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THE CONDENSATION COEFFICIENT OF WATER

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Abstract—Filmwise condensation of steam at low pressure on a vertical flat plate was investigated experimentally in order to ascertain the existence of an interfacial heat-transfer resistance and hence deduce a value of the mass accommodation or "condensation" coefficient of water. Data is presented for the condensation of saturated steam between 45 and 50°F at heat fluxes between 9000 and 12000 Btu/h-ft^2 degF. It was found that no significant interfacial resistance was present and the condensation coefficient was deduced to have a value between 0·45 and unity. That the condensation coefficient is at least greater than 0·45 ensures that the interfacial resistance will be negligible in industrial applications of filmwise condensation. The inability to determine a more exact value for the coefficient was due to an inherent limitation in the technique and the range quoted must not be taken to favor an intermediate value.

NOMENCLATURE

c,	average molecular speed $= \sqrt{(8/\pi\, RTg_0)}$;
c^*,	mean thermal speed $= \sqrt{(2RTg_0)}$;
c,	liquid specific heat;
f,	condensation or evaporation coefficient, defined by equation (1);
g,	acceleration due to gravity;
g_0,	constant in Newton's Second Law;
h_i,	interfacial heat-transfer coefficient;
h_{Nu},	Nusselt heat-transfer coefficient for filmwise condensation;
h_{fg},	latent heat of vaporization;
k,	thermal conductivity;
L,	length of condensing surface;
n,	concentration of molecules;
p,	pressure;
Pr,	Prandtl number;
q,	heat flux;
R,	gas constant;
r_0,	radius of jet;
s,	molecular speed ratio $= u/c^*$;
T,	temperature;
u,	bulk velocity of vapor normal to interface; velocity of liquid in jet;
U,	overall heat-transfer coefficient;
V_g,	vapor specific volume;
w,	net condensation rate.

Greek symbols

α,	thermal diffusivity;
Γ,	correction for bulk velocity;
ρ,	density;
ρ_v,	vapor density;
μ,	dynamic viscosity.

Subscripts

g,	vapor;
g, sat,	saturated vapor;
s,	liquid surface;
w,	wall.

INTRODUCTION

THE ACCEPTED physical model of evaporation and condensation is based on proposals of Hertz in 1882 [1] and of Knudsen in 1915 [2]. The kinetic theory of gases yields the rate at which molecules strike the condensed phase from equilibrium vapor as $n_g c_g/4$; a "condensation coefficient" may be introduced to account for the fraction of these incident molecules which enter the condensed phase, the remainder being reflected. The flux of molecules leaving the condensed phase is given by $f n_s c_s/4$ where f is the "evaporation coefficient" and is deduced to be equal to the condensation coefficient in order

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for the equilibrium situation to be attainable. During net phase change the bulk vapor velocity normal to the interface affects the molecular velocity distribution; Schrage [3] took this into account and derived the following expression for the net condensation rate:

$$w = f\left(\Gamma\frac{n_g c_g}{4} - \frac{n_s c_s}{4}\right) \text{ mol./ft}^2\,\text{s} \qquad (1)$$

where

$$\Gamma = \exp(-s^2) + (\sqrt{\pi})\, s(1 + \operatorname{erf} s). \qquad (2)$$

Introduction of the expression for molecular concentration and linearization of Γ yields

$$w = \frac{f}{1 - 0{\cdot}5f}\left[\frac{p_g}{\sqrt{(2\pi RT_g)}} - \frac{p_s}{\sqrt{(2\pi RT_s)}}\right] \quad 0 < |s| < 0{\cdot}001 \qquad (3)$$

or

$$w = \frac{f}{1 - 0{\cdot}523f}\left[\frac{p_g}{\sqrt{(2\pi RT_g)}} - \frac{p_s}{\sqrt{(2\pi RT_s)}}\right] \quad 0{\cdot}001 < |s| < 0{\cdot}1. \qquad (4)$$

For the net condensation of saturated vapor it is clear that p_s must be less than p_g, i.e. the temperature of the condensate surface must be less than the saturation temperature of the vapor. Silver [4] expressed this in terms of an interfacial heat-transfer coefficient:

$$h_i = \frac{w h_{fg}}{T_{g,\,\text{sat}} - T_s}. \qquad (5)$$

It is important to observe that this definition is in terms of the vapor saturation temperature $T_{g,\,\text{sat}}$ and not the temperature of the vapor incident on the surface, T_g. The latter differs from the condensate surface temperature, T_s, by what is essentially the temperature "jump" of the kinetic theory of gases and this difference is negligible in nearly all applications. Also there is a negligible pressure gradient normal to the surface, thus the vapor undergoes supersaturation at constant pressure before condensation. An expression for the temperature jump during phase change is derived from Grad's thirteen moment molecular velocity distribution in Appendix B of [5]. Hence $\sqrt{(T_g)}$ may be set equal to $\sqrt{(T_s)}$ in equation (3); introduction of the Clausius Clapeyron relation and combining with equation (5), yields with sufficient accuracy:

$$h_i = 778\frac{f}{1 - 0{\cdot}5f}\left(\sqrt{\frac{g_0}{2\pi RT_s}}\right)\cdot\frac{h_{fg}^2}{T_s V_g} \quad \text{Btu/h-ft}^2\,\text{degF}. \qquad (6)$$

