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DIRECT CONTACT CONDENSATION OF VAPOUR ON A SPRAY OF SUBCOOLED LIQUID DROPLETS

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Abstract—A theoretical study of the direct contact condensation of a pure vapour on a spray of subcooled liquid drops has been carried out. Theoretical considerations include the analysis of drop size distribution, motion of drops, and heat transfer rate. A mathematical model for the process is proposed. As a result, the thermal utilization (or the average spray temperature) for a given length of spray can be obtained. The influence of certain parameters (initial drop velocity, drop size, etc.) on the thermal utilization was analyzed, and the drop size found to be the most important.

The experimental investigation was performed with the steam–water system, using three full cone nozzles at different pressure drops. High values of thermal utilization (from 0.783 to 0.987 for distances from the nozzle of 42–356 mm) were obtained experimentally. The values of the thermal utilization obtained from heat transfer experimental data are compared with those predicted by the mathematical model. Good agreement between the two was obtained.

NOMENCLATURE

A,	cross-sectional area of the spray;
a,	distribution parameter, equation (8);
c_p,	heat capacity at constant pressure [$\mathrm{kJ\,kg^{-1}\,K^{-1}}$];
C_D,	drag coefficient;
D,	drop diameter [m];
D_m,	maximum drop diameter [m];
$Fo = \dfrac{4\alpha\theta}{D_i^2}$,	Fourier number;
g,	acceleration due to gravity [$\mathrm{m\,s^{-2}}$];
h,	enthalpy [$\mathrm{kJ\,kg^{-1}}$];
k,	thermal conductivity [$\mathrm{W\,m^{-1}\,K^{-1}}$];
m,	mass flow rate [$\mathrm{kg\,s^{-1}}$];
N,	number of drops per unit time;
P,	steam pressure [kPa];
ΔP,	pressure drop at the nozzle [kPa];
Q_T,	theoretical heat transfer rate [kW];
Q_z,	heat transfer rate at distance z from the nozzle [kW];
T,	temperature [K];
v,	normalized volume of drops;
V,	volume rate of drops [$\mathrm{m^3\,s^{-1}}$];
w,	velocity [$\mathrm{m\,s^{-1}}$];
y,	defined by equation (8);
z,	distance from the nozzle [m].

Greek letters

$\alpha = \dfrac{k}{\rho_l c_p}$,	thermal diffusivity of liquid [$\mathrm{m^2\,s^{-1}}$];
δ,	distribution parameter, equation (7);
ε,	thermal utilization;
λ,	heat of vaporization [$\mathrm{kJ\,kg^{-1}}$];
ψ,	defined by equation (4);
ρ,	density [$\mathrm{kg\,m^{-3}}$];
θ,	time [s];
$\phi(D) = \dfrac{\mathrm{d}N}{\mathrm{d}D}$,	where N is the number of drops.

Subscripts

d,	drop;
f,	liquid at saturation line;
i,	initial;
l,	liquid;
s,	saturation;
v,	vapour.

INTRODUCTION

THE PROCESS of vapour condensation occurs in many types of industrial equipment. In most heat exchangers vapour is condensed on a cold surface. Recently, more attention has been paid to condensation in direct contact of vapour and subcooled liquid. Great interest has been shown in the application of direct contact condensation in steam power plants [1–6], and several types of direct contact condensers have been developed [1, 7, 8]. Direct contact condensation has been a subject of theoretical investigations, as well. The approximate solutions for condensation of vapour on liquid cylindrical and flat jets, and on liquid sheets have been suggested.

Although direct contact heaters featuring condensation of vapour on a spray of subcooled liquid drops have been used, very few theoretical investigations of the process have been published. Brown [9, 10] in his analysis of the process used mean diameters (Sauter mean diameter for the heat transfer rate and arithmetic

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mean diameter for the trajectory of drops), instead of the drop size distribution which actually appears in the spray. In his experimental investigation of condensation of steam on a spray of subcooled water drops (of dia. 125–520 μm), he obtained values of the heat transfer coefficient up to 27 200 W m^{-2} K^{-1}, indicating the high efficiency of this type of condensation. In the analyses of Isachenko *et al.* [11–14], the volume mean diameter and a one parameter drop size distribution function were used. Comparing the theoretical and experimental results, they have concluded that "the observed rate of the process is higher than that theoretically predicted" [14].

