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PRESSURE DROP WITH FORCED-CONVECTION BOILING OF POTASSIUM*

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ABSTRACT

The results are given for the pressure drop with boiling potassium in forced-convection flow through a vertical, circular tube. The data, which comprise a single series of measurements (Series D), were obtained in a 6-ft long $\times$ 0.270-in.-ID tube having pressure taps at 1-ft spacings over the final 4 ft of the channel. The pressure-measurement system and its deficiencies are discussed; the primary difficulty appears to be "aging" of the diaphragms in the pressure transmitters over long exposure times at high temperatures. The pressure distribution along the boiler was estimated using both homogeneous and Lockhart-Martinelli descriptions of the flow; the calculated exit pressures compared favorably with the measured values.

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INTRODUCTION

The pressure drop associated with forced-convection boiling is of significant concern in the design of power systems utilizing this mode of cooling and has been the subject of intensive study. Most attention has focused on water in a variety of geometries at high system pressures. More recently, interests have broadened to include lower pressure systems (including subatmospheric) and other coolants (such as the alkali liquid metals).

While we have been accumulating pressure-drop data in the course of our heat-transfer studies with boiling potassium in upward forced flow through vertical circular tubes, we have described our results only briefly at these conferences.¹ The discussion which follows considers some of the difficulties we have encountered in making these measurements and gives a preliminary analysis of the data.

EXPERIMENTAL APPARATUS

The test channels utilized in our initial studies with boiling potassium provided only limited information on the pressure drop. Thus, both the "low-flux" and "high-flux" boilers² were designed with pressure taps at the inlet and exit only. As a consequence, the resulting data^{1,3} included losses (of unknown magnitude) associated with flow through an inlet mixing chamber, around exit-end thermowells, and through contractions and expansions at the exit.

An advanced design of the "high-flux" boiler, incorporating an additional pressure tap at the midpoint, was abandoned because of fabrication difficulties; and a test unit of intermediate-flux capability was constructed and installed. This new boiler was designed to provide data on both the heat transfer and the pressure drop; a schematic representation is shown in Fig. 1. The boiler was machined from a 7-ft length of 1/2-in. double extra heavy IPS type 347 stainless steel pipe (0.270-in. inside diameter, 0.839-in. outside diameter). As shown in Fig. 1, the boiler was divided into six 12-in.-long sections which were heated radiantly by individual clamshell heaters of 1 1/4-in. inside diameter. The heated portions were separated by 1 1/2-in.-long gaps in which the wall was

Fig. 1. Schematic of Medium-Flux Boiler Section.

thinned from the original 0.285 in. to 0.075 in. Pressure taps (see Fig. 2 for construction detail) were located in the final four gaps and at the boiler exit. The pressures at positions 2, 3, 5, and 6 were measured with Taylor diaphragm-type pressure transmitters having NaK-filled lines and strain-gage signal converters; at position 4, the pressure was determined with a 6-in. bronze Bourdon-tube gage maintained at the calibration temperature of 160°F. Care was taken to insure that the pressure-tap lines leaving the boiler were horizontal and liquid filled. While cooling jackets were provided on all pressure taps, it was found in practice that the normal heat loss was sufficient to assure the absence of vapor in the lines.

A pair of 0.008-in.-diam Pt-Pt 10% Rh thermocouples were also welded to the tube wall in each of the interheater gaps; the leads of these thermocouples were brought out along the pressure-transmitting lines. Fiberfrax insulation in these regions created an essentially adiabatic zone. In addition, tube-wall temperatures were obtained at the midpoint of each heated section by Pt-Pt 10% Rh couples placed in 0.035-in.-diam holes drilled radially into the tube wall; the locations of the bottoms of these wells with respect to the inside tube surface are indicated in Fig. 2. The 8-mil thermocouple wires were threaded through two-hole ceramic insulators (0.031-in. outside diameter) and individually resistance welded to the stainless steel at the bottom of the well. The inside surface temperature was obtained by extrapolation from the temperatures measured at the bottom of the thermocouple wells.

The pressure-measuring system was calibrated at 1000°F over the range 4 to 25 psia by comparison with a precision test gage. During calibration, the liquid potassium was held in the sump tank; and the system was filled with helium. Calibration was effected both before and after each run series. In addition, the electrical part of the system was checked periodically by detaching the strain-gage leads and imposing a standard millivolt signal at this point.

The Taylor devices installed on this loop were found to be unstable, exhibiting significant changes in calibration over a period of time. For example, the output of one transmitter varied over a six-month period by nearly 12%. These changes were always in a positive direction and derive possibly from "aging" of the diaphragms due to prolonged exposure at high temperatures,

THERMOCOUPLE POSITION	WALL THICKNESS (in.)
1-7	$z = 0.075$
8	$x = 0.1210, y = 0.041$
9	$x = 0.2005, y = 0.051$
10	$x = 0.2146, y = 0.018$
11	$x = 0.2070, y = 0.054$
12	$x = 0.2106, y = 0.033$
13	$x = 0.2040, y = 0.021$

Fig. 2. Thermocouple and Pressure Tap Details of Medium-Flux Boiler.

occasional imposition (as during calibration) of subatmospheric pressures, and/or corrosion in the potassium environment. While this factor does introduce some uncertainty in the final results, the major effect has been annoyance rather than loss of precision.

