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AERODYNAMICS, HEAT AND MASS TRANSFER IN VAPOUR CONDENSATION FROM HUMID AIR ON A FLAT PLATE IN A LONGITUDINAL FLOW IN ASYMMETRICALLY COOLED SLOT

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Аннотация—В статье приводятся результаты комплексного экспериментального исследования аэродинамики, тепло-и массообмена при конденсации пара из влажного воздуха на продольно омываемой им плоской пластине в предтурбулентной области его течения ($Re < 10000$). Исследование проведено при атмосферном давлении в диапазоне температур воздуха 18–61°C, влагосодержаний 5–200 г/кг сух. возд. и скоростей потока 0,5–4 м/сек. Критериальные зависимости, описывающие процессы тепло-и массообмена для ламинарного и переходного режимов течения влажного воздуха, аналогичны при параметрах воздуха, близких к используемым в промышленной практике. Одинаковое направление процессов тепло-и массообмена при поперечном потоке пара к поверхности раздела фаз приводило к уточнению пограничного слоя, а следовательно, и к интенсификации теплообмена в целом. Сложный характер указанного процесса позволил рекомендовать проведение раздельной количественной оценки обоих эффектов с последующим суммированием коэффициентов тепло-и массоотдачи для отыскания производительности вновь проектируемых аппаратов.

NOMENCLATURE

W, velocity [m/s];
d, moisture content [g/kg dry air];
t, temperature [°C];
T, absolute temperature [°K];
D, diameter [m];
ξ, hydraulic resistance factor;
p, pressure (atmospheric);
ν, kinematic viscosity coefficient [m^2/s];
γ, alternation coefficient;
a, thermal diffusivity coefficient [m^2/s];
λ, thermal conductivity coefficient [kcal/m h °C];
k, diffusion coefficient [m/h] or [m^2/s];
α, heat-transfer coefficient [kcal/m^2 h °C];
β, mass-transfer coefficient [kg/m^2 h atm);
r, heat of vaporization [kcal/kg];
R, gas constant [kg m/kg g];
ρ, density [kg/m^3];
c, specific heat [kcal/kg °C];
i, enthalpy [kcal/kg];
G, weight [kg/h];
q, heat flux [kcal/m^2 h];
L, length [m];
m, amount of condensed vapour [kg/m^2 h];
g, gravitational acceleration (m/sec^2);
f, duct cross-section area [m^2];
u, duct cross-section perimeter [m];
F, surface under study [m^2];
τ, function;
Δ, difference.

Similarity numbers

Nu, thermal Nusselt number $= \alpha D_e/\lambda$;
Nu', diffusional Nusselt number $= \beta D_e/k_p$;
Ar, Archimedes number $= gD_e^3/\nu^2 \cdot (\rho' - \rho'')/\rho'$;

Pr, Prandtl number $= \nu/a$;
Sc, Schmidt number $= \nu/k_s$;
Re, Reynolds number $= WD_e/\nu$.

Subscripts
e, equivalent;
v, vapour;
k, in the flow kernel;
w, at the wall;
S, based on the concentration gradient;
p, based on the gradient of partial vapour pressure;
α, due to heat transfer;
β, due to mass transfer;
c, conditional;
ag, alcohol–glycerine mixture;
$\leftarrow$, dry air;
$\leftarrow$, humid air;
G, inert gas (air);
O, at the interface.

PROBLEMS of heat and mass transfer in vapour–gas mixtures are encountered in the important branches of the modern science. One of them, the condensation of vapour from its mixture with inert gases is of great importance in a number of engineering problems, such as drying of humid air, condensation of waste vapour in heat power installations, recuperation of volatile solvents by the condensation method, air conditioning, etc.

At present the requirements in designing heat and mass exchangers have increased considerably. To design a perfect apparatus it is necessary to have available quite accurate methods of their thermal calculation. However, the possibilities of analytical solution of this problem are extremely limited. This is attributed to the unsatisfactory theory of transfer under heterogeneous system conditions as well as to the essential mathematic difficulties. The solutions obtained for some particular cases, and based on the Prandtl boundary-layer theory and hydrodynamic transfer theory, have rather a limited sphere of application [1–3].

