KILOWATT ISOTOPE
POWER SYSTEM
PHASE II PLAN
APPENDIX A TO VOLUME II AND VOLUME III
KIPS JET CONDENSER MATH MODEL

MASTER

Sundstrand Energy Systems

ROCKFORD, ILLINOIS
unit of Sundstrand Corporation

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KILOWATT ISOTOPE POWER SYSTEM ,

PHASE II PLAN .

APPENDIX A TO VOLUME II AND VOLUME III :

KIPS JET CONDENSER MATH MODEL

For

FLIGHT SYSTEM CONCEPTUAL DESIGN (FSCD)

And

GROUND DEMONSTRATION SYSTEM (GDS)

To

DEPARTMENT OF ENERGY

From

**SUNDSTRAND ENERGY SYSTEMS
4747 HARRISON AVENUE
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SUNDSTRAND AVIATION
DIVISION OF SUNDSTRAND CORPORATION
ROCKFORD, ILLINOIS

ENGINEERING REPORT

KIPS Jet Condenser Math Model

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APPENDIX A

JET CONDENSER DEVELOPMENT REPORT

APPENDIX A

JET CONDENSER DEVELOPMENT REPORT

I. INTRODUCTION

The purpose of this report is to document the analytical model of the condensation process occurring in a jet condenser that was developed during Phase I of the Kilowatt Isotope Power System Program (KIPS) at Sundstrand Corporation.

The analytical model is based on one dimensional gas dynamics in the vapor funnel, a kinetic theory approach to the vapor/liquid interface condensing process and an empirical correlation of the intrajet heat transfer coefficient (from the vapor/liquid interface to the bulk liquid in the jet). This model satisfactorily predicts the measured performance of the KIPS jet condenser.

The first section of this KIPS jet condenser report presents the KIPS jet condenser development history from the beginning of the program (August, 1975) through February, 1977, at which time it was apparent that the initial design was underperforming slightly. It was decided that an improved analytical model would be beneficial. The subsequent sections deal with the development of the basic experimental data taken on a variety of single jets in a steam/water jet condensation rig to obtain the intrajet heat transfer coefficients required by the analytical model. Also included is a section describing the experiments conducted on the jet condenser to determine the limit conditions under which jet condenser floodout can occur.

HISTORICAL BACKGROUND

The KIPS utilizes a jet condenser to allow satisfactory liquid/vapor interface control in all attitudes and under zero g conditions. The jet condenser is a device in which low pressure vapor is ducted into a funnel coaxially with a liquid jet(s) of subcooled working fluid. The liquid jet(s) is injected into the vapor with a high dynamic pressure and aimed at the throat of the vapor funnel as in Figure 1. The vapor condenses on the subcooled liquid jet and the combined liquid jet passes through a throat into a diffuser. The high dynamic pressure liquid jet undergoes a sudden expansion

somewhere in the diffuser to fill the entire cross-section. In the sudden expansion there is a loss of total pressure but a gain in static pressure of the liquid stream. Further recovery of static pressure occurs in passing through the diffuser. The behavior of the liquid jet in the diffuser is identical to that which occurs in the diffuser of a cavitating venturi. Thus, the outlet pressure from the diffuser can vary from essentially zero up to about 55-70% of inlet pressure depending on where the jet expands to fill the diffuser. In a jet condenser system this characteristic is used for inventory control by plumbing the diffuser outlet to a pressure controlled accumulator/reservoir.

In the development of the Sundstrand jet condenser for the 6 KWe Organic Rankine Cycle (ORC) system (circa 1967, 1968), it became apparent that the condensation process in the jet condenser is limited by the ability of vapor molecules to collide with the liquid jets, i.e., it is vapor density limited. By locating the liquid jets so that each jet has approximately the same vapor volume surrounding it, the Sundstrand 6 KWe jet condenser was capable of operating with the jet condenser outlet liquid subcooled only 5°F with respect to the mixing chamber saturation temperature or 15°F with respect to the jet condenser inlet saturation temperature.

Based on the 6 KWe jet condenser test data, the KIPS jet condenser design was extrapolated on the premise that the rate of condensation per unit surface area of jet is directly proportional to the vapor pressure in the jet condenser mixing chamber and that the outlet liquid temperature will be subcooled by ~5°F with respect to the mixing chamber pressure.

The initial design and build of the KIPS jet condenser utilized 45 liquid injector nozzles having 0.0145" diameter orifices and a liquid jet length of 10", which gave the prescribed condensing area per the extrapolation of 6 KWe data. The type of orifice used was a sharp edge type as shown in Figure 2 which results in a very well collimated stream and, thus, is relatively easily focused to pass through the throat of the vapor funnel and is resistant to brooming. The jets must pass through the throat without significant impingement on the walls of the vapor funnel. If significant impingement does occur, the liquid that does impinge can slow down and "puddle" near the throat section which then slows the other jets to the point where there is insufficient pressure recovery in the diffuser to keep the liquid flowing in the right direction, i.e., diffuser outlet pressure is less than accumulator/reservoir pressure. This condition is referred to as "floodout" as the liquid will no longer pass through the funnel and results in a system shutdown.

The first test results of the KIPS jet condenser utilizing the sharp edge orifice jets indicated a significant underperformance in condensation rate, except at very high liquid injection pressure ($\approx 2.5 \times$ design value). It was theorized that at the high injection pressures, the "laminar like" jets issuing from the sharp edge orifices were becoming more turbulent and breaking up, thus causing more surface area for liquid vapor contact.

At this time, visual observations were made of water jets produced by sharp edge orifices as well as by more conventional orifices. These conventional orifices had a rounded inlet and an orifice length to diameter ratio of ≈ 3 to 5 with turbulence promoting screw threads in the upstream feed tubes as in Figure 3.

Photographs of a water jet issuing from a sharp edge orifice at 90 psi and 195 psi injection pressure into air at 1 atm. are shown in Figure 4 and 5 respectively. It is apparent that the jet is nonturbulent at 90 psi and that it is breaking up at 195 psi. A photograph of the jet issuing from a rounded inlet orifice having a length/diameter ratio of 5 with upstream turbulence promoters is shown in Figure 6 for 90 psi injection pressure. By comparison with Figure 4, this jet is clearly more turbulent than that of the sharp edge orifice at the same pressure. Based on this visual comparison,

Figure 1

SHORT SHARP
EDGED ORIFICE

Figure 2

$L/D = 4$ THREADED
UPSTREAM

Figure 3

$\Delta P = 90$
 $N_R = 13280$

Figure 4

ΔP = 195
 N_R = 19548

Figure 5

$\Delta P = 90$
 $N_R = 13280$

Figure 6

the KIPS jet condenser injector was retrofitted with 45 conventional type orifices of 0.0145" diameter with turbulence promoting screw threads in the upstream feed tube.

Also important, it was found that this injector configuration was able to be focused through the same vapor funnel throat size as in the previous build.

Initial tests with the "turbulent" jet injectors showed improved condensing performance. However, the exact performance was not repeatable and seemed to degrade during testing.

It was known that the test loop leak integrity had been degrading with calendar time due to a significant number of minor disassemblies due to numerous loop difficulties. For convenience of build and testing, this was a nonhermetic test apparatus. Thus, it was suspected that the air inleakage into the very low pressure vapor (≈ 0.1 psia) system was causing a significant partial pressure of noncondensable gas which was causing the jet condenser to underperform by varying degrees.

The underperformance thought to be associated with the relatively large amounts of noncondensable gas was manifest in two ways: the condensing rate was deficient and the jet condenser was subject to floodout.

In order to cope with the air inleakage problem, the test loop was modified from the schematic of Figure 7 to that of Figure 8, which contained a deaerator and associated ancillary equipment. The incorporation of the deaerator eliminated the floodout problem and resulted in repeatable thermal performance of the jet condenser.

