3. RESULTS

Figure 1 presents the results of measurements of the dimensionless heat transfer coefficient, i.e. of the

NOMENCLATURE

c_p	specific heat at constant pressure [kJ kg^{-1} K^{-1}]
d	inner tube diameter, $2R$ [m]
E	buoyancy parameter, $Gr/Pr\,Re^4$
g	acceleration of gravity [m s^{-2}]
Gr	Grashof number, $g\beta d^4 q_w/\nu^2\lambda$
l	heated length [m]
l'	entire length of pipe [m]
Nu	Nusselt number, $q_w d/(T_w - T_b)\lambda$
p	pressure [N m^{-2}]
Pe	Peclet number, $Pr\,Re$
Pr	Prandtl number, $\mu c_p/\lambda$
q	heat flux [W m^{-2}]
R	radius [m]
Re	Reynolds number, du_b/ν
St	Stanton number, Nu/Pe
t	time [s]
T	temperature [K]
T_*	temperature scale, $q_w/\rho c_p u_\tau$ [K]
u	axial velocity [m s^{-1}]
u_τ	dynamic velocity, $\sqrt{(\tau_w/\rho)}$ [m s^{-1}]
v	radial velocity [m s^{-1}]
x	axial coordinate [m]
y	distance along the normal from the wall [m]
y^+	dimensionless coordinate, $u_\tau y/\nu$.

Greek symbols

β	coefficient of volumetric liquid expansion, $-(1/\rho)(\partial\rho/\partial T)_p$ [K^{-1}]
ε_τ	eddy momentum diffusivity [m^2 s^{-1}]
ε_q	eddy heat diffusivity [m^2 s^{-1}]
ε_t	turbulent energy dissipation
λ	thermal conductivity [W m^{-1} K^{-1}]
μ	dynamic viscosity [kg m^{-1} s^{-1}]
ν	kinematic viscosity, μ/ρ [m^2 s^{-1}]
ξ	friction coefficient
ρ	density [kg m^{-3}]
σ_T	intensity of temperature fluctuations, $\sqrt{(\overline{T'^2})}$ [K]
σ_u	intensity of velocity fluctuations, $\sqrt{(\overline{u'^2})}$ [m s^{-1}]
τ	shear stress [N m^{-2}].

Subscripts

b	bulk
M	coordinate of velocity maximum
w	wall
0	axis of pipe.

Superscripts

$\bar{}$	mean value
$'$	fluctuation.

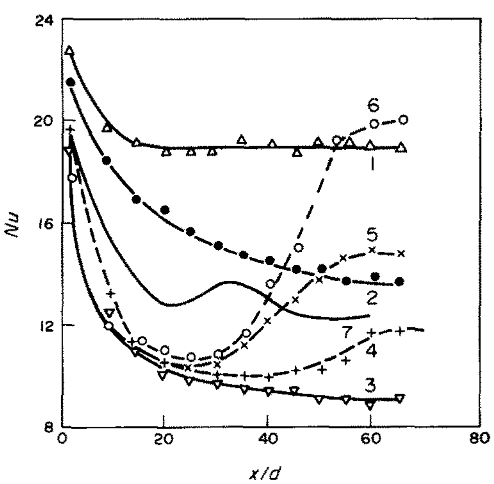

FIG. 1. Variation of heat transfer along the pipe at different $E = Gr/Pr\,Re^4$ ($Re = 5100$): 1, $\sqrt{E} \to 0$; 2, $\sqrt{E} = 10^{-4}$; 3, 1.8×10^{-4}; 4, 2.5×10^{-4}; 5, 4×10^{-4}; 6, 7×10^{-4}; 7, 4.8×10^{-4}, $Re = 4755$ [1].

Nusselt number $Nu = q_w d/\lambda(T_w - T_b)$, along the tube at different Gr. As the effect of buoyancy forces increases (increase of Gr at Re = const.), the behaviour of heat transfer along the tube is non-monotonous. First, the heat transfer decreases over the entire length (curves 2 and 3) as compared with heat transfer in the absence of the effect of body forces on the flow (curve 1). When $Gr = 1.5 \times 10^7$, a minimum heat transfer distribution sets in over the entire length. A further increase in the heat flux leads to an increase of Nu from a certain cross-section (curves 4–6); the heat transfer for high x/d ratios may exceed the forced convection heat transfer. The higher heat transfer is attributed to the development of the turbulent free convection regime.

