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EXPERIMENTAL STUDY OF HEAT TRANSFER AT THE TOTAL CONDENSATION OF MIXED VAPOURS OF MISCIBLE LIQUIDS

J. BANDROWSKI and A. BRYCZKOWSKI
Silesian Technical University, Institute of Chemical Engineering and Chemical Apparatus Construction, Gliwice, Poland

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Abstract—Experimental studies of heat transfer at the atmospheric-pressure condensation, on a horizontal tube, of binary and ternary vapours of miscible liquids, have been carried out. As a result of treatment of experimental data, a correlation equation presenting the relationship between the Nusselt number and the condensation number, was obtained.

NOMENCLATURE

A,	power exponent;
C,	constant;
c,	specific heat;
d,	diameter;
F,	surface area;
G,	mass rate of flow;
g,	acceleration due to gravity;
Q,	heat transferred;
q,	heat flux;
r,	heat of condensation;
T,	temperature;
t,	temperature;
u,	liquid-phase mass fraction of the component;
x,	liquid-phase mole fraction of the component.

Greek symbols

α,	heat-transfer coefficient;
η,	dynamic viscosity;
λ,	thermal conductivity;
ρ,	density.

Subscripts

A, B,	components of mixture;
c,	liquid phase (condensate);
i,	interface;
m,	mean value;
n,	component;
s,	element of the heat-receiving surface;
w,	cooling water circulation;
∞,	bulk of the fluid.

Superscripts

$*$,	equilibrium value.

Dimensionless groups

C_v, $= \dfrac{d^3\rho_c^2 gr}{\lambda_c \eta_c \Delta t_m}$, condensation number;

Nu, $= \left(\dfrac{\alpha d}{\lambda}\right)$, Nusselt number.

INTRODUCTION

THE TOTAL condensation of mixed vapours forming a homogeneous liquid phase is a case more often encountered in practice than condensation of single vapours, but in contradiction to the latter it has not been explained theoretically, up to now, in the full range of parameters under interest.

The number of experimental papers devoted to the problem as above is also small. This is caused by the complicated process mechanism on the one hand, and by the experimental difficulties, on the other hand.

Already the definition of heat-transfer coefficient on the condensing vapour side presents difficulties resulting from the fact of the occurrence of thermal resistances in both phases (i.e. liquid and vapour) their variation trends under the effect of heat flux variations being opposite to each other.

In view of this, the resistances in both phases should be taken into account separately. At such a formulation of the problem, the determination of the interfacial temperature, very difficult to be realised experimentally, becomes indispensable.

In turn, the values of the interfacial temperature, found by calculations using well-elaborated relationships valid for the liquid phase, would be uncertain (particularly in the range of smaller temperature differences [1]) both due to the errors committed at the calculation of mixture properties and to the amount of heat transferred in the vapour phase.

Consequently, there are various ways of approach to this problem, met in the literature. The theoretical papers [1–3] operate, as a rule, with heat-transfer coefficient, defined for the total temperature difference.

In experimental works [4–8] the heat-transfer coefficient is calculated for the liquid phase under the assumption that the interfacial temperature is equal to the bubble point of the condensate.

Hobler in his monograph [9] suggests, in the case of condensation of mixed vapours of miscible liquids, the possibility of calculating the heat-transfer coefficients by two methods. The first one, proposed by Kern [10], consists in using the relationships holding for the condensation of single vapours, the properties of the mixture being computed as weighted means with respect to the mass fractions of properties of pure components. The second one, worked out by Pressburg and Todd [4], assumes the calculation of heat-transfer coefficient for a binary vapour mixture as a mean from the value of heat-transfer coefficients for pure components, weighted according to their mole fractions.

The literature quotes two sources only [3], [11], referring strictly to the total condensation, on a horizontal tube, of mixed vapours forming a homogeneous liquid phase. However, the literature survey takes also into account other papers, concerning the cases different from that studied in the present work. The criterion of the selection of the discussed papers was the occurence of total, or approximately total, film condensation of mixed vapours of miscible liquids.

In the papers [4, 5, 7, 8] the postulate of total condensation was fulfilled only approximately. In Haselden and Platt's work [8] as well as in Pressburg and Todd's one [4], *ca* 10 per cent of vapour mixture flowing into the condenser did not condense, whereas Wallace and Davison [5] report that the amount of non-condensed vapours reached 30 per cent of the entire vapour mixture.

Theoretical elaboration of the condensation of binary vapours on vertical wall was given by Sparrow and Marschall [1]. The problem was solved basing on an analytical model, which assumed the existence, apart from the layer of flowing-down condensate, of a boundary layer (contacting with the former) in which velocity, concentration and temperature variations occur. For the condensate film the Nusselt solution was applied, whereas for the vapour layer a system of differential equations, describing the transfer phenomena, was derived. The authors have worked out an algorithm enabling the simultaneous solution of systems of equations for both phases, which makes it possible to compute the interfacial temperature and the amount of heat transferred.