In this work, a theoretical model for condensation of vapour on a spray of liquid drops is developed, and the experimental results of steam condensing on a spray of subcooled water drops are compared with this model.

THEORETICAL CONSIDERATIONS

The process of condensation of vapour on a spray of drops is very complex. In order to describe it, drop size distribution, motion of drops and condensation on drops should be analyzed.

A jet coming out of the nozzle disintegrates and breaks up into drops of various sizes. If saturated vapour is brought into contact with subcooled drops (their temperature being lower than the saturation temperature of the vapour at the given pressure), and if the size of drops is larger than the critical value, condensation of vapour on the drops will occur. Droplets smaller than the critical size tend to evaporate, rather than grow by condensation. Assuming that the drops are spheres, and ignoring variations in liquid density, the quantity of condensate deposited per unit time in a given volume of vapour–liquid mixture on drops of dimensions between $D-(\mathrm{d}D/2)$ and $D+(\mathrm{d}D/2)$ is given by the following equation [15]:

$$\mathrm{d}m_v = \frac{\pi}{2} D^2 \rho_l \frac{\mathrm{d}D}{\mathrm{d}\theta} \phi(D)\mathrm{d}D \tag{1}$$

where D is the drop diameter, ρ_l is the liquid density, θ is time and $\phi(D)$ is the drop size distribution function.

In order to solve equation (1) it is necessary to know the rate of growth of drops $\mathrm{d}D/\mathrm{d}\theta$ and the distribution function.

A review of works on the rate of growth of single drops during condensation was given by Lekic [16]. The theoretical and experimental investigations performed by Ford and Lekic [17] showed good agreement between the two. Their expression for the rate of droplet growth was

$$\frac{\mathrm{d}D}{\mathrm{d}\theta} = 2\pi^2 \frac{\alpha\psi}{D_i} \frac{\exp(-\pi^2 Fo)}{[1-\exp(-\pi^2 Fo)]^{1/2}}, \tag{2}$$

and for the change of drop diameter during condensation

$$D = D_i\{1+\psi[1-\exp(-\pi^2 Fo)]^{1/2}\} \tag{3}$$

where

$$\psi = \left[1+\frac{c_p(T_s-T_i)}{\lambda}\right]^{1/3} - 1 \tag{4}$$

and

$$Fo = \frac{4\alpha\theta}{D_i^2}, \tag{5}$$

D_i is the initial drop diameter.

It follows from equation (1) that the rate of growth of each drop (or group of drops) should be accounted for. In order to find this, the time of contact of vapour and liquid drop should be known. It varies considerably with the size of the drop and may be determined from the motion of the drop.

DROP TRAJECTORY

Although the spray of drops emerges from a nozzle, the motion of each drop is analyzed separately. In further analysis, the following assumptions are made:

There is no influence of drops on each other.

The effect of drops on vapour velocity can be neglected.

Neither coalescence nor shattering significantly influence the motion of drops and the heat transfer rate.

The density of drops does not change considerably.

The first two assumptions are justified by the fact that in most heat exchangers of this type, the void fraction is very close to unity.

The coalescence of drops may have two opposite effects on heat transfer rate. By coalescence, drops become larger and this causes a decrease in heat transfer rate. On the other hand, it can be expected that during the impact, mixing takes place, and therefore the temperature profile inside the newly formed drop is more uniform, which increases the heat transfer rate. Isachenko *et al.* [13] found no significant influence of coalescence on heat transfer rate.

