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TECHNICAL DATA RECORD		PAGE 1 OF 26	
AUTHOR E. M. Mouradian <i>EMM</i>	DEPT. & GROUP NO. 721 14	DATE 9-9-63	
TITLE Transient Thermal Start-up Analysis for CRU V		GO NO. 7562	
PROGRAM SNAP 2		S/A NO. 1230	TWR
SUBACCOUNT TITLE Start-up & Control Analysis		SECURITY CLASSIFICATION	
DISTRIBUTION Exempt from CCRP Re-review Requirements per 7/22/82 Duff/Caudle memorandum)		(CHECK ONE BOX ONLY)	(CHECK ONE BOX ONLY)
		UNCL. ☒	RESTRICTED DATA ☒
		CONF. ☒	DEFENSE INFO. ☐
		SECRET ☐	
		AUTHORIZED SIGNATURE <i>Paul D. Cohn</i> DATE 8-15-63	
STATEMENT OF PROBLEM			
<p>Determine analytically the transient thermal-hydraulic performance of the CRU-5 resulting from a non-preheated start-up of the unit.</p>			
ABSTRACT:			
<p>The transient condition occurring during the start-up period of the CRU-5 may cause damage or failure of the unit if the mercury vapor condensation rates or temperature gradients set up are sufficiently severe. In order to estimate the potential severity of these problems, a transient thermal analysis of the CRU-5 was initiated.</p>			
<p>The CRU-5 was conceptually divided into 98 nodes, including 9 vapor volume nodes, 10 liquid volume nodes, 38 solid nodes, and 41 surface nodes. An energy balance equation was set up for each node to determine its time dependent temperature. In addition, other equations were set up to determine the other mercury vapor properties and the condensation rates as functions of time. These additional equations include mass balance equations, condensation rate expressions, the mercury vapor saturation relation, and vapor flow equations. The vapor flow equations are for nozzle flow, either choked or unchoked, and constant area, or frictional flow.</p>			
<p>The equations for the various nodes are coupled to one another, and must be solved simultaneously. This is because conditions in each node are controlled not only by the mercury fluid flow conditions of upstream and downstream nodes for the fluid flow lines, but also by temperatures of all adjacent nodes which govern heat transfer rates across the thermal paths interconnecting all nodes. In all, 191 thermal paths were designated.</p>			
<p>The equations and appropriate testing systems have been set up. Much of the system parameters have been numerically evaluated. The system of equations must be programmed for computer calculation. An iterative process must be used to solve the time dependent differential equations.</p>			
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NO. NAA-SR-MEMO 8963
DATE 9-9-63
PAGE 2 OF 26

TABLE OF CONTENTS

Chapter No.	Chapter Title
I	Introduction
II	Plan of Analysis
III	Vapor Nodes
	A. Flow Equations
	B. Basic Node Equations
IV	Liquid Nodes
V	Solid Nodes
VI	Surface Nodes
VII	Modes of Heat Transfer
VIII	Combination of Equations
IX	Conclusion

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A Division of North American Aviation, Inc.

NO. NAA-SR-MEMO 8963

DATE 9-9-63

PAGE 3 OF 26

FIGURES

- 1 CRU V Components
- 2 Node Division Pattern
- 3 Schematic of Vapor Volume Nodes
- 4 Schematic of Liquid Flow Lines

TABLES

- I Node Numbering System

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A Division of North American Aviation, Inc.