The relative importance of the heat-transfer resistance $1/h_i$ will depend on the nature of the condensation or evaporation situation since the overall resistance may vary over many orders of magnitude. The interfacial resistance is large for high values of V_g, i.e. low pressures and for small values of the condensation coefficient. For water some investigators claim a low value, of the order of 0·036, while more recent experiments have indicated values between 0·40 and unity. The magnitudes involved for water are indicated by Table 1 which gives values of the interfacial heat-transfer coefficient as a function of saturation temperature for condensation coefficients of unity and 0·036.

Table 1. Values of the interfacial heat-transfer coefficient for condensation coefficients of unity and 0·036

Temperature (°F)	h_i (Btu/h-ft² degF)	
	f = 1·0	f = 0·036
110	451000	7890
90	283000	4950
70	152000	2660
50	82000	1430
30	43000	751

If condensation is the process of interest and if the condensation coefficient is of the order of unity, then the interfacial resistance is of little importance except for dropwise condensation at low temperatures. With film condensation an important contribution to the overall resistance will arise at low temperatures only if the coefficient is indeed as low as 0·036.

Table 2. Previous experimental values for the condensation coefficient of water

Investigation	Date	Reference	Temperature °C	f	Nature of the experiment
group 1					
Alty	1931	6	18–60	0·006–0·016	evaporation from a suspended drop
Alty and Nicoll	1931	7	18–60	0·01–0·02	same
Alty	1933	8	−8–+4	0·04	same
Alty and Mackay	1935	9	15	0·036	same
Pruger	1940	10	100	0·02	evaporation from a horizontal surface
Hammecke and Kappler	1953	11	20	0·045	same
Hammecke and Kappler	1955	12	?	0·100	same
Delaney, *et al.*	1964	13	0–43	0·0415–0·0265	same
group 2					
Hickman	1954	14	0	0·42	evaporation from a tensimeter jet
Nabavian and Bromley	1963	15	10–50	$0·35 \leqslant f \leqslant 1·0$	film condensation on a fluted tube
Jamieson	1965	16	0–70	0·35	condensation on a tensimeter jet
Berman	1961	17	?	near to 1·0	film condensation on a horizontal cylinder

PREVIOUS INVESTIGATIONS FOR WATER

Table 2 lists values of f for water obtained experimentally by investigators studying a variety of evaporation and condensation phenomena.

The accuracy of the reported values of f has been subject to much discussion over the years. Reasons that have been given for low measured values of f include inaccurate surface temperature measurement when large temperature gradients prevail, contamination of the surface and the presence of non-condensable gas during condensation. Detailed evaluations of the accuracy of these results have been previously given in [18, 15, 19]; the evaluation made by the present investigation is reported in [5]. In all the group 1 experiments evaporation was under consideration and the central experimental problem was the determination of the temperature of the evaporating surface, the large temperature gradient normal to the surface in the liquid phase being the complicating factor. The conclusion of the present evaluation is that in these experiments the techniques of surface temperature determination were inadequate and no value of the condensation coefficient can be deduced, furthermore, the thermal situations involved are too complex to allow re-analysis of these experiments to yield more accurate estimates of the coefficient. It remains to discuss whether the results of group 2 should be accepted at face value. Hickman and Jamieson studied evaporation and condensation from a high velocity water jet; in both cases the surface temperature of the jet was not measured but estimated. Included in the present investigation was an analysis of these experiments involving an analytical calculation of the jet surface temperature; this showed that the original estimates were erroneous and that the experiments in fact yielded values of the condensation coefficient very close to unity. Appendix B to this paper contains a detailed consideration of Hickman's experiment. The experiments of Nabavian and Bromley and of Berman were similar to that of the present investigation, thus the discussion of the present investigation will suffice to evaluate those experiments as well.

The low values of condensation coefficient obtained in the early investigations prompted attempts to obtain a theoretical explanation of this phenomenon. The most widely discussed prediction is based on transition state theory and is associated with the names of Penner [20–22], Eyring [23–25] and others. Other approaches were employed by Danon [26], Kochurova [27] and Delaney [19]. Reference [5] contains a comprehensive treatment of the theoretical aspects of condensation and evaporation and evaluates each of these theoretical approaches. The conclusion reached is that these theories are of little substance and that at the present time it is necessary to rely on experimental results.

