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Film boiling characteristics of liquid nitrogen sprays on a heated plate

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Abstract—The rates of heat transfer between a heated plate and a spray of saturated liquid nitrogen are measured at temperature differences and spray pressures ranging from 250 to 450 K and 2 to 12 bar, respectively. The analysis of heat transfer data show that the mean droplet size within a spray, the mean temperature difference and the local mass velocity of the spray are the most significant process variables controlling heat transfer at the plate surface. The effectiveness of a spray based on the thermal utilization of the liquid and fractional heat transfer area of the sprays are determined for various spray conditions investigated. A correlating equation based on dimensionless groups is proposed. As a result, it is possible to determine the effects of spray mean velocity, mean spray size and mean temperature difference on the average heat transfer coefficient between the sprays and the plate. These effects are considered in comparison with those previously reported in the literature and good agreement is obtained.

INTRODUCTION

LIQUID nitrogen is widely used for various industrial processes, including cryogenic recycling of waste materials, since it can embrittle certain metals, ceramics and polymers [1]. It is also commonly used for spray cooling of hot metals [2], rapid freezing of foods by direct contact with cryogens [3–6] and recently for the preservation of cells and tissues of human bodies shortly after clinical and legal death [7].

The use of cryogenic sprays for holding the temperature of frozen foods at about $-18°C$ or lower in refrigerated trucks for transportation has gained wide acceptance especially for long distance journeys. This is probably due to the absence of moving parts and therefore, low maintenance requirements as opposed to mechanically refrigerated trucks which are prone to failures resulting from damage caused by vibrations, fracture of pipe joints with a consequent loss of refrigerant.

The low temperature characteristics of liquid nitrogen makes it particularly suitable for rapid cooling in various industrial processes. Its boiling point at atmospheric pressure is $-196°C$ and it therefore offers a high driving force effect, ΔT, for rapid cooling rates. Since most materials generally consist of multi-component materials, liquid nitrogen and other cryogenic fluids can be used effectively in the separation of the constituents.

In the recycling of waste materials such as automobile tyres and the recovery of copper from alternators, starter motors, electric motors and solenoid coils, the materials are rapidly cooled or frozen below their brittle point before a force is applied to break the materials into small fragments. Therefore, the recovery of the desired materials for example, copper from the fragments could simply be by magnetizing the steel components away from the copper particles.

Atomization is a process used to enhance the rates of heat transfer between the materials and the cryogenic liquid. The process usually involves such stages as the hydraulics of flow in the atomizer, formation of a liquid sheet and the rapid disintegration of the sheet into small droplets.

The velocities and droplet sizes of the spray vary depending on the liquid properties, the design of the nozzle, the spray pressure used and the conditions of the fluid into which the liquid is discharged. Therefore, the rates of deceleration of the droplets as they fall from the nozzle to the solid surface depend on the drag forces of the surrounding atmosphere.

Since the sprays are ejected into a high temperature environment in comparison with the saturation temperature of the liquid, the environmental and solid surface temperatures will also influence the rates of heat and mass transfer occurring simultaneously during evaporation and their mechanisms are complex.

As seen from most published reports, the size distribution of sprays and droplet velocities are major factors controlling heat and mass transfer during evaporation [8–10] of the atomized liquid. The mechanisms associated with these factors are complex and therefore responsible for the difficulties in their measurement.

The temperature difference between the liquid and the solid being cooled is also a significant heat transfer factor. However, the temperatures used for the conditions described in this report are sufficiently high for film boiling conditions to occur where a film of vapour between the solid surface and the liquid influences the rates of the transfer processes.

The need for this study arises because most pub-

NOMENCLATURE

A	area [m^2]
C_p	specific heat at constant pressure [J kg^{-1} K^{-1}]
C_s	vapour concentration at the drop surface [kg mol m^{-3}]
C_∞	vapour concentration at mean box temperature [kg mol m^{-3}]
d	diameter [m]
d_{32}	mean diameter (Sauter) [m]
D_c	vapour diffusivity [m^2 s^{-1}]
E	electrical power input [W]
h	average heat transfer coefficient [W m^{-2} K^{-1}]
I	current flow [A]
K_c	mass transfer coefficient [m s^{-1}]
K	thermal conductivity [W m^{-1} K^{-1}]
m	mass flow rate [kg s^{-1}]
$\overset{*}{m}$	mass velocity of spray per unit area [kg m^{-2} h^{-1}]
M	molecular weight of nitrogen vapour [kg kg^{-1} mol^{-1}]
n	number of drop size classes
Nu	Nusselt number
P	pressure [bar] (1 bar = 10^5 N m^{-2})
ΔP	pressure drop at the nozzle [bar]
Pr	Prandtl number
q	heat transfer rate [W]
r	drop radius [m]
R	nitrogen gas constant [J kg^{-1} K^{-1}]
Re	Reynolds number
R_c	resistance at 0°C [Ω]
R_p	resistance at plate temperature [Ω]
R_0	characteristics gas constant [J kg^{-1} K^{-1}]
Sc	Schmidt number
Sh	Sherwood number
T	temperature [°C]
ΔT	mean temperature difference [K]
V	velocity [m s^{-1}]
We	Weber number.

Greek symbols

β	$0.75Re^{1/2}\,Pr^{1/3}$
ε_{ss}	fraction of active heat transfer area of a droplet
ε_T	thermal utilization of a spray
λ	latent heat of vaporization [J kg^{-1}]
μ	viscosity [kg m^{-1} s^{-1}]
ρ	density [kg m^{-3}]
σ	surface tension [N m^{-1}]
Σ	algebraic sum
Ψ	Stefan–Boltzmann constant.

Subscripts

a	air
i	drop size class i
p	plate
s	saturation
v	vapour.

lished work on evaporation from sprays are on the use of water as the fluid apart from that reported by Bonacina *et al.* [11] on evaporation from cryogenic sprays. Their analysis was based on the results of single droplets [12–16] extended to sprays containing uniform initial droplet sizes.

In this work, an attempt has been made to measure the rates of heat transfer between a heated plate and a spray of liquid nitrogen at different spray conditions. In addition, the parameters controlling the rates of heat transfer are investigated and use has been made of the results of Mugele [17] on the size distribution of sprays.

EXPERIMENTAL EQUIPMENT AND PROCEDURE

The equipment used can be divided into three separate units, namely, (i) an insulated box, (ii) the liquid nitrogen spray system comprising of a 100 l Polarstream PS Dewar storage vessel with a solenoid valve, a hand valve and a temperature controller/thermostat on the liquid supply pipe and (iii) a platinum plate.