However, the measured thermal performance was somewhat poorer than originally anticipated based on extrapolation from the Sundstrand 6 KWe jet condenser. Specifically, the design called for condensing 0.0306 lb-sec of Dowtherm A vapor at a pressure of 0.1 psia and a temperature of 230°F at the inlet bell on the jet condenser vapor funnel when the liquid side was supplied 0.275 lb-sec of liquid Dowtherm A at 168°F and a pressure of 80 psia. The KIPS development jet condenser with 45 "turbulent" jets condensed 0.0306 lb-sec of Dowtherm A vapor at 0.188 psia and 230°F when supplied with 0.275 lb-sec Dowtherm A liquid at 168°F. The liquid pressure required to provide the 0.275 lb-sec flow was 100 psia rather than the design value of 80 psia.

Thus this build of the jet condenser was deficient in condensing capability, as evidenced by the vapor inlet pressure of 0.118 psia rather than the design value of 0.1 psia, as well as in liquid flow area as evidenced by the 100 psia rather than the design value of 80 psia for the liquid supply pressure to provide 0.275 lb-sec. The combined effect of the higher than design turbine back pressure and increased pump work resulting from the jet condenser underperformance would have resulted in a loss in system efficiency of approximately 1 point. This loss in system efficiency was deemed undesirable and a higher performance jet condenser design was undertaken.

At this time it was clear that something better than the simple "condensation flux is proportional to the vapor density" scaling law based on the Sundstrand 6 KWe jet condenser operation was required to obtain an improved design. Specifically, more had to be known about the internal heat transfer capability of various liquid jets (from vapor/liquid interface to the bulk of the jet) as well as the sensitivity of the condensation process to the presence of noncondensable gases.

In order to investigate the thermal performance of various liquid jets, a relatively simple test rig utilizing steam and liquid water was built utilizing materials on hand or readily available. The results from this test rig and the analysis of it comprise the next section of this report.

SIMPLIFIED SCHEMATIC

Figure 7

Figure 8

The derivation of the analytical model of the jet condenser utilizing the data derived from the water/steam test rig is presented along with a comparison of predicted and actual jet condenser performance. The comparison is quite good when the jet Stanton number as originally determined on the water/steam rig is increased by a factor of 5. Analysis of the water/steam jet heat transfer data indicates that this factor of nearly 5 may be explained by the presence of a small amount of air in the steam.

The sensitivity of the jet condenser performance to noncondensable gas was determined on the development jet condenser utilizing the 45 "turbulent" jets. This is described in a subsequent section.

The capability of the jet condenser to resist floodout due to misaligned jets and liquid scavenging in the vapor funnel was determined by test and is also reported in a separate section.

II. TECHNICAL DISCUSSION

In order to achieve a better jet condenser sizing design tool, a combined analytical/experimental approach was employed. Specifically, the fluid dynamics of the vapor in the funnel and adjacent to the liquid jet(s) as well as the local condensing rate (from kinetic theory) were analyzed and modeled for both the KIPS development jet condenser and a single nozzle steam/water test rig.

A large amount of test data were taken for various nozzle types and jet lengths on the steam/water test rig. These test data were analyzed and a correlation developed for the jet heat transfer coefficient from the jet surface to the bulk liquid, for several different injector types and jet lengths. The intrajet heat transfer data correlation was then used in the math model for the KIPS development jet condenser, and when modified by a multiplier of ~ 5 , was found to predict the condensing rates actually observed in previous tests on the KIPS jet condenser. A possible explanation for the ~ 5 multiplier, noncondensable gas in the steam/water rig, is discussed in a later section.

A. Steam/Water Test Rig

In order to obtain a data base from which to develop intrajet heat transfer correlations, a test rig utilizing steam and cold water was devised. This rig is shown schematically in Figure 9. The rig was designed so that the temperature rise of a single liquid jet passing through nearly stagnant steam could be measured.

Cold water is injected through the injector nozzle and passes through a steam space and then through a focus/target plate into a Dewar flask containing a thermocouple. By measuring the liquid inlet temperature at the nozzle and the liquid temperature in the Dewar flask (see Figure 10 for Dewar thermocouple installation), the jet temperature rise was measured. A steam thermocouple as well as a manometer were used to measure the steam condition. The steam was supplied by deaerated water boiling in a hot well/tank mounted directly beneath the jet/Dewar assembly. The hot well was heated with an electrical blanket heater.

Jet length could be varied by repositioning the baffle plate which holds the Dewar flask. Another component in the rig is a cold trap, plumbed to provide a steam flow past the liquid jet in an attempt to sweep away any noncondensable gas which might accumulate around the liquid jet. These gases were then pumped overboard via a vacuum pump. The nonhermetic test rig had an air leakage rate of ~ 0.006 scc/sec.

Figure 9

Figure 10

The liquid was supplied to the injector nozzle from a bladder type accumulator with deaerated water on one side and nitrogen gas on the other side. The liquid supplied to the nozzle was cooled to the desired inlet temperature via an ice cooled heat exchanger and a bypass mixing valve. A photograph of the rig is shown in Figure 11.

Data were taken over a range of steam pressures from high to low as the heater was insufficient to supply enough heat to maintain steady state conditions. A typical run took about 15 minutes during which 9 data points were taken. Several tests were repeated and the heat transfer results (Stanton numbers) for nearly identical test conditions varied by less than 6%.

B. Steam/Water Test Rig Data Analysis

Pure Vapor Condensation Rate*

The accepted physical model of evaporation and condensation is based on proposals of Hertz in 1882 (2) and of Knudsen in 1915 (3). The kinetic theory of gases yields the rate at which molecules strike the condensed phase from equilibrium vapor as $n_g c_g / 4$;

where

n_g = concentration of molecules

c_g = average molecular speed

$$c_g = \sqrt{\frac{8 RT_g}{\pi}} \quad (1)$$

R = gas constant of vapor

T = absolute temperature of vapor

g_o = gravitation constant

A "condensation/evaporation coefficient" may be introduced to account for the fraction of these incident molecules which enter the condensed phase, the remainder being reflected. The flux of molecules leaving the condensed phase is given by $f n_s c_s / 4$ where n_s and c_s are the molecular density and speed based on saturated liquid surface conditions and f is the "evaporation coefficient" in order for the equilibrium situation to be attainable. During net phase change, the bulk vapor velocity normal to the interface affects the molecular velocity distribution; Schrage (4) took this into account and derived the following expression for the net condensation rate:

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

where

w = net condensation rate

$$\Gamma = \exp(-S^2) + \sqrt{\pi} S (1 + \text{erf}(S)) \quad (3)$$

*This treatment is taken from Ref. 1

Figure 11

(4)

$$S = \frac{u}{\sqrt{2RTg_0}}$$

u = bulk velocity of vapor normal to jet

Equation (2) can be rewritten in terms of the mass flux condensed as

$$\frac{\dot{m}}{A} \text{ cond} = f \left(\frac{\Gamma \rho_g C_g}{4} - \frac{\rho_s C_s}{4} \right) \frac{\text{lb}}{\text{ft}^2 \text{sec}} \quad (5)$$

where

ρ_g is the vapor density, $\left(\frac{\text{lb}}{\text{ft}^3}\right)$

ρ_s is the saturated vapor density based on liquid surface temperature, $\left(\frac{\text{lb}}{\text{ft}^3}\right)$

Based on the arguments of Ref. 1, f is taken as unity in this work.

For the net condensation of saturated vapor it is clear the ρ_s must be less than ρ_g , i.e., the temperature of the condensate surface must be less than the saturation temperature of the vapor. The factors that affect ρ_s (or T_s , the liquid surface temperature) are the bulk liquid jet temperature T_B , the condensation rate, w , and the ability of the liquid jet to transport heat from its surface to its bulk liquid. The factors that affect the vapor pressure near the liquid jet surface are the vapor flow area schedule in the vapor funnel, the condensation rate, w , and the amount of noncondensable gas present in the vapor.

To establish the basic relationships it will be assumed that there is no noncondensable gas present.

The situation occurring between the vapor and liquid at any axial location of the liquid jet is shown in Figure 12. At any axial location the liquid jet is characterized by its bulk temperature, T_{BL} , its mass flow rate, m_L , its bulk absolute velocity, V_L , its density, ρ_L , and its bulk enthalpy, h_L . The surface of the liquid jet is assumed to be at a temperature, T_{SL} , which is higher than the bulk and, consequently, has a liquid enthalpy, h_{LSL} , which is higher than the bulk. Also associated with the liquid surface is a saturated vapor density, ρ_{VSL} . The vapor at any corresponding axial location has temperature, T_V , pressure, P_V , density, ρ_V , and enthalpy, h_V . These are assumed to be static properties defined by the local vapor static conditions.