The results of calculations, performed by means of a digital computer, for the system methanol–water, have been represented in the form of a graph:

$$\frac{q}{q_\gamma} = f(T_v - T_s), \quad T_v,$$

where q is the heat flux calculated from a relationship holding for the condensate layer for the liquid–vapour interfacial temperature, q_γ is the heat flux referring to the vapour bulk temperature, and T_v is the saturated vapour temperature. The results refer to the temperature range $T_v - T_s = 10$–35 degC. The solution in the range of lower temperature differences, are, according to the authors, of no practical utility, as being subject to considerable errors. If the difference $T_c^* - T_s > 5$–10 degC (where T_c^* denotes the bubble point of the condensate) the calculations may be greatly simplified by assuming that the interfacial temperature $T_i = T_c^*$. The condition $T_c^* - T_s > 5$–10 degC holds only for methanol–water system; the numerical values of this difference for other systems may be determined only by performing a full computational cycle. Tamir [2] suggests, for the case of binary gravity-flow film condensation, an approximate solution in integrated form, making it possible, by trial and error, to determine the interfacial temperature T_i and the amount of heat transferred. The calculations have been carried out for methanol–water system and the results compared with the exact solution of Sparrow and Marschall [1]. The maximal deviations between the exact solution and the approximate one in the temperature difference interval $T_v - T_s = 10$–35 degC did not exceed 10 per cent, so that the author [2] recommends his method for practice.

Chaykovsky, Smirnov and Domansky [3] have presented a theoretical solution for the condensation of binary vapours on a horizontal tube. The results of calculations, carried out for mixtures of Freon 12 and 22, have been confirmed experimentally.* The mathematical model accepted in the theoretical part, assumed that Nusselt's solution holds for the condensate layer. For the vapour phase the uniform inflow of vapour, equilibrium, lack of temperature jump on liquid–vapour interface and negligibly small vapour velocity were assumed. Besides, the thermal resistance between the bulk of vapour, heat transfer due to natural convection and the thermodiffusion effect were neglected. The above-mentioned assumptions enabled to describe the vapour-phase process by a non-linear second-order differential equation. The solution of this

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*Recently, a paper of Chaykovsky and coworkers [17] appeared, devoted to the experimental determination of heat-transfer coefficient at the total condensation of mixtures of Freon 12 and 22 on a finned horizontal tube equation, taking into account the appropriate boundary conditions, together with the equation describing the behaviour of the condensate phase, makes it possible to determine the interfacial temperature and the heat flux. The results of calculations, performed by a digital computer, have been worked out in the form of graphs

$$\alpha = f[(T_v - T_s), \quad \text{composition}]$$

where:

$$\alpha = \frac{q}{t_v - t_s}. \tag{1}$$

The results of experiments, carried out in the temperature difference interval $t_p - t_s \leqslant 1$–26 degC have confirmed to the full the suitability of the analytical model assumed by the authors. The largest differences between the experimental coefficients and the calculated ones occured in the range of small temperature differences: $1\,\text{deg} < t_p - t_s < 5$ degC, in which the values of calculated heat-transfer coefficients proved to be on the lower side and the maximal error reached 20 per cent. In the interval $t_v - t_s > 5$ degC a very good agreement between experimental and calculated values is observed, the former ones being slightly higher. The authors attribute the observed tendency to the mixing phenomenon in the vapour phase.

Among previously reported experimental works, papers [4] and [5] dealt with the approximately total condensation of mixed vapours on a horizontal tube.

The results of work [5], concerned with the ethanol–water system, indicated the qualitative agreement with those obtained from Nusselt's relationship, holding for the condensation of single vapours. The heat-transfer coefficients were calculated using the bubble point of the condensate.

A similar agreement was obtained by Haselden and Prosad [6] at the study of the condensation of oxygen–nitrogen vapours on a horizontal tube.

As previously mentioned, Pressburg and Todd [4] suggest the calculation of heat-transfer coefficients at the condensation of mixed vapours as a weighted mean, with respect to the mole fractions, of the heat-transfer coefficients for single vapours, viz.

$$\alpha_m = \alpha_A x_A + \alpha_B x_B. \tag{2}$$

The coefficients α_m, α_A and α_B in equation (2) are taken for the same temperature difference, the bubble point of the condensate being assumed as the interfacial temperature.

The above relationship has been obtained on the grounds of the experimental data for the following systems: methanol–benzene, ethanol–benzene, *n*-butanol–benzene, methanol–acetone and methanol–*i*-propanol.