CONFIDENTIAL

NO. NAA-SR-MEMO 8963

DATE 9-9-63

PAGE 4 OF 26

NOMENCLATURE

<u>Symbol</u>	<u>Item</u>	<u>Units</u>
A	Area; gas flow or nozzle area, node surface or cross-sectional area	ft ²
C _p	Specific heat; for gas, C _p = 0.02475 for liquid, C _p ≈ 0.0323 at 450°F	BTU/lb _m °R
C _v	Specific heat; for gas, C _p = 0.01475	BTU/lb _m °R
f _i	Fanning Friction Factor; $\bar{T} \frac{f \rho U^2}{2 g_c}$	—
F	Force	lb _f
	Radiation View Factor,	—
$\bar{T}$	Radiation heat transfer factor, $\bar{T} = \bar{T}(F, \epsilon)$	—
g _c	Gravitational acceleration constant	ft lb _m /lb _f sec ²
h	Enthalpy; for gas, h = C _p T (°R)	BTU/lb _m
h _c	Enthalpy of condensate; h _c = (C _p T - λ)	BTU/lb _m
h	Heat transfer coefficient	BTU/ft ² sec °R
h _{con} , h _{n.c.}	Heat transfer coefficient for condensing and non-condensing vapor	BTU/ft ² sec °R
h _l	Heat transfer coefficient for liquid at pipe wall, h _l ≈ 1000 BTU/ft ² hr °R = 0.278 at bearing surface, h _l ≈ 20,000 BTU/ft ² hr °R = 5.56	BTU/ft ² sec °R
k	Specific heat ratio for mercury gas; k = 5/3	—
K	Thermal conductivity	BTU/ft sec °R
L	Length of flow channel	ft
m	Mass	lb _m

CONFIDENTIAL

ATOMICS INTERNATIONAL

A Division of North American Aviation, Inc.

~~CONFIDENTIAL~~

NO. NAA-SR-MEMO 8963

DATE 9-9-63

PAGE 5 OF 26

<u>Symbol</u>	<u>Item</u>	<u>Units</u>
mv, mom	Momentum	lb _m ft/sec or lb _f sec
Nu	Nusselt number; $Nu = \frac{hd}{k}$	—
P	Gas pressure	lb _f /ft ²
Pe	Peclet number; $Pe = \frac{vd}{\alpha} = Re Pr$	—
q	Heat flow rate	BTU/sec
$\sum q$ in	Net heat flow rate into a node by conduction, wall convection, and/or radiation	BTU/sec
Q"	Heat generation rate on a surface	BTU/ft ² sec
Q'''	Heat generation rate in a volume	BTU/ft ³ sec
$r_{\frac{a}{b}}$	Gas pressure ratio between nodes a and b; $r_{\frac{a}{b}} = \frac{P_a}{P_b}$	—
R	Ideal gas constant for mercury; R = 7.70	ft lb _f /lb _m °R
t	Time	sec
T	Temperature (absolute)	°R
T _w	Temperature of chamber wall	°R
T _{sat}	Saturation temperature of gas	°R
T _{gas} , T _{chm}	Temperature of gas in a chamber	°R
u	Internal energy; for gas, $u = C_v T$ (°R)	BTU/lb _m
V	Volume of node, either chamber or solid	ft ³
w	Gas or liquid flow rate	lb _m /sec
w _c	Condensation rate	lb _m /sec
W.P.	Wetted perimeter of flow channel	ft
W _{out}	Work removal from gas by turbine	BTU/sec
ΔX	Effective distance for thermal conduction between nodes	ft

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ATOMICS INTERNATIONAL

A Division of North American Aviation, Inc.

NO. NAA-SR-MEMO 8963

DATE 9-9-63

PAGE 6 OF 26

<u>Symbol</u>	<u>Item</u>	<u>Units</u>
ϵ	Surface emissivity for radiation	—
λ	Latent heat of vaporization; = 125	BTU/lb _m
ρ	Density	lb _m /ft ³
σ	Stefan-Boltzmann constant	BTU/ft ² sec °R ⁴

SUBSCRIPTS

ad	Adjacent node
con, nm.c.	Condensing, non-condensing
chm, gas	Gas property in chamber
in, out	Flow properties coming, going out, of chamber
liq	Liquid properties
n	Node being considered
sat	Saturation properties of gas
u, uv	Properties of upstream chamber, or flowing from upstream chamber
w	Wall properties
I	Upstream chamber
II	Chamber being considered

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ATOMICS INTERNATIONAL

A Division of North American Aviation, Inc.

NO. NAA-SR-MEMO 8963

DATE 9-9-63

PAGE 7 OF 26

I. Introduction

The Combined Rotating Unit No. 5 (CRU-5) contains all the moving parts for the SNAP 2 generator, including the mercury vapor turbine, the electrical generator, and the liquid mercury pump. While most of the thermal and mechanical analysis that has been carried on with regard to steady-state operating conditions, considerable interest has centered around the transient thermal conditions in the CRU-5 during the start-up period.