As far as the influence is concerned of transversal substance flow upon the heat-transfer rate, most investigators [2, 4–8] come to the conclusion that in the end everything is determined by the direction of the substance flow. If this direction diminishes the thickness of the boundary layer, the heat-transfer process is intensified. Otherwise, it is weakened. This conclusion is confirmed by numerous experiments. There exist, however, some data obtained, in particular, when studying drying, which are indicative of the intensifying influence of mass transfer [9] upon the heat transfer with increasing film thickness.

For quantitative estimation of heat and mass transfer in the design of the installations, separate empirical formulae are usually used, the accuracy of the majority of which cannot be considered satisfactory in the light of the modern engineering demands. At the same time it is more difficult to produce accurate experimental data using vapour–gas mixtures as the heating medium, than to do so using pure liquids or gases.

There are no reliable data for heat and mass transfer in transient flow regime usually observed at $Re < 10\,000$. The formulae [10–12] suggested in literature, do not take into consideration the effect of natural convection which is possible in this regime. At the same time they show the effect upon the results of the investigations of the individual properties of the experimental installation, which effects are very difficult to be allowed for.

The aim of the present work is a combined study of the problems of aerodynamics, heat and mass transfer in vapour condensation from humid air with its laminar and transient flow patterns over a flat plate.

The study was carried out at atmospheric pressure within the temperature range 18 to 61 °C. In the experiments the air flow velocity changed from 0·5 to 4·0 m/s and its vapour content by weight, from 0·005 to 0·17 kg/kg, which corresponded to moisture contents of 5 to 200 g/kg dry air.

The experimental installation (Fig. 1) was a closed contour consisting of an experimental condenser, a fan, heaters, a cooler and vapour generator–moistener. The experimental condenser consists of a horizontal rectangular duct 20 mm wide, 150 mm high and 600 mm long. The duct is made of a smooth polished brass plate 6 mm thick, of an area of 150 × 600 mm, steadily to the air through the sprayer made in the form of a perforated hemisphere. The humid air was delivered to a damping section, where the flow was restored after its local disturbance during moistening with vapour. From the pipeline to the experimental condenser a smooth passage was made with the help of specially

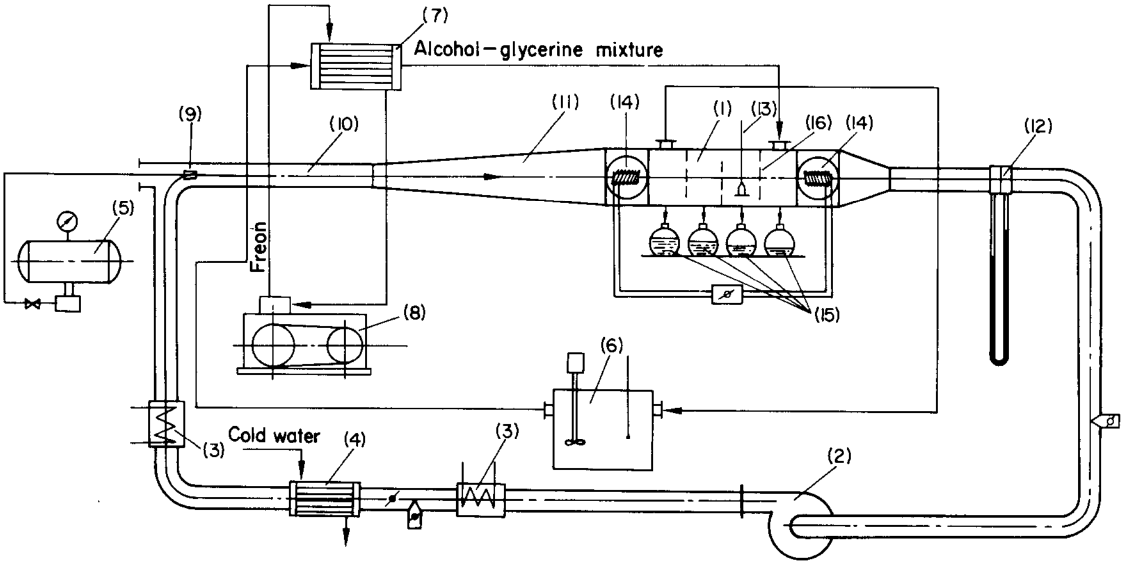

FIG. 1. Schematic drawing of the experimental unit.