The insulated box and the liquid nitrogen spray system are the same as those used in ref. [18] and will not be described in this report.

The test plate consists of three platinum plates, each measuring 10 cm long by 4 cm wide and the three plates are placed side by side to form a single test plate measuring 12 cm long by 10 cm wide by 0.03 cm thick. This plate thickness is thin enough to achieve an isothermal conduction at an instant of time [19] and a steady plate temperature. The plate is supported by strips of mica to limit the area of contact and hence reduce heat losses from the bottom of the plate to the surroundings. At a plate temperature of between 200 and 300°C, it was necessary to support the platinum plate with bricks made from clay to minimize heat losses. The methods of support in both cases are the same as those used by Batch [20].

The a.c. supply was taken from a 450 kV A transformer and was stepped down using a current transformer (ratio 1 : 150). An ammeter was connected to the current transformer to measure the current flowing through the circuit. The voltage across the plate was determined by an additional circuit with a small portion to balance with the actual voltage drop across the test portion of the plate.

A wattmeter (type G.E. serial 1293291) connected to the current transformer was used for measuring the power supplied to the test section of the plate.

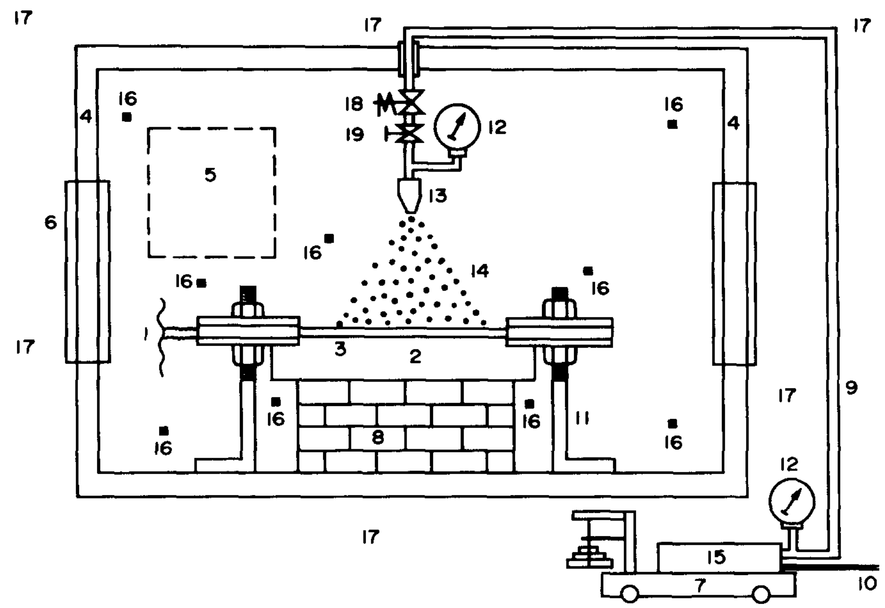

FIG. 1. Schematic diagram of experimental equipment: 1, power lead; 2, mica support; 3, platinum plate; 4, expanded polyurethane insulant; 5, venting flap; 6, viewing window; 7, platform weighing scale; 8, brick (clay) support; 9, insulated nitrogen feed pipe; 10, pressure relief line; 11, plate stand and clamp; 12, pressure gauge; 13, nozzle; 14, liquid nitrogen spray; 15, liquid nitrogen Dewar vessel; 16, thermocouple location for measuring the internal temperature of the box; 17, thermocouple location for measuring the external temperature of the box; 18, solenoid valve; 19, hand operated valve.

The platinum plate was surrounded by strips of mica plates to limit the quantity of droplets bouncing off from the plate surface during the spraying process [21] involving $We > 80$.

The cross-sectional view of the experimental set-up and the electrical circuit of the plate are shown in Figs. 1 and 2, respectively.

At the start of each experimental run, the solenoid valve was energized via the thermostat to supply liquid nitrogen through the pressure nozzle. The supply of liquid nitrogen in each case was at a preset pressure of the relief valve connected to the Dewar and indicated by two pressure gauges; one on the Dewar and the other near the nozzle. The nozzle was constructed using stainless steel material type 18–8–3 with a 6.5 mm adaptor thread for connection to a flare nut (see Fig. 3) on the liquid nitrogen supply pipe. The ratio of orifice diameter to length of nozzle is 0.17 which is less than the maximum of 0.2 required for uniform performance of pressure nozzles [22].

In order to obtain the effective spray area, A_e, of the plate covered with droplets, a thin layer of silicone grease (about 2.0 mm thick) was spread evenly over the entire plate surface without any heating.

The spray was turned on at fixed static pressures as indicated on a calibrated pressure gauge connected to a point near the spray nozzle. It was therefore possible to adjust the nozzle height vertically above the centre of the test section to maintain a uniform distribution of spray onto a fixed area, A_e, with a constant flow rate, m, and spray distribution geometries at different pressures.

After turning the spray onto the plate surface without heating, two effects are observed. First, the droplets within the spray create a depressed circular area due to impact on the plate surface and could be measured directly [23] since the diameter is constant and regular.

Second, the greased area of the plate bombarded with liquid nitrogen sprays usually freezes quite rapidly, becomes whitish and remains hard frozen for a long time, thus allowing measurements to be made. The two effects give the same A_e repeatedly and are hence reliable.

The mass flow rate of the liquid nitrogen was obtained by measuring the weight changes of the Dewar and its liquid content at time, t, of the spraying. Hence, it was possible to obtain constant mass flow rate of the liquid sprayed in kg h^{-1} m^{-2} impinging uniformly on the test section over an area, A_e.

Sixteen copper–constantan thermocouples are connected to the plate and to the temperature controller to regulate the flow of liquid nitrogen onto the test section. Ten thermocouple wires are connected in a grid form to a multiple point Comark temperature recorder for box temperature measurements. Since temperature variation in the box did not exceed $\pm 6°C$, a mean value was used in the determination of evaporation and radiation losses to the surroundings.