As the unit begins operation from a cold starting condition, transients may occur which could cause failure of the unit. Uneven heating will occur due to the introduction of hot mercury vapor into one end of the unit, as well as local heat generation as a result of bearing friction and electrical losses. This can bring about thermal stresses and differential expansion with the possible result of bore seal cracking, bearing binding, etc. High rates of mercury vapor condensation at various surfaces can harm the operation of the CRU-5. Damage can occur due to liquid drop impingement, and condensate accumulated in the alternator cavity can prevent it from coming up to speed rapidly.

In order to evaluate the extent of these problems, a thermal analysis of the CRU-5 was initiated. A transient thermal map of the unit was sought, which would include, as functions of time, the temperature, pressure, and density of the mercury vapor at points around the unit, temperatures of metal parts and liquid flow lines, and mercury vapor condensation rates at various exposed surfaces.

The work of Mr. G. Asanovich in evaluating numerically the system parameters of the CRU-5 for this project is particularly appreciated.

The design and plan of operation of the CRU-5 during start-up is well discussed elsewhere. See Reference 1. A view of the CRU-5 is shown in Figure 1.

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CRU V COMPONENTS

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1. UPPER CAN HOUSING
2. CLAMPS
3. HOUSING END CAP
4. MERCURY PUMP
5. THRUST WASHER
6. THRUST BEARING
7. ALTERNATOR BEARING
8. ALTERNATOR BEARING HOUSING
9. ALTERNATOR STATOR
10. STATOR COOLING FIN
11. SHAFT
12. ALTERNATOR ROTOR
13. UPPER TURBINE HOUSING
14. LOWER TURBINE HOUSING
15. TURBINE ROTORS
16. TURBINE BEARING ASSEMBLY
17. END CAP
18. CRADLE

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PAGE 8 OF 26

NO. NAA-SR-MEMO 8963

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NO. NAA-SR-MEMO 8963

DATE 9-9-63

PAGE 9 OF 26

II. Plan of Analysis

In the mathematical model set up, the CRU-5 was divided into 98 nodes for the thermal analysis. These nodes are of four types: vapor control volume, liquid control volume, solid, and surface. The node division pattern is shown by Figure 2.

For the vapor control volume, a series of individual control volumes were designated, through which the mercury vapor would flow. The gas properties and condensation rates in each chamber are sought as functions of time. There are nine such nodes, numbered 1 through 9. The gas temperature and density describe the conditions. Each node thus has three unknowns; mercury vapor density, ρ lb_m/ft³, temperature, T °R, and the condensation rate, w_c lb_m/sec. Three equations were therefore set up for each of these vapor nodes.

The liquid volume nodes were created on the three liquid mercury lines flowing through the CRU-5, which can transfer heat into or out of their surroundings. These three lines were divided into a total of ten control volume nodes. These three lines are the mercury pump line, alternator and thrust bearing line, and the turbine bearing line. For each of these ten nodes, the temperature is considered the only unknown. An energy balance equation was set up for each liquid volume node. These nodes were designated with two digit numbers, the first digit being 2, 3, or 4.

Solid nodes were set up for the metal portions of the CRU-5, and are numbered from 51 to 88. An energy balance equation was established for each of these solid nodes to determine its temperature.

Surface nodes were established in areas where wall convection or radiation heat transfer occurs. The temperature of each surface node is an unknown, and is to be determined from an energy balance equation on it. The surface nodes are designated by three digit numbers. The last two digits are those of an adjacent solid node.

In each node, the temperature (and gas density and pressure in vapor nodes) are assumed to be constant throughout the node at every instant, though, of course, they are variables with time.

Table I shows the node numbering scheme.

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FIGURE 2

NODE DIVISION PATTERN, CRU-5

NAA-SR-MEMO 8963
 9-9-63
 Page 10 of 26

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A Division of North American Aviation, Inc.

NO. NAA-SR-MEMO 8963
DATE 9-9-63
PAGE 11 OF 26

Table I-1
Node Numbering System

<u>Node Type</u>	<u>Number of Nodes</u>	<u>Numbering System</u>
Vapor Volume	9	1-9
Liquid Volume	10	21-43
Solid	38	51-88
Surface	41	Over 100

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A Division of North American Aviation, Inc.