(1) condenser; (2) fan; (3) heater; (4) cooler; (5) vapour generator; (6) thermostat; (7) evaporator; (8) refrigerator; (9) pulverizer; (10) humidifying section; (11) diffuser; (12) nozzle; (13) thermanemometer; (14) electric-film sensing element; (15) condensate pipe-lines; (16) membranes in the chamber.

and walls which, to allow visual observation of the condensation process, are made of two Plexiglass sheets with an air layer between them. On the back side of the plate a metal chamber with three partitions has been mounted into which chamber a cooling liquid (a mixture of 90 per cent ethyl alcohol and 10 per cent glycerine) is fed by the thermostat pump. The mixture was cooled in the evaporator by Freon coming from the refrigerator.

The fan supplied the air through a connecting pipeline to the heater and cooler, where, depending on the conditions of the experiment, it reached the prescribed temperature and then was moistened by vapour. The vapour was fed modelled diffuser. At the exit of the diffuser the flow had a smooth velocity profile across the section with no tendency to separation. The entry length L (the distance from the place of vapour moistening of air to the condenser section under test) was about 70 D_e.

The air flow rate was determined with the help of an independently calibrated nozzle with a profile of a quarter of a circle.

The temperatures of air, condenser walls and alcohol–glycerine mixture were measured by copper-constantan thermocouples. In the working section of the condenser the temperature fields were found by a planimetric method, and the mean-integral values of the temperatures used as the base of the heat- and mass-transfer coefficients were estimated.

The flow rate of the alcohol–glycerine mixture and the amount of condensed vapour were found by weighing with the help of an analytical balance.

The air humidity was measured by an electric film transmitter in which a thin film of water-absorbing salt solution covering a pure surface of the Plexiglass cylinder served as a sensitive element. Two open platinum electrodes, fed by industrial frequency voltage, were put on the film as a bifilar winding.

At present an approximate estimation of the instant of turbulence onset in a flow is made as a rule by computing the Reynolds number Re. If Re is less than some critical value, no turbulence is assumed. The value of 2300 is usually assumed as the critical value of Re obtained for long circular pipes. It is considered that at $Re <$ 2300 even the strongest disturbances in the flow will attenuate in a time.

For a more accurate definition of the instant of the onset of turbulence, we use, in practice, the experimental relations between the hydraulic resistance coefficient ξ and Re, i.e. $\xi = \xi(Re)$. The critical value of Re is determined as the lowest one for which the relation $\xi = \xi(Re)$ does not deviate from the linear Hagen–Poiseuille law.