At a preset temperature of the controller, the

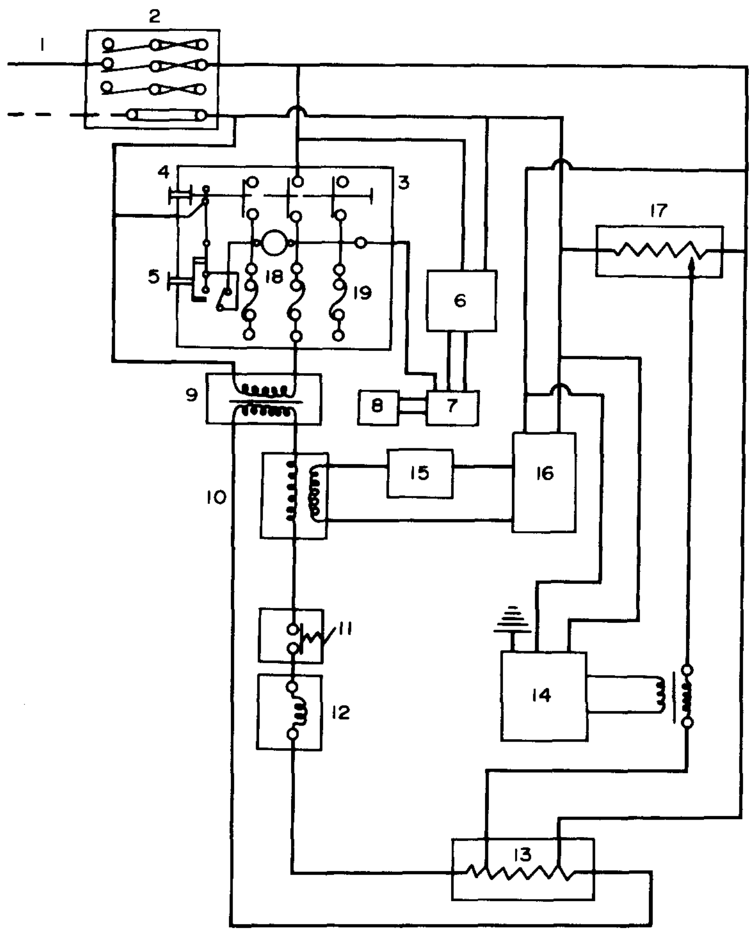

FIG. 2. Details of electrical wiring of the heating plate: 1, 220 V supply; 2, MEM isolator switch (30 A fused); 3, MEM 253 ALS/RC contactor/starter; 4, start button; 5, stop and reset button; 6, Donovan low voltage/phase failure relay; 7, Brookhirst Igranic time delay relay, 10 s off delay; 8, Planet D time clock, set point at 2 min; 9, transformer; 10, current transformer; 11, thermostat; 12, solenoid valve coil; 13, platinum plate; 14, cathode ray oscillograph; 15, ammeter; 16, wattmeter; 17, known proportional resistance; 18, starter holding coil; 19, overload protectors.

solenoid valve located on the feed pipe is switched open and liquid nitrogen flows through a hand operated valve (see Fig. 1). At the beginning of the experiments, it was necessary to regulate the flow of liquid nitrogen by adjusting the hand valve located between the solenoid valve and the spray nozzle. In this way the thermostat calls for refrigerant flow at a predetermined rate through both the solenoid valve which operates continuously, and the adjustable hand valve to avoid overflooding, and intermittent flow conditions. At the end of each experiment, the solenoid valve was switched off via the temperature controller. The actual time when the plate surface was sprayed with liquid nitrogen was recorded by using a stop watch. The time recorded was used in the computation of $\overset{*}{m}$ and the flow of liquid is switched off via the solenoid valve.

All measurements of energy input to the plate, current flow to the test plate, liquid flow rate per unit area of the plate surface sprayed, the plate temperature and spray pressures were made when the plate temperature was steady at a preset level.

EVAPORATION FOR SPRAYS

Consider an energy balance around a heated plate as shown in Fig. 4 where the quantity of vapour produced at the surface per unit time equals the rate of liquid actually hitting the plate surface.

Thus

$$q_p = q_e + q_c + q_r \quad (1)$$

where $q_p = I^2 R_p$ is the rate of energy generated within the plate, $q_e = \lambda\, dm/dt$ the rate of energy absorbed by the spray in order to evaporate, $q_c = hA_e(T_p - T_s)$ the conductive–convective heat transfer rate between the plate and the sprays, and $q_r = \Psi F A_e(T_p^4 - T_s^4)$ the rate of energy loss due to radiation from the plate to the surroundings (F is a dimensionless correction

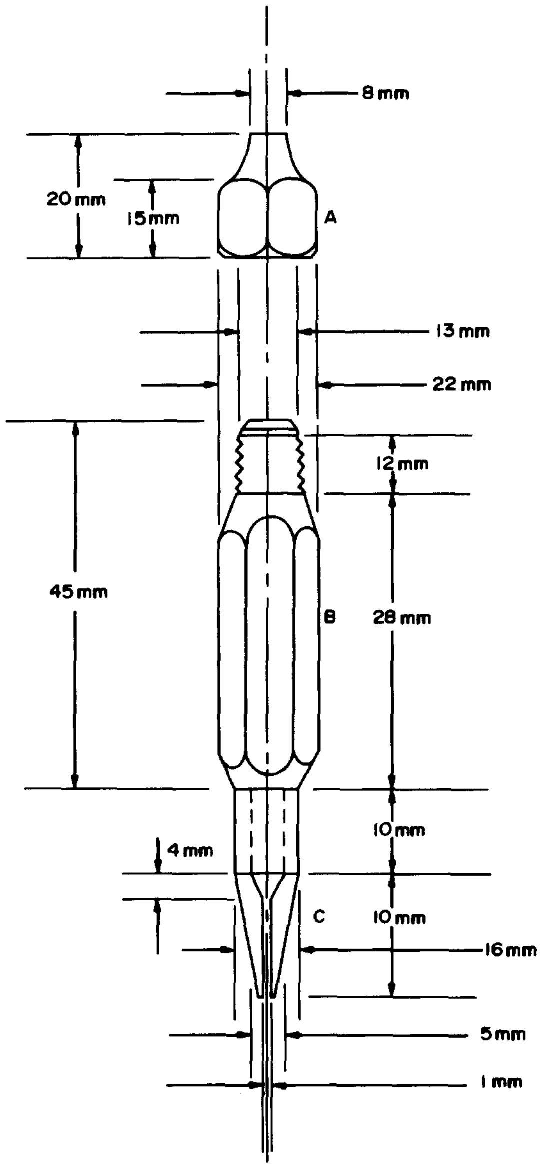

FIG. 3. Details of construction of the spray nozzle: (A) flare nut for liquid nitrogen feed; (B) stainless steel strainer housing; (C) nozzle having a diameter to length ratio of 0.17.

factor for geometry and thermal emissivity determined as given in ref. [24]).