NO. NAA-SR-MEMO 8963

DATE 9-9-63

PAGE 12 OF 26

III. Vapor Nodes

A. Flow Equations

The nine vapor nodes are shown schematically in Figure 3. Mercury vapor flows from a supply line into node 1, the first stage of the turbine, then through node 2, the scroll between stages, into node 3, the second turbine stage. Then, the vapor is released into the "head end" of the containment can, designated node 4. From there, the vapor flow divides. Most will flow over the cooling fins, nodes 5, 6, and 7, into the "rear end" of the can, node 9. Some will flow from node 4 into the gap between the rotor and stator, node 8, then on through to node 9. If, at any time, the gas pressure in node 9 is greater than in node 8, gas will flow into node 8 from both directions. A pressure ratio test was set up to determine, for each iteration, the flow direction of w_8 .

Equations were set up describing the gas flow rate from one node to the next. These flow equations are of two types, nozzle flow or constant area flow.

Nozzle flow is flow from one chamber to the next where the gas must pass through a constriction. It is either choked or unchoked (sonic or subsonic), depending on the ratio of pressures existing between the two chambers. The usual compressible flow nozzle equations were used, assuming isentropic flow and ideal gas relations. Referring to Figure 3, the flows w_0 , w_1 , w_2 , w_3 , w_4 , w_{14} , and w_9 are nozzle flows. Flow w_8 is also a nozzle flow when $P_8 \ll P_9$.

There is a constant cross-sectional flow area as the mercury gas flows from one node to the next over the cooling fins, and the nozzle equations are not applicable. The approach velocity is not negligible, and the retarding force on the fluid is the frictional shear stress at the channel walls. The flow equation is developed from a momentum balance of the form:

$$\frac{d(mv)}{dt} = g_c \sum F + [\text{mom. rate (in)} - \text{mom. rate (out)}] \quad (3-1)$$

with the force term including the pressure difference and wall shear friction. The flows w_5 , w_6 , and w_7 are governed by the "constant-area" relation, as well as flow w_8 when $P_8 > P_9$.

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9-9-63

Page 13 of 26

FIGURE 3
SCHEMATIC OF VAPOR VOLUME NODES

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A Division of North American Aviation, Inc.

NO. NAA-SR-MEMO 8963

DATE 9-9-63

PAGE 14 OF 26

The general form of the flow equations are:

CHOKED
NOZZLE
FLOW:

$$\omega_{zu} = \left[A_{zu} \left(\frac{2}{k+1} \right)^{\frac{1}{k-1}} \sqrt{\frac{2g_c k R}{k+1}} \right] (p_u \sqrt{T_u}) \quad (3-2)$$
$$\frac{P_{chu}}{P_u} < \left(\frac{2}{k+1} \right)^{\frac{k}{k-1}} = 0.4871 \quad \omega_{zu} = 11.43 A_{zu} (p_u \sqrt{T_u})$$

UNCHOKED
NOZZLE
FLOW:

$$\omega_{zu} = \left[A_{zu} \sqrt{\frac{2g_c k R}{k-1}} \right] (p_u \sqrt{T_u}) \left[\sqrt{\left(\frac{P_{chu}}{P_u} \right)^{\frac{2}{k}} - \left(\frac{P_{chu}}{P_u} \right)^{\frac{k+1}{k}}} \right]$$
$$\frac{P_{chu}}{P_u} > 0.4871 \quad \omega_{zu} = 35.21 A_{zu} (p_u \sqrt{T_u}) \left[\sqrt{(r_{chu})^{1.2} - (r_{chu})^{1.6}} \right] \quad (3-3)$$

CONSTANT
AREA
FLOW:

$$\frac{d\omega_z}{dt} = \left\{ \left[247.9 \frac{A_w}{L_w} (p_z T_I - p_z T_z) \right] - \left[\frac{f}{4} \frac{(W.P.)}{A_w^2} \frac{(\omega_z + \omega_z)^2}{(p_z + p_z)} \right] + \left(\frac{1}{A_w L_w} \right) \left[\frac{\omega_z^2}{p_z} - \frac{\omega_z^2}{p_z} \right] \right\} \quad (3-4)$$

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A Division of North American Aviation, Inc.

NO. NAA-SR-MEMO 8963
DATE 9-9-63
PAGE 15 OF 26

III. Vapor Nodes

B. Basic Node Equations

For each vapor volume node, the unknowns to be determined as a function of time are the gas properties and the mercury condensation rate. The gas conditions are to be specified by temperature and density. The other gas properties are to be determined by the ideal gas relationship. The three unknowns for each vapor node are thus the gas density, gas temperature, and the condensation rate.