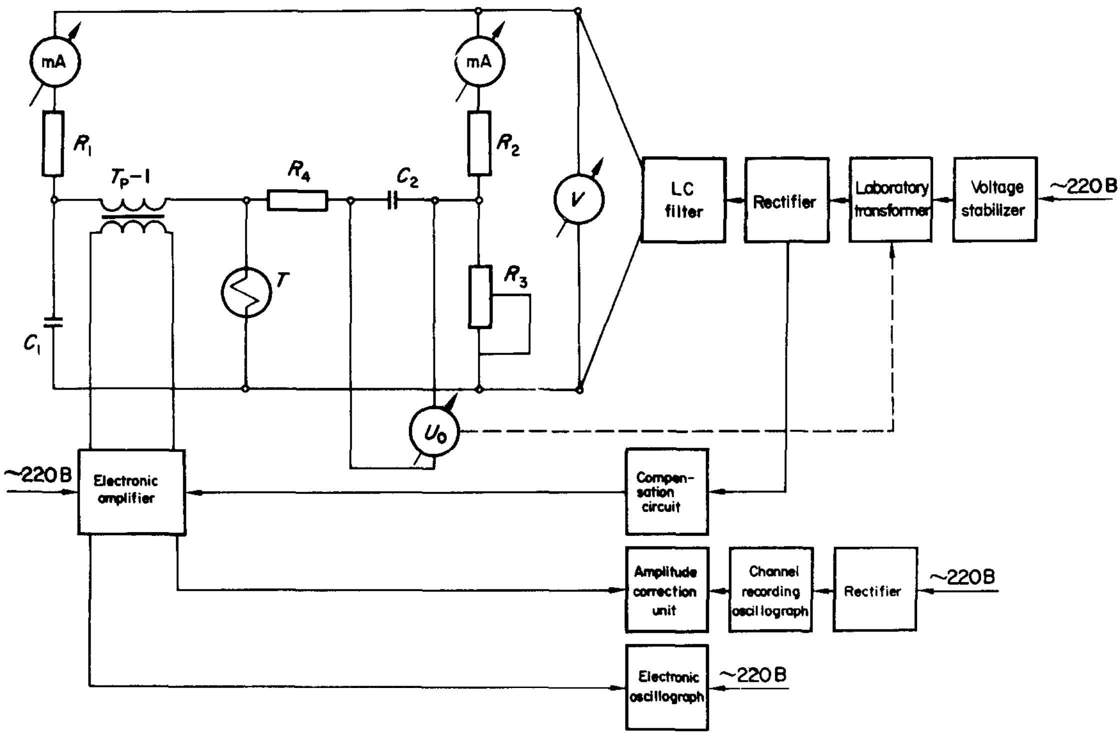

FIG. 2. Thermoanemometer block-diagram.

The above methods for the estimation of turbulence onset in a flow are proved to be of little use for ducts of industrial installations. In fact, it is typical of these ducts that they have a great number of various local resistances (sudden expansions, contractions, turns, etc.) placed rather closely to one another. It is known, however, that two or more hydraulic resistances spread over a distance smaller than the hydrodynamic entry length, present a total resistance different from the sum of the individual resistances. Moreover, the behaviour of the flow in an arbitrary duct section appreciably depends on the combined influence of the nearest local resistances and the flow is nonuniform over the greatest portion of the duct's length. Under these conditions reliable results can be obtained only by direct observation of the turbulence onset at a given position along the flow.

In the present work for the above purpose as well as for the study of the character of the transition of laminar flow into turbulent one, a special device with a thermoanemometer (Fig. 2) was designed and built. As a sensitive element, i.e. the transformer of flow velocity into electric signal, a platinum filament, T, was used 20 μ dia. and 3 mm long. The filament included in one of the equilibrium bridge arms, was heated by electric current. The bridge was fed by d.c. from the rectifier via the LC-filter. A pointer null-galvanometer, U_0, connected into the bridge diagonal through the filter. R_4C_2 served as an indicator of bridge unbalance.

The circuit is based on the combination of two principles: constant temperature with the average flow velocity and direct current with the oscillatory components of its rate. Voltage oscillations of the transformer proved to be the useful signal which was supplied to the electronic amplifier through the matching transformer $T_p - 1$. To eliminate noise compensation circuits in the form of a rectifier and RC-filter were used. In the amplifier frequency correction of the useful signal was performed and then the latter was passed in the amplitude correction unit, composed of point germanium diodes. The unit was a linearity circuit with the characteristic of reverse curvature by contrast with thermoanemometer circuit. The corrected signal corresponding to velocity pulsations was registered by a loop oscillograph. To control all the circuits an electronic oscillograph was engaged. The whole circuit was screened carefully.

Recent investigations [13] showed that transitions of the flow from laminar to turbulent state do not occur suddenly, as it was thought before. To the above transition corresponds a particular range of Re numbers within which the flow has alternate character, i.e. an irregular change is observed of laminar and turbulent flow states. To each change of states corresponds a certain velocity distribution. It was decided to describe the physical character of this alternating flow by the so-called alternation coefficient γ [14] which defines the time of the existence of the turbulent states within a fairly long time interval.

