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SNAP MERCURY RANKINE PROGRAM
MERCURY CONDENSING EXPERIMENTS

AEC Research and Development Report

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SNAP MERCURY RANKINE PROGRAM
MERCURY CONDENSING EXPERIMENTS

By
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ERRATA

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SNAP MERCURY RANKINE PROGRAM MERCURY CONDENSING EXPERIMENTS

The following corrections and additions should be incorporated into your copy of the report:

- Page 5 18. "Series A Runs" (Plural)
- Page 13 Figure 3 Slip ratio should be ϵ in the legends.
Figure 4
- Page 23 Figure 7b w is a subscript
- Page 33 Second paragraph, end of first sentence should read:
....cases for untapered tubes.
- Page 37 First sentence, first paragraph, TRW^5 should be TRW^8
- Page 40 Title: a. Derivation.....Reference 8
- Page 41 5th line from top: "deviation" should be "derivation"
- Page 43 Line before Equation 36, should read:changes
Equation 35 to
- Page 43 Equation 36 $\frac{W_t^2 \psi}{\rho A^3} \frac{dA}{dx}$ instead of $\frac{W_t^2 \psi}{\rho A^2} \frac{dA}{dx}$
- Page 44 Between Equations 40 and 41 the line should read:
....and 40 into 37,....
- Page 47 Title..... "Series A Runs" (Plural)
- Page 77 Equation A4 and A5 A_T should be A_t
- Page 80 Line above Equation B-4 W_T should be W_t
Equation B-4 W_T should be W_t
Equation B-5 W_T should be W_t
- Page 81 Equation B-10 W_{in} should be W_t
- Page 85 Below Equation C-1: $\frac{VD}{f}$ should be $\frac{\rho VD}{\mu_f}$
- Page 86 Figures 36 and 37 ID should be OD
- Page 87 Figure 38 Title 0.33-in. OD tube
- Page 90 Figure 39 Ordinate callout should read: $(T_v - T)/(T_v - T_w)$
- Page 91 Figure 40 Ordinate callout should read: $(T - T_w)/(T_v - T_w)$

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ABSTRACT

An experimental study program of the various basic operating characteristics of mercury condensers for space applications is described. Included are techniques for predicting pressure drop in a variety of environments, condenser stability (both liquid leg stability and interface stability), the effects of noncondensibles, startup dynamics, and the differences between wetting and non-wetting condensing. Although the program is primarily based on obtaining accurate pressure-drop data in single horizontal tubes to simulate the zero-gravity of space operation, tilting and multi-tube capabilities are available.

The greater emphasis of the discussion concerns existing models for predicting pressure drop. A description of the partially completed test facility is given along with the basic requirements of various pieces of equipment necessary to obtain accurate data. Tentative test plans are presented.

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I. INTRODUCTION

The (SNAP 2) Mercury-Rankine powerplant utilizes mercury (Hg) as the working fluid in a Rankine cycle to generate electrical energy from a nuclear power source. The powerplant is composed primarily of a nuclear reactor heat source and a power conversion subsystem (PCS), supported by, and contained in, a radiator condenser (RC). Energy is produced in the nuclear reactor by the fission of U^{235} . A liquid-metal, sodium-potassium alloy (NaK) serves as the heat transfer fluid and is circulated through the reactor core and a Hg boiler by a thermoelectric (TE) pump, a pump with no moving parts. The reactor heat, transferred from the NaK to the Hg in the boiler, converts the liquid Hg into superheated vapor which is then expanded through a turbine; the resulting turbine mechanical power is converted to electrical power by an alternator. The Hg-vapor turbine exhaust is condensed in the RC and pumped back to the boiler. The Mercury-Rankine powerplant is a two-loop system: The primary loop is provided to circulate the NaK and the secondary loop circulates the Hg. The powerplant is shown in Figure 1. The thermodynamic design points of the powerplant are shown in Figure 2.

The condenser consists of forty, 120-in.-long tapered rectangular tubes. The inlet tube dimensions are 0.270 by 0.254 in.; the outlet tube dimensions are 0.270 by 0.125 in. A pressure drop of 2.25 psia occurs in the tubes for an inlet pressure of 8.45 psia. The heat of condensation and subcooling of 19.8 lbm/min is radiated from 120 ft² of fin area that is attached to the condenser tubes. Further information on the SNAP 2 Mercury-Rankine system is given in References 1 and 2.

Basic to the design of Hg condensers is the accurate prediction of pressure drop in the condenser tubes. This is essential if design points of the rest of the Mercury-Rankine system are to be met. Underprediction leads to higher operating pressures in the condenser; as a result, less power is available from the turbine. Although conservative prediction (overprediction) is desirable in this regard, the inherent stability of the condenser is degraded, since sufficient pressure drop is necessary for stabilization.³ Also, overprediction is wasteful in that weight penalties occur: unnecessarily large tubes are designed. Although Hg condensers for space operation will be subjected to essentially a zero-g environment,

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Figure 1. SNAP 2 Powerplant

occasional adverse accelerations may be experienced during orbital maneuvers. The effects of these accelerations on the pressure drop must be determined to evaluate the performance of the Rankine system under these conditions.

Noncondensibles cannot be entirely avoided in the system; there will be residual gas from pumpdown, further outgassing at operating temperatures, and hydrogen that leaks out of the reactor. The heat transfer characteristics of the condenser are expected to be affected, resulting in variations in condensing length and condensing rates along the tube. These variations in turn affect the pressure drop. For these reasons, accurate means for predicting pressure drop in condenser tubes under various conditions are desired.

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Figure 2. Schematic of SNAP 2 System with Thermodynamic Design Points

A. BACKGROUND OF PRESSURE-DROP STUDIES

The problem of predicting condensing pressure drop consists of two parts: the evaluation of the frictional losses, and the method of treating the momentum recovery. The first entails the use of an appropriate two-phase flow correlation; the second, an accurate description of the flow regime. Several correlations for the first are available, those of Lockhart and Martinelli,^{4,5} Kutateladze,⁶ Sanders and Baroczy,⁷ and Koestel.⁸

The Lockhart-Martinelli correlation considers only isothermal, constant-quality, two-phase flow. The frictional pressure gradient obtained by this correlation is an average over the length of a tube. When this correlation is applied to condensing, where the quality continuously changes, it is assumed the average gradient obtained by the correlation represents the local frictional gradient in the condenser. This correlation relates the ratio of the two-phase pressure gradient to the pressure gradient of the vapor flowing alone, to the ratio of the pressure gradients of the liquid and vapor that would result if each flowed alone (fictitious liquid and vapor pressure gradients). Recently, Sanders and Baroczy modified this correlation by including the Reynold's number of the vapor as a second parameter.⁷

The Kutateladze correlation relates the two-phase frictional pressure drop to the pressure drop of the liquid if it were flowing alone. The correlating parameters are the density ratio and the superficial velocity ratio (velocity of components flowing alone in the tube).

More recently, a correlation based on a fog-flow model has been proposed by Koestel. The two-phase flow is considered to consist of a homogeneous modified fluid to which the standard pressure gradient equation applies. It is modified in that the mixture density is used. The model also considers the reduction in flow area due to the condensed droplets on the tube surface. The local Weber number is used to correlate this reduction and the ratio of two-phase pressure gradient to the pressure gradient that would result if the vapor flowed alone.

An experimental investigation of the applicability of the first three correlations to Hg condensing was conducted recently at Electro Optical Systems (EOS).⁶ These correlations were compared with data taken in horizontal condensing tests. Flow rates up to 0.5 lbm/min with various inlet qualities between 11 and 100%

were partially and completely condensed in short tubes, 16-1/4 and 20 in. long. Inlet pressure was varied between 12 and 65 psia (as indicated by inlet temperature data). The comparison of all the data taken with predicted values is very poor on the whole, differing by as much as 1100% in one case using the Baroczy-Sanders correlation. Fair agreement is obtained, however, with some runs for both the Martinelli and the Baroczy correlations. For runs with inlet qualities greater than 60% and a condensed vapor fraction greater than 0.3, the correlations of Martinelli and Baroczy are within -50 and +10% and -25 and +30%, respectively (Figures 3 and 4). A moderate portion of this spread is due to the inability to determine the momentum of the liquid. For the purpose of calculation, the liquid slip ratio was taken to be zero and one. For the slip ratio equal to zero the Martinelli correlation appears to best represent the data (-25 to +10%), but for the slip ratio equal to one the Baroczy correlation appears to be better (-25 to +25%). High-speed photographic studies from the same program indicate that the slip ratio is closer to one than zero.

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Figure 3. Comparison of Pressure-Drop Data with the Predicted Pressure Drop Using the Martinelli Correlation (From Reference 6)

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Figure 4. Comparison of Pressure-Drop Data with the Predicted Pressure Drop Using the Baroczy Correlation (From Reference 6)

Comparison of the Kutateladze correlation with a limited number of test runs showed a significant amount of scatter. It is concluded⁶ that the results do not justify acceptance of the correlation without further testing.

The applicability of the Koestel fog-flow correlation has been experimentally investigated at Thompson Ramo Wooldridge⁹ (TRW). This correlation was compared with data taken in horizontal condensing tests. Flow rates up to 3 lbm/min, with 100% inlet quality, were completely condensed in long tubes (up to 8 ft). Inlet pressures were varied between 8 and 31 psia. In these tests, local average pressure gradients were obtained at various intervals along the tube. The reduced local values of the ratio of the two-phase frictional pressure gradient to the gradient that would be obtained with the vapor flowing alone (ϕ^2) are shown in Figure 5. Although considerable scatter is present in these logarithmic plots, the trend of the data is predicted well by the correlation.

The total pressure drops of Series A of these tests is compared in Figure 6 to the pressure drops predicted by a model based on the Baroczy correlation which is currently being used by Atomics International (AI) in condenser design. The predicted pressure drops are, on the average, lower than the measured. This model, however, contains a momentum loss factor that can be varied so that predicted pressure drop can be made to match measured pressure drop and design, therefore, will have some experimental basis. This factor is assumed to be zero in Figure 6.

On the whole, the various correlations do not seem adequate to accurately predict pressure drop in condensers. It is questioned whether the fault lies in the correlations, and models based on them, or in the experimental data itself. One of the purposes of this condensing program is to obtain accurate experimental data on pressure drop so that verification or modification of the models and/or correlations can be made.

Initially, the range of test variables will be confined to the pressure levels, flow rates, tube sizes, and condensing lengths that are typical of the SNAP 2 Mercury-Rankine condenser. Although the TRW experiments used similar condensing lengths, the tube sizes, pressure levels, and flow rates were significantly higher than those required of the SNAP 2 condenser. Upon completion of the design range series, the ranges of all variables will be broadened to obtain a more complete picture of condensing pressure drop.

Various tube geometries must be considered, since tapered round and rectangular tubes have been, and are currently being, considered in the development of lightweight condensers. Verification of the applicability of the predictive

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 Figure 5. Comparison of Experimental ϕ^2 Values of Reference 9 with the Fog-Flow Correlation of Koestel (From Reference 8)

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Figure 6. Comparison of Measured Pressure Drop to Predicted Pressure Drop Using the Baroczy-Sanders Correlations for Series A Data of Reference 9

models to these geometries is also desired. In addition to horizontal pressure-drop data, data will be taken with inclined tubes. Inclined tests simulate acceleration forces that may occur during maneuvers or vehicle tumbling in space. Although Hg initially does not wet the steel condenser tubes, wetting may occur under prolonged operation. Knowledge of the pressure drop under each condition requires investigation of both wetting and nonwetting conditions.

B. STABILITY

Analytical means are available for predicting the onset of liquid leg instability under zero-g, acceleration conditions in space operation, and ground test environments. Acceleration conditions will occur during orbital maneuvers such as orientation or docking, and under tumbling in the event the stabilization

equipment of the vehicle should fail. Limits of stable operation can be defined; however, the effects of instability on a multitube system are difficult to assess. It is possible that the Rankine system can tolerate minor instabilities. Recent testing of a flat horizontal RC in the prototype system mockup (PSM-3) has indicated that the behavior of the Mercury-Rankine flight design RC is stable. Because visual observation of this condenser is not possible in this test configuration, real confirmation of the analytical model of stability cannot be made. Experimental studies, with visual observation of multitube systems, are necessary to verify the analytical means for predicting the onset of liquid-leg instability and the effects of instability on the condenser.

In general, under nonwetting conditions, liquid-leg instability occurs before interface instability when the condenser is exposed to negative accelerations in flight. These instabilities are distinctly different. Interface instability occurs in the region of demarkation between the two-phase flow and the condensate leg and is a surface tension phenomenon. Liquid-leg instability is the unstable action of the subcooled condensate leg due to unbalanced forces imposed upon it. During ground testing, however, the interface may already be unstable. In ground testing for liquid-leg instability, the knowledge of a stable interface is required so that interfacial instability does not prejudice the tests.

The effects of interface instability under wetting conditions should also be determined. Since a wetting interface covers the entire portion of the condensing section, interfacial instability will affect the condensing flow regime. If slugging occurs, serious pressure fluctuations may result. In the event noncondensibles are present, entrainment of large bubbles that could adversely effect the operation of the pump must be designed against. This may also be a problem with nonwetting but it is believed to be less serious. Qualitative if not quantitative studies are required in this area.