Due to the difference in vapor density, ρ_V and the saturation vapor density at the liquid surface, ρ_{VSL} , a net condensation flux, W_{cond}/A , occurs which is given by Equation 5 as (with $f = 1$)

$$\frac{W_{\text{cond}}}{A} = \Gamma \left(\frac{\rho_V C(T_V)}{4} - \frac{\rho_{VSL} C(T_{VSL})}{4} \right) \quad (6)$$

where $u = \frac{W_{\text{cond}}}{A \rho_V} \quad (7)$

DATA ANALYSIS

VAPOR T_v, P_v, ρ_v, h_v

$$\bullet \frac{Q}{A} \Big|_{\text{VAPOR TO SURFACE}} = \frac{Q}{A} \Big|_{\text{SURFACE TO BULK}}$$

$$\Rightarrow \frac{W_{\text{COND}}}{A} = \left(\frac{P_v}{4} C(T_v) - \frac{P_{VSL}}{4} C(T_{VSL}) \right)$$

$$U = \frac{W}{A} \frac{\text{COND}}{\rho_v}$$

$$S = \frac{U}{\sqrt{2gRT_v}}$$

$$\Gamma = e^{-S^2} + \sqrt{\pi} S (1 + \text{erf}(S))$$

$$C(T) = \sqrt{\frac{4}{\pi}} R_v g_o T$$

T_{SL}

$$\bullet \frac{W}{A} \text{COND} (h_v - h_{LSL}) = H(T_{SL} - T_{BL})$$

Figure 12

and Γ is given by equation (3)

$C(T)$ is the average molecular speed given by (1)

This net condensation flux has an associated net enthalpy of $h_V - h_{LSL}$. Thus, the liquid surface has a heat flux of Q/A vapor-surface imposed on it, where

$$\frac{Q/A}{\text{vapor-surface}} = \frac{W_{\text{cond}}}{A} (h_V - h_{LSL}) \quad (8)$$

This same heat flux must be convected from the surface of the liquid to the bulk liquid. Defining the jet convection coefficient as H , the heat flux from the liquid surface to bulk is

$$\frac{Q/A}{\text{surface-bulk}} = H (T_{SL} - T_{BL}) \quad (8a)$$

or combining (8) & (8a)

$$\frac{W_{\text{cond}}}{A} (h_V - h_{VSL}) = H (T_{SL} - T_{BL}) \quad (9)$$

It is seen from the above that the prediction of the local condensation rate, although iterative, is not difficult if a suitable value of H is available.

After some preliminary data reduction based on the assumption (Ref. 5) that H was proportional to the jet velocity, it was decided Ref. (6) that a more reasonable premise upon which to correlate H was to assume that the heat transfer coefficient, H , should be based on a characteristic velocity which is the difference between the liquid bulk velocity and the interface velocity. Subsequent to the completion of the work reported herein, it was found that Ref. (7) reported good correlation of test data of a condensing steam jet with a Stanton number based on this velocity difference. This interfacial "shear velocity" is shown in Figure 13, which is drawn for the case of zero bulk vapor velocity which is the case pertinent to the steam/water test rig. In this figure the bulk liquid jet velocity is V_L , the interface velocity is V_i and the vapor velocity is V_V , which for this case is zero far away from the jet surface.

The method of estimating the interfacial velocity is that presented in Ref. (8) which is based on equating the vapor shear stress to the liquid shear stress, including the effect of the mass transfer from the vapor to the liquid.

Defining the vapor velocity relative to the interface as V_{Vi} and the liquid velocity relative to the interface as V_{iL} it is seen that:

$$\begin{aligned} V_{Vi} &= V_i - V_V \\ V_{iL} &= V_L - V_i \end{aligned} \quad (10)$$

According to Ref. (8), the shear matching condition becomes

"SHEAR VELOCITY"

$$(\tau_i)_v = (\tau_i)_L$$

$$(\tau_i)_v = (f_i)_v \frac{\rho_v V_{vi}^2}{2g_o} + \frac{W}{A} \text{COND} \left(\frac{V_{vi}}{g_o} \right)$$

$$(\tau_i)_L = (f_i)_L \frac{\rho_L V_{iL}^2}{2g_o} + \frac{W}{A} \text{COND} \left(\frac{V_{iL}}{g_o} \right)$$

$$f_i = \frac{16}{\text{Re}_i} \quad \text{or} = \frac{.046}{\text{Re}_i^{.2}}$$

$$\text{Re}_{iL} = \rho_L V_{iL} \frac{D_j}{\mu_L}$$

$$\text{Re}_{iv} = \frac{\rho_v V_{vi} D_v}{\mu_v}$$

Figure 13

$$(\tau_i)_V = (\tau_i)_L \quad (11)$$

where:

$$(\tau_i)_V = (f_i)_V \frac{\rho_V V_{Vi}^2}{2g_o} + \frac{W_{cond}}{A} \left(\frac{V_{Vi}}{g_o} \right) \quad (12)$$

$$(\tau_i)_L = (f_i)_L \frac{\rho_L V_{iL}^2}{2g_o} + \frac{W_{cond}}{A} \left(\frac{V_{iL}}{g_o} \right) \quad (13)$$

and the friction factor, f_i is based on the relative Reynolds, number, Re_r

$$f_i = \frac{16}{Re_r} \text{ for laminar flow} \quad (15)$$

$$\text{or } f_i = \frac{0.046}{Re_r^{.2}} \text{ for turbulent flow} \quad (16)$$

$$\text{and } Re_{rL} = \frac{\rho_L V_{iL} D_i}{\mu_L} \quad (17)$$

$$Re_{rV} = \frac{\rho_V V_{Vi} D_V}{\mu_V} \quad (18)$$

where

D_j is the liquid jet diameter (nozzle metal diameter)

D_V is the equivalent diameter for the vapor passage.

For the steam/water testing, D_V was approximately 8 inches, i.e., the diameter of the acrylic cylinder.

The heat transfer coefficient was assumed to be of the form

$$H = S_t \rho_L C_{pL} V_{iL} \quad (19)$$

where S_t is a Stanton number which is assumed to be a constant for the jet.

This was done by assuming a Stanton number and then iterating the above equations for a value of T_{SL} which satisfies the heat flux equation (9) for a given value of T_{BL} . When (9) is satisfied, a heat and mass balance can be made on the liquid jet

$$\dot{m}_L C_{pL} \frac{dT_{BL}}{dX} = \frac{W_{cond}}{A} (h_V - h_{LSL}) \pi D_j \quad (20)$$

$$\frac{d\dot{m}_L}{dX} = \frac{W_{cond}}{A} \pi D_j \quad (21)$$

These equations are then integrated from the beginning to end of the jet, via a Runge-Kutta integration routine, and the process is repeated for different values of the Stanton number until the calculated outlet temperature matches the measured Dewar temperature. This process is shown in Figure 14.

In early testing, while test and data reduction procedures were being developed, a number of nozzle configurations with different turbulence promoting devices in the feed tube prior to the nozzle throat (see Figure 15) were tested. Results indicated that these upstream turbulence promoters had no substantial effect on jet performance.

In addition to finding the Stanton number that satisfied the test data, the local values of H were integrated and averaged for each test point. Jet lengths of 2 inch, 5 inch, and 10 inch were tested. The mean value and range of values of Stanton numbers for each jet length for the 0.010" and 0.0145" "turbulent" type jets are shown on Figure 16. It is apparent that there is some dependence of Stanton numbers on jet length, although for each jet length the Stanton number was assumed constant to reduce the test data. Because of this, a more realistic way of presenting the mean values of the data for the different jet lengths is as constant average values for the entire jet lengths as shown in Figure 17.