THE EXPERIMENTAL TECHNIQUE

The condensation phenomenon investigated was film condensation on a vertical flat surface from low pressure saturated steam. The condensing surface was the front face of a copper block; the opposing face was cooled by refrigerant and the remaining faces were well insulated. Measurements of the temperature distribution within the block allowed both the heat flux through the block and the surface temperature to be determined; measurement of the condensate rate and an energy balance on the coolant reinforced the heat flux determination. These measurements together with the state of the vapor sufficed for a determination of the overall heat-transfer coefficient. A description of the system is given in Appendix A, a schematic drawing and flow diagram are shown in Figs. 1 and 2.

The temperature profile across the film is shown in Fig. 3. The temperature drop across the film may be obtained from the Nusselt solution for laminar film condensation in the form proposed by Rohsenow [28] who gives the average heat-transfer coefficient as:

$$h_{Nu} = 0{\cdot}943\left[\frac{g\rho(\rho-\rho_v)k^3(h_{fg}+0{\cdot}68\,c\,\Delta T)}{L\mu\,\Delta T}\right]^{\frac{1}{4}} \tag{7}$$

where $\Delta T = T_s - T_w$.

The applicability of this formula is subject to a number of restraints. Boundary-layer analysis of the situation including the effect of vapor shear at the interface [29, 30] shows the formula to be valid for $Pr > 1$ and $c\,\Delta T/h_{fg} < 1{\cdot}0$, conditions which are met in the present investigation. Likewise [5] shows that, under the present conditions, there are negligible effects due to turbulence of the liquid, ripples on the liquid surface, vapor superheat and variation of wall temperature.

The condensation coefficient is obtained from the measured overall heat-transfer coefficient U via the interfacial heat-transfer coefficient given by equation (6), and the relation

$$\frac{1}{U} = \frac{1}{h_{Nu}} + \frac{1}{h_i}. \tag{8}$$

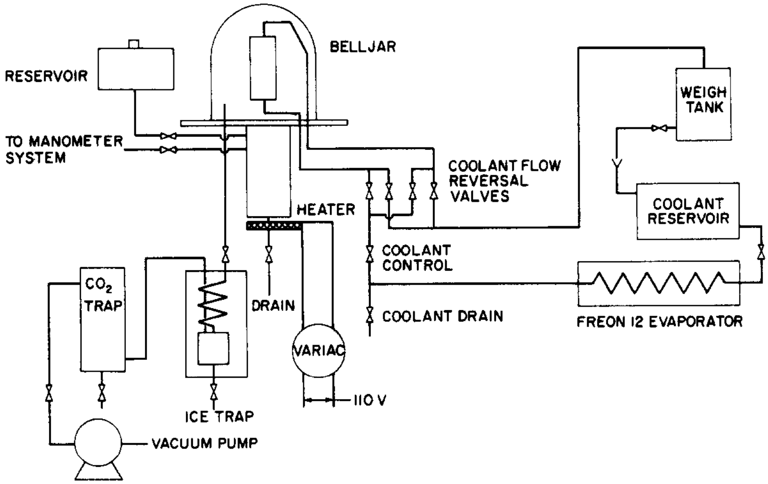

FIG. 1. Flow diagram of the experimental system.