C. NONCONDENSIBLES

Previous tests have shown that noncondensibles are generally located in front of a stable interface. Their presence increases the length needed for condensing, and consequently reduces the available subcooler length. Because vapor velocity is very small in this region slugging may become a problem under both wetting and nonwetting conditions. It is expected that the system will be sufficiently out-gassed prior to flight, however, hydrogen is continuously diffused

into the system by the reactor fuel elements and a portion of it is collected in the RC. Continuous entrainment of small amounts of hydrogen is desirable so that the partial pressure of hydrogen above the subcooler will be kept low and the potential bubbles formed by slugging will become small enough in the subcooler section to be easily swallowed by the pump. Entrainment of noncondensibles may also be encouraged by the vibration of the combined rotating unit (CRU). These problems must also be investigated.

D. STARTUP DYNAMICS

Some basic information is still to be obtained for the analysis of the transient preheat phase of the RC. The existence of choked flow at the entrance of the condenser tube, the existence of a vena contracta, its cross section and location, the condition of the vapor in this region, and the amount of pressure recovery that can be expected upon expansion is information necessary to the analysis of system startup.

System design no longer requires the use of an RC exit valve. During preheat, liquid Hg from the CRU-bearing drains fills the RC from the bottom. The existing method of analysis for evaluating the Hg inventory has been modified for this change. Tests simulating startup transients are needed to verify this model.

E. TEST FACILITY

To meet the objectives of this test program a versatile test system is being constructed. It is capable of the following:

- 1) The test section can be tilted for stability and pressure-drop investigations. The cooling system, fluoroscope assembly, and the associated instrumentation along with the test section can be tilted during operation.
- 2) An x-ray unit in conjunction with a fluoroscope screen will allow continuous observation of flow phenomena in single or multiple steel tubes. The assembly is capable of traversing the entire tube under any inclined position.
- 3) The Hg boiler is capable of generating at least 10 lbm/min so that large diameter multitube systems can be tested.

- 4) Up to 5000 cfm of cooling air is available. The cooling system is designed to provide uniform air velocities up to this rate, so that uniform condensation can be achieved in the tubes.
- 5) Accurate pressure transducers are used so that startup dynamics and the achievement of steady state can be observed.
- 6) Sampling means will be available to determine the noncondensable gas carryover through the subcooler region.

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II. DISCUSSION

In Mercury-Rankine systems of the SNAP 2 type, Hg vapor is condensed directly in an RC which is composed of an array of tubes bonded to the radiator surface. Manifolding evenly distributes the vapor to these tubes where complete condensation smoothly occurs. Under nonwetting conditions Hg vapor condenses on the tube surface and forms droplets that continue to grow until they become entrained in the flowing vapor. Observation of nonwetting condensation in glass tubes has shown that droplets are readily entrained along a good portion of the tube. Towards the end of the condensing portion some of the larger droplets coalesce and run along the bottom of the tube during ground testing. It appears that a fog type flow occurs over the bulk of the condensing length. Near the interface, the larger drops are observed to flow faster than the vapor, eventually impacting the interface with an appreciable velocity. Wetting conditions can occur in steel tubes, however, the detail of the flow is not as observable with x-ray techniques as nonwetting is with glass tubes. It is believed that a similar flow pattern exists; however, instead of droplets, an annular film of liquid exists. This film continuously breaks up to form droplets that become entrained in the vapor flow.

Static pressure drop in condensers is due to the acceleration of the vapor from the manifold into the tubes, the losses associated with the acceleration, the two-phase frictional losses along the tube, and the momentum recovery in the condensing process. This report is concerned with methods for predicting the last two. Three models for predicting this portion of the pressure drop are discussed in the following sections. In addition to condensing pressure drop, the stability of condensers, the effects of noncondensibles, and startup dynamics are discussed.

A. MOMENTUM EQUATION

Pressure drop in any model is appropriately analyzed by considering a control volume about the fluid in question and then developing the pressure-drop equation from it. This is necessary so that each term may be correctly defined in the momentum equation. The momentum equation for the fluid inside the control volume states that the forces on the control volume are equal to the

change of momentum of the fluid within the control volume. Pressure drop in the condensing process in typical Mercury-Rankine systems is essentially a one-dimensional problem, and one momentum equation along the axis of the tube (x direction) suffices to describe the system:

$$\sum \text{forces} = \text{Change in momentum of fluid within the control volume.}$$

The forces in the x direction can be:¹⁰

- 1) Pressure forces on the control volume surface whose resultant force is in the x direction;
- 2) Shear forces on the control volume surface whose resultant force is in the x direction;
- 3) Pressure and shear forces in the x direction on stationary surfaces extending into the control volume;
- 4) Pressure and shear forces in the x direction on moving surfaces within the control volume;
- 5) Body force on the fluid in question in the x direction.

The change in momentum is only for the fluid in question, that which is surrounded by the control surface. It includes:

- 1) The x component of the momentum flux into the control volume from all directions;
- 2) The x component of the momentum flux out of the control volume in all directions;
- 3) The time rate of change of the momentum within the control volume.

These possible forces on a differential control volume taken from a condenser tube are shown in Figure 7. The pressure forces are shown in Figure 7a. Here, an average pressure ($P + dP/2$) is imposed on the sides of the control volume. Although the real average may be represented with dP divided by any number other than 2, this representation suffices since the product of this term with the differential change in area (dA) is second order and will subsequently be neglected.

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Figure 7. Control Volume Forces

The shear forces on the sides of the control volume are shown in Figure 7b. The shear stress is the frictional shear of the system on the wall and for ducts is generally related to the velocity head by

$$\tau_w dA_w = \frac{f'}{D_e} \frac{\rho V^2}{2g_c} A dx, \quad \dots(1)$$

where f' is the friction factor. This equation is applicable to other geometries as well as round tubes. The hydraulic diameter can be used for the equivalent diameter D_e . Velocities however are based on the cross-sectional area.

A surface extending into the control volume is shown in Figure 7c. The surface of the control volume surrounds the extended surface so that it is not part of the system. Pressure and shear forces similar to those in Figure 7a and b can be applied to this boundary. In general, no extended surfaces exist in condenser tubes of the Mercury-Rankine system, so this force-producing mechanism need not be considered.

Figure 7d is an illustration of some moving surfaces within the outer boundaries of the control volume. Here, as in Figure 7c, the control surface surrounds these moving surfaces so that they are not included in the control volume. The pressure and shear forces on these surfaces are treated in the same way as in Figure 7a and b. Since these surfaces move in and out of the outer boundaries of the control volume, the force generated by their transit is time dependent. However, if a large number of moving surfaces continually pass through the outer boundaries of the control volume, the force generated can be considered independent of time and the control volume can be described as in Figure 7e. Here, the force generated by the continuous strings of droplets can be spread over the control surfaces connecting them. The force developed is related to the relative velocity between the moving surfaces and the velocity of the control volume fluid. The phenomenon of moving surfaces is typical of the Mercury-Rankine condenser, where Hg droplets are entrained in the flow.

If the moving surfaces or droplets are considered part of the control volume (i.e., no control surfaces are drawn about them as in Figure 7f) then the force generated by them cannot be considered in the sum of the forces on the control volume. The forces generated between the moving surfaces and the rest of the control volume fluid are reflected only in the change in momentum terms of the control volume. The fluid in the control volume, to which the momentum terms apply, is just a less homogeneous one. In the former case, where the moving surfaces were not included in the control volume and thus generated force on the control surfaces, the momentum of the moving surfaces is not included in the change of momentum terms.

The body force on the fluid in the control volume is presented in Figure 7g. Body forces in Mercury-Rankine systems are either gravitational or acceleration forces, and are represented by

$$B = \frac{a}{g_c} \rho_{cv} A, \quad \dots(2)$$

where a is the component of acceleration in the x direction. In the case where liquid is not included in the control volume, ρ_{cv} is the density of the vapor. The increased pressure drop due to the weight of the liquid is reflected in increased shear because the trajectory of the liquid droplets is modified by the inclusion of the acceleration term in the kinematics of the droplets. If the liquid is included in the control volume the density includes the liquid; however, the momentum change of the liquid is modified by the change in trajectory due to the acceleration term.

The momentum flux into and out of the control volume is shown in Figure 8.

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Figure 8. Control Volume Momentum

To be general, it is written as a summation, since two phases may be considered and each may have different velocities. In the case of Hg droplets, there may exist a spread of velocities. If only one phase, the vapor is taken to be the fluid within the control volume, the summations are over one component only, and the effects of the

liquid are reflected in the force terms. Actually the momentum of the vapor, written as WV , is only an approximation, since a velocity profile does exist across the tube and an integration is required. However, it is a good approximation for turbulent flow where the velocity profile can be represented by $(V/V_{\max}) = [1 - (r/r_o)]^{1/m}$, and where m is generally larger than 6. A plot of the ratio of the actual momentum to the momentum based on the bulk velocity is shown in Figure 9 for a range of values for m . Since a steady-state description is desired, the time rate of change of momentum within the control volume is zero. Including all the previous terms in the momentum equation leads to

$$PA + \left(P + \frac{dP}{2}\right)dA - (P + dP)(A + dA) - \tau_w dA_w - F_{is} dx - B dx$$

$$= \frac{1}{g_c} \left[\sum (W_i + dW_i)(V_i + dV_i) \right] - \frac{1}{g_c} \left[\sum V'_i dW_i \right] - \frac{1}{g_c} \left[\sum W_i V_i \right] \quad \dots (3)$$

Upon expansion, cancelling like terms, and neglecting second order terms, the equation can be reduced to

$$- A dP - \tau_w dA_w - F_{is} dx - B dx$$

$$= \frac{1}{g_c} \left[\sum W_i dV_i \right] + \frac{1}{g_c} \left[\sum (V_i - V'_i) dW_i \right] \quad \dots (4)$$

where F_{is} designates forces due to internal surfaces, and V'_i is the velocity of the i phase leaving through the side of the control volume.

Figure 9. Ratio of Actual Momentum to Momentum Based on Average Velocity for Velocity Profiles given by:
 $(V/V_{\max}) = [1 - (r/r_o)]^{1/m}$

B. PRESSURE-DROP MODELS

Three condensing pressure-drop models are discussed in the following sections. The first two use the improved Lockhart-Martinelli correlation of Baroczy; the third uses the Fog-Flow correlation of Koestel. Model No. 1 has been used with moderate success in the past by AI to predict the total pressure drop for design purposes. Experimental data has shown, however, that it does not predict the pressure profile along the tube very well. The profile is not as important though, for design purposes, as is the total pressure drop. Model No. 2 is a fog-flow model that uses the Baroczy correlation and predicts closely the same pressure drop as Model No. 1 but the pressure profile matches pressure drop data much better. The recently introduced Fog-Flow Model of Koestel has not been used by AI, as yet, to predict pressure drop. This model shows some merit, mainly due to its simplicity, and is being considered in this program.

1. Improved Lockhart-Martinelli Correlation of Baroczy and Sanders

To date, the popular method for predicting pressure drop in isothermal two-phase flow in horizontal pipes is based on the Martinelli method.^{4,5} This method relates the two-phase pressure drop to the pressure drop that would occur if the vapor phase were flowing alone by

$$\Delta P_{tp} = \phi^2 \Delta P_{vapor} \quad \dots(5)$$

The coefficient ϕ^2 is correlated to the Martinelli parameter χ^2 which is the ratio of the pressure drop of the liquid flowing alone to the pressure drop of the vapor flowing alone by

$$\chi^2 = \left(\frac{dP}{dx} \right)_{liquid} / \left(\frac{dP}{dx} \right)_{vapor} \quad \dots(6)$$

Depending on the Reynolds number of each phase the Martinelli parameter can represent four modes of flow:

- χ_{tt} - turbulent liquid, turbulent gas,
- χ_{vt} - viscous liquid, turbulent gas,
- χ_{vv} - viscous liquid, viscous gas,
- χ_{tv} - turbulent liquid, viscous gas.

In general, only viscous-turbulent and viscous-viscous modes occur within the Mercury-Rankine RC. The correlation of ϕ^2 with χ^2 was evaluated empirically for several wetting liquids flowing with air in horizontal glass tubes, where the flow was essentially developed throughout the test section.⁴ Since there are no momentum changes throughout the test section the pressure drop is completely attributed to friction.

This two-phase data has been successfully applied to predicting the frictional pressure drop in forced boiling flow by Martinelli and Nelson.¹¹ It is assumed that this data can also be successfully applied to condensing. Because the flow rates of both the liquid and gas are continuously changing along the tube, the assumption is made that the local frictional gradient in condensing flow is equivalent at each point to the pressure gradient measured over the length of the tube in the constant two-phase measurements of Martinelli for equivalent flow ratios:

$$\left[\frac{\left(\frac{dP}{dx} \right)_{tp}}{\left(\frac{dP}{dx} \right)_v} \right]_{\text{condensing}} = \left(\frac{\Delta P_{tp}}{\Delta P_v} \right)_{\text{constant quality}} \quad \dots (7)$$

at the same value of χ . This assumption is applied for the frictional portion of the condensing pressure-drop model described herein.