By realizing that the average Stanton number value of the 5" jet length is comprised of the average value over the first 2", which is known from the 2" jet data, and the average value of Stanton number over the 2" to 5" range, this latter value can easily be calculated based on equal areas. That is:

$$S_{t\ 0-2} \times 2 + S_{t\ 2-5} \times 3 = S_{t\ 0-5} \times 5 \quad (22)$$

or

$$S_{t\ 2-5} = \frac{S_{t\ 0-5} \times 5 - S_{t\ 0-2} \times 2}{3} \quad (23)$$

With $S_{t\ 0-2} = 0.102$ and $S_{t\ 0-5} = 0.094$, $S_{t\ 2-5}$

is calculated to be 0.0887.

Likewise, it is found that

$$S_{t\ 5-10} = \frac{S_{t\ 0-10} \times 10 - S_{t\ 0-5} \times 5}{5} \quad (23a)$$

and that with $S_{t\ 0-10} = 0.087$, $S_{t\ 5-10} = 0.08$.

A curve is then drawn through the three data points at their average distance from the injector, i.e., @ 1" for $S_{t\ 0-2}$ @ 3.5" for $S_{t\ 2-5}$ and at 7.5" for $S_{t\ 5-10}$. This curve then represents the Stanton number as a function of distance from the injector. The same procedure was carried out for the 0.0145 "turbulent" jets, and the resulting Stanton numbers vs. distance for both the 0.0145" and 0.010" "turbulent" jets are shown in Figure 18.

DATA ANALYSIS (CONT'D)

- $\frac{dm_L}{dx} = \frac{W}{A} \text{ COND}$
- $\frac{dT_{BL}}{dx} = \frac{W}{A} \text{ COND} \frac{(h_v - h_{LSL}) \pi D_j}{\dot{m}_L C_{PL}}$

FIND STANTON NUMBER THAT MATCHES DATA

Figure 14

Figure 15

Figure 16

Figure 17

Figure 18

With the Stanton number based on liquid interfacial relative velocity, it is clear that the heat transfer coefficient calculated by this technique will approach zero as the interfacial velocity approaches zero. It is also clear that pure conduction through the liquid jet would result in heat transfer coefficients in the 200-500 BTU/hr-ft² °F range which is an obvious minimum value. For this reason, a minimum value of the average heat coefficient for each jet length was deduced from the test data by plotting the average heat transfer coefficient as a function of average heat flux for each data point and extrapolating the data bank to the point of zero heat flux. These data are shown in Figure 19 for the 0.010" jet diameter for two different injection pressure drops for the three different jet lengths tested.

It was postulated that the form of the equation showing the X dependence of this minimum value of H should be

$$H_{MIN} = H_{MIN\infty} \left(1 + \frac{K}{X}\right) \quad (24)$$

$$\text{Where } H_{MIN\infty} = S_{t\ MIN} \cdot \rho_L \cdot C_{pL} \cdot V_L$$

Where $S_{t\ Min}$ is a Stanton number based on jet velocity and K is a constant, both to be determined from the test data.

Since the extrapolated values of H for the various jet lengths are average values over length of the jet, it is readily seen that if the value of H for the 5" jet is 10,000 BTU/hr-ft² °F and the value of H for the 10" jet is 6,000 BTU/hr-ft² °F, then the average value of H over the 5" to 10" increment of jet length is 2,000 BTU/hr-ft² °F.

The expression for H, equation (24) can be integrated over a section of jet length to be

$$\frac{H_{X1} - X_2}{H_{\infty}} = \frac{1}{X_2 - X_1} \left(1 + \frac{K}{X_2 - X_1} \ln \frac{X_2}{X_1} \right) \quad (25)$$

where

X_1 = Jet length at minimum edge of incremental jet length

X_2 = Jet length at maximum edge of incremental jet length

By utilizing a minimum value of $X_1 = 0.01"$ for the 10" jet length and its H value of 6,000 and the 5" to 10" H value of 2,000, the value of $H_{Min\ \infty}$ and K were calculated to be

$$H_{MIN\infty} = 1000 \frac{\text{BTU}}{\text{hr ft}^2 \text{ } ^\circ\text{F}} \quad (26)$$

$$K = 7.27"$$

For the test conditions with water $\rho_L = 63$, $V_L = 98$, $C_p = 1$, $H_{Min\ \infty} = 1,000$ corresponds to $S_{t\ Min} = 0.162$.

Thus the expression for $S_{t\ Min}$ for the 0.010 jet is

Figure 19

$$St_{Min} = 0.162 \left(1 + \frac{7.27}{12X + .01} \right) \quad (26) \text{ where } X \text{ is the distance from the injector in feet.}$$

Corresponding data for the 0.0145" turbulent jet is shown in Figure 20. From these data, the minimum Stanton number for the 0.0145" turbulent jet is

$$St_{Min} = 0.202 \left(1 + \frac{4.358}{12X + .01} \right) \quad (27)$$

These two expressions are shown in Figure 21.

The data for jet Stanton number based on interfacial relative velocity (shown in Figure 18) along with the values of St_{Min} were input to the data reduction program. The program was run to generate profiles of TBL and TSL as a function of jet length for various liquid inlet and saturated steam temperatures. These quantities are shown on Figure 22 and 23 for the 0.010" and 0.0145" diameter jets for an injection pressure drop of 80 psid, a liquid inlet temperature of 45°F and a steam temperature of 90°F.

Carpet plots showing the calculated jet outlet temperatures as a function of liquid inlet and steam temperature at jet lengths of 2, 5 and 10" are shown in Figures 24, 25, and 26 for the 0.010" diameter jet and in Figures 27, 28, and 29 for the 0.0145" jet. Also shown on these carpet plots are actual test points and the error in the predicted vs. measured liquid temperature rise. As is evident from the indicated error on the 2" and 5" jet lengths, the errors are small ($\pm 15\%$ or less) but systematic.

A potential cause of the systematic nature of the error will be discussed after the jet condenser math model is presented.

C. Jet Condenser Performance Mathematical Model

The thermal performance/sizing analytical model of the KIPS jet condenser is based on one dimensional gas dynamics in the vapor funnel, a kinetic theory representation of the vapor/liquid interface condensing process and an empirical correlation of the intra-jet heat transfer coefficient from the vapor/liquid interface to the bulk liquid.

The analytical model is used to predict the condenser vapor inlet pressure that is required to condense a given vapor flow rate with a specified liquid flow rate, liquid inlet temperature, liquid injector geometry, and vapor funnel geometry.

The analytical model is essentially the set of differential equations that describe the vapor momentum and continuity and the liquid energy, and continuity. These are integrated from inlet to outlet of the jet condenser for various values of vapor inlet pressure to find a value of pressure which results in the required amount of vapor condensation.

The energy equation for the vapor was not considered because there is little heat transfer except via condensation on the liquid jets. The momentum equation for the liquid was not considered, because those vapor molecules near the liquid/vapor interface which condense on the liquid jets have a bulk axial velocity not much different than the bulk liquid jet velocity. It is estimated that the momentum interaction between the vapor and liquid jets is less than 1% of the liquid jet momentum for the KIPS conditions.

Figure 20

Figure 21

0145

Figure 22

0100

Figure 23

NOTE: ARROW TIP LOCATES TEST STEAM & LIQUID TEMPS. AND PREDICTED DEWAR TEMP.

Figure 24

Figure 25

Figure 26

Figure 27

Figure 28

Figure 29

The equations pertinent to the vapor phase in what follows are written for the case in which a noncondensable gas is present along with the vapor.

The momentum equation for the vapor phase is taken from Shapiro (Ref 9) and is

$$\frac{dP}{P} + \frac{\rho V^2}{2gP} \frac{dV^2}{V^2} + \frac{\rho V^2}{2gP} \left(\frac{4f_w}{D_w} dx + \frac{dX}{\frac{\rho V^2}{2g} A} \right) + \frac{\rho V^2}{gP} (1-y) \frac{dW}{W} = 0 \quad (28)$$

where

- P = static pressure in the gaseous phase
- ρ = static density of the gaseous phase
- V = bulk axial velocity of the gaseous phase
- g = gravitational constant
- f_w = drag friction coefficient between the gaseous phase and the funnel wall
- D_w = funnel diameter
- X = the drag force exerted on the gaseous phase by the liquid jets
- A = the gaseous phase flow area
- W = the mass flow rate of gaseous phase
- $\frac{x}{L}$ = axial distance

In Shapiro's derivation dW is the total increase in mass flow of the gaseous phase and
 $y = V_i/V$ (29)

where V_i is the axial velocity of dW as it crosses the gaseous system boundary, i.e. the interfacial velocity between the liquid and gaseous phase.