Basically, the motion of a drop in a spray emerging from a nozzle is two dimensional. Kravchenko and Mukoed [18] have shown that horizontal motion of a drop is not significantly influenced by vapour velocity. Therefore, and in order to determine the height of heat exchanger, the motion of drop in the vertical direction only is analyzed.

For a drop of spherical shape, the equation of motion in the case of changing mass [19] and for counter current flow of vapour and drop is

$$\frac{D^3\pi}{6}\rho_l\frac{\mathrm{d}w_d}{\mathrm{d}\theta} = \frac{D^3\pi}{6}\rho_l g - \frac{D^3\pi}{6}\rho_v g - C_D\frac{D^2\pi}{4} \times \rho_v\frac{(w_d+w_v)^2}{2} - \frac{\mathrm{d}\left(\frac{D^3\pi}{6}\rho_l\right)}{\mathrm{d}\theta}(w_d+w_v) \tag{6}$$

where the first term on the right represents gravitational force, the second buoyancy force, the third drag force, and the last one is due to the mass change of the drop; w_d and w_v are drop velocity and vapour velocity, respectively, ρ_l and ρ_v are liquid density and vapour density, respectively, C_D is the drag coefficient. Equation (6) may be written in the following form,

$$\frac{\mathrm{d}w_d}{\mathrm{d}\theta} = \frac{\rho_l - \rho_v}{\rho_l} g - \frac{3}{4} C_D \frac{1}{D} \frac{\rho_v}{\rho_l} \times (w_d + w_v)^2 - \frac{3}{D} \frac{\mathrm{d}D}{\mathrm{d}\theta} (w_d + w_v). \tag{6a}$$

The diameter of the drop D, and the rate of growth $\mathrm{d}D/\mathrm{d}\theta$ are determined by equations (3) and (2). The expressions for the drag coefficient are chosen according to the analysis performed in [20], and can be found there. To solve equation (6a), the initial velocity of the drop is required. It is assumed that all drops have the velocity of the jet at the outlet of the nozzle. The vapour velocity is discussed later.

DROP SIZE DISTRIBUTION

It has been mentioned earlier that the jet breaks up into drops of various sizes. It is well recognized that drop size distribution has to be statistical in nature. Several equations for the presentation of drop size experimental data have been suggested in literature. An analysis performed by Lekic *et al.* [21] has shown that the drop size experimental data are best fitted by the chi-square (χ^2) distribution [22] and the upper-limit function [23] when the distribution parameters are determined as the least squares (i.e. best) estimates. Both equations are nonlinear with respect to the parameters. To find the best estimates, the method of linearization of Draper and Smith [24] and the method of Marquardt [25] were used. In further analysis, the upper-limit function is used because it has a maximum drop size (this approximates the real situation), and determination of the best estimates of the parameters for this function is easier.

From the upper-limit function [23], the normalized volume distribution is

$$\frac{\mathrm{d}v}{\mathrm{d}D} = \frac{\delta}{\sqrt{\pi}} \frac{D_m}{D(D_m - D)} \exp(-\delta^2 y^2) \tag{7}$$

where

$$y = \ln \frac{aD}{D_m - D}, \tag{8}$$

a, δ and D_m are distribution parameters. D_m represents the maximum drop diameter, a is a skewness parameter, and the parameter δ is a measure of uniformity of the spray.

HEAT TRANSFER RATE

The processes of disintegration of a jet, atomization and establishment of the drop size distribution are completed at a certain distance from the nozzle, depending on the type of nozzle, pressure drop, etc. It will be assumed that atomization takes place at the nozzle outlet, and that the drops behave as rigid spheres, as has been shown previously [17]. Then the volume of drops with diameters between $D-(\mathrm{d}D/2)$ and $D+(\mathrm{d}D/2)$ formed at the nozzle outlet per unit time is

$$\mathrm{d}V = \frac{m}{\rho_l} \frac{\delta}{\sqrt{\pi}} \frac{D_m}{D_i(D_m - D_i)} \exp(-\delta^2 y^2) \mathrm{d}D_i \tag{9}$$