Figure 2 shows the behaviour of ξ and $St = Nu/Pr\,Re$ far from the start of heating ($x/d > 50$), under the conditions close to heat exchange stabilization, as functions of $\sqrt{E} = \sqrt{(Gr/Pr\,Re^4)}$ [6], which characterizes the effect of buoyancy forces in mixed convection.

The results of calculations by the interpolation formula, which is suggested in ref. [6] and which describes the available experimental data for large x/d ratios, are also presented in this figure. It can be seen that the heat transfer results for large x/d are satisfactorily described by this formula. The behaviour of ξ coincides with experimental [3] and predicted [6, 7] results: ξ decreases at the initial stage of influence of buoyancy forces (but only by 15–20%), and then increases. This behaviour of ξ corresponds to that of St. The results presented in refs. [4, 8] do not agree with the data obtained in the present work for ξ—in contrast to St, ξ increases monotonously with Gr.

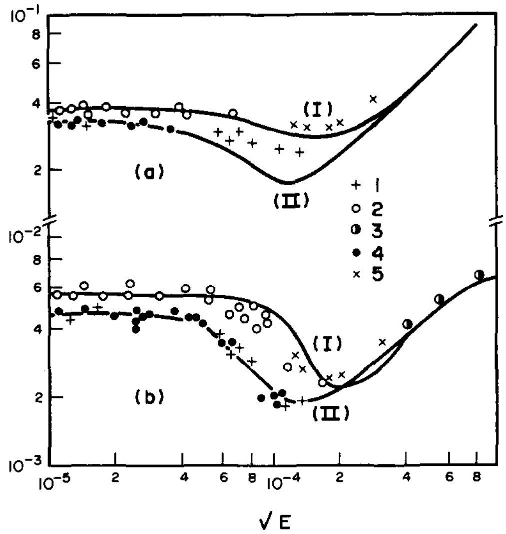

FIG. 2. Friction coefficient ξ and heat transfer St vs E: 1, $Re = (8\text{–}12) \times 10^3$; 2, $(4\text{–}6) \times 10^3$; 3, $(2\text{–}3) \times 10^3$; 4, $Re = 10^4$ [1, 2]; 5, $Re = 5100$; I, calculation [6], $Re = 5 \times 10^3$; II, calculation [6], $Re = 10^4$.

Velocity profiles also vary depending on the length of the heated section (Fig. 3). They take the M-like form. Moreover, the velocity maximum first increases, but then decreases and shifts to the flow axis. The same variation of the velocity profile with increasing Gr (Fig. 4) is also observed for a fixed $x/d = 53$. The

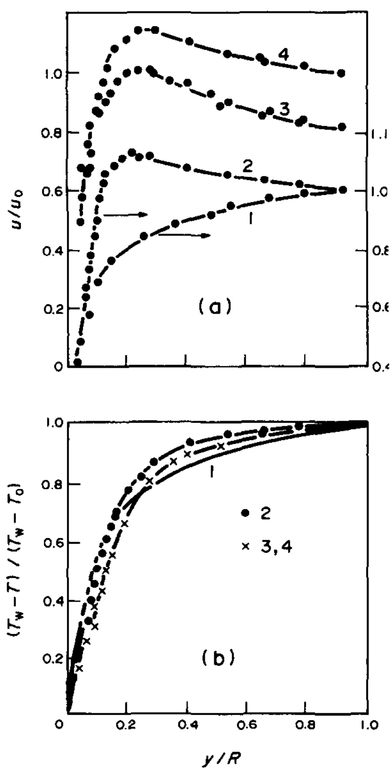

FIG. 3. Variation of velocity (a) and temperature (b) profiles at different distances from the start of heating at $Re = 5500$, $Gr = 3 \times 10^7$: 1, $x/d = 0$; 2, 29; 3, 42; 4, 53.