Mirkovich and Missen [7] investigated the condensation of mixed vapours in the following systems: dichloromethane–methanol, *n*-pentane–*n*-hexane, *n*-pentane–methanol, *n*-pentane–dichloromethane, on a nickel ring, constituting at the same time a resistance thermometer for the measurement of mean surface temperature. The results indicated that the relationship given by Pressburg and Todd does not hold for dichloromethane–methanol system. Similarly, the experimental results of Haselden and Platt [8] referring to the condensation of methanol–*n*-propanol vapours in a special system with a short path of condensate downflow, diverged from the results calculated as suggested both by Pressburg and Todd and by Kern.

Kozitsky [11] studied the condensation of mixed vapours on a horizontal tube (O.D. 20 mm and length 1 m) in the following systems: propane–isobutane–*n*-butane, isobutane–*n*-butane and ethane–propane–isobutane–*n*-butane. The experiments were carried out under the conditions of total film condensation of vapours and at the saturation pressures corresponding to the temperatures of *ca* 25–30 degC.

For the calculation of heat-transfer coefficient the relationship

$$\alpha = \frac{q}{\left(\dfrac{q}{0{\cdot}725}\right)^{4/3}\left(\dfrac{\eta_c d}{\rho_c^2 r g \lambda_c^3}\right)^{1/3} + \Delta t_v^*} \tag{3}$$

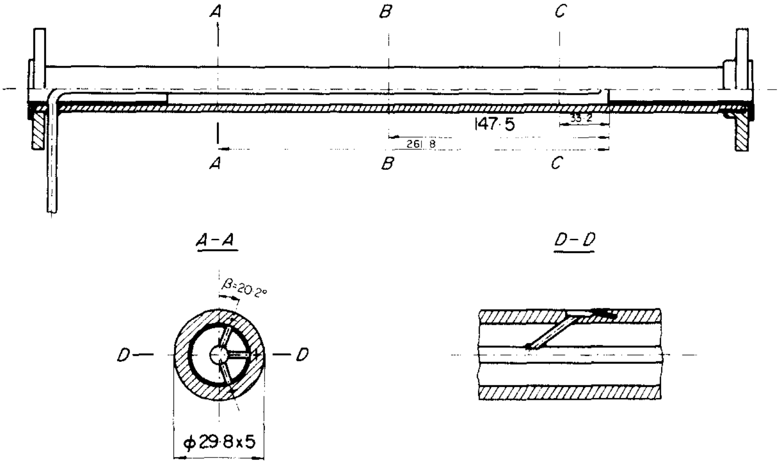
was suggested.

The heat-transfer coefficient in equation (3) was calculated using the total temperature drop $\Delta t_m = t_v - t_s$.

The quantity Δt_v^* denotes the difference between the saturation temperature of vapour and the boiling temperature of liquid having the composition equal to that of condensate. The values of heat-transfer coefficients calculated from equation (3) proved to be lower by 6 per cent, on the average, from the experimental values, the deviations being larger for smaller temperature differences. The maximal error amounted to 27 per cent.

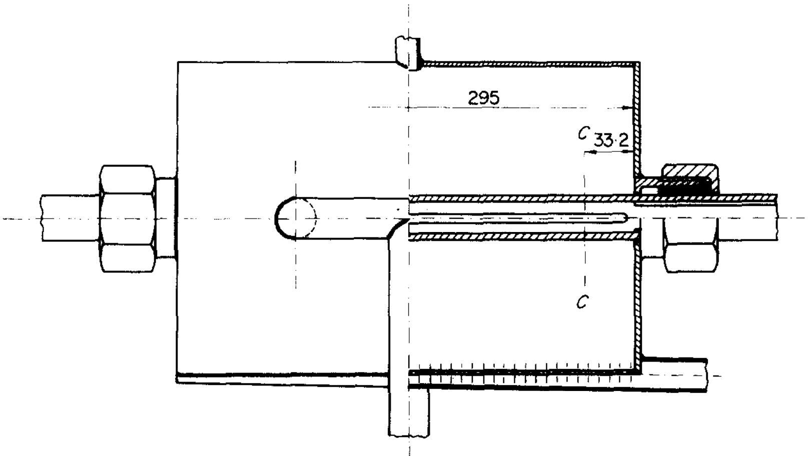
As the literature survey shows, the systematical errors and divergencies of the results obtained from the relationship worked out by individual authors enhance with the decrease of the temperature driving force. These divergencies decrease as the heat-transfer process becomes to be controlled by the thermal resistance of the condensate film.

The present work deals with the experimental determination of heat-transfer coefficients at the total condensation of mixed vapours of miscible liquids on a horizontal tube. The experiments were carried out at the pressure in the condenser equal to the atmospheric pressure. The literature does not quote any work corresponding strictly to the above-defined condensation conditions.



Sections *A, B, C* indicate the positions of thermojunctions

FIG. 1. Condenser.



Line *C–C* indicates the positions of thermojunctions

FIG. 2. Experimental tube.

