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EXPERIMENTAL STUDY OF NONSTEADY CONVECTIVE HEAT TRANSFER IN TUBES

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Аннотация—Исследовалась нестационарная теплоотдача при турбулентном течении воздуха в трубе и скачкообразном изменении тепловыделения в двух тонкостенных трубках разной толщины, а также при различных законах изменения расхода при постоянном тепловыделении в трубках.

В нестационарных условиях коэффициент теплоотдачи существенно отличается от квазистационарного расчета и зависит от $\partial T_w/\partial\tau$ и $\partial G/\partial\tau$ или от критериев K_T и K_G.

Результаты опытов обобщены в виде критериальных зависимостей (6)-(7)—для изменения тепловыделения и (8)-(9)—для изменения расхода.

NOMENCLATURE

a,	thermal diffusion coefficient;
d,	internal diameter of tube;
c, n, φ,	functions of K_T;
G,	mass-flow rate;
f,	functions;
L,	tube length;
P_0,	inlet gas pressure;
q_w,	heat flux to gas through internal tube wall;
r,	radius;
r_0,	$d/2$;
T,	temperature;
T_w,	internal tube-wall temperature;
T_b,	calorimetric mean temperature of the flow in the cross-section considered;
T_{0b}, T_{Lb},	flow temperature at inlet and outlet;
w_x,	axial velocity;
w'_x, w'_r,	turbulent fluctuations of axial and radial velocities;
x,	axial coordinates
y,	transverse coordinate with reference point at the wall.

Greek letters

α,	heat-transfer coeeficient;
λ_T,	turbulent heat-conduction coefficient;
ε_q,	turbulent thermal-diffusion coefficient;
ν,	kinematic viscosity;
ε_τ,	turbulent kinematic viscosity;
ρ,	gas density;
τ,	time;
δ,	tube wall thickness;
$\Delta\tau_l$,	time for heat transfer from wall to axis in laminar flow;
$\Delta\tau_T, \Delta\tau_G$,	time for momentum and heat transfer from wall to the bulk flow;
Nu, Nu_0,	nonsteady Nusselt number and Nusselt number calculated for nonsteady conditions from quasi-stationary relations;

$$K = \frac{Nu}{Nu_0};$$

$$K_T = \frac{\partial T_w}{\partial \tau}\cdot\frac{d^2}{(T_w - T_b)a};$$

$$K_{T_0} = \frac{\partial T_w}{\partial \tau}\cdot\frac{d^2}{(T_w - T_b)_0 \,.\, a_0};$$

$$K^*_{T_0} = \frac{\partial T_w}{\partial \tau} \times \frac{d^2}{|(T_w - T_b)_2 - (T_w - T_b)_1| \,.\, a_0};$$

$$K_G = \frac{\partial G}{\partial \tau}\cdot\frac{d^2}{\nu G};$$

	dimensionless numbers describing time changes of T_w and G;
Re, Pr,	Reynolds and Prandtl numbers.

Subscripts

1,	refers to initial state;
2,	refers to final state.

INTRODUCTION

WITHIN the recent years the demands have increased for reliable calculation methods of heat transfer in transient flow systems and with nonsteady supply of heat to heat-transfer fluids. However, as is shown in [1], these problems are studied very little and further experimentation is required. As to the quasi-stationary method widely used in engineering, it produces essential errors in many cases. In the analysis of three-dimensional nonsteady thermal process between a fluid flow and the enclosing walls the most strict formulation of the problem should be a conjugate one, since the law of time and space changes of the temperature at the interface "flow-wall" is unknown. Therefore, even in the cases when only the temperature field in the wall or only the field of parameters in the flow should be known, simultaneous solution of the set of equations for the flow and of the energy equation (heat-conduction equation) for the wall is necessary.

This is a very complicated problem since the solution, at least a numerical one, of equations for a flow with known boundary conditions alone involves tremendous difficulties in most useful practical cases, especially in turbulent flows.

The solutions of conjugate problems are treated in [2]. The number of works dealing with conjugate problems (they are usually simple) with different approximations increases continuously. Works [3] and [10] afford an example of such a problem solution.

However, for many useful practical problems of nonsteady heat transfer, the solution of conjugate problems has failed as yet.

As a rule, experimental study of a conjugate problem is undesirable. This requires large-scale experimentation since a reliable simulation cannot be obtained in most cases as similarity of a large number of criteria (dimensionless groups) should be maintained.