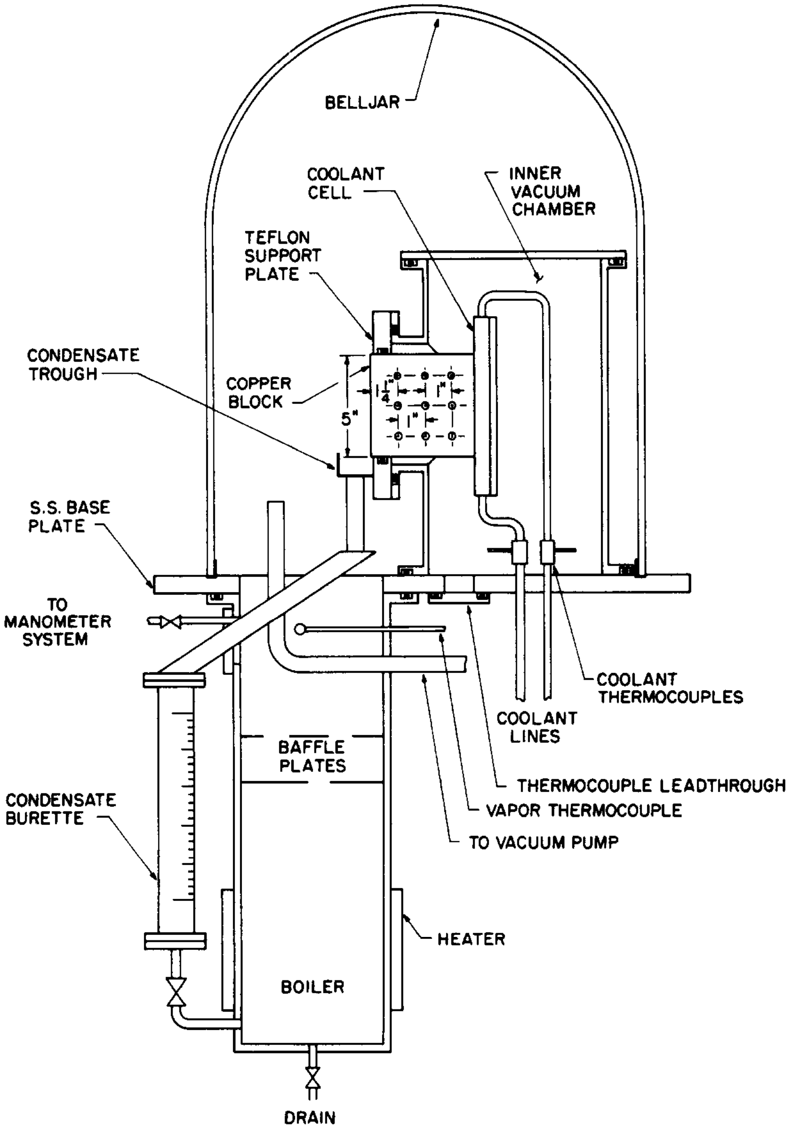

FIG. 2. Schematic drawing of the bell jar system.

Thus this technique suffers from the need to implicitly infer the surface temperature of the liquid from the measured wall temperature via the analysis for film condensation; however this method should be viewed with more confidence than the kind of estimation that needs to be employed in other experimental investigations. The chief disadvantage of this method lies in the constraint imposed by the magnitude of $1/h_{Nu}$; it is not feasible to reduce this resistance to much less than that in the present experiments. As such the method is, in principle, capable of yielding accurate estimates of low values of the condensation coefficient; on the other hand when the coefficient is greater than about 0·5 the resistance $1/h_i$ becomes too small compared to $1/h_{Nu}$ for an accurate estimate of its exact value. However knowledge

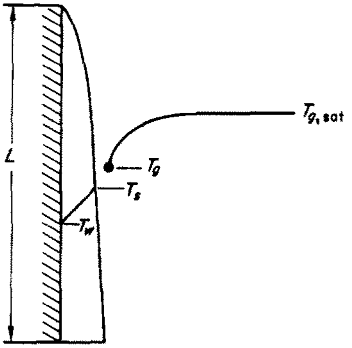

FIG. 3. Temperature profile in film condensation.

that the coefficient is indeed greater than 0·5 does mean that the interfacial resistance is negligible in practical applications of film condensation.

THE EXPERIMENTAL RESULTS

The results obtained with the present system are essentially the same as those reported for a similar system in [5]. It is important to note that the latter system was also operated at higher pressures than those reported here; under those conditions the interfacial resistance is negligible even for low values of the condensation coefficient and the Nusselt solution was duly confirmed by the experimental results.

The tests reported in Table 3 were taken over a period of two days; operation of the system was not continuous and the tests were not taken consecutively. Each test represents data averaged over about 15-min operating time. Temperature $T_{g,\mathrm{sat}}$ is the saturation vapor temperature corresponding to the measured vapor pressure while T_g is the measured vapor temperature. In most tests a slight degree of superheat is indicated, however this has negligible effect on the condensation. The temperature of the block surface, T_w, was obtained from a linear extrapolation of the temperatures measured with the block. The values of the heat flux, q, are the total transfer rate divided by the front surface area of the block. $q_{\mathrm{conduction}}$ was obtained from the temperature gradient in the copper block assuming one dimensional flow; this was regarded as the most accurate of the three independent determinations of q. $q_{\mathrm{condensate}}$ was obtained from a volumetric measurement of condensation rate and q_{coolant} from an energy balance on the coolant system. Columns H and I compare the latter two estimates with $q_{\mathrm{conduction}}$ and the ratios are reasonably close to unity. The variation about unity indicates the degree of precision of the measurements and the need to ultimately obtain an average before evaluating the condensation coefficient.

q_{Nu} is the heat flux predicted by the Nusselt solution for the measured temperature difference $(T_{g,\mathrm{sat}} - T_w)$ and is compared to $q_{\mathrm{conduction}}$ in column J. The existence of an interfacial resistance would be demonstrated by values in column J less than unity; it is clear that the scatter does not allow a significant deviation to be discerned. Another view of the situation is provided by a comparison of the measured temperature difference $(T_{g,\mathrm{sat}} - T_w)$ with that predicted by the Nusselt solution for the measured heat flux $q_{\mathrm{conduction}}$. The presence of an interfacial resistance would result in a positive difference between the former and latter, column N again shows that no significant interfacial resistance can be detected. For purposes of comparison, column M shows the interfacial temperature difference that would be expected for a condensation coefficient of 0·036, i.e. the expected value for column N. A quantitative estimate of what the degree of precision implies in terms of evaluating the condensation coefficient may be obtained from the standard deviation of column N which is calculated to be 0·50 degF. The corresponding value of the condensation coefficient for a deviation from the Nusselt solution of that magnitude is 0·45.