EXPERIMENTAL APPARATUS

The principal part of the experimental equipment was a condenser in the form of a thick-walled copper tube 29·8/20 mm dia. introduced axially into a cylinder of the length 295 mm and 200 mm dia. (Fig. 1). The temperature of the tube surface was measured by means of nine thermocouples so situated as to enable, on the grounds of performed measurements, to average the wall temperature by means of the three-point Gauss method. The experimental tube and the arrangement of thermocouples in the wall are shown in Fig. 2. The experimental solution assumed enables to maintain the homogeneity of the material in the vicinity of the given experimental point, which may be essential, particularly at high heat loads accompanying small temperature drops.

From the inside the experimental tube was provided with textolite inserts, bounding the measuring section of the length 295 mm, corresponding to the inner length of the shell. The condenser was fed with vapour in four places, symmetrically situated on both sides on the height of tube axis.

In the zone of vapour inflow to the condenser the terminal of the thermocouple for the measurement of vapour temperature was placed. This thermocouple did not contact anywhere, the condenser shell being mounted in a rubber ring. A portion of the thermocouple well, situated outside the condenser shell, was isolated. In the centre, in the highest point of the condenser, the conduit for the removal of inert gases was welded.

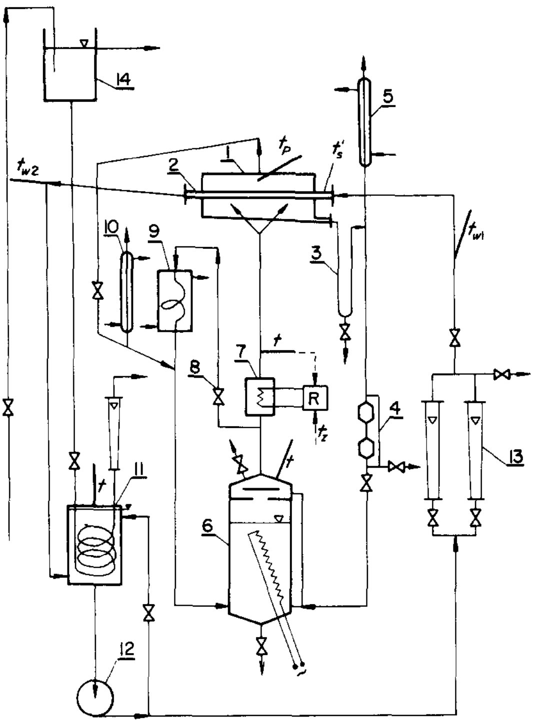


FIG. 3. Scheme of the experimental equipment: 1—condenser; 2—experimental tube; 3—siphon overflow; 4—measuring cell; 5—cooler cutting off the connection with atmosphere; 6—reboiler; 7—superheater; 8—gate valve; 9—auxiliary condenser; 10—cooler cutting off the connection with atmosphere; 11—tank with a cooling coil in the closed circulation of cooling water; 12—pump; 13—rotameters for the measurement of the rate of flow of cooling water in the closed circulation; 14—tank with overflow in the open circulation of cooling water; R—temperature controller; t_z—desired temperature.

The scheme of the experimental equipment is shown in Fig. 3. The condensate, formed in the result of the condensation of vapour on the tube, flowed down to the collector through the openings bored in the condenser bottom. Next, through a siphon overflow (3) it passed into a calibrated cell (4), connected with atmosphere through a cooler (5), and then it flowed down to the boiler (6). The latter was heated electrically by means of heaters situated in twelve vertical heating tubes of 27/21 mm dia. and *ca* 500 mm long, welded in the bottom of the apparatus. The heating power could be regulated in the range 0–10 kW.

In the upper part of the reboiler a catch-drop in the form of annular baffles was installed. The vapour formed in the reboiler passed into a superheater (7). The heater was coupled with the temperature controller having the insensitivity range 0·2 degC and maintaining a constant desired vapour temperature at the outlet from the heater, higher by 0·5 degC from the inlet temperature. From the heater the vapour passed into the condenser, thus closing the main vapour circulation in the installation. The excess of vapour formed in the reboiler was directed, from before the heater over a gate valve (8), to an auxiliary condenser (9) and returned to the reboiler in the form of condensate. The circulation of vapour in the system reboiler–auxiliary condenser–reboiler enabled the operation of the installation at heat loads of the reboiler being constant for individual experimental runs, thus stabilizing the pressure and vapour composition. This circulation was connected with atmosphere over a cooler (10), performing a double duty: cutting off the connection with atmosphere for the condensing components and condensation of vapours disengaging from the condenser (1) during the removal of inert gases from the latter. The heat of condensation was taken by water, flowing inside the experimental tube. The cooling water circulated in the system: cooling water tank (11)–pump (12)–rotameters (13)–experimental tube–cooling water tank.