Thus, urgent demand exists for the extension of one-dimensional method, which is well tested in engineering, to nonsteady heat transfer involving the heat-transfer coefficient:

$$\alpha(x, \tau) = \frac{q_w(x, \tau)}{T_w(x, \tau) - T_b(x, \tau)}. \qquad (1)$$

Here according to equation (1), α is a coefficient which describes how real processes in the three-dimensional space with real boundary conditions affect heat transfer between the fluid and the wall when they are described by a one-dimensional equation. It is impossible in principle to find the heat transfer coefficient from the solution of a one-dimensional problem. It may be obtained either from the experiment, or from the solution of a three-dimensional problem using definition (1). Certainly, α depends on the thermophysical properties of the fluid, hydrodynamics of the process and boundary conditions since, according to definition (1), it may be found from the solution of ordinary equations for viscous fluid with appropriate boundary conditions. In this sense α cannot depend on the property and the design of the enclosing walls. However, the boundary conditions (for example, T_w or q_w) depend on the wall material and design, particularly in a transient process and are unknown. However, this difficulty can be overcome. In fact, if the relation of α with the boundary conditions is studied theoretically or experimentally for typical cases, the solution by successive approximations of the heat-conduction problem for the wall with the 3rd kind boundary conditions is not difficult.

To study the effect of the boundary conditions on α, a brief analysis will be made of nonsteady phenomena occurring due to the change of the flow rate $G(\tau)$ and the wall temperature $T_w(x, \tau)$.

In [1] it is demonstrated that in an accelerating turbulent flow, the velocity profile near the wall is more uniform compared with a quasistationary case. A large velocity gradient near the wall produces the increase of $\overline{\rho w'_x w'_\tau}\, \partial w_x/\partial r$. Then, as compared with a quasistationary case, λ_T is larger at the wall and smaller in the main flow. Consequently, with a known temperature driving force, q_w and α will be higher than the quasistationary values. In a decelerating flow the behaviour of these quantities is reversed.

When the wall temperature increases with time, a heat flux due to unsteady heating of the fluid will contribute to a quasistationary convective heat transfer. This also results in an increase of α (α decreases when T_w falls).

For a tube, the time during which the past history of T_w may affect the temperature field at a certain instant τ does not exceed the time required for a heat signal to pass from the wall to the flow axis. For a tube it will be

$$\Delta\tau_l = \frac{2}{d}\int_0^{d/2} \frac{y^2\, dy}{a}$$

in laminar flow and

$$\Delta\tau = \frac{2}{d}\int_0^{d/2} \frac{y^2\, dy}{a + \varepsilon_q}$$

in turbulent flow. But in turbulent flow the transfer of heat (except in the case of liquid metals) is mainly controlled by the behaviour of the temperature field near the wall at the distance $\beta\, d/2$, where $\beta(Pr) \ll 1$. Then the past history of T_w as well as of G will affect the flow respectively by

$$\Delta\tau_T = \frac{2}{\beta d}\int_0^{\beta d/2} \frac{y^2\, dy}{a + \varepsilon_q}$$

and

$$\Delta\tau_G = \frac{2}{\beta d}\int_0^{\beta d/2} \frac{y^2\, dy}{\nu + \varepsilon_\tau}.$$

These times are usually so short for turbulent flows that all practical laws of $T_w(x, \tau)$ and $G(\tau)$ may be replaced by the first terms of their Taylor series over $\Delta\tau$ (or by the first terms of their expansion over

$$\frac{\partial^i T_w}{\partial \tau^i}\cdot\frac{d^{2i}}{(T_w - T_b)\, a^i}\ ; \qquad \frac{1}{G}\cdot\frac{\partial^i G}{\partial \tau^i}\left(\frac{d^2}{\nu}\right)^i$$

where $i = 1, 2, 3 \ldots$). Then the first terms will include

$$\frac{\partial T_w(x, \tau - \Delta\tau)}{\partial \tau} \approx \frac{\partial T_w(x, \tau)}{\partial \tau}$$

and

$$\frac{\partial G(\tau - \Delta\tau)}{\partial \tau} \approx \frac{\partial G(\tau)}{\partial \tau}.$$

Consequently, in most practical cases the dependence of α on $T_w(x, \tau)/\partial x$; $\partial T_w(x, \tau)/\partial \tau$; $\partial G(\tau)/\partial \tau$ rather than $T_w(x, \tau)$ and $G(\tau)$ should be known which considerably simplifies the study, successive treatment and application to practical calculations.

One more fact should also be included which is connected with the radial variation of the physical properties. For gases, the change in density is a controlling factor. For example when $\partial T_w/\partial \tau > 0$, the temperature gradient $\partial T/\partial r$ near the wall is higher and the density is lower compared to a quasistationary case. This will result in a decrease of $\overline{\rho w'_x w'_\tau}\, \partial w_x/\partial r$ and λ_T which will reduce α. Consequently, at transient conditions stronger effect of the temperature factor on α may be expected in comparison to the quasistationary case.