DISCUSSION AND CONCLUSIONS

The experimental results show that there is no measurable interfacial heat-transfer resistance present during the film condensation of steam at low pressures; there was no significant deviation from the Nusselt solution. The value

Table 3. *The experimental results*

Test	$T_{g\,sat}$ (°F) A	T_g (°F) B	T_w (°F) C	$q_{conduction}$ (Btu/ft^2-h) D	$q_{condensate}$ (Btu–ft^2-h) E	$q_{coolant}$ (Btu/ft^2-h) F	q_{Nu} (Btu/ft^2-h) G	$\frac{E}{D}$ H	$\frac{F}{D}$ I	$\frac{D}{G}$ J	$\Delta T_{measured}$ (degF) K	ΔT_{Nu} (degF) L	ΔT_i (degF) M	$K - L$ (degF) N
1	45·6	45·8	40·2	9900	9960	10570	8390	1·01	1·07	1·18	5·4	6·4	8·0	−1·0
2	46·6	46·9	40·6	10620	11020	10220	9030	1·08	0·96	1·18	6·0	7·1	8·3	−1·1
3	46·7	46·9	39·7	10110	10910	10570	10210	1·08	1·04	0·99	7·0	6·9	7·8	+0·1
4	48·0	48·9	40·9	10430	9900	10650	10410	0·95	1·02	1·00	7·1	7·1	7·1	0·0
5	48·5	49·5	40·6	10620	11610	11420	11170	1·09	1·07	0·95	7·9	7·5	7·8	+0·4
6	48·9	49·1	42·0	10730	11300	11770	10220	1·06	1·04	1·05	6·9	7·2	7·7	−0·3
7	49·8	50·4	42·0	10760	10850	10480	11260	1·01	0·97	0·96	7·8	7·5	7·6	+0·3
8	50·0	50·1	43·1	11050	10100	10220	10260	0·92	0·93	1·08	6·9	7·4	7·7	−0·5
9	50·0	50·4	42·9	10970	10500	11840	10490	0·96	1·08	1·04	7·1	7·4	7·6	−0·3
10	50·4	49·8	43·2	11050	11290	12650	10690	1·02	1·14	1·04	7·2	7·5	7·6	−0·3

of the condensation coefficient indicated lies between 0·45 and unity; thus this result is in accord with those reported by the experiments in group 2 of Table 2. Nothing further may be concluded from the present experiment due to the aforementioned limitation imposed by the magnitude of the resistance of the condensate film. The Nusselt analysis assumes a surface of infinite horizontal extent; the present surface was 2-in wide and a possible source of error is the thinning of the film towards the edges. No theoretical analysis of this effect has been made. Justification of assuming negligible error arises from film condensation data taken in similar systems [35, 5] where the Nusselt analysis was shown to be valid at higher pressures where there was no possibility of an interfacial resistance.

It is appropriate however to make a more general appraisal of the exact value of the condensation coefficient. Re-analysis of the experiments of Hickman and of Jamieson showed that the correct results of their experiments was a value of the coefficient close to unity. On the other hand those experiments which have yielded low values, of the order of 0·036, have been shown to be of doubtful validity; furthermore theories of phase change which predict low values of the coefficient for water are of little substance. The compilation of Paul [31] and subsequent work shows that a condensation coefficient of 1·00 is firmly established for most substances, of the remainder improvement of the experimental technique has invariably led to estimates approaching closer to unity. Thus in the light of the presently available evidence for water it seems reasonable to assert that there is little reason to believe that the condensation coefficient of water is less than unity.

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APPENDIX A

The Experimental System and Procedure

A schematic drawing and flow diagram of the system are shown in Figs. 1 and 2 respectively. The base plate was 24 × 20 × $\frac{5}{8}$-in thick 316 stainless steel, machined to support a 24-in high, 18-in dia. glass bell jar on a L-shaped Viton-A gasket. The boiler was welded from a 18-in length of 5-in dia., schedule 10, 316 stainless steel pipe, flanged at its top and bolted to the underside of the base plate with an O-ring seal. Heat was supplied to the boiler by means of two external electrical heating elements of total capacity 1·1 kW. Fill and drain lines with "Whitey" stainless steel valves were led into the boiler by "Swagelok" fittings, the pipe threads being sealed with Teflon tape. A 1-in dia. graduated glass tube was mounted over a valve on the side of the boiler and served both as a level indicator and as a condensate measuring burette, the liquid condensing on the test surface being let back to the boiler via the burette.

