General Disclaimer

One or more of the Following Statements may affect this Document

- This document has been reproduced from the best copy furnished by the organizational source. It is being released in the interest of making available as much information as possible.

- This document may contain data, which exceeds the sheet parameters. It was furnished in this condition by the organizational source and is the best copy available.

- This document may contain tone-on-tone or color graphs, charts and/or pictures, which have been reproduced in black and white.

- This document is paginated as submitted by the original source.

- Portions of this document are not fully legible due to the historical nature of some of the material. However, it is the best reproduction available from the original submission.

Produced by the NASA Center for Aerospace Information (CASI)
ABSTRACT

This report presents the results of a study to determine the effectiveness of a mercury inventory control system to regulate condensing pressure within limits bounded by mercury pump cavitation on the low side and turbine output power requirements on the high side.

The results show that the inventory control system can regulate condensing pressure within system limits although it is more effective in its ability to increase condensing pressure than to reduce it. Limitations exist involving system instability for narrow pressure regulation and the biasing effects which could result from vehicle acceleration.
Table of Contents

I  INTRODUCTION

II  DESCRIPTION OF THE INVENTORY CONTROL SYSTEM

III DESCRIPTION OF THE CONDENSER COMPUTER SIMULATION

IV  DESCRIPTION OF THE COMPUTER RUNS

V  SUMMARY AND CONCLUSIONS

VI  APPENDIX

A. Dynamic Analysis of the Mercury Inventory Control
B. Condenser Equations

TABLE I  Tabulated Results of Computer Runs

FIG. A  Condenser Inventory Control System

FIG. B  Condenser Pressure vs. Condensing Length

FIG. C  Condenser Inventory vs. Condensing Length

FIG. D  Condenser Inventory Control System Block Diagram

FIG. E  Sun - Shade Computer Run With Inventory Control

FIG. F  Sun - Shade Computer Run Without Inventory Control
I INTRODUCTION

This study was undertaken to determine the characteristics of a condenser inventory control system and its effectiveness in controlling mercury condensing pressure for all disturbances the SNAP-8 system is likely to encounter during a 10,000 hour mission. For the control system to be effective it is required to maintain the condensing pressure above a low limit dictated by the mercury pump minimum NPSH to prevent pump cavitation and below a high limit dictated by a failure to provide required turbine power output due to excessive back pressure. The control system was added to the analog computer simulation of the SNAP-8 flight reference system (Rev. B) and computer runs were made.

II DESCRIPTION OF THE INVENTORY CONTROL SYSTEM

The condenser inventory control system as shown in Figure A consists of a metal cylinder which contains a metal bellows with precharged gas on one side and liquid mercury on the discharge side. The cylinder is connected to the condenser mercury discharge manifold such that a movement of the bellows due to pressure changes causes a shift in condenser inventory and therefore condensing length. Since the condensing pressure is a direct function of condensing length, an inventory system should be able to control the condensing pressure. As an example of how this system operates a sun-shade system transient can be examined. When the main radiator goes from a sun to a shade environment, the NaK temperature of the condenser drops approximately 20°F causing the condensing pressure of the mercury to drop. This causes a pressure unbalance across the control bellows which results in mercury being shifted into the condenser until pressure equilibrium is again established. The condensing pressure increases as the condensing length decreases until the inventory control stops injecting mercury. There is some offset of the condensing pressure from the original control set point due to:

1. The bellows spring gradient
2. Change in gas pressure with bellows volume changes
3. Change in condenser pressure drop with condensing length

The offset is minimized in the direction of decreasing spring gradient and increased gas volume.
An analysis of the dynamics of the control system is shown in the appendix along with the resultant transfer function for the system. This analysis contained the following assumptions:

1. The system is operating in an earth orbit under zero "G" conditions. (there are no vehicle acceleration forces)
2. The mercury in the inventory cylinder acts as one continuous mass.
3. Fluid damping is present due to the movement of the liquid mercury through a restricted area entering or leaving the cylinder.
4. The damping is viscous (proportional to the first power of velocity).

III DESCRIPTION OF THE CONDENSER COMPUTER SIMULATION

The analog computer simulation of the SNAP-8 condenser was modified to reflect the latest information available from the "MECA" testing of the condenser. The condenser overall heat transfer coefficient was increased from 1000 to 1900 BTU/hr-ft²-°R and pressure drop vs. condensing length was inserted into the simulation. The latest -l condenser tube geometry was used; tube area and volume equations were written to accurately describe the tapered tube. Since the computer simulation depicts the flight reference "B" mercury flow rate of 9100 lbs per hour, the number of tubes was reduced to 60 to provide the same flow per tube as the -l condenser and, therefore, the same pressure drop. The condenser equations are as shown in the appendix.