The average heat transfer coefficient, h, between the spray and the plate surface could be determined from Newton's law of cooling, where

$$q_c = hA(T_p - T_s) \simeq hA\Delta T. \tag{2}$$

(A is the cross-sectional area of the spray.)

Assuming that the heat required to boil the liquid nitrogen spray comes from the effective heat transfer area of the plate, A_e, then

$$q_c \simeq hA_e\Delta T. \tag{3}$$

The actual heat flux at the plate surface is q_c/A_e and hence

$$q_c/A_e = m\lambda/A_e = h\Delta T. \tag{4}$$

Since the heat transfer occurs through a vapour film separating the plate from the droplets [11], it is necessary to modify λ as λ' where

$$\lambda'/\lambda = [1+(7/20)C_{pv}(T_p-T_s)/\lambda]^2/[1+(1/5)C_{pv}(T_p-T_s)/\lambda]. \tag{5}$$

Hence, λ' is used for the conductive–convective heat transfer coefficient values.

It is also important to consider the effective utilization of the spray in relation to heat transfer.

Thermal utilization of the spray, ε_T, can be defined as the ratio of actual heat transferred at the plate surface, q_a, to the maximum heat, q_m, which can be absorbed by the sprays per unit time assuming that such energy comprises both the latent heat required to boil the liquid and the sensible heat absorbed by the vapour produced at the plate area, A_e.

Since

$$q_a = hA_e(T_p - T_s) \tag{6}$$

and

$$q_m = m\{\lambda + [C_{pv}(T_p - T_s)]\}. \tag{7}$$

Taking the local mass velocity of the liquid sprayed per unit area as

$$\overset{*}{m} = m/A_e \tag{8}$$

hence

$$\varepsilon_T = h\Delta T/\overset{*}{m}\{\lambda + C_{pv}\Delta T\}. \tag{9}$$

(The value of C_{pv} is evaluated at a mean temperature between the plate and the spray.)

Considering the spray as consisting of mixed droplet sizes (Table 1), the mean diameter of the spray is

$$d_{32} = \sum_{i=1}^{n} d_i^3 \bigg/ \sum_{i=1}^{n} d_i^2. \tag{10}$$

The Sauter mean diameter, d_{32}, is a representative average diameter of droplets within a given spray and it is related to the specific surface area, S, of the droplets by

$$S = 6/d_{32}. \tag{11}$$

In order to calculate the Reynolds and Nusselt numbers of the spray for the heat transfer data, it is necessary to determine d_{32} for each spray condition. The results from the nomograph developed by Mugele [17] for sprays discharging through pressure atomizers can be applied to determine the maximum and mean droplet diameters of a system of sprays. The results from the nomograph are in good agreement with the experimental results and has also been applied in this work.

The experimental Nusselt number is

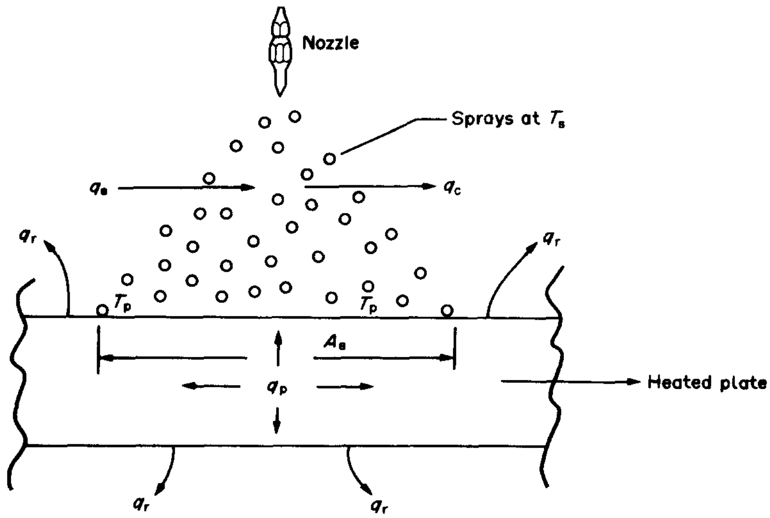

FIG. 4. Energy balance around an elemental test strip.

$$Nu = hd_{32}/K_v \tag{12}$$

and the Reynolds number is

$$Re = \rho_l V d_{32}/\mu_l. \tag{13}$$

It is also necessary to consider other heat transfer parameters of the spray, such as the latent heat of vaporization of the liquid, λ, mean temperature difference, ΔT, vapour viscosity, μ_v, and both liquid and vapour densities, ρ_l and ρ_v, respectively. These parameters may be combined either as dimensionless or dimensional groups, so

$$(K_v^2 \Delta T/\mu_v^2 C_{pv}\lambda) \quad \text{and} \quad ((\rho_l - \rho_v)/\rho_v)$$

are significant in the determination of the heat transfer coefficient of evaporating droplets [25].

The Prandtl number, Pr, relates the boundary layer thickness and temperature field at the plate surface

$$Pr = C_{pv}\mu_v/K_v. \tag{14}$$

Thus combining equations (12)–(14) together with the temperature difference and density groups, a multiple regression model can be obtained to correlate the experimental data of the spray by applying the least squares method.

EXPERIMENTAL RESULTS

The plate temperature, T_p, during each experimental run was determined by treating the platinum plate as a resistance thermometer [20] using the relationship between electrical resistance, R_p, and temperature, T_p, of the platinum plate. Hence

$$R_p = R_c[1 + (3.9788 \times 10^{-3} T_p) - (5.88 \times 10^{-7} T_p^2)] \tag{15}$$

where

$$R_c = 0.0006683\,\Omega.$$

R_c was determined separately by immersing the plate in an oil bath of constant temperature [20] and the current, I, flowing through the plate was obtained from the ammeter reading.

The energy input of the plate, E, was computed from the wattmeter reading at steady conditions and the value of R_p was obtained from the basic relationship, $E = I^2 R_p$, for the test section.

Since it was possible to obtain the quantity of liquid sprayed over a given time from the weight changes of the Dewar and its content, the mass flow rate per unit area of the plate surface subjected to the spray was recorded as $\overset{*}{m}$.

Table 1 shows the results of the spray characteristics of liquid nitrogen for the pressures investigated.

The heat flux, q_c/A_e, at the plate surface was obtained by dividing the heat transfer rate by the area of the plate covered by sprays. The heat flux data vs mean temperature difference between the plate and the boiling nitrogen droplets are shown in Fig. 5. Figure 6 shows the heat flux plotted against the mass flow rate per unit area of sprayed surface.