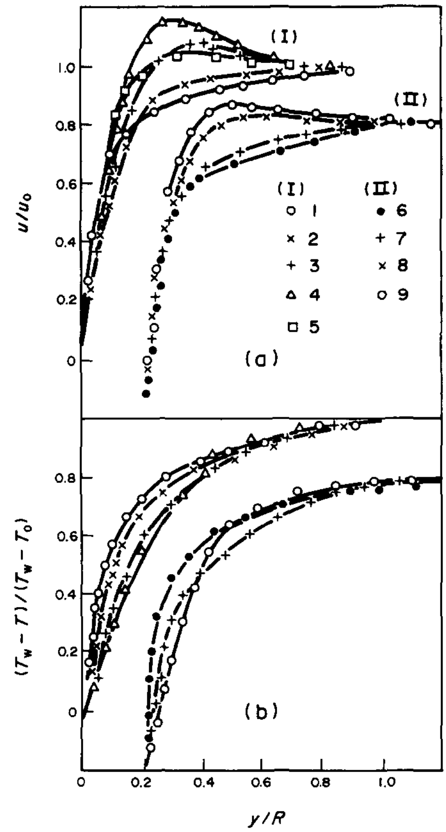

FIG. 4. Velocity (a) and temperature (b) profiles for different Gr at $x = 53d$, $Re = 5100$ (1–5) and 9000 (6–9): 1, $Gr \to 0$; 2, 9×10^6; 3, 1.3×10^7; 4, 1.5×10^7; 5, 1.8×10^7; 6, $Gr \to 0$; 7, 2.3×10^7; 8, 2.7×10^7; 9, 3.4×10^7.

profile with the most pronounced local velocity maximum corresponds to the minimum of the local Nu.

The averaged temperature profiles do not take the form of M (Figs. 3(b) and 4(b)), but their variation with x/d and Gr, just as the variation of the velocity profile, is non-monotonous.

The results of measurements of the second-order single-point fluctuation moments, which characterize the turbulent energy and the transport properties of a flow at $Re = 5100$ and 9000, are presented in Figs. 5–9. The intensities of the longitudinal and radial velocity fluctuations, just as the fluctuations of temperature, decrease (most markedly in the wall region) under the influence of buoyancy forces, thus leading to partial flow laminarization. The fragments of temperature fluctuation oscillographic traces reproduced in Fig. 10 display the intermittent character of the flow. The anisotropy of velocity fluctuations near the wall also decreases (Fig. 6) as the turbulence becomes more homogeneous over the flow cross-section. A transition to the turbulent free convection regime is detected after the deterioration of heat transfer with a further increase of Gr at Re = const.; the heat transfer coefficient will increase.

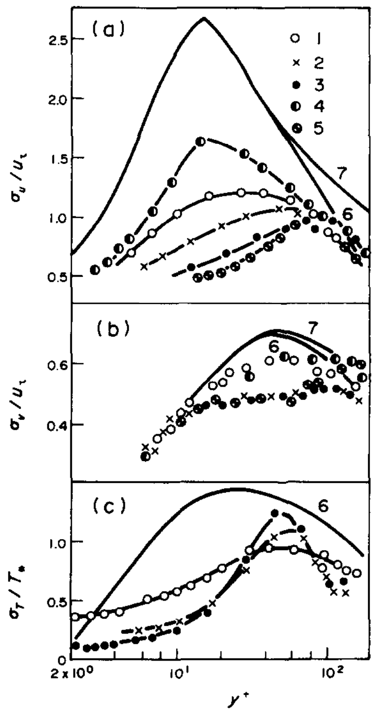

FIG. 5. Effect of buoyancy forces on the intensity of velocity (a and b) and temperature (c) fluctuations at $Re = 5100$ (1–3) and $Re = 9000$ (4, 5): 1, $Gr = 9 \times 10^6$; 2, 1.1×10^7; 3, 1.5×10^7; 4, 2×10^7; 5, 3.4×10^7; 6, $Gr \to 0$, $Re = 5540$; 7, $Gr \to 0$, $Re = 10\,500$.

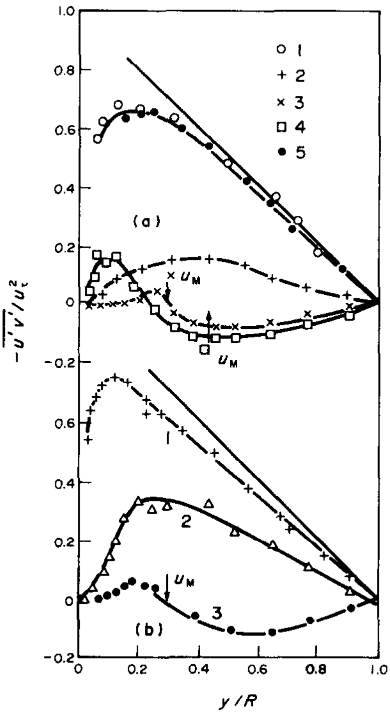

FIG. 7. Variation of Reynolds stresses under the influence of buoyancy forces. (a) $Re = 5100$: 1, $Gr = 0$; 2, 9×10^6; 3, 1.5×10^3; 4, 2.1×10^7; 5, $Gr = 0$ [11]. (b) $Re = 9000$: 1, $Gr = 0$; 2, 2×10^7; 3, 3.4×10^7.