During the tests the oscillograms of the flow velocity oscillations in the condenser channel were recorded. The flow was stabilized over the test section. The flow velocity ranged from 0·5 to 4·0 m/s which corresponded to the range of Re = 1000–10000. From the oscillograms the values of γ were calculated and the $\gamma_v(Re)$ relations plotted (Fig. 3). These relations allow

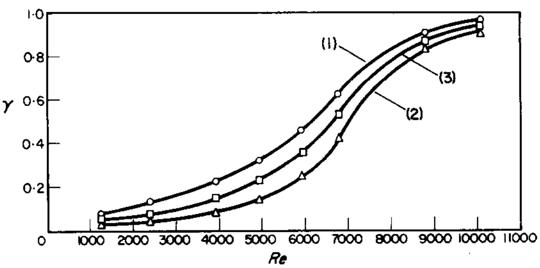

FIG. 3. Plot of the transmittance coefficient vs. Re: (1) in the middle of the channel; (1) at a distance of 400 mm from the channel middle; (3) at a distance of 65 mm from the middle of the channel.

an estimation of the onset and development of turbulence in a flow and a sufficiently accurate evaluation of the critical Re numbers, which completely agree with the laminar and turbulent flows. The above numbers are about 2000 and 10 000.

Heat and mass transfer was studied under the steady-state conditions. Within an energy test the following parameters were kept constant:

(1) Temperature, velocity and relative humidity of the air in the condenser.

(2) Temperature and flow rate of the alcohol–glycerine mixture in the condenser.
(3) Temperature of the condenser wall.

In the present study local heat- and mass-transfer coefficients were determined in the middle cross-section of the third downstream condensation test zone at a distance of ($L/D_e \approx 70$) (see Fig. 1). The specific amount of heat liberated from the air by forced convection, was expressed as

$$q_\alpha = \alpha(t_k - t_w) \tag{1}$$

and the amount of the condensed vapour, as

$$m = \beta(P_{v(k)} - P_{v(w)}). \tag{2}$$

to the heat-transfer coefficient when pure vapour is condensed

$$\alpha_c = \frac{q}{t_{(k)} - t_{(w)}} = \alpha + r\beta\frac{P_{v(k)} - P_{v(w)}}{t_{(k)} - t_{(w)}} = \alpha + r\beta\frac{\Delta P_v}{\Delta t}. \tag{5}$$

The increase of the air moisture content in the experiments was accompanied by an increase of the motive force of vapour diffusion ΔP, and hence, by the increase of the number of its molecules condensing on the interface. At the same time the conventional heat-transfer coefficient α_c also increased due to a more intensive

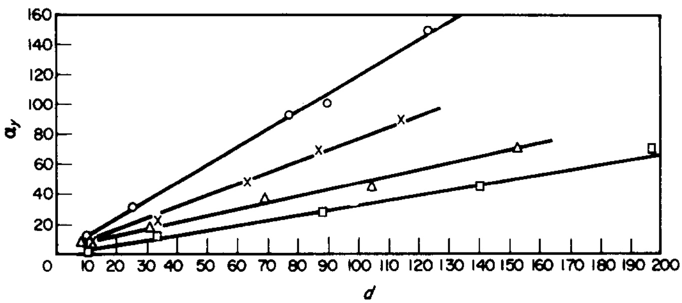

FIG. 4. Plot of conventional heat-transfer coefficient versus moisture content at $t_v = 56$–$61°C$; Δ—$W = 1{\cdot}2$–$1{\cdot}4$ m/s; □—$W = 0{\cdot}7$ m/s; ○—$W = 3{\cdot}6$–$3{\cdot}9$ m/s; ×—$W = 2{\cdot}0$–$2{\cdot}3$ m/s.

During the vapour condensation from humid air, heat- and mass-transfer processes are interconnected; however, the total heat amount liberated from the air can conditionally be divided into two parts and presented as the sum

$$q = q_\alpha + q_\beta = \alpha(t_{(k)} - t_{(w)}) + r\beta(P_{v(k)} - P_{v(w)}). \tag{3}$$

The total amount was found from the heat balance equation

$$\frac{G_{ag}C_{ag}\Delta t_{ag}}{F} = q = \frac{G_{\text{dry air}}\,\Delta i_{\text{humid air}}}{F}. \tag{4}$$

Simplified relations are often used by introducing the value of the conditional heat-transfer coefficient which is similar in estimation

heat liberation due to the change of phase. Heat- and mass-transfer intensity was also controlled by hydrodynamics: the higher the heat-transfer intensity, the higher the flow velocity (Fig. 4).