The bulk gas in a chamber can be either superheated or saturated. If the gas is in a superheated condition, the mass balance and energy balance equations are used, Equations (3-5) and (3-6), and the third equation is an expression for the condensation rate at the cold walls, Equation (3-7). If the bulk gas is saturated, the density and temperature are not independent, and an expression for the saturation relation is used as the third equation, Equation (3-8). These equations take the form:

Either superheated or saturated:

Mass balance

$$V_n \frac{d\rho_n}{dt} = w_{in} - w_{out} - w_c \quad (3-5)$$

Energy balance

$$c_v V_n \frac{d(\rho T)_n}{dt} = C_p [w_{in} T_{in} - w_{out} T_n] - w_c [C_p T_{sat} - \lambda] + \sum \dot{q}_{in} + w_{out} \quad (3-6)$$

If superheated:

$$w_c = - \sum \frac{c_{on} A (T_w - T_{sat})}{\lambda} \quad (3-7)$$

If saturated:

$$\rho_{sat} = \frac{18.70}{T_{sat}} \exp \left\{ 13.84 - \frac{12710}{T} \right\} \quad (3-8)$$

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A Division of North American Aviation, Inc.

NO. NAA-SR-MEMO 8963
DATE 9-9-63
PAGE 16 OF 26

In setting up the mass and energy balance equations, it was assumed that inlet conditions are determined by the upstream node, but that the discharge conditions of the gas are same as the average property values in the chamber. It is assumed that when condensate forms, it remains as liquid and not available for re-evaporation.

The vapor mass balance equation is developed from the form:

$$\frac{dm}{dt} = w_{in} - w_{out} - w_c \quad (3-9)$$

The left hand term represents the rate of gas mass accumulation, w_{in} and w_{out} the gas flow rates into and out of the chamber, and w_c is the rate of mass removal by condensation.

The energy equation is obtained from the form:

$$\frac{du}{dt} = w_{in} h_{in} - w_{out} h_{out} - w_c h_c + \sum q_{in} - W_{out} \quad (3-10)$$

The left side of Eq. (3-10) is the rate of accumulation of internal energy for the gas in the chamber. The first two terms on the right represent the gas enthalpy flow into and out of the node. The third term is the enthalpy of the condensate liquid. The enthalpy and internal energy of the gas are taken as $h = C_p T$ and $u = C_v T$, respectively, with temperature in degrees Rankine. The enthalpy of the liquid condensate is thus $h_c = (C_p T - \lambda)$, where λ is the latent heat of vaporization. It may be noted that at low temperatures, h_c is negative. This is merely a consequence of a scale factor occurring in the definition of the gas enthalpy expression.

A semi-empirical expression (See Reference 2) was obtained for the saturation pressure-temperature relation for mercury vapor.

$$P \text{ (psia)} = \exp \left[13.84 - \frac{12710}{T(^{\circ}R)} \right] \quad (3-11)$$

Equation (3-8) is obtained from Eq. (3-11) by means of the ideal gas relation.

The mass and energy balance equations include flow rates into and out of the particular node. These are to be expressed by the appropriate flow equations given by Equations (3-2) to (3-4). Note also that the node division is such that nodes 4, 8, and 9 may have more than just one flow stream in or one flow stream out.

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NO. NAA-SR-MEMO 8963

DATE 9-9-63

PAGE 17 OF 26

In the energy equations, the term $\sum q_{in}$ appears. This represents the heat transfer into (or out of) the vapor in the node by convection at the walls of the chamber. This heat transfer term is discussed in Chapter VII.

In the energy equation, the term W_{out} appears. This would be non-zero only in nodes 1 and 3 since it represents the energy removed from the vapor in the form of shaft work by the turbine blades. To avoid the complexity associated with inclusion of the dynamic-turbine equations, it was suggested that the work removal be initially assumed equal to the superheat entering the node representing the turbine stage.

$$W_{out} = w_{in} C_p (T_{in} - T_{sat}) \quad (3-12)$$

The energy equation (3-6) for these nodes would thus become:

$$V c_v \frac{d(P,T)_n}{dt} = C_p T_n (w_{in} - w_{out}) - w_c (C_p T - \lambda) + \sum q_{in} \quad (3-13)$$

This assumption would probably overestimate the work removal at the very beginning of start-up, and underestimate it later in time.