where

$$y = \ln \frac{aD_i}{D_m - D_i}, \tag{8a}$$

m is the liquid mass flow rate through the nozzle. (The diameters of the drops change as a result of condensation, and the subscript i stands for the initial drop diameter.) The volume and the number of drops are related, and therefore, the number of drops with diameters between $D-(\mathrm{d}D/2)$ and $D+(\mathrm{d}D/2)$ per unit time is

$$\mathrm{d}N = \frac{m}{\rho_l} \frac{\delta}{\sqrt{\pi}} \frac{D_m}{D_i(D_m - D_i)} \frac{6}{\pi D_i^3} \exp(-\delta^2 y^2) \mathrm{d}D_i \tag{10}$$

The quantity of heat absorbed by the spray per unit time from nozzle outlet to distance z (see Fig. 1), can be calculated from the following equation:

$$Q_z = \frac{\pi}{6} \rho_l \lambda \int_0^{D_m} (D^3 - D_i^3) \mathrm{d}N. \tag{11}$$

The time required for each drop to reach plane z is calculated separately, and the corresponding instantaneous drop diameter D (determined by equation (3)) is then used in equation (11).

To calculate the instantaneous drop diameter, the time for the drop to reach the distance z is required, and it can be determined by equation (6a). Equations (11), (10), (6a), (3) and (2) have to be solved numerically step by step, starting from the nozzle outlet. The time required for each drop to travel distance $\mathrm{d}z$ is:

$$\mathrm{d}\theta = \frac{\mathrm{d}z}{\frac{1}{2}[(w_d)_z + (w_d)_{z+\mathrm{d}z}]}. \tag{12}$$

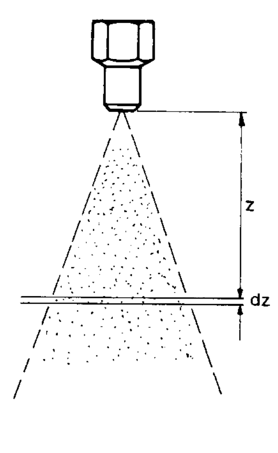

FIG. 1. Model of spray.

As the velocity $(w_d)_{z+dz}$ is not known in advance, equation (12) is solved by iteration.

The exit vapour velocity is assumed to be zero. (This assumption is fulfilled if all vapour entering the heat exchanger is condensed.) Then, the vapour velocity at any distance z is

$$w_v = \frac{Q_z}{\lambda \rho_v A} \tag{13}$$

where A is the cross-sectional area of the spray.

The total amount of heat which can be theoretically absorbed by liquid per unit time is

$$Q_T = m(h_f - h_i) \tag{14}$$

where h_i is the enthalpy of liquid at inlet and h_f that of condensate.

The thermal utilization can be defined as the ratio

$$\varepsilon = \frac{Q_z}{Q_T}. \tag{15}$$

The height of heat exchanger required to accomplish a certain thermal utilization can be determined using the above mentioned equations. (A similar approach was used by Hollands [26] for spray cooling towers, and by Katta and Gauvin [27] for spray dryers.) A computer program based on the preceding equations was developed. In order to investigate the influence of certain parameters (initial drop velocity, drop size, and liquid and vapour properties), a few cases with uniform drop size distribution were examined. (The calculations were performed for the steam–water system.) Comparing the results obtained, it was concluded that the influence of drop diameter is the most important, as shown in Fig. 2.

EXPERIMENTAL APPARATUS

The experimental investigation of direct contact condensation of steam on a spray of subcooled water drops was performed in order to examine the validity of the proposed theoretical model. The investigation essentially consisted of two parts. In the first stage the drop size distribution was measured. The splitter (S) was used to isolate a narrow strip of spray for photographic purposes, in determining the drop size distribution. Then the measurements of heat transfer rates were performed under essentially identical conditions, with the splitter removed.