The correlations $\overline{u'v'}$, $\overline{u'T'}$ and $\overline{v'T'}$, presented in Figs. 7 and 8, determine the turbulent transport of momentum and heat. They shape the averaged velocity and temperature fields. With flow laminarization the turbulent transports decrease over the entire cross-section. Thereafter, $(-\overline{u'v'})$ and $(-\overline{u'T'})$ intersect the abscissa axis in the region $y/R \approx 0.2$–0.3 and take on negative values in the flow core. These regimes are characterized by the appearance of local maxima in the averaged velocity profile. The position of the velocity maximum corresponds in this case to the position of the zero value of the correlation $\overline{u'v'}$, but it does not coincide with the zero value of $\overline{u'T'}$.

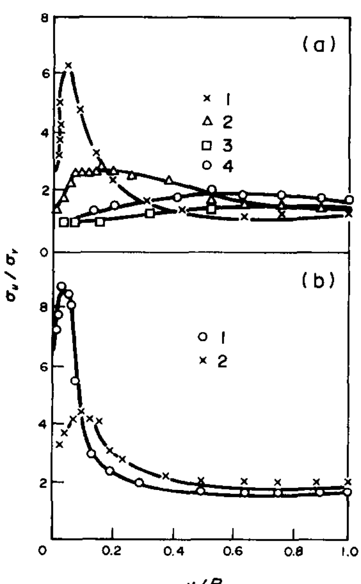

FIG. 6. Effect of buoyancy forces on the anisotropy of velocity fluctuations. (a) $Re = 5100$: 1, $Gr = 0$; 2, 9×10^6; 3, 1.5×10^7; 4, 1.8×10^7. (b) $Re = 9000$: 1, $Gr = 0$; 2, 3.4×10^7.

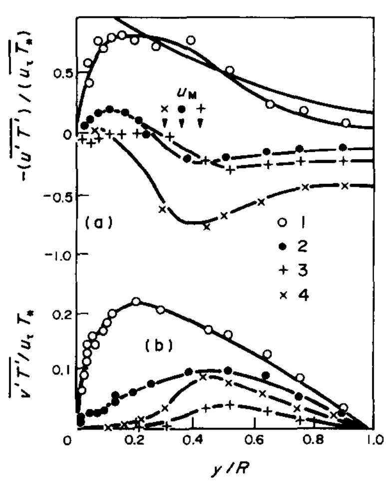

FIG. 8. Longitudinal (a) and radial (b) turbulent heat transport at $Re = 5100$: 1, $Gr \to 0$; 2, $Gr = 9 \times 10^6$; 3, 1.1×10^7; 4, 1.2×10^7; 5, 1.5×10^7.

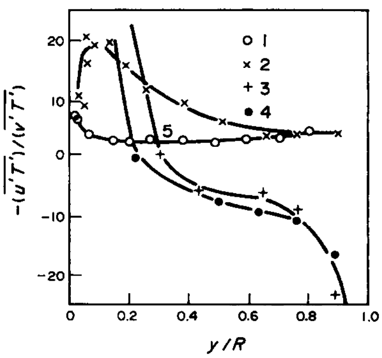

FIG. 9. Anisotropy of turbulent heat transport at $Re = 5100$: 1, $Gr \to 0$; 2, 9×10^6; 3, 1.2×10^7; 4, 1.5×10^7; 5, $Gr \to 0$, $Re = 4.2\times10^3$ [12].

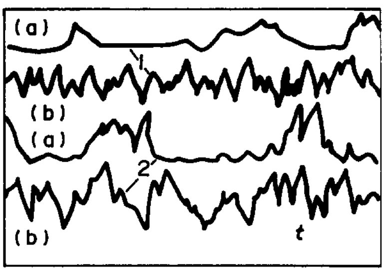

FIG. 10. Oscillographic traces of temperature fluctuations at $Re = 5600$: 1, $y/R = 0.01$; 2, 0.25; (a) with and (b) without the effect of buoyancy.

Measurements made in these regimes show that turbulent energy generation by the averaged flow $(-\overline{u'v'}\,\mathrm{d}u/\mathrm{d}y)$ and the turbulent viscosity coefficient $\varepsilon_\tau = -\overline{u'v'}/(\mathrm{d}u/\mathrm{d}y)$ remain positive throughout the entire cross-section, whereas the buoyancy-induced turbulence energy generation $(-g\beta\overline{u'T'})$ changes sign. The radial turbulent heat transport $\overline{v'T'}$ is always positive (Fig. 8(b)), corresponding to its gradient representation $\overline{v'T'} = \varepsilon_q \mathrm{d}T/\mathrm{d}y$.