The changes of average heat transfer coefficient, h, between the plate and the sprays from equation (4) as the temperature difference changes are also shown in Fig. 7 for the range of spray pressures investigated.

The thermal utilization of the spray, ε_T, determined from equation (9) has been computed for different spray pressure vs various temperature differences as shown in Fig. 8.

Obviously some evaporation from the sprays should occur while the droplets travel between the nozzle and the plate surface. Hence, the rate of evaporation, dm/dt, from the droplets into the surrounding air/nitrogen vapour has been shown [26] to be

$$dm/dt = \pi d_{32} K_a \Delta T/\lambda(2.0 + \beta d_{32}^{1/2}) \tag{16}$$

for heat transfer to spheres and

Table 1. Spray characteristics of liquid nitrogen at different pressures using a 1.0×10^{-3} m diameter nozzle

Spray pressure (absolute), P (bar)	Pressure drop, ΔP (bar)	Spray velocity, V (m s^{-1})	Reynolds number, Re	Weber number, We	Mean spray diameter, d_{32} (m × 10^{-6})	Maximum spray diameter, d_{mx} (m × 10^{-6})
2.0	1.0	10.94	55451	1740	159	406
4.0	3.0	19.03	96473	3953	120	290
6.0	5.0	24.58	124629	5903	107	254
8.0	7.0	29.25	147552	6963	90	210
10.0	9.0	33.01	167343	7961	80	195
12.0	11.0	36.50	185029	9125	75	185

$$Sh = 2.0+0.81\,Re^{1/2}\,Sc^{1/3} \quad (17)$$

for mass transfer from spheres. The Lewis number (Pr/Sc) ranged between 0.89 and 0.97. This is fairly close to unity and it could be assumed that the analogy between heat and mass transfer holds correctly [27, 28] for the model considered.

Equation (16) can be integrated to find the time required for a change of droplet diameter, d_{32}, to a new size within the spray during its motion from the nozzle to the plate surface.

Hence, the values of m used in computing the actual heat transfer coefficient, h, are less than the measured value by an amount calculated using equation (16) per droplet.

The rate of mass transfer from the top surfaces of droplets due to molecular diffusion into the surroundings in the box can also be determined for a simplified case as shown in the Appendix.

The correlating equation for predicting the actual heat transfer coefficient between the sprays and the plate surface is obtained from an IBM computer using all the experimental data

$$h = 3.7\pi\frac{K_v}{d_{32}}\left(\frac{\rho_l V d_{32}}{\mu_l}\right)^{1/2}\left(\frac{C_{pv}\mu_v}{K_v}\right)^{1/3} \times\left(\frac{\mu_v^2 C_{pv}\lambda}{K_v^2\Delta T}\right)^{1/4}\left(\frac{\rho_l-\rho_v}{\rho_v}\right)^{1/3}. \quad (18)$$

The regression line of equation (18) within the range of spray conditions investigated is shown in Fig. 9. (Note that d_{32} is an actual value reaching the plate surface.)

DISCUSSION

The range of spray pressures and temperature differences investigated are 2–12 bar and 250–450 K, respectively, using a nozzle of 1.0 mm diameter. The

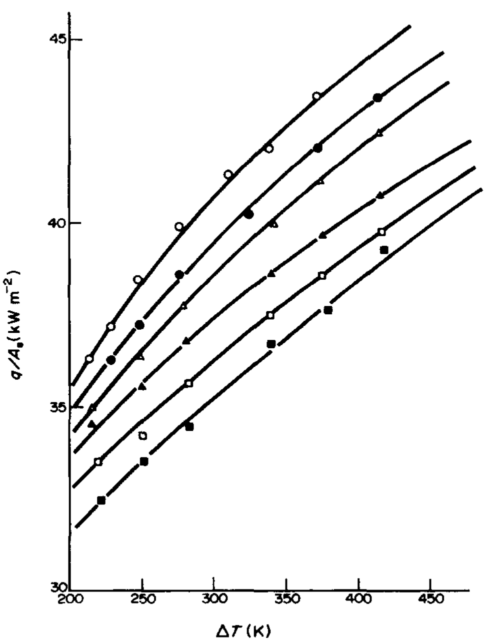

FIG. 5. Experimental heat fluxes of nitrogen sprays vs temperature differences at various spray pressures: ○, data at 12 bar; ●, data at 10 bar; △, data at 8 bar; ▲, data at 6 bar; □, data at 4 bar; ■, data at 2 bar.

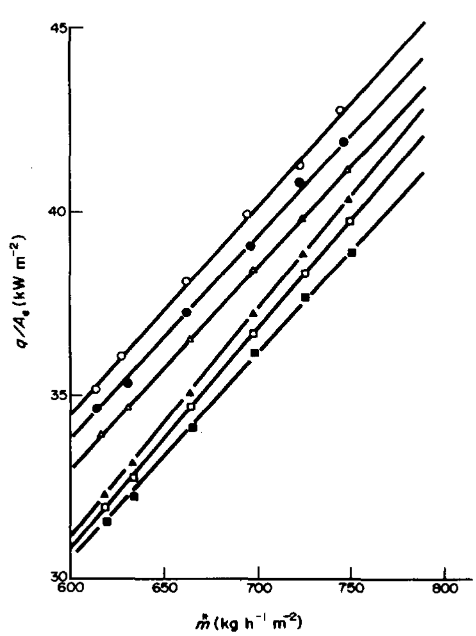

FIG. 6. Heat fluxes vs mass flow rate of liquid per unit area of sprayed surface: ○, data at $\Delta T = 450$ K; ●, data at $\Delta T = 400$ K; △, data at $\Delta T = 350$ K; ▲, data at $\Delta T = 300$ K; □, data at $\Delta T = 275$ K; ■, data at $\Delta T = 250$ K.

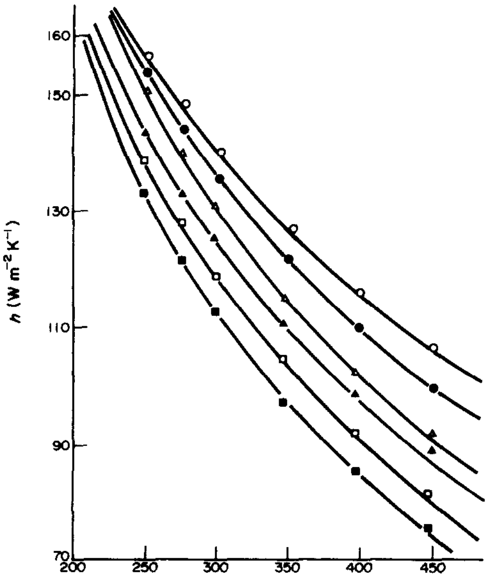

FIG. 7. Average heat transfer coefficient to the sprays vs mean temperature difference at different pressures: ○, data at 2 bar; ●, data at 4 bar; △, data at 6 bar; ▲, data at 8 bar; □, data at 10 bar; ■, data at 12 bar.