A depiction of the velocity distribution in the gaseous phase, interfacial region and in the liquid jet is shown in Figure 30 for the case where the vapor phase is moving at a higher velocity than the liquid. At this time the subscript G will be applied to the bulk velocity of the vapor (vapor + noncondensable gas) and L to the bulk velocity of the liquid. In Figure 30 the gaseous phase velocity is V_G , the interfacial velocity is V_i and the liquid velocity is V_L . The gaseous phase velocity relative to the interface velocity is V_{Gi} and the liquid velocity relative to the interface velocity is V_{iL} .

The drag force X is calculated on the gaseous phase relative interface velocity V_{Gi} , and the liquid heat transfer coefficient is based on the liquid interfacial relative velocity V_{iL} . The drag friction coefficient f_w is based on the gaseous phase velocity V_G . The method used to calculate the interface velocity is that presented in Ref 8 which involves equating the effective shear stress of the vapor and liquid at the interface.

The effective shear stress includes both the frictional component based on the velocity relative to the interface, and the momentum component associated with the mass transfer (condensation) occurring at the interface. Thus the effective gas phase shear stress on the interface is

Figure 30

$$(\tau_i)_G = (f_i)_G \rho_G \frac{V_{Gi}^2}{2g} + \frac{W_{cond}}{A} \frac{(V_{Gi})}{g} \quad (30)$$

where $(f_i)_G$ is the gas phase drag friction factor with the Reynolds number Re based on the interfacial relative velocity, V_{Gi} and the gas phase hydraulic diameter D_V based on the funnel flow area and the wetted perimeter of the funnel wall and the liquid jets.

$$(f_i)_G = \frac{16}{Re_G} \text{ or } \frac{0.046}{Re_G^{0.2}} \quad (31)$$

depending on laminar or turbulent values of Re . W/A_{cond} is the condensate mass flux from the vapor to the liquid.

The liquid effective interface shear stress is

$$(\tau_i)_L = (f_i)_L \rho_L \frac{V_{iL}^2}{2g} + \frac{W_{cond}}{A} \frac{(V_{iL})}{g} \quad (32)$$

where $(f_i)_L$ is the drag friction coefficient based on the liquid interfacial relative velocity, V_{iL} , and the liquid jet diameter, D_j . The value of V_i is that which results in $(T_i)_V = (T_i)_L$ (33)

The local condensing mass flux is calculated from kinetic theory Ref 4 as

$$\frac{W_{cond}}{A} = (\Gamma_V Y_{Vi} \frac{P M_V}{R T_V} \frac{C_V (T_V)}{4} - \rho_{VSL} \frac{C_V (T_{SL})}{4}) \quad (34)$$

$$C_V (T) = \sqrt{\frac{8 R T}{\pi M_V}} \quad (35)$$

$$\Gamma_V = e^{-s^2} + \sqrt{\pi} s (1 + \text{erf}(s)) \quad (36)$$

$$S = \frac{u}{\sqrt{\frac{2 R g_0 T_V}{M_V}}} \quad (37)$$

$$U = \frac{W_{cond}}{A} \frac{R T_V}{M_V P} \quad (38)$$

where

P = static pressure of vapor/non condenser gas mixture in funnel

Y_{Vi} = vapor mole fraction at vapor/liquid interface

M_V = molecular weight of vapor

R = universal gas constant

T_v	=	vapor temperature in funnel
$C_v(T)$	=	average vapor molecule molecular speed
ρ_{vsL}	=	saturated vapor density at temperature T_{sL}
T_{sL}	=	jet liquid surface temperature
U	=	bulk vapor velocity normal to jet surface
S	=	ratio of bulk vapor velocity normal to jet to mean molecular speed
Γ_v	=	factor which biases the molecular collision rate with the liquid jet to account for the bulk vapor flow toward the jet surface. (Ref 4)

The condensing mass flux, W/A cond., has a net enthalpy of $h_v - h_{LSL}$ associated with it

where

h_v	=	vapor enthalpy
h_{LSL}	=	liquid enthalpy at the liquid/vapor interface.

This mass flux enthalpy difference product is a heat flux which is imposed on the liquid surface and must be convected to the interior of the liquid jet. Defining H as the jet convection heat transfer coefficient, from surface to bulk liquid, the equality of heat flux from the vapor to the surface and from the surface to the bulk liquid is given as

$$W/A \text{ Cond. } (h_v - h_{LSL}) = H (T_{LSL} - T_{BL}) \quad (39)$$

where

T_{BL}	=	liquid jet bulk temperature
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The local condensing rate is known when eq. (39) is satisfied.

The value of H is based on empirical data generated on a steam/water single jet test rig for which a correlation of Stanton number, St , was obtained for various jet configurations as a function of jet length. This Stanton number was based on the liquid interfacial relative velocity, V_{iL} . Thus H is defined as

$$H = St \cdot \rho_L \cdot C_{pL} \cdot V_{iL} \quad (40)$$

where

C_{pL}	=	liquid specific heat
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and

$$S_t = S_t(x)$$

where

$$x = \text{jet length from injector (Feet)}$$

In the rig testing Stanton numbers were obtained for several different nozzle types. The Stanton numbers are given in Figure 31 for the 0.010" diameter and 0.0145" dia. injectors.

The H as calculated using this Stanton number was compared to a minimum value $H_{\min}$, calculated from

$$H_{\min} = S_{t\min} \rho_L C_{pL} V_L \quad (41)$$

which is based on the bulk jet velocity and a corresponding minimum Stanton number $St_{\min}$ shown in Figure 32 which was determined from extrapolating test data to the point where $V_{iL} = 0$. The larger of these two values is used in the calculations.

The value of the vapor mole fraction at the jet interface, Y_{Vi} , will now be addressed. It is assumed that the noncondensable gas is not soluble in the liquid.

At any axial location in the vapor funnel, the bulk gas (vapor plus noncondensable gas) has a local bulk mole fraction of $Y_{NC\infty}$. The noncondensable gas is carried along with the vapor molecules in the bulk flow toward the liquid jet surface, the bulk flow velocity being U . This situation is depicted in Figure 33. Since, by assumption, the liquid interface does not absorb noncondensable gas molecules, the noncondensable gas sets up a concentration gradient which results in a diffusion flux away from the interface which exactly balances the bulk flow flux of noncondensable gases toward the interface.

The bulk flow flux of noncondensable gas is the local product of bulk velocity and noncondensable mole fraction while the diffusional flux is the product of the gradient of mole fraction and the diffusion coefficient.

Thus

$$U Y_{NC} = -D_{V-NC} \frac{dY_{NC}}{dy} \quad (42)$$

holds from the bulk free stream to the liquid interface.

Where

$$D_{V-NC} = \text{the vapor-noncondensable gas diffusion coefficient}$$

$$\frac{dY_{NC}}{dy} = \text{gradient of noncondensable mole fraction away from the interface}$$

Figure 31

Figure 32

Figure 33

The definition of the mass transfer coefficient (similar to heat transfer coefficient) is

$$H_D = \frac{-D_{V-NC} \frac{dY_{NC}}{dy}}{(Y_{NCi} - Y_{NC\infty})} \quad (43)$$

using the definition of H_D (equation No. 43) in equation (42) results in

$$U Y_{NCi} = H_D (Y_{NCi} - Y_{NC\infty}) \quad (44)$$

or upon rearranging

$$Y_{NCi} = Y_{NC\infty} \left(\frac{H_D}{H_D - U} \right) \quad (45)$$

The problem now is how to evaluate H_D . The approach taken here is to view the situation of the vapor condensing on the liquid jets and the noncondensable gas diffusing away from the liquid jets as analogous to the situation of mass transfer in laminar flow over a flat plate with suction at the wall. This problem is treated in the classical boundary layer - stream function approach in numerous texts. See, for example, Ref. 10. The resulting differential equations, shown in Appendix B, have been solved numerically. The results are presented in Figure 34 as the ratio H_D/H_{DO} as a function of the suction parameter, $V_0/V_\infty \sqrt{R_{ex}}$ for various Schmidt numbers, S_C .