As indicated above, the vapor in a node can be either superheated or saturated, and each regime has its set of three equations to determine P , T , and w_c . To learn which regime is applicable, and when switching would occur, a test is set up. The gas is initially superheated and remains so as long as the chamber temperature is greater than the saturation temperature (given by Eq. 3-8) for the existing gas density.

When the gas temperature becomes equal to the saturation pressure, the switch is made to the saturation regime equations. The bulk gas remains saturated as long as the heat coming into the chamber in the form of gas superheat is less than the net heat being removed at the walls. That is, saturation exists as long as:

$$\left[C_p w_{in} (T_{in} - T_{sat}) - \sum q_{in} \right] < 0 \quad (3-14)$$

When the left side of Eq. (3-14) becomes positive, the switch is made back to the superheated regime.

In the two turbine stages, nodes 1 and 3, if it is assumed that the work removal is equal to the incoming superheat, as given by Eq. (3-12), then the gas in those two nodes is always in the saturated condition, and no superheat-saturation test or switching is necessary.

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0310507030

NO. NAA-SR-MEMO 8963

DATE 9-9-63

PAGE 18 OF 26

IV. Liquid Nodes

In the CRU-5, there are three lines in which liquid mercury flows; the liquid mercury pump line, the line supplying the alternator and thrust bearing, and the turbine bearing line. These lines are shown schematically in Figure 4. Each line is divided into several volumes, and the nodes are considered to be the volume of liquid within that node at any one time. Line 20 is the mercury pump line and is divided into three nodes; node 21, inlet; node 22, pump interior; and node 23, outlet. Line 30 is the alternator and thrust bearing line, and has four nodes; node 31, inlet; node 32, thrust bearing interior; node 33, alternator bearing interior; and node 34, outlet. Line 40 is the turbine bearing line with three nodes, 41, 42, and 43, which represent the inlet, bearing interior, and outlet, respectively.

The temperature of the liquid within each node is the unknown to be determined, and an energy balance is set up for each node. This energy balance equation takes the form:

$$\frac{dT_n}{dt} = \left(\frac{1}{V_n}\right) \left(\frac{1}{\rho C_p}\right) \left\{ C_p w_n (T_{in} - T_{out})_n + \sum q_{in} + (V_n Q_n''') \right\} \quad (4-1)$$

The term on the left-hand side of Equation (4-1) represents the rate of heat accumulation in the node. The term $C_p w_n (T_{in} - T_{out})_n$ is the heat entering and leaving the node due to liquid flow. The heat convected into or out of the node at the chamber walls is expressed by $\sum q_{in}$ which is expanded to $\sum h_L A_i (T_{wi} - T_n)$. Heat generation within the node is accounted for by the term $V_n Q_n'''$. This occurs in nodes 22, 32, 33, and 42 due to pump and bearing friction. It is assumed that the temperature of liquid entering and leaving a node is related to the average node temperature by the relation:

$$T_n = \frac{(T_{in} - T_{out})_n}{2} \quad (4-2)$$

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FIGURE 4

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SCHEMATIC OF LIQUID FLOW LINES

LINE 20 LIQUID PUMP

LINE 30 THRUST & ALTERNATOR BEARINGS

LINE 40 TURBINE BEARING

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NO. NAA-SR-MEMO 8963

DATE 9-9-63

PAGE 20 OF 26

V. Solid Nodes

The metal of the CRU-5 was divided into 38 solid nodes of various shapes and sizes. For each of these nodes, the temperature as a function of time is an unknown. Therefore, an energy equation is set up which takes the general form:

$$(m C_p)_n \frac{dT_n}{dt} = \sum q_{in} + (Q''' V)_n \quad (5-1)$$

The temperature derivative term represents the heat accumulation rate in the solid. The $\sum q_{in}$ term is the net heat transferred into (or out of) the node by conduction, wall convection, or radiation. The final term is to account for heat generation within the solid. This occurs in the several nodes representing the alternator rotor and stator which electrical (I^2R) losses occur.

The temperature of each node is a variable with time, though it is assumed that at every instant, the same temperature prevails throughout the node. The smaller sizes the nodes, the less inaccuracy is introduced by this assumption.

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A Division of North American Aviation, Inc.