The increase in the air temperature was accompanied by the decrease in heat-transfer intensity (Fig. 5). This is attributed to the increase of the temperature difference Δt, other conditions being equal. Experimental data for turbulent motion of vapour–gas medium obtained by various investigators both in our country and abroad showed the existence of the approximate analogy between heat and mass transfer [15–19]. This allows the following to be supposed as a preliminary hypothesis which will be verified further: "in the present study the existence of the above analogy is also thought possible".

Consider a forced steady-state flow in the channel of the experimental condenser. Let us consider the air to be an incompressible

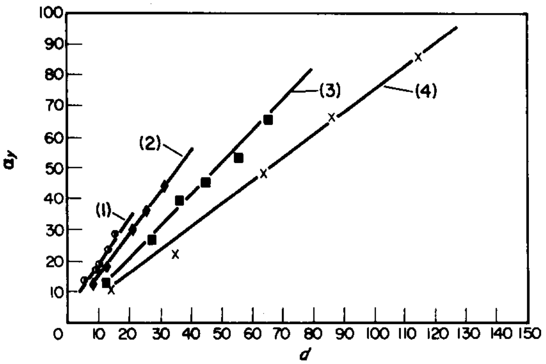

FIG. 5. Plot of conventional heat-transfer coefficient versus moisture content at $W = 2{\cdot}0$–$2{\cdot}3$ m/s. 1, $t_v = 18{\cdot}9$–$20°C$; 2, $t_v = 28{\cdot}5$–$31{\cdot}9°C$; 3, $t_v = 41{\cdot}6$–$44{\cdot}6$; 4, $t_v = 56$–$61°C$.

medium and its flow to be determined by the pressure drop and the motive force. Heat and mass transfer between the air and cooling liquid proceeds through the wall. The direction of the heat flow is not accounted for. All physical properties of air except density are assumed constant and, in particular, independent of temperature. Also, the heat- and mass-transfer processes are taken to be steady.

In addition to the geometric similarity the main conditions for the existence of an approximate analogy are as follows:

(a) Satisfying the equality $a = k_s = k_p R_v T$ or $Pr = Sc$.
(b) Similarity of boundary conditions for steady processes, the similarity in the starting flow section and that of the parameter distribution on the interface.
(c) Low concentration of an active mixture component (vapour) and small difference of its partial pressures ΔP_v.

By accepting these assumptions and eliminating the influence of the substance (vapour) flow on the fields of velocity, temperature and partial pressure, heat- and mass-transfer processes can be described with the help of differential equations of motion, thermal conductivity, diffusion and transfer across the phase interface. Then, confining ourselves to the direction of the channel axis x we can write equation of motion

$$W_x \frac{\partial W_x}{\partial x} + W_y \frac{\partial W_x}{\partial y} + W_z \frac{\partial W_x}{\partial z} = -\frac{1}{\rho}\frac{\partial P_k}{\partial x} + \nu\left(\frac{\partial^2 W_x}{\partial x^2} + \frac{\partial^2 W_x}{\partial y^2} + \frac{\partial^2 W_x}{\partial z^2}\right) + \frac{\Delta \rho g}{\rho} \tag{6}$$

heat-conduction equation

$$W_x \frac{\partial t}{\partial x} + W_y \frac{\partial t}{\partial y} + W_z \frac{\partial t}{\partial z} = a\left(\frac{\partial^2 t}{\partial x^2} + \frac{\partial^2 t}{\partial y^2} + \frac{\partial^2 t}{\partial z^2}\right) \tag{7}$$

diffusion equation

$$W_x \frac{\partial P_v}{\partial x} + W_y \frac{\partial P_v}{\partial y} + W_z \frac{\partial P_v}{\partial z} = k_s\left(\frac{\partial^2 P_v}{\partial x^2} + \frac{\partial^2 P_v}{\partial y^2} + \frac{\partial^2 P_v}{\partial z^2}\right) \tag{8}$$

transfer equations at the interface

$$\alpha \Delta t = -\lambda \left(\frac{\partial t}{\partial y}\right)_{y=0} \tag{9}$$

and

$$\beta \Delta P_v = -k_p \left(\frac{\partial P_v}{\partial y}\right)_{y=0} \frac{P}{P_{G(O)}}. \tag{10}$$