Since the properties of liquid Hg differ significantly from the fluids used in the discussed correlation, tests were conducted with Hg and nitrogen to certify that the same correlating technique was valid.¹² The correlation technique was verified; however, it was found to be improved with the introduction of another parameter, the vapor Reynolds number.⁷ The smoothed curves of this correlation are shown in Figure 10.

a. Condensing Pressure-Drop Model No. 1^{7,13}

For Model No. 1 the following assumptions are made:

- 1) The control volume includes both the liquid and vapor flow; however, the momentum of the liquid is assumed to be negligible, so that only the vapor is considered in the momentum terms.
- 2) The liquid volume fraction is very small so that the vapor flow area is taken to be the cross-sectional area of the tube.

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Figure 10. Two-Phase Factor of Baroczy and Sanders
(From Reference 7)

- 3) No body forces are present since space operation with no acceleration is assumed.
- 4) The velocity of the liquid is assumed to be negligible.

With these assumptions in mind, Equation 4 is written as

$$-A dP - \tau_w dA_w = \frac{1}{g_c} W dV + \frac{1}{g_c} V(1 - y) dw, \quad \dots(8)$$

where

B = zero

$$y \equiv \frac{V'}{V}$$

(Subscript i is omitted since only the vapor phase is considered).

The shear force term $\tau_w dA_w$ is related by

$$\frac{f'}{D_e} \frac{\rho V^2}{2g_c} A dx,$$

where

$$f' = f\phi^2$$

f = the single-phase friction factor based on the vapor flow alone.

In gradient form, Equation 8 is

$$-\frac{dP}{dx} - \frac{\phi_{2f}^2 \rho V^2}{D_e 2g_c} = \frac{1}{g_c A} \left[W \frac{dV}{dx} + V(1 - y) \frac{dW}{dx} \right]. \quad \dots(9)$$

When one introduces

$$V = \frac{W_t \psi^*}{\rho A}, \quad \dots(10)$$

$$W = W_t \psi, \quad \dots(11)$$

$$\frac{dW}{dx} = W_t \frac{d\psi}{dx}, \quad \dots(12)$$

$$\frac{dV}{dx} = \frac{W_t}{\rho A} \frac{d\psi}{dx} - \frac{W_t \psi}{\rho^2 A} \frac{d\rho}{dx} - \frac{W_t \psi}{\rho A^2} \frac{dA}{dx}, \quad \dots(13)$$

$$\frac{d\phi}{dx} = \frac{d\rho}{dP} \frac{dP}{dx}, \quad \dots(14)$$

Equation 9 becomes

$$\frac{dP}{dx} \left(\frac{W_t^2 \psi^2}{A^2 \rho^2 g_c} \frac{d\rho}{dP} - 1 \right) = \frac{\phi_{2f}^2}{D_e} \frac{W_t^2 \psi^2}{2\rho A^2 g_c} + \frac{W_t^2 \psi}{\rho A^2 g_c} (2 - y) \frac{d\psi}{dx} - \frac{W_t^2 \psi^2}{\rho A^3 g_c} \frac{dA}{dx}. \quad \dots(15)$$

With the knowledge of the vapor pressure at a particular point, the vapor temperature, density, and variation of density with pressure are known. Also at the particular point, the quality, quality gradient, area, area gradient, and the equivalent diameter are known. The fictitious liquid and vapor pressure gradients and the vapor Reynolds number can be calculated for this point so that ϕ^2 can be evaluated. With the foregoing, and an assumed value for y , the

* ψ is used for vapor quality throughout this report instead of the standard X so that it cannot be confused with the Martinelli parameter χ or length x .

pressure gradient can be evaluated. A node-wise integration of this sort allows one to calculate the pressure drop in a condenser tube.

In the derivation of the momentum equation for Model No. 1, a momentum term, $\frac{1}{g_c A} V(1 - y) \frac{dW}{dx}$, was obtained. This term represents the momentum that can be recovered from the vapor that is about to be condensed. If the vapor molecule at the main stream velocity (V) leaves the control volume at an axial velocity greater than zero, less momentum becomes available to the rest of the main stream. Physically, the molecule, with an initial stream velocity (V), loses some of its momentum to the main stream, and collides with the condensing surface with a velocity V'_v . If V'_v is equal to V , ($y = 1$), no momentum can be recovered. The momentum recovery which does occur is due to the diffusion of the noncondensing vapor at that point. If V'_v is zero ($y = 0$) all the momentum is recovered. Since a molecule must make its way through the boundary layer to be condensed, it is thought that it has ample opportunity to give up most of its momentum in the boundary layer before colliding with the condensing surface.

Although it is believed that y is close to zero, y remains as a correction parameter in the pressure-drop computer program. The purpose of the y factor is to match the predicted pressure drop with experimental data so that condenser tube design has some experimental basis. A value of y equal to 0.57 has been used in the past for design purposes. This value was obtained in the attempt to match some low-pressure (approximately 8 psi inlet) condensing runs tested at TRW.⁹ An average for the four runs was used. The measured and predicted pressure profiles for these runs are shown in Figure 11.

However, y is not a constant correction factor. Evaluation of y for various other runs at higher pressures indicates a decrease in y with increase in pressure (Figure 12). Since the mechanism necessary to make y a function of pressure is not obvious, one is lead to suspect the ζ^2 correlation of the model or the experimental data itself. If the y factor is truly a function of pressure, as indicated, it should also vary along the tube length. Inclusion of this variation just further complicates the model. This question can only be resolved with accurate pressure-drop tests.

The y -type loss may be considerable in a very rapid condensing process. When the vapor condenses rapidly the boundary layer is "sucked" away

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Figure 11. Comparison of Predicted Pressure Profiles with $y = 0.57$ Using Model No. 1 for Low Pressure Series A Data of Reference 9

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Figure 12. Momentum Loss Factor (y) vs Pressure. Used to Correlate Model No. 1 Prediction with Series A Data of Reference 9

to some extent, making it possible for vapor to leave the control volume at an appreciable velocity. This may occur during startup transients when the temperature of the RC is extremely low.

Considerations of Model No. 1

In Model No. 1 only the vapor was considered in the control volume. The presence of the liquid merely served to increase the frictional portion of the pressure drop; no account was taken of the forces on the control volume due to the change in momentum of the liquid along the tube. Only the increase in the frictional wall shear due to the presence of the liquid is reflected in the ϕ^2 factor. To be consistent with the Martinelli correlation, no momentum changes can be included in the ϕ^2 factor. Two momentum terms are missing from the momentum equation: the change in the velocity of the liquid flowing in the middle of the tube as it tries to follow the velocity of the vapor; and the entrainment term for the locally condensed liquid. The effect of the first of these terms is to increase the static pressure due to the deceleration of the liquid. Use of the second term decreases the static pressure since work is necessary to entrain the condensate from the wall.

The effects on the integrated pressure drop of neglecting the liquid are not as serious as one might think, since the momentum recovered is the same in both cases. This is discussed later in this section.

The momentum equation for Model No. 1 should be

$$-\frac{dP}{dx} - \frac{\phi_f^2}{D_e} \frac{\rho V^2}{2g_c} + F_1 + F_2 = \frac{1}{g_c A} \left[W \frac{dV}{dx} + V(1 - y) \frac{dW}{dx} \right], \quad \dots(16)$$

where F_1 and F_2 are the two previously mentioned force terms in pressure gradient form. F_1 is the force on the control volume due to the decelerating liquid already entrained in the flow, and F_2 is the retarding force on the control volume due to the entrainment of the locally condensed liquid. Since the liquid starts from zero velocity and eventually ends up with zero velocity, the momentum gain upon entrainment, represented by F_2 , is given up upon deceleration in its transit toward interface, which is represented by F_1 . Assuming that the velocity of the liquid follows the vapor velocity, the liquid velocity is zero at

the interface and all its momentum is recovered. If, however, the slip ratio is greater than one, not all of the momentum can be recovered in the two-phase region, but it is recovered upon impact with the interface. Frictional losses, however, may occur in the liquid at the interface and somewhat reduce this recovery. Assuming that these losses are negligible, no real harm is done by neglecting to add F_1 and F_2 to the equation, since their integration from inlet to a point just inside the interface results in a value of zero. Since the momentum changes of the liquid are equal to these forces, the forces can be represented in the equation by including the liquid on the momentum side.

A derivation of the momentum equation, including the momentum of the liquid, follows.

b. Derivative of Momentum Equation for Model No. 2

When both the liquid and vapor are considered, Equation 4 becomes

$$-A dP - \tau_w dA = \frac{1}{g_c} \left[W_v dV_v + W_l dV_l + (V_v - V'_v) dW_v + (V_l - V'_l) dW_l \right], \quad \dots(17)$$

where the subscripts v and l designate the vapor and liquid phases. Here the shear force term is assumed to be represented by the Baroczy and Sanders correlation. The velocity of the liquid is associated with the velocity of the vapor via the slip ratio $V_l/V_v = \epsilon$. In gradient form, Equation 17 is

$$-\frac{dP}{dx} - \phi^2 f \frac{\rho V^2}{2g_c D_e} = \frac{1}{g_c A} \left[W_v \frac{dV_v}{dx} + W_l \left(\epsilon \frac{dV_v}{dx} + V_v \frac{d\epsilon}{dx} \right) + (V_v - V'_v) \frac{dW_v}{dx} + (\epsilon V_v - V'_l) \frac{dW_l}{dx} \right] \quad \dots(18)$$

where V'_l represents the velocity of the condensed liquid entering the control volume. Since the liquid has essentially no initial velocity of its own, and must be entrained by the vapor, V'_l can be omitted. The slip ratio and its gradient must come from experimental data. It is generally expressed as

$$\epsilon = \frac{\rho_\ell}{\rho_v} \frac{\psi}{1-\psi} \frac{1-R_\ell}{R_\ell}, \quad \dots(19)$$

where the liquid volume fraction (R_ℓ) is obtained experimentally. Assuming for simplicity that the slip ratio is unity (fog flow), Equation 18 reduces to

$$-\frac{dP}{dx} - \phi^{2f} \frac{\rho V^2}{2g_c D_e} = \frac{1}{g_c A} \left(W_t \frac{dV_v}{dx} - \frac{dW}{dx} y V_v \right), \quad \dots(20)$$

since

$$W_v + W_\ell = W_t,$$

and

$$dW_v = -dW_\ell.$$

When one introduces Equations 10 through 14, Equation 20 becomes

$$\begin{aligned} \frac{dP}{dx} \left(\frac{W_t^2 \psi}{\rho^2 A^2 g_c} \frac{d\rho}{dP} - 1 \right) = \frac{1}{g_c} \left(\phi^{2f} \frac{W_t^2 \psi^2}{D_e^2 \rho A^2} + \frac{W_t^2}{\rho A^2} \frac{d\psi}{dx} \right. \\ \left. - \frac{W_t^2 \psi}{\rho A^3} \frac{dA}{dx} - \frac{W_t^2 \psi}{\rho A^2 g_c} y \frac{d\psi}{dx} \right) \quad \dots(21) \end{aligned}$$

Note the difference in the momentum equation in Equations 9 and 20. The first right hand term of Equation 20 includes the total weight flow, whereas in Model No. 1 only the vapor flow is included. Since the vapor flow decreases along the length of the tube, less momentum is recovered in the first term of Model No. 1 for the same velocity gradient. Assuming y equal to zero for both models, the second term in Model No. 2 is zero and in Model No. 1 it is equivalent to the first term. Figure 13 is a normalized plot of the momentum terms for the two models vs vapor quality. For simplicity, incompressible flow in untapered tubes is assumed. Although the total momentum to be recovered is the same for both models (same area under curves) greater momentum recovery occurs at first

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Figure 13. Momentum Recovery Terms
for Incompressible Flow vs Quality.
Comparison of Model No. 1
with Fog Type Flow

in Model No. 1, thus reducing the slope of the pressure profile. Towards the end of the condenser, little momentum recovery is left. In Model No. 2, the momentum recovery is spread evenly over the length of the tube. The initial pressure gradient is steeper than in Model No. 1, but towards the end it is less steep. In most cases the momentum recovery towards the end is larger than the frictional gradient so that the pressure gradient

becomes distinctly positive, with the minimum pressure located approximately two-thirds of the condensing length.

In incompressible flow, either model would suffice for the total pressure drop since ϕ^2 , V , etc., would vary similarly along the tube and the momentum recovery is the same. But in compressible flow, the density drops more rapidly in Model No. 2 since the pressure gradient is steeper. The local vapor flow rates, however, are similar, so that higher velocities occur in Model No. 2, hence an unrecoverable steepening in the pressure gradient. The overall effect is to increase the pressure drop over that predicted by Model No. 1. The increase is more noticeable when the ratio of the pressure drop to the pressure level is large.