The test surface was the 5 × 2-in front face of a $4\frac{1}{2}$-in long copper block. The copper was oxygen free, high conductivity copper obtained from the American Brass Co. Thermal conductivity data [32, 33, 34] indicates values of 225 Btu/h degF at 0°F and 220 Btu/h ft °F at 100°F. A brass coolant cell was soft soldered to the back of the block; the width of the flow passage could be varied in order to obtain sufficient cooling without excessive coolant flow rates. The block was fitted with nine thermocouples located in three rows, 1, $2\frac{1}{2}$ and 4 in from the top of the block and $1\frac{1}{4}$, $2\frac{1}{4}$ and $3\frac{1}{4}$ in from the front face. The thermocouples were of 30 gage copper–constantan duplex, glass covered, wire, led through 1-in long, $\frac{3}{16}$-in dia. copper plugs and soft soldered to the plug tips. The plugs were inserted as push fits in holes drilled in the block. The copper block had $\frac{1}{4}$-in dia. rounded corners to allow use of two O-ring seals in the 1-in thick Teflon supporting plate. The Teflon plate was in turn bolted onto the stainless steel inner chamber via an O-ring seal. The inner chamber was constructed from a 16-in length of 9-in dia. schedule 10 stainless steel with $\frac{1}{4}$-in thick flanges and cover plates, all joints having O-ring seals. Coolant lines were led into the inner chamber to the coolant cell and were fitted with 22 gage copper constantan thermocouples for measuring the inlet and outlet coolant temperatures. These thermocouple wires together with those from the block were led out the inner chamber via a vacuum seal. During operation the inner chamber was always maintained at a pressure below 1 mm Hg, this ensured (i) that there were no leaks into the test chamber since the pressure there was always higher (ii) there was a minimum of heat transfer by natural convection between the sides of the copper block and the walls of the inner chamber.

The coolant was a "Dowtherm" glycol in water solution and was cooled to about 0°F before entering the coolant cell. The plumbing was arranged so that the coolant could enter at either the top or the bottom of the cell; it was invariably led into the top since this proved to give a more even heat flow through the copper block. During operation vapor was continually removed from the system through a $\frac{1}{2}$-in dia. tube; in this manner any air entering the system via leaks or dissolved in the water did not accumulate in the system.

Measurements

1. *Pressure.* A butyl phthalate manometer was used with one leg connected to a vacuum pump maintaining a vacuum of less than 20 μ. The butyl phthalate levels were read using a cathetometer.

2. *Temperatures.* The thermocouple e.m.f.'s measurements were made with a Leeds and Northrup Cat. No. 8686 potentiometer, least count 0·005 mV. A cold junction kept in melting ice was used as reference. The accuracy of individual temperature measurements is appraised to be within 0·10 degF. The important value of the surface temperature of the block, however, involved the extrapolation of a number of temperatures read within the block, and thus the average value of that temperature was probably known only to 0·5 degF.

Operating procedure

At the commencement of a series of tests the boiler was filled with triple distilled water and the valve leading to the vacuum pump fully opened. The system was allowed to run at a low boiling rate for at least four hours to deaerate the water. At the same time the system temperature was lowered below ambient to about 45°F by passing iced water over the bell jar and base plate. The exhausting rate was then decreased and the heat input to the boiler increased until saturated steam at the desired pressure was condensing, usually at the maximum rate limited by the coolant supply. When steady-state conditions were indicated by the recording potentiometer, test data was taken. With the system condensing saturated steam at the lowest desired temperature a new condition was obtained by increasing the electrical input to the boiler heater; the steam temperature would then increase until a steady state at the new temperature was obtained.

Problems encountered

1. A major problem was the attainment of perfect filmwise condensation. An earlier system used in the investigation reported in [5] had proved very troublesome in this respect and it was thought that contamination of the steam by various insulating materials was the cause. In the present system the only materials in contact with the steam in the test chamber were stainless steel, glass, Teflon and Viton A. All components were cleaned following procedures recommended by Welch [35] and in this way the absence of dropwise condensation was ensured. The copper test surface itself was polished to 2/0 grade emery cloth and cleaned with a solution of one part nitric acid in three parts by volume of water. Final rinsing was with triple distilled water. However, it was found that the preferred mode of condensation was one of rivulets and a peculiar downward flow of many small drops of small contact angle on what visually appeared to be a wetted surface. In this mode the heat-transfer coefficients were about twice those predicted by the Nusselt solution. On some occasions this mode would appear at commencement of operation changing to a film about 12 h later, on other occasions the reverse would happen. A procedure which often proved useful for obtaining a film from rivulets or removing areas of proper dropwise condensation was to cease the boiling and hence allow the water to freeze on the test surface; subsequent melting usually resulted in perfect filmwise condensation.

2. Earlier experience [5] had shown the importance of ensuring the absence of non-condensable gases. In the present system the problem was solved by elimination of leaks, deaeration of the water and the continuous exhausting of an appreciable amount of steam.

3. Even though the boiler was filled with

stainless steel wire gauze it was found that, after about 48 h continuous operation, the nucleation sites would be completely depleted and the water would not boil in a steady manner; rather 20–30 degF of superheat would build up over a period of a few minutes and then, due to the low pressure, the vapor would be released explosively. It was not possible to take data under those conditions.