In the cooling water tank a coil cooler, stabilizing the temperature of circulating water, was placed. Heat was absorbed by water, flowing inside the coil and taken from the tank with overflow (14), supplied by the water-pipe network. The whole installation was made of brass, whereas the pump in the cooling water cycle had a brazen body, and a bronze rotor. The installation as a whole, except pump, was insulated with a double layer of asbestos rope, forming an envelope *ca* 15 mm thick, wrapped by a silk insulating tape from the outside. The condenser was additionally insulated by placing it in a box of the size 500 × 500 × 400 mm, made of foamed vinyl polychloride plates, *ca* 50 mm thick. The free spaces in the box were filled with cotton wool. The conduit delivering the vapour to the condenser (1) and that one discharging the inert components were additionally insulated with a layer of felt, *ca* 10 mm thick.

EXPERIMENTS

For the measurements methanol, *n*-propanol and water as well as their binary and ternary mixtures were used. The mutual combinations of the above substances yielded series of mixtures with very differentiated properties.

The temperature of the tube surface was measured by means of thermocouples. For the measurements the thermocouples copper–constantan 2 × 0·1 mm, in insulation made of impregnated glass fibre, were used. The measurements have been carried out by the current method by means of "Norma" galvanometer, model 251 J, class 2.

The whole system of temperature measurement was calibrated according to the indications of platinum

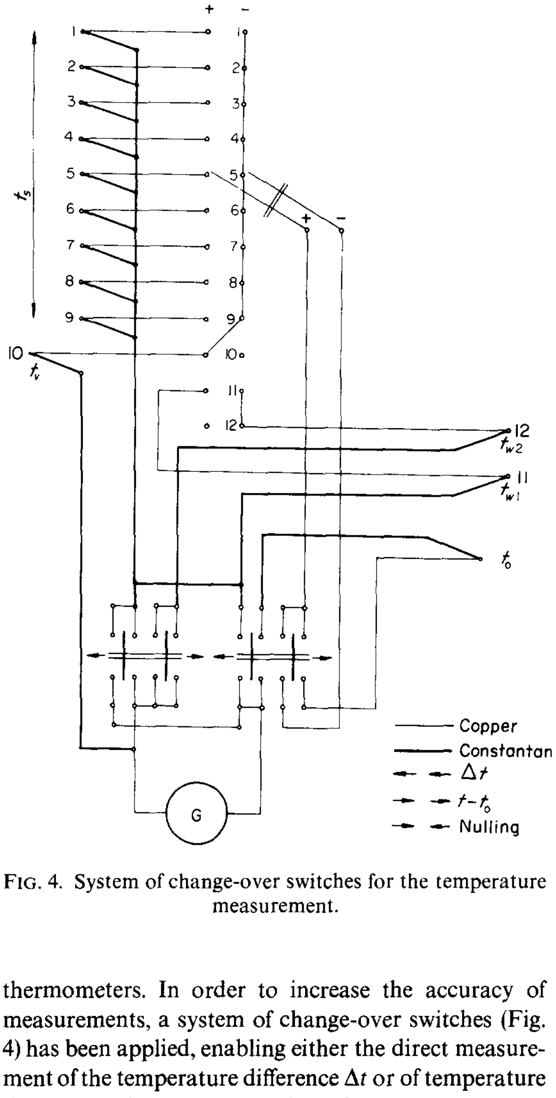


FIG. 4. System of change-over switches for the temperature measurement.

thermometers. In order to increase the accuracy of measurements, a system of change-over switches (Fig. 4) has been applied, enabling either the direct measurement of the temperature difference Δt or of temperature difference with respect to the reference temperature $(t-t_0)$.

The rate of flow of cooling water was measured by two rotameters connected parallelly; it varied in the range 20–1000 kg/h. In the reboiler and in the cooling water tank the laboratory glass thermometers were installed and at the outlet of water from the coil cooler—a rotameter; their indications were approximate and facilitated the control of operation of the installation. The composition of the condensate was determined by the analysis of samples taken after temperature and cooling water flow-rate readings. The composition of *n*-propanol–methanol mixtures was determined refractometrically at constant temperature of 20 degC.

The compositions of remaining binaries were determined densometrically by means of a picnometer with the measurement of solution temperature at the same time. The composition was read from the tables [12] (Vol. III, pp. 115, 119) with linear interpolation of the tabular values with respect to the density and temperature.

The composition of ternary mixtures was determined chromatographically, by comparing the result of the analysis of a sample of unknown composition with the result of the analysis of a standard sample.

The measurements were preceded by a preliminary start-up period of *ca* 2 h. During this period the installation attained thermal equilibrium and the inert components were removed. Before each measurement the condenser was blown through with steam.