It must be stressed that the ϕ^2 factor cannot reflect the force of the liquid on the control volume when the control volume includes the vapor only. To be consistent with the Martinelli correlation, ϕ^2 only reflects the increase in pressure drop due to the presence of the liquid. It is as if the combination of the liquid and vapor produced a modified fluid whose shear stress on the wall can be represented by the pressure-drop equation:

$$\left(\frac{dP}{dx}\right)_m = f_m \frac{1}{D_{e_m}} \frac{1}{2} \rho_m V_m^2, \quad \dots(22)$$

where m stands for modified; ϕ^2 then represents

$$\phi^2 = \frac{f_m \frac{1}{D_{e_m}} \frac{1}{2} \rho_m V_m^2}{f_v \frac{1}{D_{e_v}} \frac{1}{2} \rho_v V_v^2}. \quad \dots(23)$$

2. Fog-Flow Condensing Pressure-Drop Model

A new means for predicting the frictional pressure gradient in condensing flow has been proposed by A. Koestel of TRW.⁵ This model considers a more accurate physical description of Hg condensing flow. Due to the simplicity of the model an analytic value of the two-phase factor is achieved:

$$\phi^2 = \left(\frac{D_t}{D_c} \right)^{4.75} \frac{1}{\psi^{0.75}}, \quad \dots(24)$$

where

$$\left(\frac{D_t}{D_c} \right) = f(We_t), \quad \text{the tube Weber number.} \quad \dots(25)$$

The condensing process with nonwetting Hg consists of the growth of droplets on the surface of the condenser tube. When the droplets have reached some critical size they are entrained in the main stream flow, the core. Here, core is the tube area reduced by the layer of droplets that grow on the tube surface. There is experimental evidence to show that the critical droplet size is dependent on the local velocity head of the main flow and on the ratio of the droplet diameter to tube diameter (curvature effect). The droplet size is correlated by its Weber number (We_δ) which is a function of the diameter ratio and tube diameter (Figure 14):¹⁴

$$We_\delta = \frac{\rho V^2 \delta_{cr}}{2\sigma g} = 4E_\sigma = f\left(\frac{\delta_{cr}}{D}, D\right), \quad \dots(26)$$

where σ is the surface tension of Hg.

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Figure 14. Droplet Weber Number in Terms of E_σ for Entrainment of Droplets into Nitrogen Flow from Tube Surface (From Reference 14)

For small droplet sizes ($\delta_{cr}/D < 0.03$), the value for E_σ approaches the constant value $E_\sigma = 0.0464$ that was obtained for the tilted plate test (Figure 15)¹⁴ where:

$$\frac{g}{g_c} \sin \theta \rho_f \frac{2}{3} \frac{\delta_{cr}^2}{\sigma} = 4E_\sigma \quad \dots(27)$$

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Figure 15. Droplet Size Prior to Movement Down a Tilted Glass Plate (From Reference 14)

Since droplets of these sizes will occur over a good portion of the tube (typically 80%) it is expedient to consider E_σ as constant.⁸ Since ρV^2 is a smooth decreasing function of the distance along the tube, a smooth increase in droplet size occurs along the tube. This Weber number relationship is questionable near the interface where the vapor velocities are low and the predicted drop size approaches infinity (Figure 16). But along most of the tube where velocities are high, this relationship is expected to hold and a model for this portion can be generated. Most of the frictional pressure drop occurs in this portion so that this model may be justified.

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Figure 16. Nonwetting Mercury Droplet Size as a Function of Vapor Velocity Along a Tube

The droplets of critical size become entrained in the core flow and it is assumed that the droplets rapidly achieve the core velocity and maintain the local core velocity along the tube (i.e., slip ratio equal to 1). A study of the trajectory of droplets entrained in a typical Hg condenser showed this to be a fairly good assumption (Figure 17). Since the droplets follow the core flow it is expedient to consider both the liquid and vapor in the core as the fluids for the control volume as in Figure 7f.

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Figure 17. Entrained Droplet Trajectory Along Tube

The core flow is assumed to act as a modified fluid having a density (ρ_m), a viscosity (μ_m), a core diameter (D_m), and to behave in the same manner as conventional fluids. The friction factor is based on the Blasius relationship with Reynolds number for the mixture Re_m .

In Equation 24, ϕ^2 is shown to be a function of the diameter ratios. This relationship was developed for round tubes, but cannot be used for rectangular tubes where an equivalent hydraulic diameter might be substituted. The development of ϕ^2 for rectangular tubes is treated later in this section. The derivation for round tubes is presented in the following.

a. Derivation of ϕ^2 from Reference 5

The ratio of the two-phase frictional pressure gradient in the core to the frictional pressure gradient of the vapor flowing alone in the bare tube is ϕ^2 . The subscript m designates the properties of the core flow:

$$\phi^2 = \frac{\left. \frac{dP}{dx} \right|_m}{\left. \frac{dP}{dx} \right|_v} = \frac{\frac{1}{2} \frac{f_m}{D_m} \frac{\rho_m V_m^2}{g_c}}{\frac{1}{2} \frac{f_v}{D_t} \frac{\rho_v V_v^2}{g_c}} \quad \dots (28)$$

Since the density ratio ρ_ℓ/ρ_v is extremely large (at least 3000 to 1 and as high as 8000 to 1) for the pressure levels of interest, the ratio of the velocity of the

mixture in the bare tube to the velocity of vapor flowing alone is practically one. Therefore the ratio of the velocity of the mixture in the core to the velocity of the vapor flowing alone in the tube is proportional to $(D_t/D_m)^2$.

The ratio of the densities ρ_m/ρ_v is very closely proportional to $1/\psi$. The deviation of this relationship is shown in Appendix A. The ratio of the friction factors for turbulent flow is

$$\frac{f_m}{f_v} = \frac{0.316/\text{Re}_m^{0.25}}{0.316/\text{Re}_v^{0.25}} = \left(\frac{\rho_v V_v D_t / \mu_v}{\rho_m V_m D_m / \mu_m} \right)^{0.25} \quad \dots(29)$$

Since viscosity is volume fraction dependent the mixture viscosity is approximately equal to the vapor viscosity ($\mu_m \cong \mu_v$). Equation 29 becomes

$$\frac{f_m}{f_v} = \left(\psi \frac{D_m}{D_t} \right)^{0.25} \quad \dots(30)$$

Introducing the ratio values leads to

$$\phi^2 = \left(\frac{D_t}{D_m} \right)^{4.75} \psi^{-0.75} \quad \dots(24)$$

Assuming that the thickness of the layer of droplets at any point is equal to the critical drop size, the core diameter is

$$D_m = D_t - 2\delta_{cr} \quad \dots(31)$$

The critical drop size is related by

$$\delta_{cr} = \frac{8E_\sigma \sigma g_c}{\rho_v V^2} \quad \dots(32)$$

Furthermore, the ratio of the diameters can be related to the tube Weber number by further algebraic manipulation:

$$We_{\text{tube}} = \frac{D_t \rho_v V_v^2}{2g_c \sigma} = \frac{8E_\sigma}{\left(\frac{D_t}{D_m}\right)^4 - \left(\frac{D_t}{D_m}\right)^3}, \quad \dots(33)$$

so that

$$\phi^2 \psi^{3/4} = f(We_t). \quad \dots(25)$$

This relationship is unnecessary for the computation of ϕ^2 , since the direct evaluation of the critical drop size is more expedient. It is, however, a good method for displaying ϕ^2 data. This relationship, along with measured $\phi^2 \psi^{3/4}$ values of Reference 9, is shown in Figure 5. Although data scatter is present, the trend in the data is predicted well by this model. Both wetting and non-wetting condensing are shown, and it appears that pressure drop under wetting conditions can be predicted as successfully as that of nonwetting condensing.

b. Pressure-Drop Equation for Fog-Flow Model

The assumptions for the Fog-Flow Model are the same as for Model No. 2:

- 1) The control volume includes the vapor flow and the entrained droplet flow which, under steady-state conditions, is the total flow. The momentum term includes both the liquid and vapor.
- 2) The flow area for the momentum terms is the tube area. Reduction of the cross-sectional area due to the droplets on the wall is not considered.
- 3) No body forces are present.

With these assumptions in mind, Equation 4 is written as

$$-A dP - \tau_w dA_w = \frac{1}{g_c} (W_t dV + V_y dw_v), \quad \dots(34)$$

where

$F_{is} = \text{zero}$ since no extended surfaces are present and the entrained droplets are part of the control volume

$B = \text{zero}$

$$\sum W dV = W_t dV, \text{ since } dV_l = dV_v \text{ and } W_l + W_v = W_t$$

$$\sum V_i dW_i = 0, \text{ since } dW_l = -dW_v \text{ and } V_v = V_l$$

$$y = (V'_v / V_v)$$

$$V'_l = 0.$$

The shear force term $\tau_w dA_w$ is related by

$$\frac{f'}{D_e} \frac{\rho V^2}{2g_c} A dx,$$

where $f' = f\phi^2$ and f is the single-phase friction factor based on vapor flowing alone in the bare tube. In gradient form, Equation 32 is

$$-\frac{dP}{dx} - \frac{\phi^2 f}{D_e} \frac{\rho V^2}{2g_c} = \frac{1}{g_c A} \left(W_t \frac{dV}{dx} - V_y \frac{dW}{dx} \right). \quad \dots(35)$$

Introducing Equations 10 through 14 changes Equation 33 to

$$\frac{dP}{dx} \left(\frac{W_t^2 \psi}{\rho^2 A^2 g_c} \frac{d\rho}{dP} - 1 \right) = \frac{1}{g_c} \left(\frac{\phi^2 f}{D_e} \frac{W_t^2 \psi^2}{2\rho A^2} - \frac{W_t^2 \psi}{\rho A^2} y \frac{d\psi}{dx} - \frac{W_t^2 \psi}{\rho A^2} \frac{dA}{dx} + \frac{W_t^2}{\rho A^2} \frac{d\psi}{dx} \right) \dots(36)$$

Note that this equation is the same as that of Model No. 2 except for the value of ϕ^2 . As in Section II.B.1, a node-wise integration can be made with this equation to obtain the pressure drop along a tube.

c. Pressure Drop in Rectangular Tubes

Following the method of Koestel, a value for ϕ^2 can be obtained for rectangular tubes:

$$\phi^2 = \left(\frac{\rho_{v_t} V_{v_t} D_{v_t}}{\rho_{m_c} V_{m_c} D_{m_c}} \right)^{0.25} \left(\frac{D_{v_t}}{D_{m_c}} \right) \left(\frac{\rho_{m_c}}{\rho_{v_t}} \right) \left(\frac{V_{m_c}}{V_{v_t}} \right)^2 . \quad \dots(37)$$

The density ratio is related by:

$$\frac{\rho_v}{\rho_m} = \frac{1}{\psi} . \quad \dots(38)$$

The velocity ratio is equal to the ratio of the flow areas

$$\frac{V_{m_c}}{V_{v_t}} = \frac{A_t}{A_c} . \quad \dots(39)$$

The diameter ratio, in the case of geometries other than round tubes, is the ratio of the hydraulic diameters, where the hydraulic diameter is defined by

$$D_h = \frac{4A}{p} . \quad \dots(40)$$

Inserting Equations 38, 39, and 40 into 35, changes ϕ^2 to

$$\phi^2 = \frac{1}{\psi^{3/4}} \left(\frac{p_c}{p_t} \right)^{1.25} \left(\frac{A_t}{A_c} \right)^3 , \quad \dots(41)$$

where

$$\frac{p_c}{p_t} = \frac{2(w + h - 4\delta_{cr})}{2(w + h)} \quad \dots(42)$$

and

$$\left(\frac{A_t}{A_c} \right) = \frac{wh}{(w - 2\delta_{cr})(h - 2\delta_{cr})} \quad \dots(43)$$

d. Low-Velocity Region

Close to the interface (roughly the last 20% of the condensing length) the predicted diameter of the entrained droplet becomes very large, eventually exceeding the diameter of the tube and approaching infinity at the interface where the velocity is zero. A typical case is shown in Figure 16. In this region, ϕ^2 rapidly approaches infinity due to the reduction in the core diameter (D_c), and then becomes incalculable due to the negative value for the core diameter ($D_c = D_t - 2\delta_{cr}$). As a temporary remedy for this problem, limits have been set on the droplet diameter so that this model can be currently used to predict pressure drop. For the predictions presented in this report, ($\delta_{cr} \leq 0.1D_t$) has been used.

Furthermore, the Reynolds numbers for this region are generally less than 2000, indicating that viscous flow exists. A laminar fog-flow, two-phase factor can also be obtained by the same method. In the development, the quality term vanishes and ϕ^2 becomes a function of (D_t/D_c):

$$\phi^2 = \left(\frac{D_t}{D_c} \right)^4 \quad \dots(44)$$

But still, one is plagued with the same problem of the diameter ratios. The assumption $u_m = u_v$ is questionable here also.

Further analysis and experimental observation of the entrained droplet sizes and growth of the surface droplets will result in the modification of the value of the droplet Weber number so that the same development for ϕ^2 may be used. Or possibly a Martinelli correlation can be used in this region.

e. Flow Regime Under Wetting Conditions

Under wetting conditions, an annular film of condensate exists on the tube. This film is prone to instabilities, wherein waves are produced that continue to grow along the tube until they break up into the core flow or until they bridge the tube and form slugs. The subject of film stability is a complex one and most investigations have considered only constant quality isothermal conditions. Film stability during condensing is further complicated by mass and heat transfer. A discussion of film stability as it is related to the condensing process is found in References 9 and 15.

It has been proposed by A. Koestel that the film breaks up rapidly and produces a fog in the core. It is also assumed that the reduction in flow area is similar to that under nonwetting conditions so that the same correlation may be used. Pressure-drop tests^{9,15} conducted under wetting conditions indicate that this correlation can be used to predict wetting condensing pressure drop as well as it does nonwetting. The experimental data is compared with the correlation in Figure 5c.