APPENDIX B

Re-analysis of Hickman's Experiment [14]

In this experiment water was evaporated from a high speed tensimeter jet into a chamber maintained at 1 mm Hg vacuum. Hickman defined an uncorrected evaporation coefficient as:

$$f_{\text{uncorrected}} = \frac{\text{rate of evaporation}}{p_s/\sqrt{(2\pi R T_s)}} \tag{9}$$

where p_s and T_s were evaluated at the average bulk temperature along the jet. However, although the initial surface temperature equals the bulk jet temperature, it rapidly falls below the bulk temperature as the latent heat of evaporation is supplied. The evaporation rate is given approximately by equation (4). Strictly speaking, since in Hickman's experiment the molecular speed ratio exceeded 0:1, the linearization of the bulk velocity correction factor Γ used to obtain this equation is not accurate; however, the discrepancy is small and must be ignored. Comparing equations (4) and (9) and integrating along the length of the jet there results:

$$f_{\text{uncorrected}} = \frac{f}{1 - 0{\cdot}523\, f}\,\frac{1}{t_0}\int_0^{t_0} \frac{p_s - p_g}{p_b}\,\mathrm{d}t$$

where t_0—residence time of water in jet.

p_b—saturation pressure evaluated at the average bulk temperature along the jet.

The surface temperature of the jet must be calculated in order to determine p_s. The energy equation describing the temperature distribution in the jet is:

$$u\frac{\partial T}{\partial x} = \alpha\left(\frac{\partial^2 T}{\partial r^2} + \frac{1}{r}\frac{\partial T}{\partial r}\right)$$

with initial condition $T = T_{b,e}$ the entering bulk temperature and boundary condition:

$$k\frac{\partial T}{\partial r}\bigg|_{r=r_0} = h_i(T_s - T_{g,\text{sat}})$$

The following assumptions are made:

(i) The jet velocity u is constant across its cross-section.
(ii) The flow is laminar.
(iii) Constant fluid properties.
(iv) The jet cross-sectional area is constant.
(v) Heat conduction in the axial direction is negligible.
(vi) A suitable average value of h_i may be used.

The problem then reduces to a transient heat-conduction problem dealt with by Carslaw and Jaeger [36] on p. 201. However, the residence times in the jet are small enough to allow a further simplification; since the temperature near the center line of the jet remains close to the entering bulk temperature, the finite size of the jet is of no consequence and the solution for the corresponding problem for a semi-infinite solid may be used. Carslaw and Jaeger deal with this problem on p. 71 and the solution may be written:

$$\theta = \frac{T_s - T_{g,\text{sat}}}{T_{b,e} - T_{g,\text{sat}}} = \exp(h^2\alpha t)\,\operatorname{erfc} h\sqrt{(\alpha t)}$$

where $h = h_i/k$, k being the thermal conductivity of the liquid. Let $\xi = h(\sqrt{\alpha t})$ then θ is tabulated for values of ξ up to 3·0 on p. 485 of Carslaw and Jaeger. For larger values of ξ the following formula is accurate:

$$\theta = \frac{1}{(\sqrt{\pi})}\left(\frac{1}{\xi} - \frac{1}{2\xi^3} + \ldots\right).$$

The only real computational difficulty is choice of the correct average temperature for the evaluation of h_i; as the surface temperature

Table 4. Computations for re-analysis of Hickman's experiment

Residence s, t_0	$h\sqrt{\alpha t}$	θ	T_s (°F)	P_s (mm Hg)	$\frac{p_s - p_g}{p_{b,e}}$	$\frac{1}{t_0}\int_0^{t_0} \frac{p_s - p_g}{p_{b,e}} \mathrm{d}t$	$f_{\text{uncorrected}}$
10^{-6}	0·0727	0·923	41·8	6·7	0·730		
10^{-5}	0·230	0·785	35·1	5·2	0·539		
10^{-4}	0·727	0·497	21·1	2·91	0·245		
5×10^{-4}	1·63	0·301	11·6	1·91	0·117	0·198	0·425
10^{-3}	2·30	0·227	8·0	1·62	0·0795	0·146	0·307
2×10^{-3}	3·25	0·174	5·44	1·43	0·0551	0·106	0·223
3×10^{-3}	3·98	0·142	3·88	1·35	0·045	0·087	0·186

Evaluate h_i at 10°F giving 21 400 Btu/h-ft² degF for $f = 1$.
Evaluate k, α at 15°F, i.e. $k = 0{\cdot}310$ Btu/h-ft °F