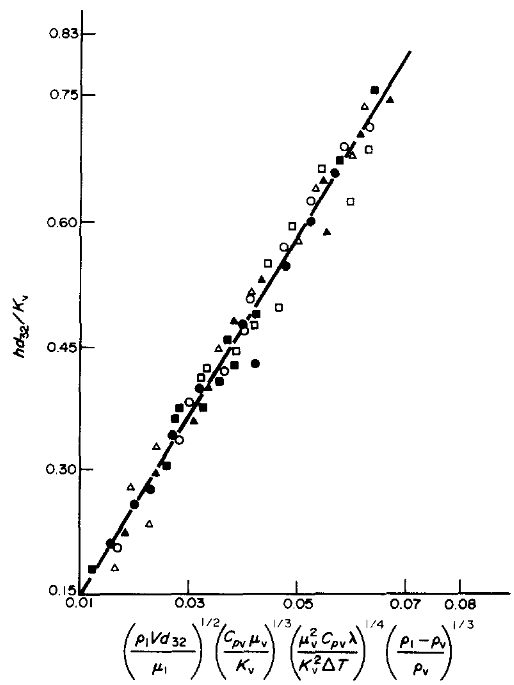

FIG. 9. Dimensionless correlation of experimental heat transfer data: ○, data at 250 K; ●, data at 275 K; △, data at 300 K; ▲, data at 350 K; □, data at 400 K; ■, data at 450 K.

Sauter mean diameter, d_{32}, of the sprays ranged from 159 μm at 2 bar to 75 μm at 12 bar.

As expected in Fig. 5, the heat flux increases as the mean temperature difference increases for a given pressure and as the mass flow rate per unit area of liquid nitrogen spray increases. Essentially the heat transfer rates will increase with increasing flow rate as shown in Fig. 6 for various mean temperature differences.

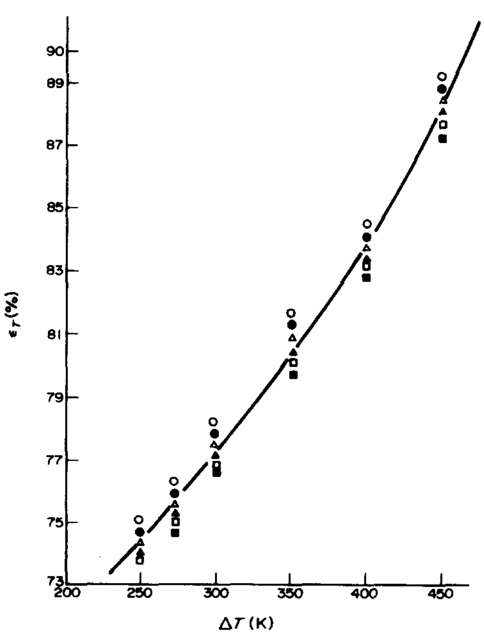

FIG. 8. Thermal utilization of sprays at various temperature differences: ○, data at 2 bar; ●, data at 4 bar; △, data at 6 bar; ▲, data at 8 bar; □, data at 10 bar; ■, data at 12 bar.

The plate temperatures used are much higher than the Leidenfrost temperature of liquid nitrogen [15], therefore occurrence of stable film boiling is indicated in Fig. 7 where the average heat transfer coefficient decreases with increasing temperature difference. This suggests that precooling of materials prior to freezing or cooling and the use of moderate temperature differences are required to enhance the rates of heat transfer to liquid nitrogen sprays.

The thermal utilization of the liquid spray on the plate surface depends on temperature difference, ΔT, mass flow rate or liquid per unit area of plate surface, $\overset{*}{m}$, and the heat transfer coefficient, h between the plate and sprays.

From Fig. 8 it could be seen that the thermal utilization, ε_T, of a spray increases as the spray pressure decreases for a given temperature difference. Since the mean spray size increases as the spray pressure decreases, therefore the thermal utilization of a spray will increase as the mean spray size and plate temperatures increase [11, 29].

In this report, ε_T values ranged between 0.74 and 0.89 even after correcting the mass flow rate, m, of liquid nitrogen for evaporation losses between the nozzle and the plate and also the radiation losses from the plate. This suggests that there was no overflooding of the plate which is in agreement with observation during the experiment.

Since the maximum value of ε_T is 0.89, there is a possibility that some droplets are bouncing off from the plate surface because ε_T is dependent on the amount of liquid delivered at the plate surface. These are droplets hitting near the edge of the plate. Such droplets will eventually evaporate into the surrounding air in the box and will undoubtedly contribute to the lowering of the box temperature.

In this preliminary study, it was not possible to quantify the flow rate of liquid due to bouncing off as some of the droplets are likely to rebound before bouncing off and some will just roll on the plate depending on their size and the plate temperature. The droplets that roll on the plate will contribute to raising the ε_T value than those bouncing off, although this contribution depends on how long they roll on the plate surface.

The Weber number, We, of the sprays vary between 1700 and 5900 (see Table 1) which is far greater than 80, so break up will occur [21]. Incidentally, film boiling conditions favour the rolling of droplets [30] and bouncing of droplets [21]. In either case, a contact effect occurs between the plate and the droplets. The time required for such contact is short as reported in ref. [21] but it enhances heat transfer and ε_T rather than those droplets bouncing off entirely from the plate surface.

In addition to ε_T, it is significant to consider the fraction of effective heat transfer area, ε_{ss}, of the spray. This has been done by Bonacina *et al.* [11] as

$$\varepsilon_{ss} = \overset{*}{m}\lambda'/h\Delta T. \tag{19}$$

Equation (19) refers to a uniform initial size distribution of droplets characterized by a mean size, d_{32}, under steady-state condition.

But Bonacina *et al.* [11] did not measure $\overset{*}{m}$ values in their theoretical analysis and since this is a critical variable, it has been measured in this work for each spray condition. The model used by Bonacina *et al.* [11] looks good and if accepted for the case studied here, the fraction of active heat transfer area, ε_{ss}, would be between 0.20 and 0.31. Although they reported lower values ranging from 0.13 to 0.19 in food freezing using liquid nitrogen sprays, it is interesting to note that both works and others [11, 19, 29, 31] confirmed an increase in the ε_{ss} value as the mean spray size, d_{32}, increases or pressure decreases (see Fig. 10). Also, the large droplets contained in the spray (being far from uniform in Table 1) should normally enhance higher values of ε_{ss}.