3. Comparison of Models

A comparison of the three models (with y equal to zero) with experimental pressure drops taken from Reference 9, is shown in Figure 18. The effect of the difference in the momentum terms of Models No. 1 and 2 is clearly seen. The pressure gradient for Model No. 1 is not as steep due to the high initial momentum recovery, and the profile is fairly flat towards the end of the tube where the frictional gradient and momentum recovery are small. The opposite is true of Model No. 2, the initial gradient is steep but the recovery towards the end is large.

The Fog-Flow pressure profile shows the same trends as does Model No. 2, since their equations are similar except for the evaluation of the two phase factor, ϕ^2 . Note also that the experimental data shows the same trends as Model No. 2 and the Fog-Flow Model, a steep initial profile, a minimum point, and a considerable rise in pressure. The difference between Model No. 2 and the Fog-Flow Model is attributed to the larger values of ϕ^2 in the latter. Note that the gradients near the interface are steeper for Model No. 2 than the experimental data and less steep for the Fog-Flow Model. This is attributed to the lower ϕ^2 factor in the improved Lockhart-Martinelli correlation, and the extremely high ϕ^2 factors in the Koestel correlation. The ϕ^2 inferred from the data appears to be somewhere between the two. Some of the discrepancies between the models and the data may be due to flow measurement error. This data must be verified by accurate pressure-drop tests, the results of which will be used to verify or modify the models currently being considered.

C. EFFECT OF ACCELERATION ON PRESSURE DROP

When axial acceleration or body forces are imposed on the condenser, an additional force term

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Figure 18. Comparison of Pressure Profiles Predicted by Models 1, 2 and Fog Flow Model with Arbitrarily Selected Series A Run of Reference 9

$$\frac{a}{g_c} \rho_{cv} A dx$$

is present in the momentum equation. If the vapor alone is considered as part of the control volume, ρ_{cv} is the density of the vapor. The effect of the body forces on the liquid are reflected in the change in the interaction of the liquid with the control volume from that under zero-g conditions. For illustration purposes, assume that under zero-g the slip ratio is unity. When the tube is tilted so that gravity opposes flow (or a negative acceleration due to orbital maneuvers), the droplets of liquid are retarded somewhat so that the slip ratio is less than one. An increase in the interaction with the control volume then occurs due to the relative velocity between them. Thus, F_1 and F_2 are increased. The momentum equation for vapor alone is

$$-\frac{dP}{dx} - \phi^2 f \frac{1}{2} \frac{\rho V^2}{g_c} + F'_1 + F'_2 - \frac{a}{g_c} \rho_v = \frac{1}{g_c} [W_v dV_v + V_v(1 - y) dW_v], \dots (45)$$

where the primes designate the increased interaction due to the body forces on the liquid. Here again, F_1 and F_2 are equal to the sum of the momentum changes of the liquid and the body force on the liquid. By including the liquid in the control volume, the momentum change and body force of the liquid are separated in the equation:

$$-\frac{dP}{dx} - \phi^2 f \frac{1}{2} \frac{\rho V^2}{g_c} - \frac{a}{g_c} \rho_{cv} = \frac{1}{g_c} [W_v dV_v + V_v(1 - y) dW_v + W_\ell dV_\ell + V_\ell dW_\ell], \dots (46)$$

where

$$\rho_{cv} = R_v \rho_v + R_\ell \rho_\ell. \dots (47)$$

The change in the trajectory of the liquid due to these body forces must be obtained before the evaluation of the liquid momentum term can be made. A kinematic analysis backed by experimental work is necessary for this evaluation.

The value for ϕ^2 is not expected to remain constant from a condition without body forces to one where body forces are present. If retardation of the liquid is great, an increase in liquid holdup will affect the ϕ^2 factor. The accurate evaluation of the momentum terms and body-force term locally appears to be a formidable task. Further analysis will determine the limitations of analytical models. In any event, the overall effects of tilting can be determined in the experiments.

D. EFFECT OF NONCONDENSIBLES

Noncondensibles arise from several sources in the flight system: (1) hydrogen leakage from the reactor, (2) residual pumpdown gas in the Hg system, and (3) outgassing from Hg system components at operating temperature. The single tube condensing test will use argon and hydrogen to study the separate effects to be expected in the flight system. Argon is used in place of nitrogen since it has been found¹⁶ that bottled nitrogen contaminates Hg. Deliberate introduction of fixed volumes of argon will take place, and temperatures, pressures and flows will be observed. A fluoroscope will be used in the location of the interface and in the observation of flow perturbations. Hydrogen will be introduced from a capsule at a constant rate, rather than in discrete volumes, since it diffuses out of the RC tube itself. The rate will attempt to simulate the leakage that is expected of a typical flight system. Sampling means will be provided to determine the amount of noncondensable carryover into the subcooler region. Design margin requirements will be gained from these tests.

E. TENTATIVE PRESSURE-DROP TEST OBJECTIVES

Initially, condensing pressure-drop tests will be conducted with glass tubes so that flow phenomena can be observed visually. These will be nonwetting tests because Hg does not wet glass. The phenomena of interest is the growth of droplets on the tube surface and their subsequent entrainment into the core flow. The critical droplet size, as predicted in Reference¹⁴ for tilted plates and nitrogen-Hg flow, will be measured. The verification of the prediction in the high-velocity region and its modification in the low-velocity region will be made. Photographic methods will be used in this investigation. Approximately 1/4-in.-ID tubes will be used with pressure taps at each end.

Nonwetting tests with steel tubes of three different sizes will be conducted to obtain local pressure drop data. It has been proposed that 10-ft-long tubes, with approximate diameters of 0.15, 0.26, and 0.4 in. be used. The large variation in diameter (approximately 3 to 1) is essential in verifying the dimensionless aspect of the pressure-drop correlations. Since current Rankine-system development calls for a fairly low condenser pressure (8.5 psia inlet) the inlet pressure levels will be around this value, with a maximum of 20 psia and a minimum of 6 psia. Pressure taps will be located at uniform intervals (approximately 1-1/2 ft in the 10-ft tubes) for the seven pressure transducers that are available. The interval length will be reduced for tests with short condensing lengths (less than 5 ft). Prior to testing with Hg, the smoothness of the tube, with the taps in place, will be verified with argon, so that the smooth-pipe friction-factor basis of the two-phase correlations will not be affected. Flow rates and condensing lengths which will produce pressure drops of up to 5 psi will be employed. Although the primary objective of this program is to obtain horizontal (zero-g) data, at least 20% of the runs in this phase of the program will be under inclined conditions. These tests will be conducted up to the angles at which instabilities occur.

Upon satisfactory verification or modification of the pressure-drop prediction with the round untapered tubes, tests will be run with rectangular tubes to verify the applicability of the hydraulic diameter relationship in the prediction. The ability to predict pressure drop in tapered tubes will be checked by tests on round tapered tubes. One tube for each of these phases will be used.

A tapered rectangular tube of the size to be used in the flat radiator of the system tests (PSM-3) is available to this program. Tapered copper fins are attached to the tube to simulate the heat rejection of a single tube of the RC. The fin is designed for heat rejection by radiation, but can be modified for air cooling. Tests will be conducted on this fin-tube assembly to further verify the prediction methods. But primarily, the characteristics of this assembly will be obtained to enhance the understanding of the PSM-3 tests.

F. EQUATION FOR THE REDUCTION OF DATA

It appears from comparison with previous data that inclusion of the liquid momentum is required for the accurate prediction of the pressure profile. For

this reason Equation 18 will be used to reduce pressure-drop data. Upon rearranging Equation 18 and introducing Equations 10 through 14, ϕ^2 is

$$\begin{aligned} \phi^2 = & \frac{2\rho A^2 D_e g_c}{f W_t^2 \psi^2} \left[\frac{W_t^2}{\rho^2 A^2 g_c} \frac{d\rho}{dP} (\psi^2 - \epsilon \psi^2 + \epsilon \psi) - 1 \right] \frac{dP}{dx} \\ & - \frac{2D_e}{f \psi^2} \left[(2\psi - 2\epsilon \psi - y\psi + \epsilon) \frac{d\psi}{dx} - \frac{1}{A} (\psi^2 - \epsilon \psi^2 + \epsilon \psi) \frac{dA}{dx} \right. \\ & \left. + (\psi - \psi^2) \frac{d\epsilon}{dx} \right]. \end{aligned} \quad \dots(48)$$

Since ϵ is expected to be very near one along most of the condensing length, the first attempts at evaluating ϕ^2 will be with $\epsilon = 1$. Also, y will be assumed to be zero for small heat rejection rates (long condensing lengths); thus

$$\phi^2 = \frac{2\rho A^2 D_e g_c}{f W_t^2 \psi^2} \left(\frac{W_t^2 \psi}{\rho^2 A^2 g_c} \frac{d\rho}{dP} - 1 \right) \frac{dP}{dx} - \frac{2D_e}{f \psi^2} \left(\frac{d\psi}{dx} - \frac{\psi}{A} \frac{dA}{dx} \right). \quad \dots(49)$$

For Equation 49, the geometry of the tube and the total weight flow are known for each test run. The quality and its gradient along the tube are readily calculated by methods discussed in Appendix B. In each run the local pressure is measured at several points along the tube. The average pressure gradients between these points can be used to approximate the actual pressure gradient at the midpoints, thus dP/dx is known. Since saturated conditions exist during condensing, the properties of Hg can be related to the local pressure; the average pressure between the taps will be used. With the foregoing, ϕ^2 can be evaluated. The correlating parameters of interest, the Martinelli parameter, the vapor Reynolds number, and the tube Weber number are also obtained.

G. STARTUP DYNAMICS

During the SNAP 2 System startup phase, in which condensing is used to pre-heat the RC, it is believed that the pressure in the tubes is essentially zero and

causes the flow to be choked. In order to choke, the vapor must accelerate from its inlet velocity to sonic velocity at the throat of a vena contracta. It is questioned whether supersonic or subsonic flow occurs after the throat. The amount of frictional pressure drop, hence pressure recovery, is dependent on the mode of flow.

In the process described, there are several unknowns which must be determined for an accurate transient analysis of the RC preheat. Among these are: (1) the verification of choked flow, (2) the effect of the vena contracta, (3) the condition of the vapor in this region (i.e., Does equilibrium condensing occur due to the temperature drop that accompanies the pressure drop?), and (4) the amount of pressure recovery that can be expected in this type of process. The first three items determine the method of analysis that must be used to describe this process, the last item is required input to the selected analysis.

The presence of choked flow can be determined by measuring the variation of flow rate with stagnation pressure. If the flow is choked, the flow rate should increase linearly with an increase in stagnation pressure. The pressure distribution in the vena contracta region, as well as the pressure recovery downstream, can be found by modifying the inlet manifold for the glass-tube tests to accommodate an externally positioned search tube. Accurate means for determining the size of the vena contracta and the existence of equilibrium condensing in the vena contracta are still to be determined.

The second major subject about which the glass-tube condensing tests can provide useful information concerns the experimental verification of the existing analytical model which predicts the required Hg inventory for the preheat of the RC. This empirical code, which is based on PSM-3 test results, has not been verified experimentally for the current reference Mercury-Rankine system startup scheme. Mercury inventory requirements involved in the preheat of the RC are now supplied from both the CRU bearing drains (which flow into the bottom of the RC), and from condensing Hg vapor at the top of the RC. For further information on startup see Reference 2. This scheme can be simulated in a glass-tube test.

H. RC STABILITY

Two types of instability can occur in the condenser, interface instability and liquid-leg instability. The first is concerned with the unstable action in

the region of demarkation between the two-phase flow and the pure liquid in both wetting and nonwetting condensing, and in the region of two-phase flow where an annular liquid layer occurs during wetting conditions. The second is concerned with the bulk movement of the condensate leg due to unstable forces imposed upon it.

1. Interface Stability (Nonwetting)

Under nonwetting conditions, the stability of the interface is dependent mainly on the local acceleration imposed upon it, and to a lesser extent on the flow regime upstream of it (impacting droplets). Analysis and experimental work have fairly well defined the stable limits. The criterion¹⁴

$$D \leq 1.836 \sqrt{\frac{\sigma}{a(\rho_l - \rho_v)}} \quad \dots(50)$$

is sufficient to define the stable limits when the interface is exposed to acceleration or body forces. These forces are present during maneuvers in space and during ground testing.

During horizontal ground testing, with tubes greater than the critical diameter, the interface is no longer vertical but tends to spread out over the length of the tube (Figure 32). Restoring forces in the form of frictional drag, momentum interchange between the interface and the condensing vapor, and impacting droplets are present to reduce the length of the interface. Short interface lengths are not detrimental to the system if they do not result in further instabilities. The length of these interfaces is dependent on the momentum transfer which can be related to the inlet momentum of the vapor flow. Interface length data, reported in Reference 9, are plotted against inlet momentum in Figure 19.¹⁷ There appears to be a good correlation between the two. With sufficiently high inlet momentum, one can be assured that no long troublesome interfaces will be present under horizontal conditions.