ELABORATION OF THE EXPERIMENTAL RESULTS

The tube surface temperature was averaged by means of the three-point Gauss method of numerical integration [13]. The thermocouples were placed in three sections along the tube axis, three thermocouples on the perimeters of each section, which, at the designation of the experimental points shown in Fig. 5, leads to the following formula for the mean temperature drop in the points of attachment of thermojunctions

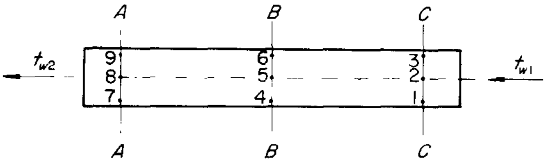


FIG. 5. Designation of the positions of thermocouples on the experimental tube.

$$\Delta t'_m = 0{\cdot}0771605(\Delta t_1+\Delta t_3+\Delta t_7+\Delta t_9) + 0{\cdot}123457(\Delta t_2+\Delta t_4+\Delta t_6+\Delta t_8) + 0{\cdot}197531\,\Delta t_5 \quad (4)$$

where $\Delta t'_m$ is the mean temperature difference without allowance for the correction for the temperature drop in the tube wall and $\Delta t_1, \ldots, \Delta t_9$ are temperature differences measured.

The mean difference between the vapour bulk temperature and mean temperature of tube surface was calculated from the relationship

$$\Delta t_m = \Delta t'_m - (t_s - t'_s) \quad (5)$$

where $(t_s - t'_s)$ is the temperature drop in the wall between the point of fixing of thermojunctions and the tube surface.

For the steady state, after introducing constant values of parameters into the equation for heat conduction in the tube wall, one obtains

$$t_s - t'_s = 0{\cdot}0003150Q, \quad \left(Q \text{ in } \frac{\text{kcal}}{\text{h}}\right). \quad (6)$$

The amount of heat transferred was calculated from the balance of cooling water circulation

$$Q = G_w c_w \Delta t_w. \tag{7}$$

Heat-transfer coefficient was determined from its defining relationship

$$\alpha = \frac{Q}{F \Delta t_m}. \tag{8}$$

Correlation and results

The experimental data were correlated for two forms of equations, viz.

$$\alpha = C \Delta t^A, \quad \left(\alpha \text{ in } \frac{\text{kcal}}{\text{m}^2\text{h deg}}\right) \tag{9}$$

$$Nu = C C_v^A \tag{10}$$

where

$$C_v = \frac{d^3 \rho_c g r}{\lambda_c \eta_c \Delta t_m}; \quad Nu = \frac{\alpha d}{\lambda_c}.$$

The physical properties appearing in the dimensionless groups as above refer (except the heat of condensation) to the film of liquid condensate and were calculated for the mean film temperature t_f. The latter has been assumed as the arithmetic mean of the mean temperature of the tube surface and that of the bulk of vapour, which after introducing the mean temperature difference Δt_m yields the relationship

$$t_f = t_v - \frac{\Delta t_m}{2}. \tag{11}$$

The heat of condensation was calculated for the vapour bulk temperature. The physical properties of the mixtures were determined on the computational way from the composition of mixture and the properties of pure components, according to the following rules:

$$\frac{1}{\rho_c} = \sum_1^n \frac{u_n}{\rho_{cn}} \tag{12}$$

$$\lambda_c = \sum_1^n u_n \lambda_{cn} \tag{13}$$

$$\log \eta = \sum_1^n x_n \log \eta_{cn} \tag{14}$$

$$r = \sum_1^n u_n r_n. \tag{15}$$

The data referring to the pure components were taken from [12], [14] and [15]. The data were correlated by the method of least-squares for the above-presented forms of correlation functions (9) and (10) by means of a digital computer ZAM-41. The results of correlation according to the relationship

$$\alpha = C \Delta t^A \left(\frac{\text{kcal}}{\text{m}^2\text{h deg}}\right)$$

for the individual experimental series are presented in Table 1.

Table 1. Results of correlation according to the relationship $\alpha = C\Delta t^A \left(\frac{\text{kcal}}{\text{m}^2\text{h deg}}\right)$ for individual experimental series

Composition			C	A	Mean error (%)	Maximal error (%)	Percentage of points with the error exceeding 15%
n-propanol			3532	−0·314	7·84	21	13·2
methanol			4788	−0·282	4·64	17·2	2·2
water			19831	−0·321	7·15	25	8·6
x_m = 0·1682	x_p = 0·8318		2907	−0·266	3·33	17	3·1
x_m = 0·5354	x_p = 0·4646		2259	−0·186	3·55	24·4	3·1
x_m = 0·7603	x_p = 0·2397		2266	−0·140	1·77	5·75	0·0
x_p = 0·781	x_w = 0·219		4094	−0·335	4·01	17·1	2·5
x_p = 0·5482	x_w = 0·4518		5120	−0·311	3·69	17·2	2·6
x_p = 0·3058	x_w = 0·6942		5854	−0·273	6·59	19·2	4·8
x_m = 0·967	x_w = 0·033		6828	−0·405	4·47	17·6	4·7
x_m = 0·883	x_w = 0·117		7625	−0·327	5·34	30·7	3·4
x_m = 0·864	x_w = 0·136		9117	−0·370	4·81	14·8	0·0
x_m = 0·809	x_w = 0·191		6375	−0·201	7·81	22·5	8·3
x_m = 0·2379	x_p = 0·4516	x_w = 0·3105	4400	−0·268	3·28	13·4	0·0
x_m = 0·2332	x_p = 0·3357	x_w = 0·4311	3888	−0·257	1·69	5·63	0·0
x_m = 0·1922	x_p = 0·4392	x_w = 0·3686	3277	−0·211	3·35	11·4	0·0
x_m = 0·4500	x_p = 0·1235	x_w = 0·4265	6007	−0·331	3·91	11·4	0·0
x_m = 0·3860	x_p = 0·1641	x_w = 0·4498	6083	−0·331	1·93	7·57	0·0