IV DESCRIPTION OF THE COMPUTER RUNS

The initial control system gain was found to be excessive for the system since the system became unstable when the control was switched into the loop. The initial gain was derived to provide for ± 1 psi pressure regulation (see appendix) for ± 50 lbs of inventory change. The gain was reduced to approximately ± 5 psi for ± 50 lbs of inventory change as shown in Figure B. This gain does not represent the highest gain which could be obtained since the
damping factor and natural frequency of the system could be changed to provide for increased stability margin. Figure B shows the locus of operation of the control with changes in condenser NaK outlet temperatures. The intersection of the control set point pressure and rated NaK temperature shows the nominal rated operating point of the condenser. The resultant condensing pressure for a change in NaK temperatures would correspond to the vertical intersection with the temperature curve if no inventory control were present. In other words, the locus of pressure with changes in NaK temperatures occurs at a constant condensing length for no control. With an inventory control, the condensing length varies with changes in control inventory and pressure is controlled along the slanted line which results in less total pressure deviation for given NaK temperature changes. The higher the control gain the more nearly horizontal the control line becomes and the smaller the resultant pressure variations for temperature changes. The control line in Figure B below the normal operating line shows the effect of 20 lbs loss of mercury from the inventory cylinder which includes 10 lbs of leakage past the TAA space seals and a 10 lb shift of mercury to the boiler. The drop in the operating line is caused by reduced gas pressure in the bellows when it expands to occupy the volume left by the 20 lbs of mercury. This variation in pressure could be minimized by sizing the gas volume larger initially since the pressure change is inversely proportional to initial gas volume. Leakage of 20 lbs without a control is represented by an increase in condensing length corresponding to the 20 lbs reduction in condenser inventory. The resultant condensing pressure is then the intersection of the new condensing length with the rated NaK temperature line. A limitation on the effectiveness of the inventory control is apparent on close examination of Figure B. The pressure profiles of the condenser approach a horizontal asymptote at long condensing lengths which coincide with the vapor pressure of mercury corresponding to the NaK outlet temperature. This occurs because the minimum mercury condensing temperature cannot be less than the NaK coolant temperature. Therefore the minimum condensing pressure obtainable is dictated by the coolant and not the control. Any attempt to reduce the pressure to that minimum would result in a complete evacuation of the mercury inventory in the condenser.
Computer runs which were made on the computer are summarized in Table I and include sun-shade, variations in NaK exit temperature of the condenser, and 20 lbs leakage from the control cylinder. Vehicle load changes of 35 kw were made but are not reported since the steady state changes in pressure are small with or without the control. A sun-shade computer run with and without the inventory control system is shown by Figures E and F respectively.

V SUMMARY AND CONCLUSIONS

The condenser inventory control system is a feasible means of controlling condensing pressure although there are limitations on its effectiveness. These limitations involve the requirement of maintaining system stability which may require increased pressure regulation limits by a necessity to reduce the control gain. The lack of control of condenser NaK exit temperature also limits the minimum obtainable condensing pressure regardless of control gain. The inventory control therefore is more effective in increasing condensing pressure than reducing it. A biasing effect on the operation of the inventory control would be the effect of acceleration forces on the liquid mercury columns during orbit departures. This effect could be somewhat minimised by equalising the column heights by proper location of the control cylinder in relation to the condenser nominal interface position of the mercury.

Advantages of the condenser inventory control involve its simplicity, reliability and ability to maintain a relatively fixed inventory in the condenser with mercury leakage and variable boiler inventory.
Dynamic Analysis of a Mercury Inventory Control of the Snap-8 Condenser Pressure

**Accumulator Dynamic Equations**

**Force Balance on Accumulator Bellows**

\[ \Sigma F = 0 \]

\[ F_i + F_d + F_s = (F_a - F_c) \]

\[ M \ddot{S} + D \dot{S} + K S = (F_a - F_c) A \]

Since \( \dot{WA} = \frac{dW}{dt} = \frac{d}{dt} (PA S) = PA \dot{S} \)

\[ S = \int \dot{S} \, dW \]

\[ \ddot{S} = \frac{W}{EA} \]
\[
\frac{1}{\partial A} \left[ M \ddot{W}_a + D \dot{W}_a + k \int \dot{W}_a \, dt \right] = (P_a - P_0) A
\]