High inlet momentum, however, may not be sufficient to restore the interface under tilted conditions. Tilting increases the head of Hg along the length of the interface, and a further increase in length is necessary to provide more surface area to accommodate the restoring momentum transfer. A point

will be reached when the increase in head is greater than the restoring forces and the liquid will continue to run down the tube. Surface waves, which may result in plug flow or slugging, could be generated on this long interface.

Figure 19. Nonwetting Interface Length in Tube Diameters as a Function of Inlet Momentum for Various Condensing Lengths

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Future Rankine-system testing calls for tilting a flat horizontal RC to simulate the adverse accelerations that may cause instability of the condensate leg during space operation (tumbling, orientation, docking, etc.). An analysis¹⁸ has shown that liquid-leg instability occurs before interface instability during orbital tumbling conditions. If liquid-leg instability is to be simulated, knowledge of a fairly stable short interface is required or the results of the test may be prejudiced by interface instability.

Tests (Nonwetting)

The nonwetting interface will be observed during horizontal and inclined pressure drop studies. Interface length, angle of tilt, Hg contact angle and inlet momentum will be recorded. Data will be taken up to the point of incipient instability. Instability of the liquid leg will be inhibited by the exit valve so that only the one form of instability can be observed. Continuous observation of the interface in the steel tubes will be maintained via the fluoroscope.

2. Interface Stability (Wetting)

When Hg wets the condenser tube, an annulus of liquid is formed in the condensing region. Instability of this annulus results in film breakup or slugging.

Fine grained film breakup is not considered serious; however, large scale slugging can result in undesirable pressure fluctuations. Prediction of annular film instability for the purpose of understanding the flow regime is a difficult matter. Heat and mass transfer in the condensing process further complicates any analysis. A discussion of the various types of instabilities that may occur in the film, and a discussion of the work by other investigators on noncondensing film stability, is found in Reference 15. An understanding of wetting film stability and the condensing flow regime is best gained from observation.

Tests (Wetting)

Observation of the flow regime during wetted pressure-drop testing with horizontal and inclined tubes will be made. A qualitative description of the flow regime, the effects of instability and quantitative data on pressure fluctuations will be recorded.

3. Liquid-Leg Stability

Liquid-leg stability concerns itself with the bulk movement of the liquid leg. This instability is bidirectional in that the liquid can fill the tube and enter the vapor manifold or the liquid leg can recede, drawing vapor into the liquid manifold. In the first case, serious pressure fluctuations may develop due to large slugs of liquid entering and leaving the manifold. In the second case, uncondensed vapor in the liquid manifold may eventually enter the condensate pump with the possibilities of depriming and cavitation. Also, both phenomena may occur at the same time in adjacent tubes (similar to inverting a U-tube manometer). The criterion¹⁸ for this stability is

$$\frac{d\Delta P}{dL} > 0$$

at operating point, where ΔP is the pressure drop between the liquid and the vapor manifolds and L is the condensing length. Included in ΔP are the entrance acceleration and its associated losses, the condensing pressure drop, and the pressure drop in the liquid leg. This criterion is for static stability; stable oscillations about the operating point can occur, however.

Under zero-g conditions and horizontal ground operation, the pressure drop in the liquid leg is essentially zero, since frictional losses are negligible.

For inclined ground testing, it is equal to

$$\rho \frac{g}{g_c} \sin \theta (r_1 - r_2) ,$$

and for tumbling in orbit

$$\rho \omega^2 (r_1^2 - r_2^2) ,$$

where r_1 and r_2 represent the distances of the interface and liquid manifold from the center of gravity in space vehicles, or from some point along the tube under ground testing.

Analyses³ have shown that zero-g and horizontal ground operation are inherently stable. This is due to the high pressure drop associated with typical Hg condensers for space use. However, under tilted and rotating conditions, limits on tilting and rotating¹⁸ exist when the accelerations are opposite the flow direction, and beyond these limits the above criterion cannot be met. Typical steady-state operating lines with stability limits for rotation and tilting, are shown in Figures 20 and 21. Condenser pressure drop for these cases was predicted by Model No. 1 of Section II.B.1.

The onset of instability during startup is not predictable since the pressure-drop models do not adequately describe the process and no model is currently available for this purpose. An understanding of stability during startup must come from experimentation.

Tests

A flat condenser consisting of three or more parallel glass tubes joined by manifolds will undergo startup and steady-state testing under tilted conditions (including horizontal tests for reference). Tubes, 0.26-in. ID and 10-ft-long, will be used to simulate the condenser tubes of typical space condensers. Various flow rates and inlet pressures giving steady state pressure drops in the neighborhood of 2.5 psi will be used.

In the event interface instability tests show that no short semistable interface can be maintained in the 0.26-in. tube (critical diameter 0.15 in.), tubes smaller than 0.15 in. will be used. Flow rates and condensing lengths will be scaled down to produce steady-state pressure drops around 2.5 psi.

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Figure 20. Typical Operating Lines for the SNAP 2 Flight Design Condenser Under Tumbling Condition in Orbit

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Figure 21. Typical Operating Lines for a Flat Mockup of the SNAP 2 Flight Design Condenser Under Tilted Ground Environment

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III. TEST APPARATUS

Horizontal single-tube condensing tests have been conducted at AI in the past. The results of these tests have shown that the equipment and test facility were not conducive to accurate testing. Among the many problems were: the lack of space due to the location of other system tests, the dependency on the equipment of other tests (notably the use of a system test boiler), the limited capacity of the air cooling system, the inability to easily position the fluoroscope system, and the problem of inducing noncondensibles in various parts of the system upon clearing the test section with nitrogen. For these reasons the Hg-condensing apparatus was moved to another laboratory that provides sufficient space and independence from other tests.

The test apparatus was modified to facilitate testing and to expand the capabilities of the test. Among these capabilities are: greater cooling for higher flow rates and shorter condensing length, test section tilting for draining and stability studies, rapid positioning of the fluoroscope system, future accommodation of tubes up to 15 ft long, constant monitoring of pressure via pressure transducers, and the capability of multitube studies.

A. MERCURY LOOP

The Hg loop basically consists of a boiler, liquid separator, immersion thermocouple, test section, exit valve or pressure regulator, flow-measuring vessel, drain tank, and liquid pump in series (Figure 22). A boiler-pressure relief line and a vacuum line are auxiliary to the loop. The partially completed loop is shown in Figure 23.

1. Boiler

A cross-sectional view of the boiler is shown in Figure 24. Eighteen electrical heating elements located at the bottom are capable of generating 30 kw or at least 10 lbm/min of Hg. A baffle is located in front of the vapor exit line to inhibit liquid droplet exit from the boiler. The Hg level is monitored by a pressure transducer located on the side of the boiler. When full, a maximum of 270 lbs of Hg is available for testing before the heating elements are exposed to Hg vapor. For most testing, around 1 lbm/min, 4-1/2 hr of batch-wise operation are available. For higher flow rates or longer testing time, the boiler may be continuously fed.

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Figure 22. Schematic of Mercury Loop

Figure 23. Mercury Loop Apparatus - Partially Completed

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Figure 24. Mercury Boiler

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2. Separator

To restrict Hg droplets from entering the test section, a liquid separator (Figure 25) is provided between the boiler and the test section. Liquid droplets are swirled out toward the walls of the container and are collected at the bottom. A float valve controls the level of the liquid at the bottom. Excess liquid is collected in the drain tank. Line heaters and sufficient insulation will be provided on the piping between the boiler and test section to further restrict condensation and to superheat the vapor slightly.

3. Immersion Thermocouple

A superheat measuring system (Figure 26) is located in front of the test section. It consists of a 1-in. pipe that accommodates an immersion thermocouple and a pressure tap. The immersion thermocouple is sufficiently shielded to prevent radiation transfer to the chamber wall. Shielding is a necessary

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Figure 25. Mercury Liquid Separator

Figure 26. Superheat Measuring Device

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requirement, since radiation heat transfer at the test temperature (approximately 600°F) is generally larger than convection heat transfer with Hg vapor. An unshielded thermocouple of any length suspended in the vapor stream will measure the chamber wall temperature very much better than the vapor temperature. Since the chamber wall temperature will be purposely kept higher than the vapor temperature to restrict condensation, shielding becomes a necessary requirement. A discussion of thermocouple shielding and length requirements is found in Appendix D. The 1/8-in. sheath-type thermocouple used is removable for calibration.

4. Pressure Regulator

The pressure level of the liquid leg is controlled by either of two methods. The most common is the exit valve across which the pressure is developed. A constant pressure, however, can only be maintained under extremely steady-state conditions. Instability of the liquid leg in a single-tube arrangement is damped by the restricting valve, so that the pressure level in the liquid leg fluctuates with the instability and little gross movement of the condensate leg occurs. The second method employs a constant head of Hg to maintain the pressure level (Figure 27). Under stable and unstable conditions, a fairly constant pressure can be achieved.

Figure 27.
Constant Head Device

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Either method can be used in horizontal pressure-drop studies, whereas the first method must be used during interface instability studies where the liquid leg may be inherently unstable, and the second method must be used during liquid leg instability studies where simulation of a pressure regulator is required.

5. Flow Measuring Tank

The flow rate of Hg is measured with a calibrated steel vessel over a period of time. The Hg level in the vessel is determined with electrical contact probes imbedded in the side of the vessel (Figure 28). A timing device is triggered when contact is made. Three vessels for three ranges of flow rates, 0.3, 1.0, and 3.0 lbm/min are provided. The vessels are evacuated during operation.

Figure 28. Flow Measuring Vessel

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6. Drain Tank

A drain tank located below the rest of the system accepts liquid from both the separator and the flow measuring vessels. A total of 800 lbms can be accommodated in the drain tank so that during shutdown the boiler can be drained of Hg. In addition to the drain lines, vacuum and pressurized argon lines are provided. During operation, the drain tank is evacuated. Pressurized argon will be used to transfer the liquid to the boiler in the event of pump failure or leakage.

7. Pump

A gear pump transfers Hg from the drain tank to the boiler. Continuous operation can be provided, if necessary, although most testing will be with low flow rates, and batch wise operation suffices.

8. Relief Valve and Quench Tank

A Crosby-Ashton JUMB 1-1/2 relief valve is provided on the boiler outlet and is set for 16 psid. The relief line is fed to a quench tank which is partially filled with Hg and water. The Hg seals the evacuated relief line from the water and the water is provided as the heat sink for the Hg vapor in the event the boiler should become over-pressurized.

9. Vacuum System

The Hg loop is evacuated at several points, the boiler exit, the pressure regulator, and/or measuring tank, the drain tank, and pressure relief line. A freeze trap (Figure 29) for condensing any Hg vapor that should enter the vacuum line is located in front of the vacuum pump. A refrigerating unit cools the freeze trap.

Figure 29. Freeze Trap for Vacuum Line

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B. COOLING SYSTEM

The tube is cooled by air directed vertically upward from the outlet duct that parallels the tube. This configuration was chosen so that each tube in a horizontal array of multiple tubes would be subjected to similar air velocities and temperatures. Also, greater tilting angles can be achieved with the plenum below the tubes. The outlet duct is fed by a tapered plenum chamber which in turn is fed by a 5000 cfm blower. The plenum is tapered to insure that little momentum recovery occurs along the plenum so that the pressure drop through the outlet duct is uniform along the duct and uniform outlet velocities occur. Also, plenum velocities are purposely kept low to further insure against maldistribution of flow. The knowledge of a uniform velocity profile is a mandatory requirement since cooling rates, hence quality, along the tube must be known quite accurately. To further insure a uniform velocity profile, screens in the outlet duct can be provided.

The width of the outlet duct is variable so that an optimum outlet velocity can be achieved for various sizes and numbers of tubes. The outlet width is typically 1 in. for single tubes, which allows a maximum outlet velocity of 100 fps. For three tubes in parallel, a 2-in. slot will be used. Air jet widths are purposely kept large in comparison to the tube sizes so that the tube or tubes are submerged in the core of the jet and are subjected to a uniform velocity profile across the width. Cooling with narrow jets within 20 jet widths must be avoided since misalignment of the jet and the tube would subject the tube to nonuniform velocities along the length of the tube.

Although a temperature drop accompanies the pressure drop inside the tubes, the heat transfer coefficient for nontapered tubes is essentially constant, provided that a uniform velocity profile exists. The temperature drop along the tube is small in comparison to the temperature difference between the tube and the cooling air, so that the film properties of the air are essentially constant. The heat-rejection rate is then a function of the temperature differences along the tube. The effect of this small temperature variation on the pressure-drop calculations is shown in Appendix B. A considerable variation in the heat-rejection rate, however, occurs along tapered tubes. This also is discussed in Appendix B. Predicted cooling rates for this system are found in Appendix C.

C. TILTING ASSEMBLY

The tiltable support frame, shown in Figure 30, forms a cage that accommodates the tapered cooling air plenum and provides support for the test section, the fluoroscope system, the pressure transducers, and associated plumbing. At present, this cage is capable of being tilted from $+20$ to -12° due to the ceiling and floor limitations. Analysis has shown that this range should be sufficient for stability testing. If greater angles are found to be required, the pivot point of the cage can be raised or lowered. The plenum is fed from a duct (not shown) that passes through the rear wall and is connected to the blower via a diffuser, both located outside the building.