m—methanol; *p*—propanol; *w*—water.

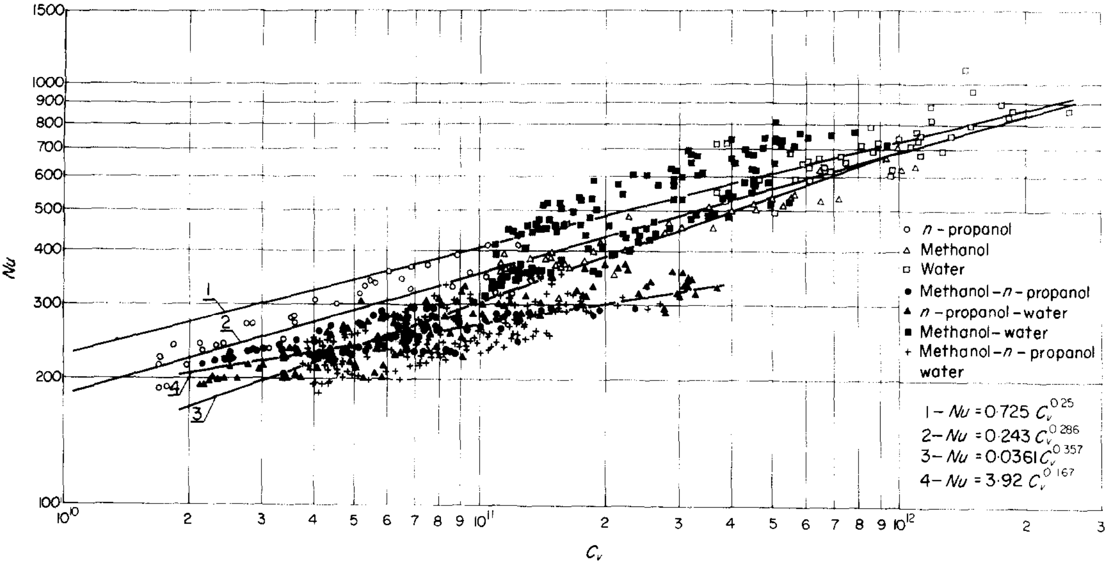


FIG. 6. Correlation $Nu = f(C_v)$ vs experimental points.

Table 2* presents the data for correlation according to equation (10). The correlation of experimental data for pure components, according to equation (10), leads to the relationship:

$$Nu = 0{\cdot}243C_v^{0{\cdot}288}. \tag{16}$$

The mean (arithmetic) error is equal to 8·08 per cent, the maximal error amounts to 29·7 per cent and 90·8 per cent of the points displayed the error lower than 15 per cent. Equation (16) is shown graphically (Fig. 6, line 2) and compared with the line 1, corresponding to the theoretical Nusselt equation $Nu = 0{\cdot}725C_v^{0{\cdot}25}$.

Heat-transfer coefficients for mixtures have been correlated by means of equation

$$Nu = 0{\cdot}0361C_v^{0{\cdot}357}. \tag{17}$$

The mean error is equal to 15·8 per cent, the maximal error amounts to 66 per cent and for 90·9 per cent of points the error does not exceed ±30 per cent.

Equation (17) is represented by line 3 in Fig. 6.

Figure 6 indicates that the points referring to methanol–water mixtures lie pronouncedly higher than remaining experimental points, which may be attributed to a incompletely-filmwise condensation regime.

Such an interpretation may be supported by observations [5, 7, 16] concerning the occurence of dropwise condensation of mixed vapours. In the latter case the increase of the values of heat-transfer coefficients compared to those for film condensation was much smaller, viz. *ca* twofold, than in the case of dropwise respective film condensation of single vapours. Besides, there may exist intermediate states in a rather broad range of temperature differences [7], the pure film condensation occuring only for sufficiently large temperature drops. The above premises induced to disregard the points for methanol–water mixtures at the elaboration of the final correlation, which in the effect led to the following equation:

$$Nu = 3{\cdot}92C_v^{0{\cdot}167}. \tag{18}$$

The relationship (18) is represented in Fig. 6 by line 4 and the scatter of experimental points is shown in Fig. 7.