An experimental apparatus (Fig. 3) was built for this purpose. The cylindrical condensing chamber was 305 mm dia. by 850 mm high. Subcooled water was supplied to the nozzle through tubing welded to the top flange of the apparatus. Three different nozzles were used: 0.64 cm (1/4 G 6.5) and 1.27 cm (1/2 G 16) nozzles of Spraying Systems Company, and a 0.95 cm (3/8 BXF093) nozzle of Delavan–Watson Limited. All nozzles were full cone type.

The water and steam flow rates were regulated by valves on the supply lines. The liquid level in the apparatus was kept constant by the valve on the condensate discharge line. In this way, steady-state flow of condensate was ensured, and the height of the spray was constant. To measure the heat transfer rates at different distances from the nozzle, three extension tubes of different lengths were made. These tubes were placed between the top flange of the apparatus and the nozzle. In order to reduce the heat transfer between steam and water flowing through the extension tubes, the tubes were insulated with a 50 mm thick layer of glass wool, and a 5 mm thick asbestos tube.

To determine the heat transfer rates, the temperatures of water and steam at the inlet, and condensate at the outlet, were measured. The temperatures were measured by iron–constantan thermocouples calibrated using an NBS certified mercury thermometer. The water flow rate was measured by calibrated rotameters. The difference between the steam pressure in the apparatus and atmospheric pressure was mea-

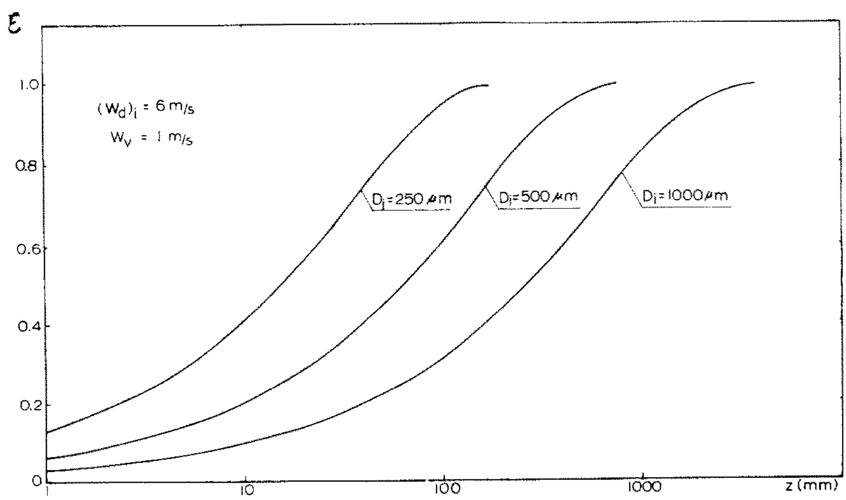

FIG. 2. Influence of drop size on thermal utilization.

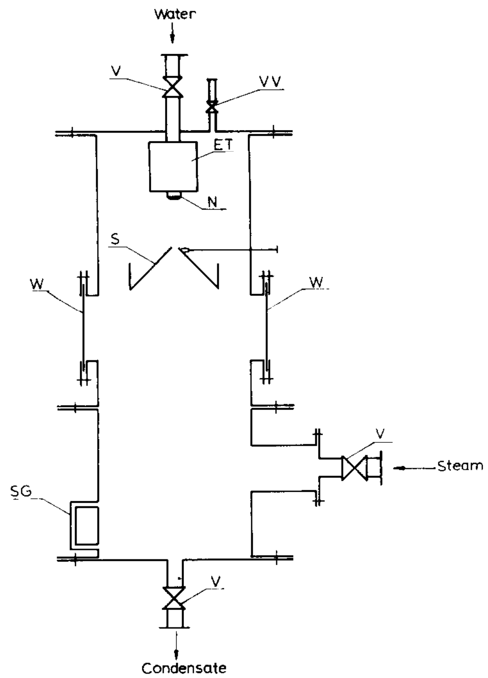

FIG. 3. Schematic diagram of apparatus: ET—extension tube; N—nozzle; S—splitter; SG—sight glass; V—valve; VV—venting valve; W—window.

sured by mercury differential manometer. The pressure drop through the nozzle was measured by a pressure gauge. The method of drop size distribution measurement and the results obtained are given elsewhere [20].