The above analysis shows that nonsteady change of the boundary conditions will affect nonsteady heat transfer through the parameters $\partial G/\partial \tau$ and $\partial T_w/\partial \tau$.† Dimensional analysis allows

† In case of a fluid temperature change at the inlet, the parameter $\partial T_{0b}/\partial \tau$ is added and the appropriate number

$$K_{T_0} = \frac{\partial T_{0b}}{\partial \tau}\cdot\frac{d^2}{T_{0b}\, a}$$

is included.

some dimensionless groups corresponding to these parameters to be found. The following seem to be the most reasonable ones in a physical sense

$$K_T = \frac{\partial T_w}{\partial \tau} \cdot \frac{d^2}{(T_w - T_b)\,a}; \tag{2}$$

$$K_G = \frac{\partial G}{\partial \tau} \cdot \frac{d^2}{Gv}. \tag{3}$$

In works [5–7] it has been shown that the change of Nu with Re, Pr and x/d in a nonsteady case is, within experimental error, the same as in the steady case, and the effect of T_w/T_b on Nu is stronger than that on Nu_0. Therefore the experimental data have been correlated by the relation

$$K = \frac{Nu(Re, Pr, x/d, T_w/T_b\, K_T, K_G)}{Nu_0\,(Re, Pr, x/d, T_w/T_b)} = f\left(K_T, K_G, \frac{T_w}{T_b}\right). \tag{4}$$

EXPERIMENTAL PROCEDURE AND APPARATUS

In the present work a well-tested procedure described in [1, 4, 5, 8] was used. In this procedure the temperature of the internal wall and heat flow from the wall to the gas were determined as a function of the distance coordinate and time from the reverse solution of the heat-conduction problem, the temperature and heat flux at the external wall being prescribed as the boundary conditions. The external wall temperature was measured by thermocouples in 20 sections of the tube and the heat flux was found by calibration and from the heat flow to the screens which was negligibly small.

The calorimetric mean temperature of gas was determined from the solution of the energy equation which included a known heat flow to the gas and was checked by measurements at the outlet from the working section. The experimental unit, as compared to that described in [4, 5], employed a more perfect measurement system and allowed to vary the flow rate in different ways. The unit was an open loop with air supply from the tanks to a working section heated by electric current.

In [4–7] the working section was a steel tube with $d = 5{\cdot}39$ mm; $\delta = 0{\cdot}32$ mm and $L = 1085$ mm. In the present experiments a tube of the same steel with $d = 5{\cdot}56$ mm; $\delta = 0{\cdot}22$ mm; $L = 1076$ mm was used.

This allowed the range of K_T and K_G to be extended.

The quasistationary heat transfer was determined, as in [4, 5], by the formula

$$Nu_0 = 0{\cdot}021\, Re^{0\cdot8}\, Pr^{0\cdot4} \left(\frac{T_w}{T_b}\right)^{-0\cdot5} F(x/d) \tag{5}$$

where $F(x/d)$ was taken from [9]. The experiment on steady-state heat transfer at the working section is well approximated by (5).

EXPERIMENTAL RESULTS AND THEIR ANALYSIS

In the present section the results of new experiments on the tube with $\delta = 0{\cdot}22$ mm are described. The experiments were carried out under two types of nonsteady conditions:

1. Stepwise increase and decrease of electric load at constant flow rate. Increase of load was carried out both with initially isothermal flow ($q_{w_1} = 0$) and from an already existing initial load ($q_{w_1} \neq 0$). The decrease in load was analogous occurring at $q_{w_2} = 0$ and $q_{w_2} \neq 0$.

2. Stepwise and continuous changes of the gas flow rates following different laws at constant heat release.

In the analysis and treatment of the experimental data, results of earlier work [4–7] were also used.

Stepwise heat release in the tube was achieved at constant mass flow rate of the gas. First the experiments were carried out with increasing load from $q_{w_1} = 0$ and decreasing load up to $q_{w_2} = 0$. The ranges of the main parameters in these experiments were as follows: the Reynolds number before switching on $Re_1 = 3\,.\,10^4$–$4{\cdot}12\,.\,10^5$; after switching off (a fully developed regime) $Re_2 = 5{\cdot}7\,.\,10^4$–$3{\cdot}32\,.\,10^5$, the temperature factor in the fully developed regime $(T_w/T_b)_2 = 1{\cdot}25$–$1{\cdot}6$, air pressure upstream of the working section

$$P_0 = (7\text{–}19)\,.\,10^5\ \text{N/m}^2.$$

The time variation of the wall temperatures in the different sections and of the flow at the outlet and of K with stepwise increase ($q_{w_1} = 0$) and decrease ($q_{w_2} = 0$) are shown in Fig. 1. Use of a thinner tube reduced the time of transient processes (compared to [6]) proportionally to the decrease of the wall thickness. This allowed $\partial T_w/\partial\tau$ to be decreased in the initial instants of time at up to 300–360 deg/s with stepwise increase of the load and up to −(330–390) deg/s when the load was decreased. K increased up to 1·5–1·6 in the case of increasing load and decreased to 0·5–0·6 with decreasing load. Obviously, the transient time is shorter at higher Re and smaller q_w.