The quality at the boiler exit was determined by passing the effluent vapor-liquid mixture through a separator and measuring the flow rates of the resulting vapor (after condensation) and liquid streams both volumetrically in calibrated hold tanks and dynamically by electromagnetic flow meters.

Other features of the experimental system have been described previously,^{1,3}

EXPERIMENTAL AND CALCULATED RESULTS

The data obtained in the Series D experiments for the axial pressure distribution with boiling potassium flowing through a vertical, circular tube are summarized in Table 1 along with the inlet mass flow, exit quality, and heat input. Results for two typical runs are shown graphically in Figs. 3 and 4. In the first of these (Run D-4, Fig. 3), a low mass flow-high exit quality condition existed; while in the second (Run D-15, Fig. 4), the situation was reversed. Both figures present measured and predicted pressures, fluid temperatures, and calculated qualities as a function of distance from the boiler inlet; the locations of the six boiler heaters are also indicated.

The quality variation along the boiler was calculated from an energy balance which neglected the wall friction and the static head but retained the kinetic energy term. A trial-and-error solution was then effected in which the magnitude of the slip ratio was systematically adjusted to generate an exit quality agreeing as closely as possible with the measured value; a zero inlet quality was assumed. The results are shown by the open circles in the lower portion of Figs. 3 and 4; the measured exit qualities are given by the filled circles. Significant changes in the slip ratio were found to have only minor effects on the calculated exit quality. This result is not surprising in that for the conditions of this experiment the enthalpy term dominates the kinetic energy term in the energy balance. As observed for the

Table 1. Measured and Calculated Pressures for Forced-Convection Boiling of Potassium within a Vertical Circular Channel.

Run No. (Series D)	Inlet Mass Flow (lb/hr)	Exit Quality (wt % Vapor)	Total Heater Power (Kw)	Measured Boiler Pressure (psia)					Calculated Exit Pressure (psia)	
				PE-6*	PE-5	PE-4	PE-3	PE-2	Lockhart- Mortelli	Homo- geneous
4	20.5	87.4	5.77	23.3	23.1	22.6	20.8	19.8	19.0	19.9
5A	70.1	19.4	3.84	21.1	21.0	20.4	19.2	17.7	14.4	13.0
6C	76.2	9.95	1.83	16.6	15.8	14.9	13.4	11.9	12.6	11.0
7	72.8	20.1	3.85	25.1	24.1	23.4	20.9	18.1	18.5	17.4
8	64.8	23.5	3.96	23.9	23.1	22.5	19.8	17.4	17.8	16.9
9	62.0	36.9	5.68	30.4	28.7	28.0	24.6	22.1	23.6	23.3
10	55.2	40.4	5.69	29.7	28.1	27.4	24.1	21.7	22.7	22.8
11	49.3	43.3	5.59	27.7	26.5	25.9	22.9	20.7	21.6	21.6
12	43.6	47.3	5.59	26.6	25.2	24.9	22.6	19.9	20.4	20.6
13	51.4	53.0	7.73	34.2	32.5	32.0	28.2	25.7	26.4	27.5
14	56.9	48.7	7.46	35.4	33.5	33.0	28.9	26.3	27.6	28.6
15	63.7	42.2	7.45	37.0	35.0	34.4	30.1	27.2	28.5	29.3

* Pressure tap locations, distances measured from boiler inlet:

PE-6, 2.156 ft; PE-5, 3.234 ft; PE-4, 4.313 ft; PE-3, 5.391 ft; PE-2, 6.469 ft.

Pressure (atm)

Temperature (°C)

Quality X (Dimensionless)

- △ Measured
- Post from T_{sot}
- ⊗ Lockhart-Martinelli
- ✕ Homogeneous

PRESSURE

TEMPERATURE

QUALITY

Heater Locations

I II III IV V VI

0 1 2 3 4 5 6 7

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PER L E 20

Position

two cases graphed, the calculated exit quality is always somewhat lower than the measured exit quality. Since the slope of the axial quality profile is fixed by the known heat input, the inlet quality could be obtained by extrapolation back from the measured value. The inlet qualities thus obtained were 15% for Run D-4 and 10% for Run D-15. These estimated values are markedly greater than the qualities believed, on a qualitative basis, to have existed at the boiler inlet and may reflect in part errors (on the high side) in the measured qualities due to vaporization between the boiler exit and the measurement station. In any event, the pressure-drop estimations discussed below were based on the calculated qualities without adjustment for the indicated deviation from the measured exit quality.

The central portion of both Figs. 3 and 4 show the fluid temperature distribution along the boiler. The data plotted are wall temperatures measured in the thin-walled (0.075 in.), adiabatic zones (1.5 in. long) between the heaters. It is presumed, under these circumstances, that the wall temperature is essentially the same as the fluid temperature.