Transposing to the Laplacian Domain

\[
\frac{1}{\partial A} \left[ MS + D + \frac{k}{S} \right] W_a(s) = [P_a(s) - P_0(s)] A
\]

\[
\frac{W_a(s)}{P_a(s) - P_0(s)} = \frac{\frac{\partial A^2}{M S + D + \frac{k}{S}}}{\frac{\partial A^2}{M S^2 + D S + k}}
\]

\[
= \frac{\frac{\partial A^2 S}{M S^2 + D S + k}}{\frac{\partial A^2 S}{M S^2 - D S + k}}
\]

\[
= \frac{\frac{\partial A^2 S}{M S^2 - D S + k}}{\frac{\partial A^2 S}{M S^2 + D S + k}}
\]

\[
\frac{W_a(s)}{P_a(s) - P_0(s)} = \frac{K_s S}{S^2 + a S + b}, \quad \text{Lbs/sec}
\]

\[
\frac{\partial}{\partial A} = \text{Density of mercury liquid} \approx 800 \text{#/ft}^3
\]

\[
A = \text{Area of bellows} = 0.785 \text{ft}^2 \quad \text{(for } D = 1.0 \text{ ft)}
\]

\[
M = \text{Mass of mercury liquid in accumulator}
\]

\[
M = \frac{50 \text{ lbs}}{32.2 \text{ ft/sec}^2} = 1.55 \text{ slugs}
\]

\[
D = \text{Damping factor} = \text{Lbs/ft/sec}
\]

\[
\kappa = \text{Spring rate} = \text{Lbs/ft}
\]

\[
F_{\text{max}} = \kappa S_{\text{max}} = \kappa \left( \frac{\Delta V_{\text{max}}}{A} \right) = \kappa \left( \frac{W_{\text{max}}}{\partial A} \right)
\]

\[
\kappa = \frac{F_{\text{max}} \times \partial \times A}{W_{\text{max}}} = \frac{113 \times 800 \times 0.785}{50 \text{ lbs}} = 1420 \text{ Lbs/ft}
\]

\[
F_{\text{max}} = 1441 \times 0.785 \text{ ft}^2 = 113 \text{ Lbs}
\]

\[
\text{For 1 psi steady state error}
\]
\[ b = \frac{K}{m} = \frac{1420}{1.55} = \frac{916}{sec^2} = \text{un}^2 \]

Natural frequency = \((916)^{\frac{1}{2}} = 30.3 \text{ rad/sec}\)

\[ q = \frac{D}{M} = \frac{2 \text{ films}}{\text{un}} \]

For critical damping (fastest response with no overshot) let \( f = 1.0 \)

\[ q = \frac{D}{M} = 2(1.1)(30.3) = 60.6 \text{ sec}^{-1} = q \]

\[ D = 60.6(1.55) = 94 \text{ Lb./ft/sec} \]

\[
\frac{W_a(s)}{P_a(s) - P_o(s)} = \frac{318.5}{s^2 + 6065 + 916}
\]

Static gain = \( K_0 / \text{un}^2 = \frac{318}{916} = 0.347 \text{ Lbs/Hg/Lb/ft}^2 \)

\[ K_0 = \frac{\rho A^2}{M} = \frac{800(1.785)^2}{1.55} = \frac{318}{1.55} \]

Let \( W_a(s) = \frac{W_a}{s} = \text{inventory, Lbs} \text{ change} \)

\[
\frac{W_a(s)}{P_a(s) - P_o(s)} = \frac{318}{s^2 + 6065 + 916} \text{ Lbs/Hg/Lb/ft}^2
\]

Accumulator pressure

\[ P_a(V_a + \Delta V) = P_a V_o \]

or \[ P_a = \frac{P_a V_o}{V_a + \Delta V} \]

Assuming isothermal expansion (T=C)
\[ \Delta V_a = \frac{1}{c} \int W_a \, dt = \frac{W_a(s)}{c} \]

\[ P_a(s) = \frac{P_a \cdot V_{ao}}{V_{ao} + W_a(s)} \]

where \( P_a = \text{original accumulator gas pressure} \)
\( V_{ao} = \text{original accumulator gas volume} \)