Higher values of heat transfer coefficients are possible with lower mean sizes of a spray for any given ΔT, and an increase in the mass flow rate of liquid, therefore, higher heat transfer rates will require higher spray pressures in order to produce smaller droplet sizes and hence, as h increases, ε_{ss} decreases.

The effect of vapour bubbles breaking out of the sprays during impact at high plate temperatures (greater than the minimum required for vapour nucleation) [31] may contribute to lower values of ε_{ss} since the overall heat transfer area of the spray may be reduced.

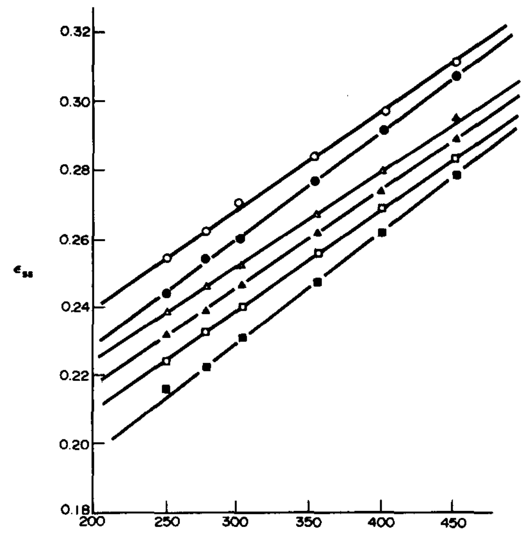

FIG. 10. Fractions of active heat transfer area, ε_{ss}, of a spray vs temperature differences at various pressures: ○, data at 2 bar; ●, data at 4 bar; △, data at 6 bar; ▲, data at 8 bar; □, data at 10 bar; ■, data at 12 bar.

The transfer of heat in spray cooling is largely by a conductive–convective mechanism. The radiation losses from the plate are about 0.8 and 3.5% of the energy input to the heater at plate temperatures of 150 and 255°C, respectively.

The influence of box and plate temperatures on evaporation losses from the sprays while moving from the nozzle towards the plate surface ranged between 16 and 37% of the overall mass of liquid nitrogen sprayed at plate temperatures of 55 and 255°C, respectively. These losses contribute to the lowering of the box temperature.

As shown in the Appendix, the rates of mass transfer from sprays due to molecular diffusion are about 5–12% of the rates due to the conductive–convective effect at the plate temperatures used even though equation (A6) is independent of plate temperature, T_p. In other words, it is simply assumed that the actual diffusion coefficient is the same as the molecular diffusivity of liquid nitrogen at saturation temperature, T_s. However, at high plate temperatures, the effect of molecular diffusion is relatively low when compared with the conductive–convective heat transfer between the plate and the sprays.

The results from equation (18) show that $h \propto d_{32}^{-1/4}$ which was obtained for heat transfer to individual droplets evaporating on solid surfaces [14, 25, 32]. Thus, the heat transfer coefficient to the sprays and the heat fluxes at the plate surface will increase if the mean diameter, d_{32}, of the spray is decreased and higher pressures are required.

Equation (18) also shows that $h \propto \Delta T^{-1/4}$ which is in agreement with those reported earlier [14, 25, 32] for single droplets on a solid surface. So, to get a high rate of heat transfer coefficient, h, the stable film boiling regime should be avoided wherever possible by reducing the mean temperature difference, ΔT, between the plate and the sprays.

It is also evident from equation (18) that the Nusselt number is given by

$$Nu = Re^{1/2}\,Pr^{1/3}.$$

This is similar to heat transfer involving moving droplets or spheres in a fluid stream [8, 9, 28, 34, 35].

The prediction of h values from equation (18) are in good agreement with the experimental data and within $\pm 10\%$ deviation. The maximum scatter point is also within $\pm 15\%$ of the experimental data.

CONCLUDING REMARKS

An experimental investigation of heat transfer using liquid nitrogen sprays has been carried out. The mean spray diameter, d_{32}, the mean temperature difference between the plate and the liquid, ΔT, and the mass flow rate of the liquid sprayed per unit area, $\overset{*}{m}$, are the most significant parameters controlling the rates of heat transfer during spray cooling processes using liquid nitrogen.

The computed values of the fractional coverage of a spray, ε_{ss}, ranged between 0.20 and 0.31 compared with 0.13 and 0.19 reported by Bonacina *et al.* [11] for freezing food using liquid nitrogen sprays. The differences between the values are probably due to surface roughness of the plate in comparison with food. The food surface is generally rougher and therefore, the rougher surface may tend to trap the sprays such that the Leidenfrost phenomenon [30] predominates and this limits the effective contact between the sprays and the plate surface. However, both works confirmed an increase in the fractional coverage, ε_{ss}, of a spray as the mean spray size, d_{32}, increases and the heat transfer coefficient, h, decreases.

The values from the predictive equation developed for the average heat transfer coefficient to the sprays are in good agreement with the values obtained experimentally and are also of the same order of magnitude as those reported in the literature for similar operations.

However, some areas requiring further investigation are given below.

(1) The behaviour of droplets on hitting the plate surface should be studied even though this may be difficult to do because of the foggy nature of liquid nitrogen sprays when in contact with moist atmospheric air. This fog greatly limits the field of vision around the spray area and therefore makes a photographic study almost impracticable. But such a study should concentrate on the residence time of droplets and their velocity components during bouncing, rebounding and rolling periods.

(2) The mass flow rate of the sprays bouncing off the plate surface should be measured so as to determine the overall mass of the sprays responsible for heat transfer at the plate surface after allowing for losses due to evaporation into the surrounding air.

(3) The effective coverage of droplets, ε_{ss}, in relation to the size distribution at the plate surface requires further investigation since droplets may flatten out upon impact and hence, increases their contact areas. Therefore, the assumption of spherical droplets for d_{32} may not be quite correct when droplets actually hit the plate surface. Indeed, some droplets may break into smaller droplets or explode upon contacting the hot plate, depending on the plate temperature during the impact. Even when droplets break up, their rates of heat transfer should generally increase (from equation (18)). Some droplets within a given spray may collide with one another to form larger size droplets which may assume a new shape far from spherical.