Pressures measured at five positions along the boiler are shown by the open triangles in the upper portion of Figs. 3 and 4; as noted in Fig. 1, the first pressure tap (PE-6) was located in the adiabatic zone between the second and third heaters. The data show somewhat more scatter than previously observed in run Series C;⁴ this may result from the uncertainties associated with the pressure-measurement system as discussed above in the section on apparatus. An arbitrary smooth curve has been drawn through the data.

The lower pressure curve in both figures gives the saturation pressure (open circles) as derived from the fluid temperature data. The pressure distribution thus derived shows a trend with distance along the boiler that agrees closely with the curve through the measured values but falls significantly and consistently below the latter. This discrepancy in magnitude results in part from the use of the vapor pressure correlation of Lemmon et al.⁵ in this analysis; at a given T_{sat} , Lemmon and his co-workers found P_{sat} to be less than the value indicated by other available correlations. Thus, using the equation developed by Ewing et al.,⁶ the saturation pressure calculated at position PE-4

in Run D-4 is 1.47 atm; this compares with a value of 1.39 atm calculated from the Lemmon et al. equation and 1.54 atm measured. Similarly, for Run D-15 at the same measurement station, $P_{\text{sat}} = 2.14$ atm according to Lemmon, 2.22 atm according to Ewing, and 2.34 atm measured. The remaining difference between the measured pressure and that calculated from T_{sat} could derive from low temperature readings due to heat loss in the "adiabatic" zone or from a systemic error in the pressure measurements.

For purposes of comparison, the pressure drop along the boiler was also estimated using both the homogeneous and the Lockhart-Martinelli⁷ models for the flow. Thus, the filled triangles in Figs. 3 and 4 show the results obtained by assuming the fluid to be homogeneous, while the closed circles give corresponding values obtained by the Lockhart-Martinelli procedure with corrections for changes in the kinetic and gravity heads.

The Blasius friction-factor equation, $f_B = 0.0396 N_{\text{Re}}^{-0.25}$, was used in the homogeneous calculations; while the analogy relation, $f_B = 0.023 N_{\text{Re}}^{-0.20}$, was used in the Lockhart-Martinelli computation. The following further comments are pertinent to the estimation of the pressure drop by the Lockhart-Martinelli procedure:

1. Since the slip ratio was unknown, a value of unity was assumed in the calculations of the kinetic and gravity head corrections.
2. For convenience in the computer calculations, an empirical equation was fitted to the Lockhart-Martinelli parameters; thus,

$$\ln \phi_{\text{ltt}} = 1.478 - 0.5403 \ln X_{\text{tt}} + 0.05194 (\ln X_{\text{tt}})^2 + 0.000698 (\ln X_{\text{tt}})^3,$$

where ϕ_{ltt} is the square root of the ratio of the total two-phase pressure drop, $(dP/dL)_{\text{TP}}$, to the pressure drop for the liquid phase only, $(dP/dL)_\ell$; and

$$X_{\text{tt}} = \left(\frac{1}{x} - 1 \right)^{0.9} \left(\frac{\mu_\ell}{\mu_v} \right)^{0.1} \left(\frac{\rho_v}{\rho_\ell} \right)^{0.5},$$

x being the mean quality in each heated section. Over the range of the experimental data, $0.01 < X_{\text{tt}} < 1.00$, this empirical equation overestimates ϕ_{ltt} ; at $X_{\text{tt}} = 1.0$, the error is ~5%, and at $X_{\text{tt}} = 0.01$, ~10%.

3. Since the pressure was not measured at the boiler inlet, the starting point for the pressure calculations was assumed to be P_{sat} as estimated from the wall temperature measurements. This is probably not an unreasonable assumption in that the fairly long loop section immediately preceding the boiler is essentially adiabatic; and, hence, the wall temperature measured at the end of this region should equal the fluid temperature. The data shown in Figs. 3 and 4 suggest this conclusion.

DISCUSSION

In view of the error discussed above in the evaluation of $\phi_{g,tt}$, the Series I data have not been presented in the usual graphical form of the pressure-drop ratio versus the Lockhart-Martinelli parameter, X_{tt} . These calculations will be repeated in the near future using a newer, more precise computer procedure for evaluating $\phi_{g,tt}$. Thus, any comparison of the present data with results obtained with water under similar experimental conditions must be deferred.

The final three columns of Table 1 provide a comparison of measured and calculated exit pressures; in general, the agreement between the tabulated values is good. However, there does not appear to be any consistent trend in these comparisons. Thus, for Run D-4, Lockhart-Martinelli predicts an exit pressure less than measured, while the homogeneous model yields a value greater than measured. In Run D-8, the situation is reversed; while in Run D-15, both calculations indicate exit pressures greater than measured. As discussed above, these comments must be qualified somewhat in respect to the Lockhart-Martinelli calculation. An overestimate in the pressure drop by 15% on the average is indicated; this results in predicted exit pressure which are 3 to 5% low.

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