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Figure 30. Tilting Support Structure

An electrically driven cable-type lifting mechanism is used to tilt the cage. Locks are provided to inhibit vibration or swaying. Stops are used to locate the horizontal position accurately during horizontal testing.

D. FLUOROSCOPE ASSEMBLY

A Muller MG 150 constant potential x-ray machine is being procured for the fluoroscope system. This unit is continuously adjustable in the ranges from 25 to 75 and from 50 to 150 kv-dc. Continuous rated output of the apparatus is 3 kw. The x-ray tube is a Norelco MG 150/1 double-focus, beryllium-window, 150-kv-dc tube. This particular unit has been successfully used recently in some dynamic void fraction measurement experiments at AI. In addition to the x-ray tube and power supply, a line voltage regulator and scintillation detectors were used in these experiments. A description of the test setup is found in Reference 19. The primary purpose of the fluoroscope system is to detect the location of the interface and to observe any unstable flow phenomena. For this purpose x-ray apparatus requirements are very much less severe than those of void fraction measurements.

The carriage that accommodates the x-ray tube and the fluoroscope screen is shown in Figure 31. A shadow box is provided in front of the screen to enhance visual observation. The test section is mounted within a few inches of the back of the screen so that a fairly sharp image, approximately to scale, is produced. This carriage is capable of traversing the entire length of the test section under any tilted orientation. Positioning of the carriage is obtained by a rack and pinion system driven by a bidirectional motor.

During single-tube testing the x-ray head will be set level with the tube so that an unshadowed profile of the interface can be seen. This requirement will be relaxed in multitube testing since the x-ray head must be rotated up around the tubes somewhat to reduce the shadowing effect of one tube on another. A fluoroscopic image of a typical nonwetting interface in a single, over-critical-size tube is shown in Figure 32. This photograph was obtained in a previous horizontal single-tube condensing test. Here the fluoroscopy was limited in that the x-ray beam had to pass through the cooling air duct, and also fixed positioning of the x-ray head was required. One of the vertical pressure taps is also shown. This tap is obviously filled with noncondensibles that were

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Figure 31. Fluoroscope Carriage and Screen

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Figure 32. Fluoroscopic Image of Interface in 0.26-in. ID Tube
(Overcritical Size)

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forced into the tap by a high-pressure fluctuation that required a large change in Hg inventory in the manometer. This observation has prompted the use of small volumetric pressure transducers.

Since the accurate reduction of pressure-drop data requires a knowledge of the void fraction (or liquid-volume fraction) the x-ray technique may be used to gain it. The elaborate equipment of the dynamic void fraction tests is available for this purpose. An analysis is required, however, to assure that this equipment will meet our objectives.

E. PRESSURE INSTRUMENTATION

Measurements of static pressure in the tube are made via 1/8-in., Hg-filled pressure taps located at various intervals along the tube. The taps, connected to differential pressure transducers (Pace P7D), are welded horizontally at the centerline level of the tube. This insures against pressure fluctuations in the event large droplets should roll along the bottom of the tube. This configuration further guards against trapping noncondensibles in the pressure taps in that droplets as large or larger than the tap hole will usually run along the bottom of the tube. In addition, the taps are used as supports for the tube (Figure 33).

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Figure 33. Pressure Transducer System

Pressure transducers are used for three reasons: (1) in contrast to Hg manometers, there is little change in transducer volume with fluctuations in pressure (ΔV full scale = 0.0003 in.³). This discourages noncondensibles from entering the pressure taps during a fluctuation in pressure during studies with noncondensibles or when small amounts of noncondensibles are inadvertently present; (2) readout is done with greater facility in comparison to manometers. One easily read voltmeter can service all transducers; and (3) the startup dynamics can be observed by recording all data. The achievement of steady-state conditions is readily noted with recorded data.

The Pace P7D is a variable reluctance transducer powered by a 3-kc carrier source. Its linearity is within $\pm 3/4\%$. Transducer output is amplified, demodulated, rectified, and filtered by a demodulator and results in a full-scale output of ± 5 v-dc.

The transducer case and diaphragm are made of 416 stainless steel to minimize the corrosion effect of Hg. Both the positive and the reference side are capable of operating with Hg since the transducer is symmetrical and either side can be used as the positive side. In this report, positive is used to indicate the test side of the transducer although negative pressures can be imposed upon it. Recording the negative pressure is achieved by reversing the output leads. Changes in the pressure ranges are easily made by changing diaphragms.

The transducers are mounted approximately 30 in. below the tap so that the positive side of the transducer will always be at atmospheric pressure or above. This allows the reference side to be pressurized with a positive pressure. All the transducers will have this reference pressure in common. The reference pressure is supplied by pressurized argon through a precision pressure regulator. A Hg manometer is used to measure the reference pressure. This configuration is the best suited for tilting since the common nitrogen reference will vary little between transducers. The head of Hg on the positive side is reduced by tilting, and the reference pressure must be reduced accordingly. The transducer is mounted in a horizontal position originally so that upon tilting, nonlinearities due to the variation in pressure difference across the diaphragm are minimized.

Calibration

Transducer calibration can be done two ways; during shutdown, or during a test. During shutdown, the test section is filled with Hg and a head of Hg is applied; the reference pressure is then increased to match the total known head on the transducer. At this point, the carrier demodulators are nulled, the reference pressure is reduced by a full-scale amount, and the demodulator sensitivity is adjusted to full-scale output. Calibration within the full-scale reading can then be made by varying the reference pressure. During testing, one or more of the transducer-demodulator combinations can be calibrated with the use of the Hg supply. The valve between the transducer and the test section is closed and the supply valve opened. A known head of Hg is applied to the Hg supply. Renulling and sensitizing, as well as calibration, can be done by this method.

F. OTHER INSTRUMENTATION

The partially completed instrument and control panel which will service the following is shown in Figure 34.

1) Boiler

- a) Continuous recording of boiler pressure
- b) Heater voltage and current
- c) Liquid level transducer
- d) Outlet vapor temperature

2) Line Heaters

Heater current and voltage

3) Superheat Measuring Device and Test Section

- a) Intermittant monitoring of pressure transducers with a pen recorder, using a stepping switch to select transducer
- b) Individual digital voltmeter readings for final accurate pressure measurements
- c) Immersion thermocouple and test-section thermocouples for heat-rejection computations using an accurate potentiometer

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Figure 34. Instrumentation

d) Test-section temperature profile monitoring with cathode ray tube scanner for noncondensable studies

4) Flow Rate

Electrically triggered timer for flow measuring vessel.

G. SAFETY

Due to the poisonous nature of Hg vapor, a secondary enclosure is required to protect personnel in the event Hg vapor should leak out of the system. A plexiglass wall is used to separate the personnel area from the Hg loop. It is located a few inches in front of the fluoroscope screen (not shown in photographs). A negative pressure is maintained in the enclosed portion of the room by an exhaust fan, and the exhaust is ducted out above the building.

REFERENCES

1. R. T. Lancet, et al., "SNAP 2 Performance Analysis," NAA-SR-9087 (Classified Report), February 15, 1964
2. P. M. Gresho, et al., "SNAP 2 Startup Analysis," NAA-SR-9078 (Classified Report), January 20, 1964
3. R. Forslund, "SNAP 2 R/C Stability Under Zero "G" Conditions," NAA-SR-TDR-9126, October 24, 1963
4. R. C. Martinelli, L. M. K. Boelter, T. H. M. Taylor, E. G. Thomsen, and E. H. Morsin, "Isothermal Pressure Drop for Two Phase, Two Component Flow in Horizontal Pipe," Trans. ASME, Vol. 66, p 139, 1944
5. R. W. Lockhart and R. C. Martinelli, "Proposed Correlation of Data for Isothermal Two Phase, Two Component Flow in Pipes," Chemical Engineering Progress, Vol. 45, p 39, 1949
6. L. Hays, "Investigation of Condensers Applicable to Space Power Systems, Part I Direct Condensers," EOS Report 1588, August 15, 1962
7. C. J. Baroczy and V. D. Sanders, "Pressure Drop for Flowing Vapors Condensing in a Straight Horizontal Tube," NAA-SR-6333, June 1961
8. A. Koestel, "Fog Flow Mercury Condensing Pressure Drop Correlation," Preprint prepared for Third Annual High Temperature Liquid Metal Heat Transfer Technology Conference, Oak Ridge National Laboratory, September 4, 1963
9. R. G. Gido and A. Koestel, "Mercury Wetting and Non-wetting Condensing Research," Progress Report No. 3, TRW Report No. ER-5214, January 1963
10. A. Shapiro, Fluid Dynamics, Compressible Flow, Vol. I, (1st ed., Ronald Press), 1953
11. R. C. Martinelli and D. B. Nelson, "Prediction of Pressure Drop During Forced-Circulation Boiling of Water," Trans. ASME, Vol. 70, 1948
12. R. Kiraly and A. Koestel, The SNAP 2 Power Conversion System Topical Report No. 8, "Condenser Development and Design Study," TRW Report No. ER-4104, June 1960
13. R. P. Forslund, "Pressure Drop Program for Condensing Mercury in the SNAP 2 Condenser," NAA-SR-TDR-9064, September 17, 1963
14. C. T. Jaenke, A. Koestel, and J. G. Reitz, The SNAP 2 Power Conversion Topical Report No. 13, "Offbase Testing," TRW Report No. ER-4670, November 1961

15. A. Koestel and C. M. Smith, "Radiator Design Limitations for Dynamic Converters," Preprint prepared for the Sixth AGARD Combustion and Propulsion Colloquim, Cannes, France, March 1964
16. F. G. Hammitt, "Fourth Monthly Progress Letter," University of Michigan, March 1964
17. R. P. Forslund, "Interface Lengths for Non-Wetting Condensing Mercury in One "G" Horizontal Steel Tubes - Test Results," NAA-SR-TDR-8484, May 8, 1963
18. R. P. Forslund, "Stability Analysis of the SNAP 2 Radiator-Condenser with Vehicle Tumbling During Steady State Operation," NAA-SR-TDR-8843, August 5, 1963
19. J. D. Kendron, E. E. Stoner, and G. M. Taylor, "Dynamic Void Fraction Measurement System," NAA-SR-7875, May 15, 1963
20. H. Schenck, Jr., Heat Transfer Engineering, (1st ed., Prentice Hall), 1959
21. W. H. McAdams, Heat Transmission, (3rd ed., McGraw-Hill Book Co., Inc., New York), 1954

APPENDIX A EVALUATION OF THE RATIO OF THE FOG-FLOW MIXTURE DENSITY TO THE VAPOR DENSITY

The average mixture density is given by

$$\rho_m = R_v \rho_v + R_\ell \rho_\ell . \quad \dots (A-1)$$

The volume fractions are related by

$$R_v = 1 - R_\ell = \frac{1}{\left(\frac{1-\psi}{\psi}\right)\left(\frac{\rho_v}{\rho_\ell}\right)\left(\frac{V_v}{V_\ell}\right) + 1} . \quad \dots (A-2)$$

Inserting Equation A-2 into A-1, and noting that $V_v/V_\ell = 1$ for fog flow and rearranging, Equation A-1 becomes

$$\frac{\rho_m}{\rho_v} = \frac{\left(\frac{1}{\psi}\right)}{\left(\frac{1-\psi}{\psi}\right)\left(\frac{\rho_v}{\rho_\ell}\right) + 1} . \quad \dots (A-3)$$

Since $\rho_v \ll \rho_\ell$ the first term in the denominator is negligible along most of the condensing length. In the pressure range of interest, $3000 < \rho_\ell/\rho_v < 8000$. At $\rho_\ell/\rho_v = 3000$, ψ must be less than 4% before the denominator changes by 1% from unity.

Equation A-2 is easily derived from the ratio of

$$W_\ell = W_t(1 - \psi) = A_T R_\ell \rho_\ell V_\ell , \quad \dots (A-4)$$

and

$$W_\ell = W_t(\psi) = A_T R_v \rho_v V_v . \quad \dots (A-5)$$

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APPENDIX B

CALCULATION FOR QUALITY

Although a uniform cooling-air velocity, hence heat-transfer coefficient, can be assured over the length of a constant diameter tube, the heat-transfer rate is not uniform due to the temperature drop along the tube that accompanies the pressure drop within the tube. Variation of the fluid properties with this change in temperature is negligible, so the coefficients are essentially constant. This temperature variation is especially true for condensation at the low pressures typical of the Rankine system. Although the heat transfer nonuniformity is small, however, its effects on pressure drop make it worth taking into consideration. Because the heat transfer is fairly close to uniform, the temperature drop through the tube wall should be uniform so that the tube temperature variation should follow the vapor temperature variation. This variation is nonlinear since $T_{\text{vapor}} = f(\log P_{\text{vapor}})$ and the pressure variation along the tube is nonlinear, but a second-order curve fit for the tube temperature will closely approximate it:

$$T_{\text{tube}} = T_{\text{in}} + ax + bx^2. \quad \dots(\text{B-1})$$

The heat-transfer rate is

$$q(x) = H\pi D(T_{\text{in}} + ax + bx^2 - T_{\text{air}}) \text{ Btu/hr-ft}.$$

Total heat rejected at any point along the tube is proportional to the Hg condensed:

$$Q(x) = \int_0^x H\pi D(T_{\text{in}} + ax + bx^2 - T_{\text{air}}) dx \quad \dots(\text{B-2})$$

$$= H\pi D\left(T_{\text{in}}x + \frac{a}{2}x^2 + \frac{b}{3}x^3 - T_{\text{air}}x\right) = W(x)_{\text{cond}}\lambda' \quad \dots(\text{B-3})$$

Although the Hg may be superheated at the tube entrance, condensation occurs immediately. Convective heat-transfer coefficients are so small that

the Hg temperature cannot be reduced to saturation temperature immediately and nonequilibrium condensation occurs. It is assumed that the sensible heat of superheat is rejected uniformly along the condensing length, hence λ' is used to represent both the heat of vaporization and the sensible heat. To minimize the weight of this assumption, the amount of superheat must be kept as low as possible.