(4) Owing to the low surface tension characteristics of liquid nitrogen, it is necessary to determine more accurately the active heat transfer area, A, of the droplets upon contacting the plate. In this work, the area, A, has been assumed to be less than the surface area of the lower half of the sphere but greater than the projected area of a sphere [14]; that is

$$\pi r^2 < A < 2\pi r^2$$

so

$$A = (\pi r^2 + 2\pi r^2)/2$$
$$\simeq \tfrac{3}{8}\pi d_{32}^2.$$

There is little evidence to justify this assumption for liquid nitrogen and so, the heat transfer coefficients reported here should be applied in conjunction with this area. Thus, whether the area is valid or not, the heat flow rate into the droplet will still be given by

$$q = (hA)\Delta T.$$

A photographic technique is suggested for the determination of this heat transfer area over a wide range of droplet sizes and mean temperature differences.

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APPENDIX. RATES OF MASS TRANSFER FROM DROPLETS DUE TO MOLECULAR DIFFUSION

The rate of mass transfer, $\dot{d}m/dt$, due to molecular diffusion from the sprays impinging on the plate could be determined by considering the general mass transfer theory [36]

$$dm/dt = MK_c A(C_s - C_\infty). \tag{A1}$$

From the results of Froessling [8], Ranz and Marshall [9] and Fuchs [37], the dimensionless correlation for mass transfer from spheres in a gas stream is

$$Sh = a + b\,Re^{1/2}\,Sc^{1/3} \tag{A2}$$

where a is approximately equal to 2.0 and b is 0.6 for water droplets and 0.81 for liquid nitrogen droplets [26]. For zero relative velocity between the droplets and the surrounding gas

$$Sh \cong 2.0. \tag{A3}$$

Therefore

$$K_c \cong D_c/r. \tag{A4}$$

Assuming that the diffusing medium is an ideal gas and also $C_\infty = 0$, then equation (A1) reduces to

$$dm/dt = K_c P_s A/RT_s \tag{A5}$$

where A is the surface area, $\frac{5}{8}\pi d_{32}^2$, of the upper surface of a sphere which rests on its bottom surface of area $\frac{3}{8}\pi d_{32}^2$ (as defined in ref. [14]) and P_s is the partial pressure of the liquid droplet in equilibrium with its surrounding gas, i.e. P_s is the atmospheric pressure. Actually the gas venting window was designed to open whenever the box pressure reaches a maximum of approximately 5% higher than atmospheric pressure. This ensures that the box pressure is maintained at approximately 1 atm. Hence, T_s is approximately equal to the saturation temperature of liquid nitrogen (77 K) at 1.013 bar absolute.

For all gases, the product $MR = R_0$ and hence, for each droplet

$$dm/dt = MK_c P_s A/R_0 T_s. \tag{A6}$$

It is assumed that the bottom surface area $\frac{3}{8}\pi d_{32}^2$, is used for conduction–convection heat transfer at the plate surface hence, in equation (A6)

$$A = \tfrac{5}{8}\pi d_{32}^2.$$

CARACTERISTIQUES DE L'EBULLITION EN FILM DE L'AZOTE LIQUIDE ATOMISE SUR UNE PLAQUE CHAUDE

Résumé—Le transfert de chaleur entre une plaque chaude et un brouillard d'azote liquide saturé est mesuré à différents écarts de température et différentes pressions allant de 250 à 450 K et de 2 à 12 bar. L'analyse montre que la taille moyenne des gouttes dans le brouillard, la différence de température moyenne et la vitesse locale sont des paramètre les plus significatifs. L'efficacité du brouillard basée sur l'utilisation thermique du liquide et la fraction d'aire de transfert thermique sont déterminés pour différentes conditions. On propose une équation basée sur des groupes sans dimension. Il est ainsi possible de déterminer les effets de la vitesse moyenne du brouillard, de la taille moyenne des gouttes et de la différence de température moyenne sur le coefficient moyen de transfert de chaleur entre le brouillard et la plaque. Ces effets sont comparés à ceux antérieurement publiés et on obtient un bon accord.

EIGENSCHAFTEN DES FILMSIEDENS EINES SPRÜHNEBELS AUS STICKSTOFF AN EINER BEHEIZTEN PLATTE

Zusammenfassung—Für Temperaturdifferenzen zwischen 250 und 450 K und Drücken von 2 bis 12 bar wird der Wärmeübergang zwischen einer beheizten Platte und einem Sprühnebel aus gesättigtem Stickstoff gemessen. Die Ergebnisse zeigen, daß die mittlere Tropfengröße, die mittlere Temperaturdifferenz und die lokale Massenstromdichte des Sprühnebels die wesentlichen Einflußgrößen sind und den Wärmeübergang an der Plattenoberfläche bestimmen. Die Wirksamkeit des Sprühnebels, die auf der an die Flüssigkeit übertragenen Wärme und der anteiligen Wärmeübertragungsfläche des Sprühnebels beruht, wird für verschiedene Versuchsbedingungen bestimmt. Es wird eine dimensionslose Korrelationsgleichung vorgeschlagen. Als Ergebnis kann man den Einfluß der mittleren Geschwindigkeit des Sprühnebels, der mittleren Tropfengröße und der mittleren Temperaturdifferenz auf die durchschnittlichen Wärmeübergangskoeffizienten zwischen Sprühnebel und Platte bestimmen. Diese Einflüsse werden mit Beziehungen aus der Literatur verglichen, wobei sich eine gute Übereinstimmung ergibt.

ХАРАКТЕРИСТИКИ ПЛЕНОЧНОГО КИПЕНИЯ РАСПЫЛОВ ЖИДКОГО АЗОТА НА НАГРЕТОЙ ПЛАСТИНЕ

Аннотация—Измерена интенсивность теплопереноса между нагретой пластиной и факелом, полученным при распыле жидкого азота при разности температур и давлений, соответственно равных 250–450 К и 2–12 бар. Анализ данных по теплообмену показывает, что средний размер капли в факеле распыла, средняя разность температур и локальная массовая скорость в факеле распыла являются наиболее существенными для описания процесса переменными, которые определяют перенос тепла на поверхности пластины. Эффективность использования распыла, выраженная чержез теплофизические характеристики жидкости и площадь теплообмена струй, определялась для различных исследуемых условий. Предложено обобщенное уравнение в безразмерных критериях. Определено влияние средней скорости и среднего размера капель в факеле распыла, а также средней разности температцур на средний коэффициент теплопереноса между факелами распыла и пластиной. Проведено сравнение с ранее опубликованными в литературе результатами и получено хорошее соответствие.