In Fig. 2 the experimental data are plotted as $K = f(K_{T_0})$ where K_{T_0} includes the initial temperature driving force $(T_w - T_b)_0 = (T_w = T_b)_1$ in case of decreasing load and the final one $(T_w - T_b)_0 = (T_w - T_b)_2$ in case of increasing load, a_0 being determined by T_{0b}.

It is of importance to note that with equal T_w/T_b and K_{T_0}, the values of K for tubes of different wall thicknesses (i.e. for different variation of T_w) agree well.

Figure 3 represents the experimental data with stepwise increase of the load on two tubes as $K = f(K_{T_0}, T_w/T_b)$. This correlation is obtained by choice of points with similar values of T_w/T_b. The scatter of K for different temperature factors is easily observed. The effect of the temperature factor is stronger in comparison to the quasi-stationary case. The deviation of heat transfer from that under quasi-steady conditions may be attributed to the nonsteady change of $\partial T/\partial r$ at the wall because of rearrangement of the temperature profile which is included in the dimensionless number K_{T_0}. The experimental data on nonsteady heat convection with variable heat flux are correlated by the relations:

$$K = 1 + [2{\cdot}12 - 1{\cdot}12(T_w/T_b)]\,[\exp(0{\cdot}01913\,K_{T_0} - 0{\cdot}000243\,K_{T_0}^2) - 1] \qquad (6)$$

when $K_{T_0} = 0$–30; $T_w/T_b = 1$–$1{\cdot}4$,

$$K = 1 + [1{\cdot}08(T_w/T_b) - 0{\cdot}62]\,[\exp(0{\cdot}02015\,K_{T_0} + 0{\cdot}000352\,K_{T_0}^2) - 1] \qquad (7)$$

when $K_{T_0} = -30$–0; $T_w/T_b = 1$–$1{\cdot}5$.

To verify the possibility of extension of relations (6) and (7) to more complicated cases of heat flux variations, experiments have been carried out with partial stepwise increase ($q_{w_1} \neq 0$) and decrease ($q_{w_2} \neq 0$) of electric load. The ranges of the main parameters were: before increase of the load

$$Re_1 = (1{\cdot}15\text{–}2{\cdot}7)\,.\,10^5;\ (T_w/T_b)_1 = 1{\cdot}05\text{–}1{\cdot}16;\quad P_0 = (6\text{–}14)\,.\,10^5\ \text{N/m}^2$$

after increase of the load

$$Re_2 = 8{\cdot}5\,.\,10^4\text{–}2{\cdot}09\,.\,10^5;\ (T_w/T_b)_2 = 1{\cdot}35\text{–}1{\cdot}50;\quad P_0 = (6\text{–}14)\,.\,10^5\ \text{N/m}^2.$$

Figure 4 shows the change in the wall temperature at seven sections, and of the outlet flow temperatures and the dimensionless number K with partial increases and decreases of the load. In transient regimes with partial change of the load, the rate of change of the wall temperature T_w and of the heat flux q_w is less in comparison to the case of complete increase ($q_{w_1} = 0$) or complete decrease ($q_{w_2} = 0$) of the load and depends on the ratio of the loads in the initial and final steady-state conditions (if final and initial steady-state conditions are equal for stepwise increase and decrease, respectively).

The experimental data with partial change of the load have been correlated by the dimensionless number $K_{T_0}^*$ involving absolute value of the initial and final temperature driving forces. The results have been compared with relations (6) and (7). In Fig. 5 the data on partial increase and decrease of the load are plotted including the temperature factor taken from (6) and (7).

Agreement of the results on partial increases and decreases of the load correlated with the