**Block Diagram**

\[ P_{ao} \cdot V_{ao} = 2250 \cdot 1.24 = 2765 \text{ lb-ft} \]

\[ P_a(s) = \frac{2765}{1.24 + W_a(s)} = \frac{2.21 \times 10^6}{992 + W_a(s)} \]
RESULT OF 20# OF MERCURY LEAKAGE FROM ACCUMULATOR

\[
M' = \frac{30}{32.2} = 0.932 \\
\frac{M}{M'} = \frac{1.55}{0.932} = 1.665
\]

\[
q' = \frac{\alpha M'}{M'} = 60.6 \times 1.665 = 101
\]

\[
b' = \frac{b M'}{M'} = 916 \times 1.665 = 1525
\]

\[
K_0' = K_0 \times \frac{M}{M'} = 318 \times 1.665 = 530
\]

\[
\frac{W_a}{P_a - P_o} = \frac{530}{5^2 + 1015 + 1525}
\]

STATIC GAIN = \frac{530}{1525} = 0.347 \, \text{lbs/ft}^2

NATURAL FREQUENCY = (1525)^{\frac{1}{2}} = 39.1 \, \text{rad/sec}

DAMPING FACTOR = \frac{101}{2(39.1)} = 1.29

RESULT: INCREASE IN STABILITY MARGIN FOR LOSS OF 20# FROM ACCUMULATOR. THE OPERATING PRESSURE WOULD DECREASE 0.4 PSI.

where

\[
\begin{align*}
\frac{P_a}{V_a} &= \frac{P_{a0} V_{a0}}{V_{a0} + \Delta V} = \frac{2765}{1.24 + \frac{30}{800}} = \frac{2765}{1.24 + 0.025} = 2185 \, \text{psf} \\
\frac{P_a}{P_o} &= 15.1 \, \text{psi} \\
\Delta P_a &= 15.5 - 15.1 = 0.4 \, \text{psi} \checkmark
\end{align*}
\]
CONDUCTOR EQUATIONS

1. \((LMTD)C = \frac{C1 \times (\omega_{16} - \omega_{14})}{Ac}\)  \(\Rightarrow \) Solve for \(T_{202}\)

2. \((LMTD)C = \frac{(T_{203} - T_{303}) - (T_{202} - T_{302})}{\ln \left(\frac{T_{203} - T_{303}}{T_{202} - T_{302}}\right)}\)

3. \(T_{302} = \frac{C2 \times A_C \times (LMTD)_C - C3 \times \omega_{3} \times (T_{302} - T_{303})}{C4 \times L_C}\)

4. \(T_{303} = C6 \times \omega_{306} \times (T_{203} - T_{204}) - C3 \times \omega_{3} \times (T_{303} - T_{304})\)
   \(\Rightarrow (C4 + C7) \times LSC\)

5. \(P_{202} = f_1(T_{202})\) from Saturation Curve for Mercury

6. \(P_{203} = P_{202} - \Delta P\)

7. \(\Delta P = f_2(L_C)\) see Fig. 1

8. \(T_{203} = f_3(P_{203})\) from Saturation Curve for Mercury

9. \(T_{204} = T_{304} + C9\)

10. \(L_C = C5 - LSC\)

11. \(LSC = \int \left(\omega_{16} - \omega_{306} + \omega_A\right) d\theta / C10 \times \left[\frac{V_{7SC}}{LSC}\right]\)

12. \(\frac{V_{7SC}}{LSC} = f_4(LSC)\) see Fig. 2

13. \(A_C = f_5(L_C)\) see Fig. 3

14. \(LSC = G_8 + LSC\)
Constants

\[ C_1 = \frac{h_{fg}}{U_c} = \frac{126.4}{0.528} = 239.5 \text{ sec-ft}^2\text{-of/Lb} \]

\[ C_2 = \frac{U_c}{0.528} = 0.528 \text{ Btu/sec-ft}^2\text{-of} \]

\[ C_3 = \frac{C_{PN}}{0.21} = 0.21 \text{ Btu/Lb-of} \]

\[ C_4 = \frac{(MCP)_{HT}}{LT} = \frac{13.64}{4.295} = 3.18 \text{ Btu-of-ft} \]

\[ C_5 = 3.215 \text{ ft} \]

\[ C_6 = \frac{C_{PMH}}{0.0324} = 5.0 \text{ Btu/Lb-of} \]

\[ C_7 = \frac{(MCP)_{NH}}{LT} = \frac{2.98}{4.295} = 0.694 \text{ Btu-of-ft} \]

\[ C_8 = 1.075 \text{ ft} \]

\[ C_9 = 492 \text{ deg} \]