It follows from the equation for the kinetic energy of turbulence 'b' that

$$\frac{\partial b}{\partial t} + \frac{\partial}{\partial y}(\overline{\tfrac{1}{2}u'_\alpha u'_\alpha + p'/\rho)v'} = -\overline{u'v'}\frac{\partial u}{\partial y} + g\beta\overline{u'T'} - \varepsilon_t. \quad (1)$$

The change of sign of $\overline{u'T'}$ shows that the buoyancy forces decrease the turbulent kinetic energy near the wall and increase it in the far region.

The anisotropy of the turbulent transport of heat changes sign in the flow core (Fig. 9).

A further increase in the effect of buoyancy (transition to a free convective regime) results in a situation when the position of the zero Reynolds stress does not coincide with that of the velocity maximum (Fig. 7(a)). The eddy viscosity is then negative in the region between the two, i.e. the energy of fluctuations is transmitted to the averaged flow.

These flow regimes are not amenable to the description by gradient models. The detected aspects of the turbulence structure determine the character of turbulent exchange in the gravity field and should be incorporated into the models.

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DEVELOPPEMENT DU TRANSFERT TURBULENT DE CHALEUR LE LONG DE TUBES VERTICAUX EN PRESENCE DE CONVECTION MIXTE D'AIR

Résumé—On présente des résultats expérimentaux de transfert turbulent thermique dans des tubes verticaux circulaires en présence de convection mixte, à des nombres de Reynolds $Re = 5100$ et 9000. Des mesures de corrélations ($\overline{u'v'}$ et $\overline{u'_t T'}$) sont faites pour la première fois. On montre que le transport turbulent de chaleur près de la paroi est supprimée à un plus grand degré que celui de quantité de mouvement, ce qui est responsable d'une forte réduction du transfert de chaleur avec un faible changement du coefficient de frottement. Les forces d'Archimède influencent la turbulence différemment dans la section droite du tube: suppression de la turbulence près de la paroi et sa génération dans le centre. L'anisotropie du transfert thermique longitudinal et radial subit un changement de signe dans le coeur de l'écoulement.

WÄRMEÜBERGANG ENTLANG VERTIKALER ROHRE BEI TURBULENTER MISCHKONVEKTION

Zusammenfassung—Es wird über eine experimentelle Untersuchung des Wärmeübergangs und des turbulenten Transports in beheizten vertikalen kreisrunden Rohren bei Mischkonvektion mit Reynolds-Zahlen von $Re = 5100$ und $Re = 9000$ berichtet. Erstmals wurden Messungen des turbulenten Transports ($\overline{u'v'}$ und $\overline{u'_t T'}$) durchgeführt. Es wird gezeigt, daß der turbulente Wärmetransport in Wandnähe stärker unterdrückt wird als der Impulstransport, was für eine merkliche Verringerung des Wärmeübergangs bei einer nur geringen Änderung des Reibungskoeffizienten verantwortlich ist. Die Antriebskräfte beeinflussen die Turbulenz über dem Querschnitt unterschiedlich—Unterdrückung der Turbulenz in Wandnähe und Erzeugung derselben im Strömungskern. Es wird gezeigt, daß die Anisotropie von longitudinalem und radialem turbulenten Wärmetransport im Strömungskern das Vorzeichen wechselt.

РАЗВИТИЕ ТУРБУЛЕНТНОГО ТЕПЛОПЕРЕНОСА ПО ДЛИНЕ ВЕРТИКАЛЬНЫХ ТРУБ ПРИ СМЕШАННОЙ КОНВЕКЦИИ

Аннотация—Представлены результаты экспериментального исследования процессов турбулентного переноса в вертикальных обогреваемых круглых трубах при смешанной конвекции с числами Рейнольдса $Re = 5100$ и 9000. Измерения выполнены с использованием ЛДА и миниатюрного термометра сопротивления. Впервые выполнены измерения турбулентных переносов $\overline{u'v'}$ и $\overline{u'_t T'}$ в непосредственной близости от стенки, где градиенты скорости и температуры максимальны. Показано, что турбулентный перенос тепла около стенки подавляется в большей степени, по сравнению с переносом импульса, что объясняет сильное уменьшение теплоотдачи при слабом изменении кеэффициента сопротивления трения. Влияние сил плавучести на турбулентность по сечению неодинаково—вблизи стенки происходит подавление турбулентности, а в ядре течния—ее генерация. Анизотропия турбулентного переноса тепла в продольном и радиальном направлениях меняет свой знак в ядре течения.