In the boundary layer analysis V_0 is the bulk velocity normal to the wall, and is identical to U , the velocity normal to the jet surface, and V_∞ is the free stream axial velocity, which is identical to V_G the bulk vapor and noncondensable gas axial velocity in the funnel. The Reynolds number, R_{ex} is based on the bulk gas velocity and the distance, x , from the inlet.

H_{DO} is calculated from

$$H_{DO} = .332 \frac{D}{x} S_C .343 \sqrt{R_{ex}} \quad (46)$$

$$\text{Where } S_C = \frac{\mu}{\rho D} \quad (47)$$

and μ is the gaseous phase viscosity

ρ is the density

D is the diffusion coefficient.

Knowing Y_{NCi} allows Y_{Vi} to be calculated via

$$Y_{Vi} = 1 - Y_{NCi} \quad (48)$$

Figure 34

At this point in the analysis it can be seen that if the values of vapor velocity, V_G , pressure, P , liquid velocity, V_L , free stream noncondensable gas mole fraction, $Y_{NC\infty}$, vapor temperature T_V , bulk liquid temperature, T_{BL} are known at any axial location, x , then the values of the interfacial velocity, V_i liquid surface temperature, T_{SL} , and the condensing mass flux, $\frac{W_{cond}}{A}$ can be found

which mutually satisfy equations (30) through (48). Finding these values is iterative task but is easily done with the digital computer.

Once $\frac{W_{cond}}{A}$ is known, an energy balance on the liquid streams results in the bulk liquid differential equation

$$\frac{d T_{BL}}{dX} = \frac{W_{cond}}{A \dot{m}_L C_{pL}} (h_v - h_{LSL}) N \pi D_j \quad (49)$$

and the mass balance equations

$$\frac{dW}{dX} = - \frac{W_{cond}}{A} N \pi D_j \quad (50)$$

$$\frac{d \dot{m}_L}{dX} = - \frac{dW}{dX} \quad (51)$$

where

W = the gaseous phase mass flow rate

$\dot{m}_L$ = liquid mass flow rate

N = number of liquid jets

D_j = diameter of individual jets.

Assuming that the noncondensable molecules have the same bulk velocity and temperature leads to

$$Y_{NC\infty} = \frac{\frac{\dot{m}_{NC}}{M_{NC}}}{\frac{\dot{m}_V}{M_V} + \frac{\dot{m}_{NC}}{M_{NC}}} \quad (52)$$

from which

$$\frac{d Y_{NC\infty}}{dX} = \frac{\frac{-\dot{m}_{NC}}{M_{NC}}}{\frac{\dot{m}_V}{M_V} + \frac{\dot{m}_{NC}^2}{M_{NC}^2}} \frac{dW}{dX M_V} \quad (53)$$

where

$$\begin{aligned}\dot{m}_{NC} &= \text{mass flow rate of noncondensable} \\ M_{NC} &= \text{molecular insight of noncondensable} \\ M_V &= \text{molecular weight of vapor} \\ \dot{m}_V &= \text{vapor mass flow rate}\end{aligned}$$

The molecular weight of the gaseous phase is

$$M = Y_{NC\infty} M_{NC} + (1 - Y_{NC\infty}) M_V \quad (54)$$

At this point it is convenient to update and simplify the momentum equation eg. (28) utilizing V_G to denote the gaseous phase velocity.

Rewriting equation 1 results in

$$\frac{dP}{dx} = -\left[\frac{\rho}{2g} \frac{dV_G^2}{dx} + \frac{\rho V_G^2}{2g} \left(\frac{4f_w}{D_w} + \frac{1}{\frac{PV_G^2}{2g} A} \frac{dX}{dx} + \frac{2}{w} \left(1 + \frac{V_i}{V_G} \right) \frac{dw}{dx} \right) \right] \quad (55)$$

In this equation $\frac{dw}{dx}$ is given by (50)

dX/dx can be calculated from the gaseous phase relative velocity V_{Gi} and the interfacial friction coefficient $(fi)_G$ as

$$\frac{dX}{dx} = \pm (fi)_G \frac{\rho V_{Gi}^2}{2g} \cdot N \pi D_j \quad (56)$$

where the sign is + when the $V_G > V_L$ and - when $V_G < V_L$.

The wall drag friction coefficient, f_w is calculated based on the velocity V_G and the wall diameter D_w as

$$f_w = .046 \left(\frac{\rho V_G D_w}{\mu} \right)^{-.2} \quad (57)$$

The quantity V_G^2 is calculated from the mass flow rate, pressure, temperature, molecular weight and flow area as

$$V_G^2 = \frac{W^2 R^2 T_V^2}{P^2 M^2 A^2} \quad (58)$$

It is assumed that $T_V = \text{constant}$ throughout the funnel since the vapor velocities are relatively low (dynamic temperatures $\approx 50^\circ\text{F}$). The absolute temperature level is $\sim 700^\circ\text{R}$ and the vapor is not superheated very much ($\sim 20^\circ\text{F}$) so that little error is involved by assuming it constant. With T_V assumed constant eq (58) is easily differentiated to show that

$$d(V_G^2) = 2 V_G^2 \left(\frac{dW}{W} - \frac{dP}{P} - \frac{dM}{M} - \frac{dA}{A} \right) \quad (59)$$

Substituting (59) & (58) into (55) and rearranging results in

$$\frac{dP}{dx} = \frac{\rho \frac{V_G^2}{g} \left[\frac{1}{A} \frac{dA}{dx} + \frac{1}{M} \frac{dM}{dx} - \frac{1}{W} \frac{dW}{dx} \left(2 - \frac{V_i}{V_G} \right) - \frac{2f_W}{D_W} \right] - \frac{1}{A} \frac{dX}{dx}}{1 - \frac{M V_G^2}{g R T_V}} \quad (60)$$

where

$$\rho = \frac{M P}{R T_V} \quad (61)$$

By specifying a funnel geometry, the value of A and dA/dX are known. The value of dM/dX is easily derived by differentiating eq (54)

$$\frac{dM}{dX} = (M_{NC} - M_V) \frac{dY_{NC\infty}}{dX} \quad (62)$$

Having defined differential equations for dT_{BL}/dx (eq 49), dW/dx (eq 50), $d\dot{m}_L/dx$ (eq 51), $dY_{NC\infty}/dx$ (eq 53), dP/dx (eq 60), dA/dx from the funnel geometry and dM/dx (eq 62), these differential equations can be integrated by any suitable numerical method. A 4th order Runge-Kutta routine was incorporated into a digital computer program which performed the iterative process to determine the values of T_{SL} , W/A cond and V_i to allow the differentials to be evaluated. These differentials then integrated via the Runge-Kutta routine.

The problem is integrated from the jet liquid injection plane where the mass flows and temperatures are specified.

The integration is carried out for various gaseous phase inlet pressures until that value which allows the full vapor flow to be condensed within the jet condenser vapor funnel is determined.

The jet condenser math model with 45* .0145" jets was exercised in an attempt to match the measured performance of the KIPS development jet condenser. It was found necessary to increase the intra jet heat transfer coefficients (eq 40 & 41) by a factor of ~ 5 in order to match the measured jet condenser data. Only those data where the presence of noncondensable gas was very low (< 5 ppm) were used and the model was used with zero noncondensables.

Detailed results of the jet condenser math model are given in Figure 35 for the development jet condenser where the vapor mass flow, vapor velocity, vapor pressure bulk liquid and jet surface temperatures and vapor funnel geometry are shown as a function of the distance from the injector. It should be noted that the pressure at the injector plane is $\sim .83$ times the regenerator outlet/jet

* Actually, 38 jets were used in this analysis since it was found that about 7 nozzles were effectively plugged by bits of "O" ring when jet condenser performance measurements were taken.

Figure 35

condenser inlet bell pressure due to the area ratio of the inlet bell to injector plane.