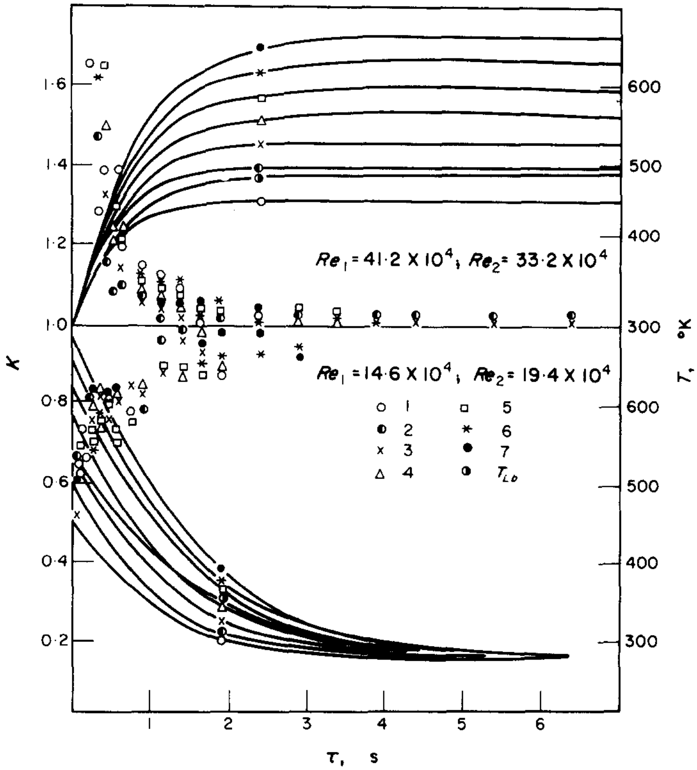

FIG. 1. Variation of T_w, T_{Lb} and K with time in the case of increase ($q_{w_1} = 0$) and decrease ($q_{w_2} = 0$) of electric load. Points with $K > 1$ correspond to increase of the load, with $K < 1$ to decrease of the load. 1–7: x/d = 17·1, 44·0, 71, 98, 126, 152, 179, respectively.

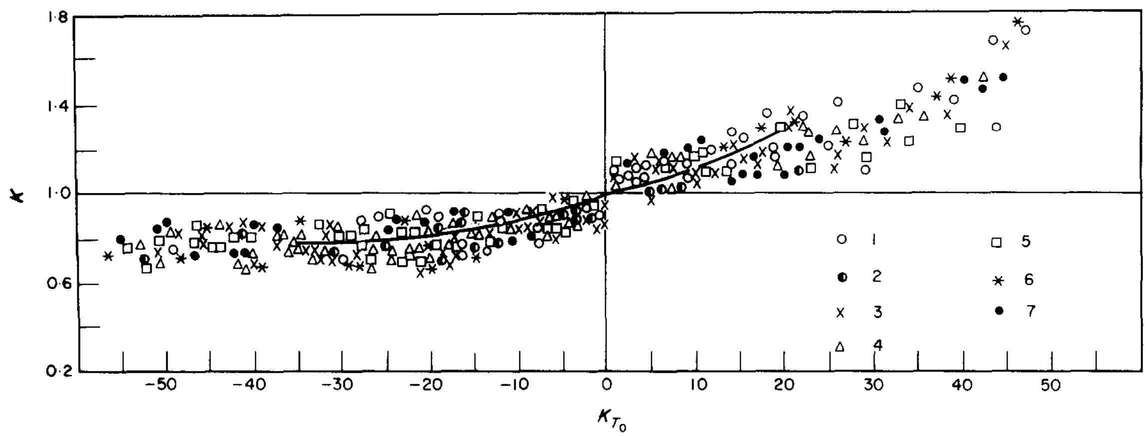

FIG. 2. Dependence of K on K_{T_0} for increase and decrease of electric load. Temperature driving force in K_{T_0} is determined for the initial instant of time in case of decreasing load and for final instant of time in case of increasing load. Solid line denotes the correlation of experimental points for the tube with $\delta = 0{\cdot}32$ mm. Legend is the same as in Fig. 1.

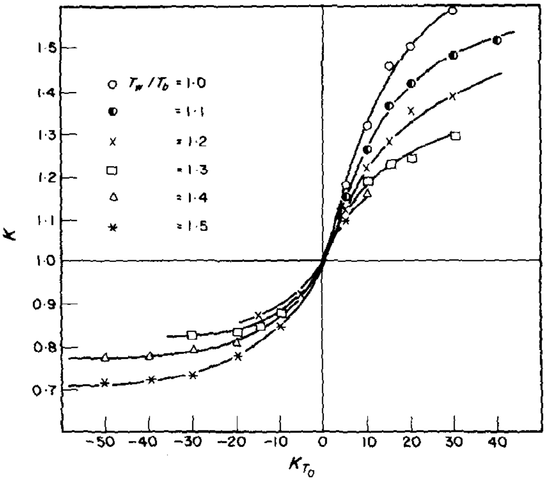

FIG. 3. Averaging relations between K and K_{T_0} with T_w/T_b as parameter.