EXPERIMENTAL RESULTS AND DISCUSSION

As mentioned earlier, the experimental investigation was performed with three different nozzles. The water flow rate and drop size distribution depend on the pressure drop through the nozzle, thus this was included as an experimental variable. The pressure drop through the nozzle was chosen to be in the range 30–80 kPa. The initial drop velocities were determined from the water flow rate, and ranged from 6 to 10 m s^{-1}. Saturated steam (at two different pressures, $\sim$99 and $\sim$138 kPa) was used in all investigations. The inlet water temperature was in the range 10–15°C. (It was concluded earlier [16] that the influence of water inlet temperature on heat transfer rate was not significant.) The maximum drop diameter ranged from 800 to 1100 μm, depending on the nozzle, pressure drop through the nozzle and steam pressure. (The Sauter mean diameter varied from 433 to 585 μm, and the volume mean diameter from 353 to 453 μm.)

The water (spray + condensate) temperature leaving the condensing chamber was measured at distances 42–356 mm from the nozzle. Values of thermal utilization [defined by equation (15)] obtained experimentally, ranged from 0.783 to 0.987. This indicates high heat transfer rates even at relatively short distances from the nozzle.

The comparison of experimental results with those obtained by the proposed model was performed in two ways. For the first comparison, the normalized volume distribution obtained from the experimental data was used. The thermal utilization as a function of the distance from the nozzle obtained by the model, and values obtained from the heat transfer measurements, are shown in Fig. 4. It can be seen from the figures that the values obtained by the model and by the experiment are very close for the 0.95 and 1.27 cm nozzles. For the 0.64 cm nozzle, the agreement between model and experiment is satisfactory for higher values of thermal utilization (which are of the primary interest). The differences appearing at lower values might have been due to small drop velocities being lower than those predicted in the model. The assumption that all drops had the outlet spray velocity could cause higher values of the thermal utilization experimentally than that predicted by theory. On the other hand, at high values of the thermal utilization, incorrect choice of initial velocity of small drops does not materially affect the result.

A further comparison between the model and experiment has been performed for a few cases, using the drop size distribution given by equation (7), with the best estimates of the parameters, instead of the experimental values. No significant difference in results have been found at higher values of the thermal utilization. Therefore, this comparison shows that equation (7) (when available), can be used in the model instead of experimental data on the drop size distribution.

CONCLUSIONS

A mathematical model for condensation on a spray of drops has been developed. Droplet diameter was found to be the most important parameter, affecting thermal utilization. The experimental investigation of steam condensing on a spray of subcooled water drops

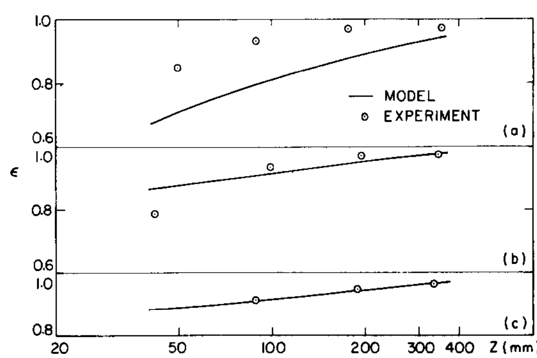

FIG. 4. Thermal utilization as a function of distance for various nozzles. P = 99.0 k Pa. (a) 0.64 cm nozzle, ΔP = 34.0 kPa, (b) 0.95 cm nozzle, ΔP = 30.6 kPa, and (c) 1.27 cm nozzle, ΔP = 34.0 kPa.

has been performed. Comparison of the values of thermal utilization obtained experimentally (from 0.783 to 0.987 for distances from the nozzle of 42–356 mm), and those predicted by the model showed good agreement. The values of thermal utilization obtained indicate that this is a highly efficient way of condensing pure steam.