A comparison of the analytically predicted jet condenser inlet bell vapor pressure as a function of liquid inlet temperature with measured values from the development jet condenser for liquid mass flow rates of $\sim .28$ lb/sec and vapor flow rates of $\sim .032$ lb/sec are shown in Figure 36. The agreement of this prediction and test data are quite close.

The predicted jet condenser bell inlet pressure as a function of liquid inlet temperature for the 90 nozzle .010" diameter GDS jet condenser is shown in Figure 37. The detailed results of the math model for this design are shown in Figure 38.

The effect of noncondensable gas on jet condenser operation was not investigated with the jet condenser math model, although the model was written to allow such investigation. This was due to the fact that by the time the math model was ready to be exercised, test data were available which showed the sensitivity of the development jet condenser to noncondensable gas. These test data indicated the need to keep noncondensable gas concentrations (air) to rather low levels < 10 ppm, so that at that time analytical effort was focused on devising and sizing a suitable noncondensable gas separator.

D Noncondensable Gas — A Potential Explanation for Needing the Stanton Number Multiplier

As indicated earlier, the need for the multiplier on the Stanton number to match the test data from the KIPS development jet condenser may be due to noncondensable gas effects in the basic steam/water rig test data.

Test data for the two inch jet lengths are presented in Figure 39 for the 0.010 in. and 0.0145 in. diameter jets. As is evident from the data, the deduced Stanton number increases as the heat flux is decreased. The lower heat fluxes occur at lower steam pressures. The value of the diffusion coefficient increases as the pressure is decreased, so it is possible that the increasing diffusion coefficient lessens the amount of noncondensable gas poisoning.

Two data points for the 0.010 in. jet were analyzed utilizing the noncondensable poisoning equations presented in the previous section and treating the jet as one increment.

One data point was at 0.5 psia (79°F) and the other at 0.7 psia (91°F). Using one inch as the characteristic length, the suction parameter ($V_0/V_\infty \sqrt{R_{ex}}$) was 48.9 for the higher pressure and 36.2 for the lower pressure data point. It was also assumed that the unpoisoned Stanton number was 0.55 which is $\sim$ five times the average for the two inch jet. With this assumption, it was possible to calculate the average liquid surface temperature from which the interfacial noncondensable gas concentration is then calculable from equation (34) and (48). This resulted in interfacial gas mole fractions of 0.33 and 0.39 for the 0.5 psia and 0.7 psia pressures.

The value of the diffusion coefficient of steam/air is 2.15×10^{-2} ft²/sec. and 1.54×10^{-2} ft²/sec at the .5 and .7 psia pressure levels.

Utilizing eq (46) with a $S_C = .226$, which pertains to these conditions, values of H_{D0} of 2.135 ft/sec and 1.827 ft/sec correspond to the .5 and .7 psia data points. The data from Figure 34 were

Figure 36

Figure 37

Figure 38

Figure 39

rearranged to show $\left(\frac{h_D - V_O}{h_{DO}}\right)$ as a function of the suction parameter and is shown in Figure 40. From this figure it is estimated that $\left(\frac{h_D - V_O}{h_{DO}}\right) = .242$ and $.261$ respectively for the .5 and .7 psia points.

The suction velocities are 87.12 and 98.43 ft/sec respectively for the .5 and .7 psia points. Thus from eq (45) the free stream to interface concentration ratios are calculated to be 170 and 207 for the .5 and .7 psia data points.

The resulting estimate for free stream noncondensable gas concentration for air in the steam is .001904 and .00195 for those two points.

The fact that they are about the same level lends some credence to the supposition that gas poisoning may be responsible for the steam/water test results being too low.

No vapor sample was taken from the rig while operating on steam and water. However, when the rig was converted over to use Dowtherm vapor it was impossible to get useful data due to the presence of air. A gas sample was taken at that time and the concentration of air was about .03, which is of the proper magnitude, per the analysis technique used herein to assess noncondensable gas effects, to cause the observed poor jet performance.

Figure 40

APPENDIX A TEST RESULTS

GENERAL

All jet condenser tests utilizing Dowtherm as the working fluid were performed on the test loop which is schematically presented as Figure A-1. With this loop, it was possible to accurately set all state points surrounding the jet condenser. Component names and valve numbers referred to in the following procedures are as schematically shown.

BACKGROUND

The liquid interface in a jet condenser is the location where the high velocity injected jets undergo a sudden expansion to fill the diffuser. Normally, this interface is stably positioned well downstream of the throat. However, under some abnormal conditions, this interface can move upstream in the diffuser and through the throat which causes the jet condenser to flood out.

There are three primary causes for floodout: scavenge flow, misalignment and noncondensable gases.

Jet condenser performance degradation and tolerance limits due to each cause were experimentally determined. Tolerances to scavenge flow and misalignment meet the design requirements. The jet condenser thermal performance was found to be sensitive to noncondensable gases and therefore a noncondensable removal system has been incorporated to obviate this potential problem.

All floodout tests were performed with a jet condenser comprised of a 45 at .0145 inch diameter element liquid injector, a compound angle vapor funnel, a .133 inch diameter throat and a compound angle diffuser.

SCAVENGING

It is required that the bearing flow be returned to the system. To achieve minimum losses, the scavenging should be done at low pressure. Return at the jet condenser is the obvious choice. In addition during startup, some condensation will occur on the cold housings. In a one g field, this fluid will drop out to the jet condenser where it must be scavenged without causing a perturbation to system operation. Tests were made to determine the jet condenser tolerance to these scavenge flows.

The test configuration jet condenser and scavenge locations are shown in Figures A-2 and A-3. Three locations were tested: throat, funnel wall and funnel center. At the throat, the flow is returned through a gap .010 inch wide. The flow here is very similar to that in a jet pump.

The procedure was to establish jet condenser conditions with no scavenge flow and then to introduce scavenge flow in small steps until floodout occurred. Tests were run with and without vapor flow and at varying levels of jet condenser recovery.

There was no discernable change in the scavenging capability with vapor flowing or without vapor flowing. Even more significantly, there was no change in scavenging capability with increasing recovery to within approximately two percentage points of the maximum available recovery.

Figure A-1

Figure A-2

Figure A-3

The maximum scavenging capability was not determined due to instrumentation limitations. In all locations, scavenge flow capability exceeded 133% of the design bearing flow. One test indicated ability to scavenge up to 300% of the bearing flow. In addition, tests were made with flow of 133% bearing flow concurrently scavenged at the funnel wall and at the throat.

The results indicate that the bearing flow can be successfully scavenged at any jet condenser location. Scavenge on the funnel wall or funnel center depends on vapor drag to assure proper return of the scavenge to the throat during operation in zero g. Calculations show that the drag is marginally adequate in normal operation but in the presence of even slight accelerations opposite the flow direction, the scavenge will be perturbed and fluid will be "stored" in the vapor space. This in itself is not troublesome but when the acceleration is relieved, this "stored" inventory tries to flush through the throat. This attempted scavenge of large amounts of fluid has caused floodout on the rig. Scavenging at the throat requires the bearing cavity scavenge system to generate some pressure (2.75 psi at design flow - refer to Figure A-4). This occurs at a slight cost in system power; however, since the throat scavenge relies only on liquid momentum, it will function in zero g as well as in moderate acceleration fields generated in the direction opposite to flow such as exist during launch or orbit maneuvers. Scavenge at the throat has been selected for the KIPS system.

MISALIGNMENT

During launch or orbit transfer, an induced environment is established comprised of superimposed accelerations and vibration. In the jet condenser, the result of this environment is a deflection of the liquid jets relative to the throat. Acceleration causes a continuous shift proportional to the g level. Vibration causes an oscillating shift, centered about the constant acceleration shift. The deflections are illustrated in Figure A-5. These relative deflections must not cause the jet condenser to flood out.

The test configuration jet condenser is illustrated in Figure A-6. Two dial indicators were used at the throat location to measure the deflection relative to the initial, as focused (see TP-414), location.