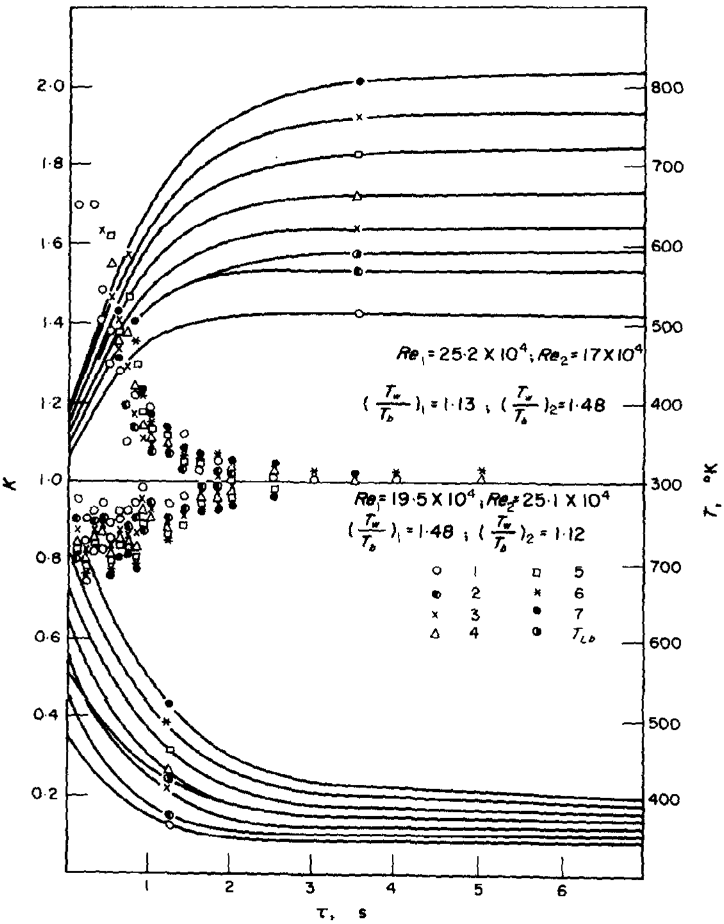

FIG. 4. Variation of T_w, T_{Lb} and K with time in case of partial increase ($q_{w_1} \neq 0$) and decrease ($q_{w_2} \neq 0$) of the load. Legend as in Fig. 1.

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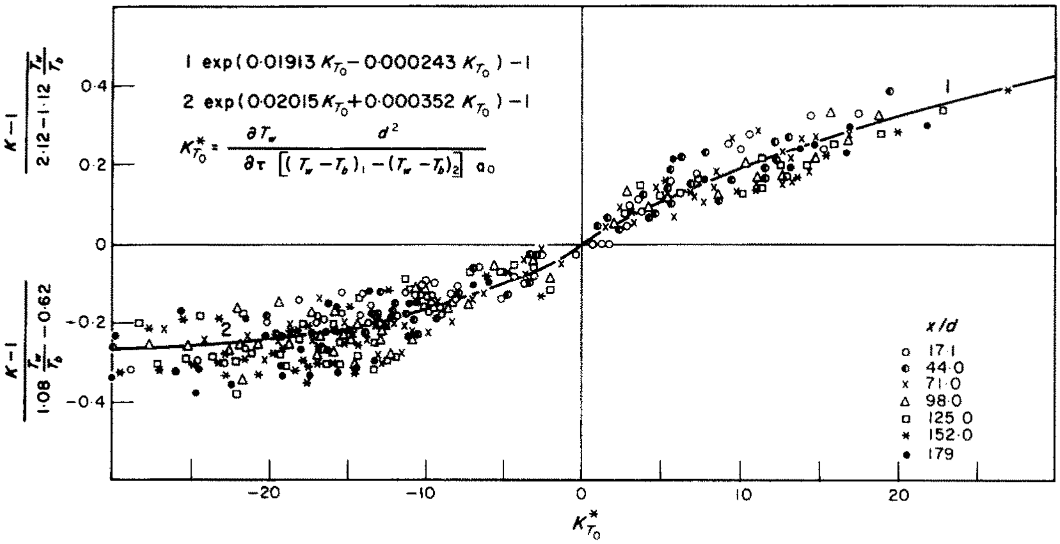

FIG. 5. Dependence of K on $K^*_{T_0}$ for partial increases and decreases of the load.

help of the $K^*_{T_0}$ and the experimental data on total increases and decreases correlated by the number K_{T_0} shows that the established relations (6), (7) describe correctly the essence of the phenomena; the nature of the number $K^*_{T_0}$ is more general in comparison to the number K_{T_0} which may be regarded as a special case of the number $K^*_{T_0}$. Relations (6) and (7) may be used with the number $K^*_{T_0}$ at different ratios of the load in the initial and final steady-state regimes.

In Fig. 6 typical variations of the air-flow rate and the corresponding changes of K for one of the tube sections are shown.