Since only complete condensation of saturated or slightly superheated vapor is considered,

$$W(L)_{\text{cond}} = W_T,$$

and

$$W_T \lambda' = H \pi D \left(T_{\text{in}} L + \frac{a}{2} L^2 + \frac{b}{3} L^3 - T_{\text{air}} L \right). \quad \dots(B-4)$$

The vapor quality is then

$$\psi(x) = 1 - \frac{W(x)_{\text{cond}}}{W_T} = 1 - \frac{\left(T_{\text{in}} x + \frac{a}{2} x^2 + \frac{b}{3} x^3 - T_{\text{air}} x \right)}{\left(T_{\text{in}} L + \frac{a}{2} L^2 + \frac{b}{3} L^3 - T_{\text{air}} L \right)}. \quad \dots(B-5)$$

This relationship for quality can be used for untapered rectangular tubes also. Although the heat-transfer coefficient is different between round and rectangular geometries, it is missing from the final equation for quality. When tapered round tubes are used the product HD must be included in the integration. Upon inserting

$$HD = k_f \left[B_1 + B_2 \left(\frac{\rho V D}{\mu_f} \right)^m \right], \quad \dots(B-6)$$

from Equation C-1, and

$$D = D_{\text{in}} + \frac{D_{\text{out}} - D_{\text{in}}}{TL} x \quad \dots(B-7)$$

in Equation B-3,

$$Q(x) = \pi k_f \left\{ B_1 \left[(T_{in} - T_{air})x + \frac{a}{2}x^2 + \frac{b}{3}x^3 \right] + B_2 \left(\frac{\rho V}{\mu_f} \right)^m \left[\frac{T_{in} - T_{air}}{K_D} \left(\frac{z^{m+1}}{m+1} \right) + \frac{a}{K_D^2} \left(\frac{z^{m+2}}{m+2} - \frac{D_{in} z^{m+1}}{m+1} \right) + \frac{b}{K_D^3} \left(\frac{z^{m+3}}{m+3} - 2 \frac{D_{in} z^{m+2}}{m+3} + \frac{D_{in}^2 z^{m+1}}{m+3} \right) \right] \right\} \dots (B-8)$$

where

$$K_D = \frac{D_{out} - D_{in}}{TL}$$

$$z = D_{in} + K_D x$$

In the event $\rho V D / \mu_f$ is both greater and less than 1000 along the tube the integration must be done in two sections.

Quality is calculated in the manner previously described. The condensed liquid at any section is

$$W(x)_{cond} = \frac{Q(x)}{\lambda'} , \quad \dots (B-9)$$

and the total condensed

$$W_{in} = W(L)_{cond} = \frac{Q(L)}{\lambda'} . \quad \dots (B-10)$$

The quality

$$\psi(x) = 1 - \frac{Q(x)}{Q(L)} . \quad \dots (B-11)$$

These equations are best evaluated with the aid of the computer. Unlike the calculation for quality in untapered tubes, heat-transfer coefficients are present in these equations.

The variation in heat-transfer rates of tapered rectangular tubes is hard to predict accurately. This is especially true when only one dimension is tapered. No further consideration is given this prediction since no direct air cooled tests are planned for tapered rectangular tubes.

The following is an illustration of the importance of accurate evaluation of the heat-transfer rates. The example considers the condensation of 0.5 lb/min of Hg in a 0.26-in.-ID, untapered round tube with an inlet pressure of 6 psia and a 100-in. condensing length. Under the assumption that uniform condensation takes place, a pressure drop of 1.732 psi results. Accompanying this is a temperature drop shown in Figure 35a. A second-order curve fit of this temperature was then applied in Equation B-5 for calculating the quality profile. The difference between this profile and the assumed linear profile is notable, as much as 1.7%. The ratio of the two is shown in Figure 35b. Shown in Figure 35c are the ratios of the frictional gradient, momentum recovery gradients and the total pressure gradients. Because the quality profile decays more rapidly in the irregular case (faster condensing rate) the momentum recovery is greater.

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Figure 35. Effect of Incorrect Evaluation of Heat Rejection on Pressure Drop Calculations

With smaller flow rates of vapor the absolute value of the frictional gradient is smaller. The net effect is to reduce the absolute value of the total pressure gradient, as much as 5%. The gradient ratios shown in Figure 35c are not weighted so they are not additive. The total pressure drop for this irregular case is 1.672 psi or 3.5% less than the uniform case. Pressure drop in this illustration was predicted by Model No. 1.

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APPENDIX C

COOLING RATES

Cooling rates in Btu/hr-ft of tube as a function of air velocity for several test-section tube sizes are shown in Figure 36. These data were obtained for tube air temperatures of 600°F and 70°F, respectively, from the correlation²⁰

$$\frac{HD}{k_f} = B_1 + B_2 \left(\frac{\rho VD}{\mu_f} \right)^m, \quad \dots (C-1)$$

where

B_1	B_2	m	$\frac{VD}{f}$
0.32	0.43	0.52	< 1000
0	0.24	0.60	> 1000

This data is in agreement with data points taken in previous single-tube tests.

For cooling lengths up to 10 ft, velocities of at least 100 ft/sec are obtained with a 1-in. jet at 5000 cfm. The envelopes of Hg flow rates and condensing lengths that can be accommodated for the three tube sizes are shown in Figure 37. Typical flow rates and condensing lengths to be tested are indicated by the dashed lines. Although it is not shown in this figure, a small increase in heat rejection can be obtained for short condensing length by shutting off the jet in the subcooler region. Jet velocities for the various tubes can be optimized by varying the jet width and length.

A second cooling method, which employs parallel air flow in a concentric duct, was considered for the test section. The heat-rejection rate for this method, vs flow rate of air for the 0.35-in. OD test section with various duct sizes is shown in Figure 38. This data was obtained with the correlation²¹

$$\frac{HD_e}{k_b} = (0.020) \left(\frac{\rho_f V_b D_e}{\mu_f} \right)^{0.8} Pr_b^{0.4} \quad \dots (C-2)$$

Figure 36. Heat Rejection Rate for Various Diameter Test Sections vs the Velocity of Air Blown Normal to the Tube

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Figure 37. Condensing Capabilities of Single Tube Apparatus

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for the same tube and air temperatures. Shown also is the region where the frictional pressure drop is above 10 in. H_2O and the region where velocities in the duct are greater than 100 and 500 ft/sec.

Figure 38. Heat Rejection Rates from a 0.33-in. ID Tube Cooled by Annular Flow of Air in a Concentric Duct

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Although the heat-rejection rates are comparable, blower requirements are excessive since air velocities are very high. Furthermore, only a small temperature rise in the air can be tolerated so that uniform heat rejection may be obtained along the tube. At least 1000 cfm are required for each lbm of Hg condensed if heat rejection is to be kept within 1%.

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APPENDIX D

IMMERSION THERMOCOUPLE REQUIREMENTS

Because thermal radiation predominates over convective heat transfer in a Hg vapor system, multiple radiation shields are necessary for immersion thermocouples. Not only do the shields inhibit radiation but they provide considerable surface area for convection heat transfer. The greater the number of shields and the lower their emissivity, the more closely does the inner shield approach the temperature of the vapor. The thermocouple then reads the temperature of the inner shield by radiation. The linearized heat transfer equations governing infinite concentric shields are

$$H_1 A_1 (T_v - T_1) = 4\sigma T_w^3 F_{12} A_1 (T_1 - T_2) \text{ for the thermocouple, } \dots (D-1)$$

and

$$\begin{aligned} (H_i + H_{i+1}) A_i (T_v - T_i) &= 4\sigma T_w^3 F_{i,i+1} A_i (T_i - T_{i+1}) \\ &+ 4\sigma T_w^3 F_{i,i-1} A_i (T_i - T_{i-1}) \text{ for the shields, } \dots (D-2) \end{aligned}$$

where

$$F_{ij} = \frac{1}{\left(\frac{1}{\epsilon_i}\right) + \frac{A_i}{A_j} \left(\frac{1}{\epsilon_j} - 1\right)}. \dots (D-3)$$

Results of a simultaneous solution of these equations are in the form

$$\begin{aligned} (T_{\text{thermocouple}} - T_{\text{vapor}}) / (T_{\text{tube wall}} - T_{\text{vapor}}) \\ = f(\text{number of shields, emissivity}). \dots (D-4) \end{aligned}$$

This ratio vs emissivity for 0 to 3 shields is shown in Figure 39 for the following assumptions. Values less than 0.1 are desirable for this ratio:

- 1) Shields equally spaced in 1-in.-ID pipe,

- 2) Laminar heat transfer coefficient $Nu = 3.66$ used (conservative assumption),
- 3) Conduction loss through thermocouple lead neglected (infinite system),
- 4) Conduction loss through shield supports neglected.

Figure 39. Effect of Radiation Shielding on Measurement of Vapor Temperature

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Because conduction losses through the leads are high in comparison to radiation and convection heat transfer the leads must be sufficiently long to prevent reading the root temperature of the lead. The linearized differential fin equation

$$-K \frac{d^2 T}{dx^2} = H\pi D(T_v - T) + 4\sigma T_w^3 F\pi D(T_s - T) \quad \dots(D-5)$$

is used with the assumption that the lead temperature at the entrance of the shield is equal to the pipe wall temperature.

Results of the integration for x equal to the inserted length are in the form

$$\begin{aligned} & (T_{\text{thermocouple}} - T_{\text{wall}}) / (T_{\text{vapor}} - T_{\text{wall}}) \\ & = f(\text{thermocouple size, shielded length}). \quad \dots(\text{D-6}) \end{aligned}$$

This ratio vs shield length for three sheath thermocouples is shown in Figure 40 for the following. Values of the ratio greater than 0.85 are desirable.

- 1) A three-shield system used in the prior analysis with $\epsilon = 0.5$ is assumed.
- 2) Each shield is uniform in temperature; no end effects.

A three-shield system with an 1/8-in. thermocouple inserted 3 in. into the shields is accepted for the superheat measuring device.

Figure 40. Effect of Thermocouple Length on Vapor Temperature Measurement

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NOMENCLATURE

- A = tube cross-sectional area
 A_w = tube wall surface area
 a = local acceleration in pressure-drop equation or coefficient in temperature equation
 B = body force on control volume
 B_1, B_2 = coefficients in heat transfer relationship
 b = coefficient in temperature equation
 D = diameter
 E_σ = constant for droplet Weber number
 f = friction factor for single-phase fluid
 f' = friction factor for two-phase fluid
 F_{is} = force on control volume due to surfaces within the outer boundaries of the control volume
 F_1, F_2 = force on the control volume due to the entrainment and deceleration of the condensed liquid
 g = 32.2 ft/sec²
 g_c = 32.2 lbm-ft/sec²-lbf
 h = tube height or Hg head
 H = heat transfer coefficient
 K = thermal conductance of thermocouple
 K_D = coefficient for tapered tubes $\frac{D_{out} - D_{in}}{TL}$
 k = thermal conductivity of air
 L = condensing length
 m = exponent in velocity profile equation or cooling rate equations
 P = static pressure
 p = tube perimeter

Pr_b = Prandtl number for the bulk flow

q = heat rejection rate per foot of tube

Q = integrated heat rejection rate

r_o = tube radius

R = volume fraction

T = temperature

TL = tube length

V = velocity

W = flow rate

w = tube width

We = Weber number

x = axial distance along tube

$y = \left(\frac{V'_v}{\bar{V}_v} \right)$, ratio of axial velocity of condensing vapor to local average
vapor velocity

$z = D_{in} + \frac{D_{out} - D_{in}}{TL}$

χ^2 = Martinelli parameter

δ_{cr} = critical droplet diameter

ϵ = liquid-slip ratio or emissivity

λ = heat of condensation

μ = viscosity

ψ = quality of vapor

ρ = density

σ = mercury surface tension or Stephan-Boltzmann constant

τ_w = shear stress at the wall

θ = tilting angle

ϕ^2 = two-phase factor

ω = rotation rate

SUBSCRIPTS

avg = average

b = bulk

c = core

cv = control volume

cr = critical

e = equivalent

f = film

h = hydraulic

in = inlet

ℓ = liquid

m = mixture or modified fluid

max = maximum

out = outlet

t = total or bare tube

tp = two phase

tt = turbulent liquid — turbulent vapor

tv = turbulent liquid — viscous vapor

v = vapor

vt = viscous liquid — turbulent vapor

vv = viscous liquid — viscous vapor

w = wall

δ = droplet