The jet condenser was established at conditions representative of overspeed system operation. It is preferred that launch always be taken at this higher speed in order that the benefits of stiffer bearings and jet condenser liquid streams be realized. It should be noted that electric power can be supplied in overspeed operation. As the pressure available at the jet condenser liquid inlet is dependent on the electric load, several pressures bracketing the expected value were tested. Pressure recovery was set at 20% (the system accumulator will maintain an essentially constant jet condenser outlet pressure) and system power loop flow was set at 115% of nominal (as in system operation at overspeed).

The vapor funnel was then deflected relative to the injector by moving one jacking screw causing the funnel to pivot about the other two. Deflections were measured with the dial indicators. The deflection was increased in steps until floodout occurred. The funnel was then returned to its original position and the deflection repeated in the opposite direction.

The test results are presented in Table A-A. As expected, the higher injection pressures achieve greater deflection capability. At the minimum pressure (160 psig), the capability is $\pm .060$ inch. It is further indicated that 133% bearing scavenge flow at the throat has no effect on the misalignment

Figure A-4

ACCELERATION AND VIBRATION

Figure A-5

Figure A-6

Table A-A Misalignment

INJECTION PRESSURE (psig)	$\dot{m}_v$ (lb/sec)	RECOVERY	BEARING FLOW AT THROAT (lb/sec)	MAX DEFLECTION (in)
200	.0345	20%	—	+.080
180	.0345	20%	—	+.080
160	.0345	20%	—	+.080
200	.0345	20%	—	-.070
180	.0345	20%	—	-.065
160	.0345	20%	—	-.055
200	.0345	20%	.012	+.080

Figure A-7

capability. Figure A-7 are the results plotted as throat static pressure against deflection and injection pressure. It can be seen that deflections of $\pm .020$ inch have essentially no effect. The constant deflection due to acceleration is predicted to be .020 inch and the vibration deflection to oscillate about this "null".

This is significant in that deflections greater than .020 inch, when deleterious effects (increased bearing scavenge pressure) begin to accrue, will be sinusoidal in nature and not continuous as in the test. Temporary excursions of greater amplitude than the measured continuous deflections may be acceptable.

The throat has been maintained at .133 inch diameter based on these results.

NONCONDENSABLE GASES

Dowtherm undergoes a slight pyrolytic degradation due to being at the peak cycle temperature. In this degradation, several products are formed as the carbon rings and chains are broken. Some of these products are noncondensable gases, mainly hydrogen and methane. These gases tend to concentrate in the condenser due to natural separation there (See Figure A-8). If present in significant quantities, these gases can cause a performance degradation due to the occlusion of the liquid streams. The performance degradation is in the form of an increased vapor pressure required to condense a given vapor flow rate. In severe cases of gas accumulation, floodout can occur due to brooming. This is due to the entrained gas expanding and disrupting flow when accelerated through the liquid injectors.

The jet condenser test configuration was as described above.

The test consisted of establishing an equilibrium concentration of air, then measuring the concentration and thermal performance (vapor inlet pressure). The minimum concentration was achieved by forcing all flow through the deaerator. Higher concentrations were achieved by bypassing some flow around the deaerator.

The concentration was measured by opening an evacuated ullage volume to the jet condenser vapor inlet. Condensation occurred on the walls of the ullage until the sub-cooled walls were all occluded by the separated noncondensable gases. By measuring the volume of collected liquid and knowing the total available volume and pressure, the concentration can be calculated. This technique is illustrated in Figure A-9. The pressure was measured utilizing a Dowtherm liquid manometer.

The results are presented as Figure A-10. It can be seen that at a concentration of 30 ppm, the system back pressure level is $\sim 15\%$ higher than with zero ppm. At a concentration of approximately 90 ppm, floodout occurs.

Due to this sensitivity, a noncondensable gas removal system has been incorporated into the KIPS system.

X = FREE STREAM NONCONDENSABLE CONCENTRATION

ASSUMING INSOLUBILITY OF NONCONDENSABLE IN LIQUID

Figure A-8

CALCULATION OF NON CONDENSABLE CONCENTRATION

$$C = \frac{\text{cc NC}}{\text{cc Dow}} (\text{PRVO}) (K)$$

$$\frac{\text{mole NC}}{\text{mole Dow}} \quad \text{psia}$$

$$\text{cc N} = \text{total available cc} - \text{cc Dow}$$

$$K = \left[\frac{1}{66} \quad 1728 \quad \frac{1}{.06102} \quad \frac{166}{453.6} \right] \left[\frac{1}{22.4 \times 10^3} \right] \left[\frac{1}{14.4} \right]$$

$$\left[\frac{\text{ft}^3}{\text{lbm}} \quad \frac{\text{in}^3}{\text{ft}^3} \quad \frac{\text{cc}}{\text{in}^3} \quad \frac{\text{lbm}}{\text{mole}} \right] \left[\frac{\text{mole}}{\text{cc}} \right] \left[\frac{1}{\text{psia}} \right]$$

$$K = 4.868 \times 10^{-4} \quad \left[\frac{\text{cc}}{\text{mole}} \right]_{\text{Dow}} \quad \left[\frac{\text{mole}}{\text{cc}} \right]_{\text{NC}} \quad \left[\frac{1}{\text{psia}} \right]$$

Figure A-9

Figure A-10

APPENDIX B KIPS JET CONDENSER NONCONDENSABLE GAS BOUNDARY LAYER ANALYSIS

For the steady, two-dimensional flow of an incompressible fluid over a porous flat plate, the simplified boundary-layer equations of momentum, energy, and mass constitute a set of four nonlinear partial differential equations. Introducing the concept of the stream function and using a similarity transformation this can be reduced to a set of three nonlinear, ordinary differential equations:

$$\frac{d^3 f}{d\eta^3} + \frac{1}{2} \cdot f \cdot \frac{d^2 f}{d\eta^2} = 0$$

$$\frac{d^2 \theta}{d\eta^2} + \frac{1}{2} \cdot Pr \cdot f \cdot \frac{d\theta}{d\eta} = 0$$

$$\frac{d^2 \phi}{d\eta^2} + \frac{1}{2} \cdot Sc \cdot f \cdot \frac{d\phi}{d\eta} = 0$$

subject to the following boundary conditions:

$$\text{at } \eta = 0; \frac{df}{d\eta} = \theta = \phi = 0; f = -2 \cdot \frac{V_o}{V_\infty} \sqrt{Re_x}$$

$$\text{at } \eta = \infty; \frac{df}{d\eta} = \theta = \phi = 1$$

This constitutes a nonlinear two-point boundary value problem, which can be solved with a digital computer. The solution technique used involves continuously varying the initial conditions on

$$\frac{d^2 f}{d\eta^2}, \quad \frac{d\theta}{d\eta} \quad \& \quad \frac{d\phi}{d\eta}$$

until the final (steady state) values of

$$\frac{df}{d\eta}, \quad \theta \quad \& \quad \phi$$

equal unity.

Back transforming the above solution to find the mass transfer coefficient results in:

$$h_D \cdot x = Re_x^5 \cdot \left(\frac{d\phi}{d\eta} \right)_0$$

For the suction parameter equal to zero the Sherwood number is given by:

$$\frac{h_{D0}}{D} \cdot X = .332 \cdot S_C^{.343} \cdot Re_X^{.5}$$

The ratio of the mass transfer coefficients is:

$$\frac{h_D}{h_{D0}} = 3.012 \cdot S_C^{.343} \cdot \left(\frac{d\phi}{d\eta} \right)_0$$

The results of the digital solution are given in Figure 34 of the main report.

For values of the suction parameter less than -10, an approximate linear solution exists based on the fact that f doesn't change significantly from its value at η equals zero.

$$\phi \doteq 1 - \exp \left(S_C \cdot \frac{V_0}{V_\infty} \cdot \sqrt{Re_X} \cdot \eta \right)$$

$$\left(\frac{d\phi}{d\eta} \right)_0 \doteq - S_C \cdot \frac{V_0}{V_\infty} \cdot \sqrt{Re_X}$$

Thus, the ratio of the mass transfer coefficients is approximately:

$$h_D \doteq -3.012 \cdot S_C^{.657} \cdot \frac{V_0}{V_\infty} \cdot \sqrt{Re_X}$$

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