Similar changes of the flow rate were carried out within the following ranges of parameters:

$$Re_1 = (2{\cdot}7\text{–}9{\cdot}7).10^4;\ Re_2 = 10^5\text{–}2{\cdot}2\ 10^5;$$
$$G_1/G_2 = 0{\cdot}24\text{–}0{\cdot}55;\ (T_w/T_b)_1 = 1{\cdot}18\text{–}1{\cdot}45;$$
$$(T_w/T_b)_2 = 1{\cdot}15\text{–}1{\cdot}25.$$

In the experiments on a thick tube [5] the flow rate changes followed curves 1 and 4 in Fig. 6 when the numbers K_T and K_G had a parallel effect that resulted either in increase or decrease of heat transfer. Thus in case 1 a sharp flow-rate jump (0·3–0·5 s) is followed by its smooth (1–5 s) decrease up to a new steady-state value. Within this main period $\partial G/\partial \tau < 0$ and $K_G < 0$ which results in decreased heat transfer. However, if the mass-flow rate increases

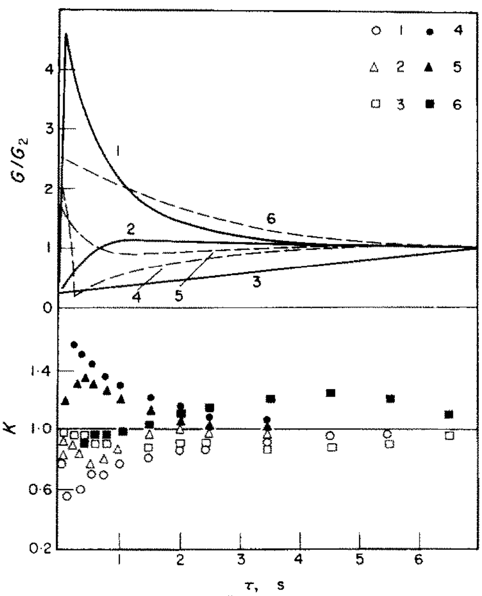

FIG. 6. Variation of mass-flow rate and K with time for different changes of the mass-flow rate (for x/d = 44).

in comparison to its initial value, then the wall temperature falls ($\partial T_w/\partial \tau$ and K_T are negative) which also reduces the heat transfer. When the mass flow rates follow curves 3, 2, 5 and 6 (Fig. 6), K_T and K_G have opposite effects on heat transfer and the net effect is reduced.

The experimental data in cases of increase and decrease of mass-flow rate have been correlated as K vs. K_T for different values of the parameter K_G (Fig. 7) as in [5]. The data on

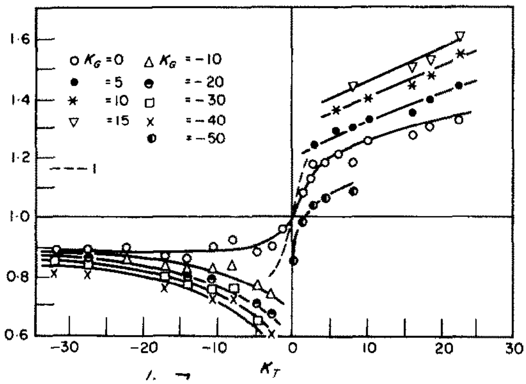

FIG. 7. Dependence of K on K_T and K_G for the case of sharp decrease of the mass-flow rate. The dotted line represents the case of $K_G = 0$ for smooth change of the mass-flow rate.

heat transfer with sharp changes of the flow rate for the same signs of $\partial T_w/\partial \tau$ and $\partial G/\partial \tau$ agree satisfactorily with the data for a thicker tube with equal K_G and K_T. The points corresponding to the regions with different signs of $\partial T_w/\partial \tau$ and $\partial G/\partial \tau$ are not numerous, and they were neglected in the correlation of experimental data. As can be seen (Fig. 7) the effect of K_G and K becomes stronger as K_T decreases. This may be attributed to the temperature factor effect whose influence could not be separated in the present experiments. The experimental data on two tubes of different thicknesses, with sharp changes of the flow rate, are correlated by the relations

$$K = 1 + 0{\cdot}1155\,(K_T)^{0{\cdot}353}$$
$$(0{\cdot}0213 + 0{\cdot}000415\,K_T)\,(K_G)^{0{\cdot}75}$$
$$\text{when } K_T = 0\text{–}25;\ K_G = 0\text{–}15. \qquad (8)$$
$$K = \exp\,[\varphi(K_T)\,.\,K_T] - C\,.\,(-K_G)^n;$$

when $K_T = -3\text{–}-32;\ K_G = -40\text{–}0.$
where $\varphi(K_T) = 0{\cdot}044\,(-K_T)^{-0{\cdot}5}$ with $-6{\cdot}2 < K_T < -3{\cdot}2;\ \varphi(K_T) = 0{\cdot}0829\,(-K_T)^{-0{\cdot}915}$ at $-32 < K_T < -6{\cdot}2$

$$C = 0{\cdot}132\,.\,(-K_T)^{-0{\cdot}8};\ n = 0{\cdot}424\,(-K_T)^{0{\cdot}14} \quad \text{at } -14{\cdot}1 < K_T < -3$$

$$C = 239\,(-K_T)^{-3{\cdot}66};\ n = 0{\cdot}0669\,(-K_T)^{0{\cdot}84} \quad \text{at } -32 < K_T < -14{\cdot}1.$$

The present experiments covered mainly the ranges with the same signs of K_T and K_G. Conversely, the experiments with a smooth change of the flow rate covered the ranges with opposite signs of K_T and K_G, the absolute values of K_T and K_G being considerably smaller. These experimental data are presented in Fig. 8.