$$\alpha = 1{\cdot}38 \times 10^{-6}\,\text{ft}^2/\text{s}.$$

$$h = \frac{h_i}{k} = 6{\cdot}19 \times 10^4\,\text{ft}^{-1}.$$

$$T_{b,e} - T_{g,\text{sat}} = 48{\cdot}5\,°\text{F}.$$

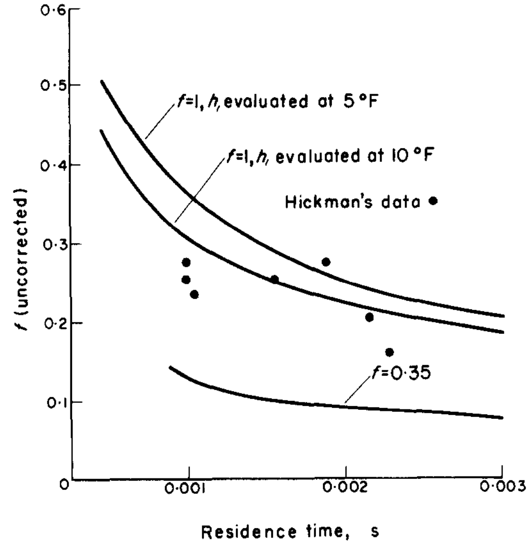

FIG. 4. Comparison of measured and predicted values of the uncorrected evaporation coefficient for Hickman's experiment.

falls off very rapidly this average temperature is close to the vapor temperature. One or two iterations are sufficient to decide on a suitable value of this temperature. Table 4 shows the results of computations for h_i evaluated at 10°F and $f = 1$. Figure 4 shows curves of $f_{\text{uncorrected}}$ against residence time for h_i evaluated at 10°F and at 5°F for $f = 1$ and for $f = 0{\cdot}35$, the value concluded by Hickman as the result of his experiment. It can be seen that $f = 1$ is indeed the best conclusion from the experimental data; $f = 0{\cdot}35$ yields evaporation rates which are 50 per cent of those measured while a value of $f = 0{\cdot}036$ would be impossible. The calculations also show that the surface temperature of the water must have dropped as low as 6°F. This is borne out by two experimental observations:

(i) The seven runs reported by Hickman were taken from well over one hundred attempts; in the rest, the jet iced up within 5 s preventing any measurements. The successful runs followed long periods of water deaeration and had durations of 20–496 s. It appears that the water could remain supersaturated without freezing provided that there were no nucleation sites arising from dissolved gas.

(ii) The temperature of the vapor adjacent to the jet was measured to be 5°F which compares well with the predicted surface temperature.

Résumé—La condensation par film de la vapeur d'eau à faible pression sur une plaque plane verticale a été étudiée expérimentalement afin de prouver l'existence d'une résistance interfaciale au transport de chaleur et d'en déduire une valeur du coefficient d'accommodation massique ou de "condensation" de l'eau. On présente des résultats pour la condensation de la vapeur d'eau saturée entre 7 et 10°C à des flux de chaleur compris entre 51,2 et 68,3 $kW/m^2 °C$. On a trouvé qu'il n'existait aucune résistance sensible au transport de chaleur et l'on en a déduit que le coefficient de condensation est compris entre 0,45 et l'unité. Le fait que le coefficient de condensation est au moins supérieur à 0,45 assure que la résistance interfaciale sera négligeable dans les applications industrielles de la condensation par film. L'incapacité de déterminer une valeur plus exacte pour le coefficient était due à une limitation inhérente à la technique et la gamme citée ne doit pas être prise afin de favoriser une valeur intermédiaire.

Zusammenfassung—Es wurde Filmkondensation von Wasserdampf an einer senkrechten Wand bei niedrigem Druck experimentell untersucht. Zweck der Untersuchungen war es, einen Wärmeübergangswiderstand an der Phasengrenze nachzuweisen und daraus einen Wert für den Kondensationskoeffizienten abzuleiten. Es werden Versuchsergebnisse wiedergegeben für die Kondensation von gesättigtem Dampf bei Temperaturen von 7,2°C bis 10°C und bei Wärmestromdichten von 50000 bis 68000 W/m^2. Es ergab sich kein nennenswerter Wärmeübergangswiderstand an der Phasengrenzfläche. Daraus kann geschlossen werden, dass der Kondensationskoeffizient zwischen 0,45 und 1 liegt. Bei industrieller Anwendung der Filmkondensation kann deshalb der Wärmeübergangswiderstand an der Phasengrenzfläche vernachlässigt werden. Den Kondensationskoeffizienten genauer zu bestimmen, war mit der verwendeten Versuchstechnik nicht möglich; aus den angegebenen Grenzen lässt sich nicht auf einen Mittelwert schliessen.

Аннотация—Экспериментально исследовалась конденсация пара типа пленочной при низком давлении на вертикальной плоской пластине для определения межфазного сопротивления теплообмену и, следовательно, для установления величин коэффициента аккомодации или «конденсации» воды. Представлены данные для конденсации насыщенного пара при температуре 45–50° и тепловых нагрузках 9000–12000 БТЕ/час-фут^2 °F. Обнаружено отсутствие значительного сопротивления на поверхности раздела, и установлено, что коэффициент конденсации равен 0,45–1. Величина коэффициента конденсации выше 0,45 гарантирует незначительное сопротивление на поверхности раздела в промышленных установках пленочной конденсации. Невозможность определения более точного значения коэффициента обусловлена ограничением методики.