Here y is the direction normal to the surface.

Using the similarity method, we can write the solution of the above equations in the general form conformable to heat- and mass-transfer coefficients used in the calculations of heat transfer

$$Nu = \tau(Re, Ar, Pr) \tag{11}$$

and of mass transfer

$$Nu' = \tau'(Re, Ar, Sc). \tag{12}$$

Treatment of the experimental data has shown that the difference between Nu and Nu'

is negligible and within the range of the experimental error (Fig. 6). This allowed the above hypothesis of the existence of the approximate analogy between heat and mass transfer to be assumed valid and the functions τ and τ' to be approximated by the following equations:

Laminar regime (Re = 1000–2000):

Heat transfer

$$Nu = 4{\cdot}55\,(10)^{-3}\,(Re)^{0{\cdot}36}\,(Ar, Pr)^{0{\cdot}4} \quad (13)$$

Mass transfer

$$Nu' = 4{\cdot}55\,(10)^{-3}\,(Re)^{0{\cdot}36}\,(Ar,Sc)^{0{\cdot}4}. \quad (14)$$

Transient regime (Re = 2000–10,000):

Heat transfer

$$Nu = 6{\cdot}48\,(10)^{-5}\,(Re)^{0{\cdot}92}\,(Ar\,Pr)^{0{\cdot}4}. \quad (15)$$

Mass transfer

$$Nu = 6{\cdot}48\,(10)^{-5}\,(Re)^{0{\cdot}92}(Ar\,Sc)^{0{\cdot}4}. \quad (16)$$

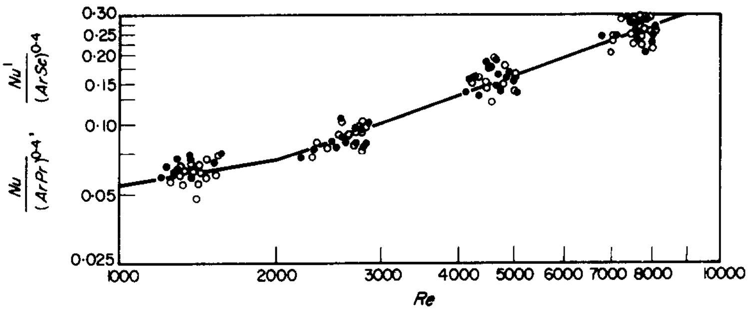

FIG. 6. Heat and mass transfer in condensation of vapour from humid air in a rectangular duct ($4D_e$ = 70).

In the above relations the equivalent channel diameter $D_e = 4f/u$ was used as the reference diameter, and the mean temperature in the cross-section of the condenser under study, as the reference temperature. All physical constants refer to the air–water mixture.

The physical meaning of the solution proposed comes down to the following: "under laminar and transient flow regimes of humid air, heat transfer process depends entirely on the nature of the velocity field, its correlations with the temperature and partial pressure fields and on the relationship between the buoyancy and the inertia forces.

CONCLUSIONS

1. With moisture contents 6–10 $\leqslant d \leqslant$ 20–25 g/kg of dry air and flow velocities W = 0·5–4 m/s the influence of heat- and mass-transfer upon the heat-transfer intensity is approximately the same. With moisture contents d > 25 g/kg of dry air, the mass transfer predominates in the total effect. Thus, for example, with d = 202 g/kg of dry air and W = 0·69 m/s, the mass transfer contributes about 90 per cent of the total heat transfer.