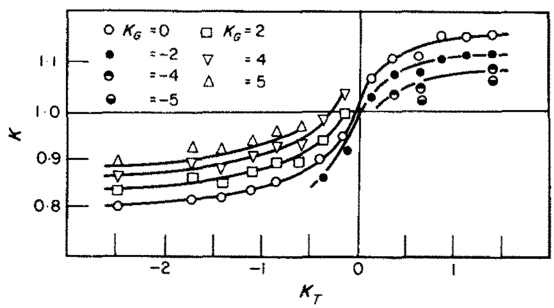

FIG. 8. Dependence of K on K_T and K_G for the case of smooth change of the mass-flow rate.

As is seen from Figs. 7 and 8, the experimental data with sharp and smooth change of the mass-flow correlated by $K = f(K_T, K_G)$ agree quantitatively. The comparison of the curves $K = f(K_T)$ at $K_G = 0$ shows that with smooth change of the mass-flow rate the effect of K_T on K is somewhat stronger. This may be attributed to the temperature factor effect. The points which are shown by the curves $K = f(K_T)$ at $K_G = 0$ with a smooth change of the mass flow rate correspond to a smaller wall temperature compared with the case of a sharp change. With a smooth change they correspond to $T_w/T_b = 1{\cdot}25$, and in case of a sharp change to $\sim 1{\cdot}4$.

Thus, the data on nonsteady heat-transfer obtained on two tubes with various wall thicknesses are correlated by the same dimensionless relations. The heat-transfer coefficient in transient regime may differ essentially from the quasi-stationary data. The experimental results may be reasonably explained by the model described in the Introduction. The dependence of heat transfer on Re and x/d at nonsteady and quasi-steady conditions is the same within the experimental error. The effect of T_w/T_b on heat transfer at nonsteady conditions is stronger. In the correlation of the present experiments on the variation of the mass-flow rate, this effect was included via K_T since K_T and T_w/T_b were interdependent, and it is the aim of further experimentation to find the correlating equation

$$K = f(K_T, K_G, T_w/T_b).$$

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Abstract— An experimental study is described of nonsteady heat transfer under conditions of turbulent air flow and stepwise heat flux change in two thin-walled tubes of different wall thicknesses for different methods of changing the mass-flow rate.

Under nonsteady conditions the experimental heat transfer coefficient differs considerably from the predicted quasi-stationary one and depends on $\partial T_w/\partial \tau$ and $\partial G/\partial \tau$ or on the dimensionless numbers K_T and K_G. The experimental data are correlated by dimensionless relations (6) and (7) for the heat flux variation and (8) and (9) for flow rate variation.

ETUDE EXPÉRIMENTALE D'UNE CONVECTION THERMIQUE INSTATIONNAIRE DANS DES TUBES

Résumé—Cet article décrit une étude expérimentale de transport de chaleur en régime transitoire sous des conditions d'écoulement turbulent d'air et d'un changement de flux de chaleur en échelon dans deux tubes à parois minces d'épaisseurs différentes et avec différentes lois de changement de débit massique.

Sous des conditions transitories, le coefficient expérimental de transport de chaleur diffère considérablement du coefficient quasi-stationnaire prévu et dépend de $\partial T_w/\partial \tau$ et de $\partial G/\partial \tau$ ou des nombres sans dimensions K_T et K_G. Les résultats expérimentaux sont corrélés par les relations sans dimensions (6) et (7), pour le changement de flux de chaleur, et par (8) et (9) pour le changement de débit.

EXPERIMENTELLE UNTERSUCHUNG DES NICHTSTATIONÄREN WÄRMEÜBERGANGS IN ROHREN

Zusammenfassung—Untersucht wurde die instationäre Wärmeübertragung bei turbulenter Luftströmung in einem Rohr und bei einer sprungartig veränderten Wärmeerzeugung in zwei dünnwandigen Röhren verschiedener Dicke, sowie bei verschiedenen Methoden der Durchsatzänderungen für konstante Wärmeentwicklung in den Rohren.

Im instationären Fall unterscheidet sich die Wärmeübergangszahl wesentlich von der quasistationären Berechnung und hängt von $\partial T_w/\partial \tau$ und $\partial G/\partial \tau$ oder von den Kriterien K_T und K_G ab.

Die Versuchsergebnisse waren als Kriterialabhängigkeiten (6) und (7) für die Wärmeentwicklungsänderung und (8) und (9) für die Durchsatzänderung verallgemeinert.