\[ C_{10} = \frac{\text{C}_{\text{HCL}}}{79.3} = 79.3 \text{ Lb/ft}^3 \]
RATED CONDITIONS

\[ T_{202} = 680^\circ F = 1140^\circ R \]
\[ T_{302} = 665^\circ F = 1125^\circ R \]
\[ T_{203} = 680^\circ F = 1140^\circ R \]
\[ T_{303} = 503^\circ F = 963^\circ R \]
\[ T_{204} = 499.9^\circ F = 959^\circ R \]
\[ T_{304} = 495^\circ F = 955^\circ R \]
\[ W_A = 0 \]
\[ \Delta P_c = 0 \]
\[ P_{302} = 15.5 \text{ psig} = \]

\[ \omega_{216} = \omega_{302} = 2.53 \text{ ft}^3/\text{sec} \]
\[ \omega_3 = 8.89 \text{ ft}^3/\text{sec} \]
\[ X_t = 0.948 \]
\[ A_c = 8.68 \text{ ft}^2 \]
\[ \frac{U_c}{S} = 0.528 \text{ Btu/sec-ft}^{2} \text{-of} \]
\[ L_e = 1.44 \text{ ft} \]
\[ L_{scf} = 1.775 \text{ ft} \]

STEADY STATE CHECK OF EQUATIONS

1. \[ C_{MTC} = \frac{0.948(126.4)(2.53)}{0.528 \times 8.68} = 66^\circ F \]
2. \[ C_{MTC} = \frac{(680 - 503)}{(680 - 665)} = 142 \]
3. \[ 0.528 \times 8.68 \times 66 = 0.31(142)(142) \]
4. \[ 0.528(2.53)(181) = 0.21(0.88)(8) \]
\[ \frac{14.84}{14.94} = \]
Glossary of Terms

\( T_{202} = \text{Hg condenser inlet temp} \)  \(^\circ\text{F}\)
\( T_{203} = \text{" exit } \)  \(^\circ\text{F}\)
\( T_{204} = \text{" subcooler } \)  \(^\circ\text{F}\)
\( T_{302} = \text{N.AK condenser } \)  \(^\circ\text{F}\)
\( T_{303} = \text{" inlet } \)  \(^\circ\text{F}\)
\( T_{304} = \text{" subcooler } \)  \(^\circ\text{F}\)
\( W_{16} = \text{Hg flow into condenser} \)
\( W_{306} = \text{Hg flow from pump} \)
\( \delta_{9} = \text{Hg heat of condensation} \)
\( X_{7} = \text{Hg quality at condenser inlet} \)
\( A_{c} = \text{condenser heat transfer area} \)
\( (\text{LMTD}) = \text{log mean temp. diff. for condenser} \)
\( W_{3} = \text{N.AK flow} \)
\( L_{c} = \text{condensing length} \)
\( L_{s} = \text{subcooling length} \)
\( L_{Sc} = \text{subcooler tapered length} \)
\( P_{202} = \text{condenser pressure corresponding to } T_{202} \)  \(\text{lb/ft}^2\)
\( P_{203} = \text{" } T_{203} \)  \(\text{lb/ft}^2\)
\( \Delta P_{c} = \text{condenser pressure drop } (P_{202} - P_{203}) \)  \(\text{lb/ft}^2\)
\( V_{Sc} = \text{volume of tapered section of subcooler} \)
\( U_{c} = \text{overall heat transfer coefficient for condenser } \)  \(\text{BTU/sec-ft}^2\)  \(^\circ\text{F}\)
\( C_{pn} = \text{specific heat at constant press. for N.AK} \)  \(\text{BTU/lb-} \)  \(^\circ\text{F}\)
\( C_{pnl} = \text{" Hg liquid } \)  \(\text{BTU/} \)  \(\text{lb-} \)  \(^\circ\text{F}\)
\( L_{T} = \text{total condenser + subcooler length} \)
\( (MC_{P})_{T} = \text{total N.AK + condenser metal capacitance } \)  \(\text{BTU/} \)  \(^\circ\text{F}\)
\( (MC_{P})_{Hg} = \text{total possible Hg liquid capacitance } \)  \(\text{BTU/} \)  \(^\circ\text{F}\)
\( \rho_{Hg} = \text{density of mercury liquid } \)  \(\text{lb/ft}^3\)
# Table I