2. The higher the air velocity and the lower the temperature drop, the more intensive is the total transfer process. When heat transfer is intensive enough within the temperature range W= 0·5–4 m/s no noticeable moisture entrainment is observed; this makes this range to be considered as one of the most suitable for real processes.

3. We suggest the use of the Nusselt power equations similar for heat and mass transfer and valid under the conditions of laminar and transient regimes of a hydrodynamically stabilized ($L/D_e \approx 70$) flow of vapour–gas medium. It is recommended to calculate heat- and mass-transfer coefficients by these two equations with

their subsequent summation for the determination of the performance of the heat exchanger.

4. A method is proposed for the estimation of the gas flow regime by means of the alternation coefficient.

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Abstract—In the present paper results are reported of a combined experimental study of aerodynamics and heat and mass transfer in the condensation of vapour from humid air on a flat plate in a pre-turbulent region of a longitudinal flow ($Re < 10000$). The study has been carried out at atmsopheric pressure within the range of air temperatures 18 to 61°C, moisture content 5 to 200 g/kg of dry air and flow velocities 0·5 to 4 m/s. The critical relations describing heat- and mass-transfer processes for laminar and transient flow patterns of humid air are similar when the air parameters are close to those used in industry. Similar directions of heat- and mass-transfer processes when the interface is in a transverse vapour flow led to thinning a boundary layer and hence to intensifying heat transfer in the total process. The complicated nature of the above process allowed separate quantitative estimations of both effects and a subsequent summation of heat- and mass-transfer coefficients to determine the efficiency of newly designed installations.

Résumé—Dans le présent article, on expose des résultats d'une étude expérimentale combinée de l'aérodynamique et du transport de chaleur et de masse dans la condensation à partir de l'air humide sur une plaque plane dans une région pré-turbulente d'un écoulement longitudinal (Re < 10 000). L'étude a été conduite à pression atmosphérique dans la gamme de températures de l'air allant de 18 à 61°C, de quantité d'humidité allant de 5 à 200 g/kg d'air sec et de vitesses d'écoulement allant de 0,5 à 4 m/s. LEs relations sans dimensions décrivant les processus de transport de chaleur et de masse pour des configurations d'écoulement laminaire et transitoire d'air humide sont semblables lorsque les paramètres de l'air sont voisins de ceux employés dans l'industrie. Des directions semblables des processus de transport de chaleur et de masse lorsque l'interface est dans un écoulement transversal de vapeur ont conduit à l'amincissement de la couche limite et ainsi à intensifier le transport de chaleur dans le processus total. La nature compliquée du processus ci-dessus a permis des estimations quantitatives séparées des effets ainsi qu'une sommation consécutive des coefficients de transport de chaleur et de masse pour déterminer le rendement d'installation nouvellement conçues.

Zusammenfassung—Eine experimentelle Untersuchung der aerodynamischen Vorgänge kombiniert mit denen des Wärme- und Stoffaustausches wird behandelt. Untersucht wurde die Kondensation von Dampf aus feuchter Luft an einer abenen Platte im nichtturbulenten Bereich einer Längsanströmung (Re ∠ 10 000). Die Untersuchung geschah bei Atmosphärendruck, Temperaturen zwischen 18 und 61°C, Feuchtigkeitsgraden zwischen 5 und 200 g/kg trockene Luft und bei Strömungsgeschwindigkeiten zwischen 0,5 und 4 m/s. Die Beziehungen, die den Wärme- und Stoffaustausch von feuchter Luft für laminare Strömung und für Übergangsströmung beschreiben, sind ähnlich, wenn die Luftparameter nahe den in der Industrie verwendeten liegen. Ähnliche Richtungen von Wärme- und Stoffaustauschprozessen bei querangeströmter Grenzfläche führten zu einer Verkleinerung der Grenzschicht und somit zu einer Verstärkung des Wärmetransports im ganzen Prozess. Die komplizierte Natur des obigen Prozesses erlaubte getrennte quantitative Abschätzungen beider Effekte und eine Addition von Wärme- und Stoffaustauschkoeffizienten, um die Wirksamkeit von neu entworfenen Anglagen zu bestimmen.