Figure 8 presents, as an example, the relationship $\alpha = f/\Delta t_m$ for methanol–*n*-propanol mixtures (full lines) and corresponding functions calculated by Pressburg and Todd's method [4] (broken lines).

In the case of *n*-propanol–water mixtures (system with much differing values of heat-transfer coefficients for pure components) the heat-transfer coefficients calculated from Pressburg and Todd's equation assume much higher values, than those obtained experimentally, in the whole range of temperature differences examined.

CONCLUSIONS

1. In the case of total condensation of mixed vapours of miscible liquids the over-all thermal resistance on the side of condensing mixture consists of the condensate resistance as well as of vapour film resistance. In consequence, the inlet conditions of vapour and the

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*Table 2 is available from the authors upon request.

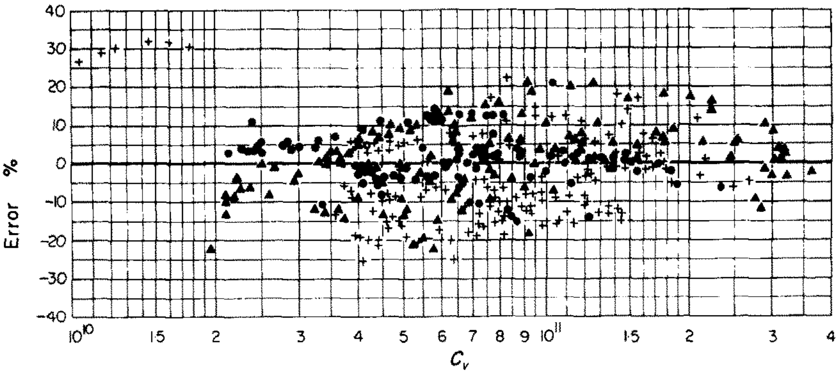

FIG. 7. Scatter of experimental points for mixtures according to the relationship $Nu = 3{\cdot}92C_v^{0{\cdot}167}$.

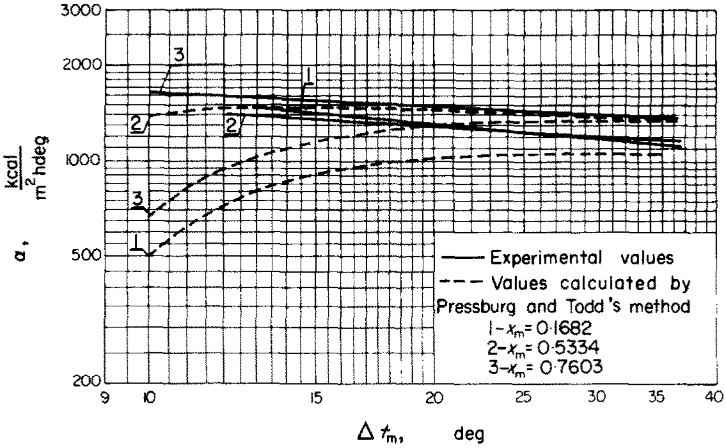

FIG. 8. Comparison of the present results with those obtained from Pressburg and Todd's relationship for methanol–*n*-propanol mixtures.

geometry of the vapour space of the condenser exert an effect upon the values of heat-transfer coefficient at the condensation of mixed vapours. This effect will sink with the rise of the temperature driving force, which may be one of the reasons of the observed increase of the divergencies of results obtained by various authors with diminishing temperature difference.

2. Assuming for calculations the temperature driving force $\Delta t_m = (t_v - t_s)$, the final correlation equation (18):

$$Nu = 3{\cdot}92C_v^{0{\cdot}167}$$

was obtained. This equation was examined in the range

$$\Delta t_m = 5\text{–}45{\cdot}6\,\text{degC};\quad C_v = 1{\cdot}16\times10^{10} - 3{\cdot}68\times10^{11}.$$

The mean error is equal to 8·46 per cent, the maximal error amounts to 33·8 per cent and for 85 per cent of experimental points the error does not exceed 15 per cent.

Equation (18) was correlated on the grounds of experiments performed for binary systems (methanol–*n*-propanol, *n*-propanol–water) and for ternary system: methanol–*n*-propanol–water. The relationship (18) is valid in the case of controlling condensate film resistance in the total thermal resistance on the side of condensing vapour mixture. In the range of small temperature differences this state is subject to change and the vapour-phase resistance begins to influence significantly the total thermal resistance. For this reason the above-presented correlation equation has a sharp restriction imposed upon the lower value of the temperature difference interval.

3. There have been confirmed the observations of Mirkovich and Missen [7] and of Haselden and Platt [8] concerning the deviations, which may exist between the measured heat-transfer coefficients and those calculated from equation given by Pressburg and Todd [4].

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