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CONDENSATION PAR CONTACT DIRECT D'UNE VAPEUR ET D'UN BROUILLARD DE LIQUIDE SOUS-REFROIDI

Résumé—On étudie théoriquement la condensation par contact direct d'une vapeur pure avec un brouillard de gouttes liquides sous-refroidies. L'analyse tient compte de la distribution de la taille des gouttes, du mouvement des gouttes et du transfert thermique. Un modèle mathématique est proposé. On peut obtenir de ce modèle l'utilisation thermique (ou la température moyenne du brouillard) pour une longueur donnée. L'influence de certains paramètres (vitesse initiale des gouttes, dimension des gouttes, etc...) sur l'utilisation thermique est analysée et la taille de la goutte est le paramètre le plus important.

L'étude expérimentale concerne le système eau-vapeur d'eau, avec trois tuyères coniques et différentes pressions. On obtient des valeurs élevées de l'utilisation thermique (de 0,783 à 0,987 pour des distances en aval de la tuyère entre 42 mm et 356 mm). Les valeurs obtenues pour l'utilisation thermique à partir des essais sont comparés avec celles calculées à partir du modèle. On obtient un accord satisfaisant.

DAMPFKONDENSATION DURCH DIREKTEN KONTAKT MIT VERSPRÜHTEN UNTERKÜHLTEN FLÜSSIGKEITSTROPFEN

Zusammenfassung—Eine theoretische Untersuchung der Kondensation eines reinen Dampfes durch direkten Kontakt mit versprühten unterkühlten Flüssigkeitstropfen wurde durchgeführt. Die theoretischen Betrachtungen schließen eine Analyse der Tropfengrößenverteilung, der Tropfenbewegung und des Wärmeübergangs ein. Für den Prozeß wird ein mathematisches Modell entwickelt. Es ist möglich, für eine gegebene Sprühlänge den thermischen Nutzfaktor oder die durchschnittliche Spraytemperatur zu ermitteln. Der Einfluß bestimmter Parameter (anfängliche Tropfengeschwindigkeit, Tropfengröße, etc.) auf den thermischen Nutzfaktor wurde untersucht, wobei sich die Tropfengröße als sehr wichtig erwies.

Die experimentelle Untersuchung wurde an einem Dampf-Wasser-System durchgeführt; dabei wurden drei Vollkegel-Düsen bei verschiedenen Druckverhältnissen verwendet. Beim Versuch ergaben sich für den thermischen Nutzfaktor hohe Werte (0,783 bis 0,987 bei einer Entfernung von der Düse zwischen 42 und 356 mm). Die Werte für den thermischen Nutzfaktor, die durch Wärmeübergangsmessungen gewonnen worden sind, werden mit denen verglichen, die mit Hilfe des mathematischen Modells ermittelt wurden. Hierbei wurde gute Übereinstimmung erzielt.

КОНТАКТНАЯ КОНДЕНСАЦИЯ ПАРА НА ФАКЕЛЕ РАСПЫЛА КАПЕЛЬ НЕДОГРЕТОЙ ЖИДКОСТИ

Аннотация — Теоретически исследовалась контактная конденсация чистого пара на факеле распыла капель недогретой жидкости. Проведен анализ распределения капель по размерам, движения капель и интенсивности теплообмена. Предложена математическая модель процесса, позволяющая определять величину регенерации тепла или среднюю температуру распыла при заданной длине факела. Исследовалось влияние ряда параметров (начальной скорости капель, размера капель и т. п.) на величину регенерации тепла и найдено, что важнейшим параметром является размер капель.

Эксперименты проводились с системой пар–вода при истечении из трех конических сопел с различными перепадами давлений. Получены высокие экспериментальные значения величин регенерации тепла (от 0,783 до 0,987 при расстояниях от сопла 42–356 мм). Проведено сравнение экспериментальных данных с результатами, рассчитанными с помощью математической модели, и получено хорошее соответствие.