## Tabulated Results of Computer Runs

<table>
<thead>
<tr>
<th>Disturbance to Condenser</th>
<th>Condensing Pressure Changes</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Without Control</td>
</tr>
<tr>
<td>Sun-Shade</td>
<td>$-3.0 \text{ psi}$</td>
</tr>
<tr>
<td>Nak Exit Temp. of Cond. = 680°F</td>
<td>$+2.2 \text{ psi}$</td>
</tr>
<tr>
<td>Nak Exit Temp. of Cond. = 620°F</td>
<td>$-5.0 \text{ psi}$</td>
</tr>
<tr>
<td>20 lbs Leakage</td>
<td>$-1.5 \text{ psi}$</td>
</tr>
</tbody>
</table>

---

**Aerojet General**  
**Snap-8 Division**  
**Von Karman Center**
Fig. C

CONDENSOR INVENTORY VS CONDENSING LENGTH

NOTE: 60 TUBES

SHIFT WITH 20 LBS LEAKAGE (NO CONTROL)

NOMINAL OPERATING POINT (BEFORE LEAKAGE)

\[ \frac{20 \text{#/ft}}{45} = \frac{1}{k_c} \]

\[ \frac{20 \text{#/ft}}{\text{condensing length in ft}} = \frac{1}{k_c} \]

AEROJET GENERAL

SNAP-8 DIVISION - VON KARMAN CENTER
CONDENSER INVENTORY CONTROL SYSTEM

BLOCK DIAGRAM

\[ K_0 = \frac{\text{Eng} A_t^2}{\text{Mnca}} \]
\[ \alpha = \frac{D}{\text{Mnca}} \]
\[ b = \frac{k}{\text{Mnca}} \]
\[ \text{Mnca} = \text{mass of Hg in accumulator} \]
\[ k = \text{spring constant} \]
\[ D = \text{damping} \]

\[ W_s = \int (\text{Hg}_{in} - \text{Hg}_{out}) \, dt \]

CHANGE IN INVENTORY DUE TO TRANSIENT FLOW CHANGES

\[ W_A = \int W_a \, dt \]

\[ V_{ao} = \text{const.} \]

\[ P_{ao} V_{ao} = \text{const.} \]

\[ \Delta V = \text{const.} \]

\[ P_{ao} - P_{co} \]

\[ W_A, \text{LBS Hg} \]

\[ \Delta l_{sc} \]

\[ l_{sc} \]

\[ \text{SUBCOOLING LENGTH} \]

\[ \text{INITIAL SUBCOOLER LENGTH, L}_{sc} \]

\[ \text{ACCUMULATOR} \]

\[ \text{CONDENSER EXIT PRESSURE} \]

\[ \text{CONDENSER Simulation} \]

\[ \text{PRESSURE, } P_a \]

\[ \frac{d}{dt} \]

\[ \text{Eng A} \]

\[ \text{E} = \text{const.} \]

SNAP-8 DIVISION - VON KARMAN CENTER
FIG. 1

SUN TO SHADE RUN WITH INVENTORY CONTROL

4-18-44

Mercury Flow Rate, 1000, LBS PER SEC

Mercury Condenser Inlet Temp, 200, OF

Mercury Condenser Inlet Pressure, 300, LBS PER SQ FT (N2)

Condensing Length, LC, FT

Inventory Control, Mercury Output, Wt., LBS

N.A. Exit Temp. of Condenser, 1200, OF

Mercury Inlet Temp. To Evaporator, 350, OF

Mercury Exit Temp. To Evaporator, 650, OF

N.A. Exit Flow, LBS PER SEC
FIG. 1

CONDENSER PRESSURE DROP VS. CONDENSING LENGTH
BASED ON EXPECTED -1 CONDENSER PERFORMANCE

CONDENSING LENGTH, Lc Ft.

CONDENSER PRESSURE DROP, $\Delta P_{\text{cond}}$ lbf/ft$^2$
SUBCOOLER TAPE RED TUBE
VOLUME & LENGTH VS LENGTH

\[ V = \frac{60 + 1000 \times 10^{-6} \left(2 + 0.775 \cdot e^{-1} + 0.145 \cdot e^{-2} \right)}{\text{LST}} \]

\[ \text{LST} = \text{FX} \]
CONDENSING AREA VS LENGTH

FIG. 3

17
16
15
14
13
12
11
10
9
8
7
6
5
4
3
2
1
0
0
10
20
30
40

AC ~ CONDENSING AREA ~ FT²

LC ~ CONDENSING LENGTH ~ FT

NOTE: B = 60 (15L - 0.01L²)

RATED

1.05'

2.05'

L0 ~ CONDENSING LENGTH ~ FT