NO. NAA-SR-MEMO 8963
DATE 9-9-63
PAGE 21 OF 26

VI. Surface Nodes

Forty-one surface nodes have been established. These are located at various surfaces of the solid, metal, nodes. In addition to their role in taking a part in determining the temperature distribution within the CRU-5, they have a special purpose in the determination of the actual metal surface temperature at various points, since it is these temperatures which must be used in calculating convection and radiation heat transfer terms.

These surface nodes are essentially the same as solid nodes, except they have no mass assigned to them, and as a result, have no heat storage capability. The energy balance for a surface node has the form:

$$0 = \sum q_{in} + (Q'' A_s) \quad (6-1)$$

The term $\sum q_{in}$ represents the net heat transferred into or out of the node. The $(Q'' A_s)$ term represents heat generation at the surface. In almost all cases, this term is equal to zero. However, for the several nodes on the surface of the rotor and inside surface of the stator, the windage heat generation produced in the bore seal gap has been assigned as surface heat generation at the appropriate surface nodes.

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NO. NAA-SR-MEMO 8963
DATE 9-9-63
PAGE 22 OF 26**VII. Modes of Heat Transfer**

The energy balance equation written for each node contains a term, $\sum q_{in}$, which represents the net heat transferred into the node from its adjacent nodes. The number of thermal paths into a particular node is dependent on the geometry of CRU-5 and the manner of division into nodes, and varies from one to ten for various nodes. A listing of adjacent nodes and associated thermal paths into each individual node was made. In all, there are 191 thermal paths established, 110 conduction, 27 radiation, 14 liquid wall convection, and 40 vapor wall convection.

The heat transfer across each thermal path occurs by one of the four mechanisms; conduction through solid, convection from solid to liquid, convection from solid to vapor, or radiation. Heat carried into a liquid or vapor node by the flowing fluid is not included in the $\sum q_{in}$ term, but is accounted for by a separate term in the energy equation.

The conduction heat transfer is considered to occur between the centers of two adjacent nodes, and follows the expression:

$$q = \frac{(T_{ad} - T_n)}{\sum \frac{\Delta x}{kA}} \quad (7-1)$$

When conduction occurs from one solid node to another, there are two thermal resistance terms, $\frac{\Delta x}{kA}$, to be accounted for, but when conduction occurs between a solid and surface node, only one $\frac{\Delta x}{kA}$ term is necessary.

The liquid convection term representing heat flow between a surface and liquid node is expressed as:

$$q = h_L A (T_{ad} - T_n) \quad (7-2)$$

The values chosen for the heat transfer coefficient, h_L , are 20,000 and 1,000 $\frac{\text{BTU}}{\text{hr. ft}^2 \text{ of } ^\circ\text{F}}$ for bearing surfaces and pipe surfaces, respectively. These estimates were made on the basis of mercury properties, flow and bearing surfaces velocities, and the liquid metal heat transfer correlation:

$$\text{Nu} = 0.625 (\text{Pe})^{0.4} \quad (7-3)$$

which is given in Reference 3.

The heat convection to a vapor is assumed to occur according to:

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ATOMICS INTERNATIONAL

A Division of North American Aviation, Inc.

NO. NAA-SR-MEMO 8963

DATE 9-9-63

PAGE 23 OF 26

$$q = h_{\text{con}} A (T_w - T_{\text{sat}}) + h_{\text{N.C.}} A (\Delta T)_{\text{N.C.}} \quad (7-4)$$

where

$$h_{\text{con}} = 0 \quad \text{if} \quad T_w > T_{\text{sat}} \quad (7-5)$$

$$h_{\text{con}} > 0 \quad \text{if} \quad T_w < T_{\text{sat}}$$

and

$$(\Delta T)_{\text{nc}} = (T_{\text{sat}} - T_{\text{gas}}) \quad \text{if} \quad T_w < T_{\text{sat}} \quad (7-6)$$

$$(\Delta T)_{\text{nc}} = (T_w - T_{\text{gas}}) \quad \text{if} \quad T_w > T_{\text{sat}}$$

The expression of Eq. (7-4) is used since it can take into account the higher heat fluxes that occur during condensation.

Radiation heat transfer occurs between some surface nodes, and the expression followed is:

$$q = \sigma (\epsilon, F) A (T_{\text{ad}}^4 - T_n^4) \quad (7-7)$$

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ATOMICS INTERNATIONAL

A Division of North American Aviation, Inc.

NO. NAA-SR-MEMO 8963
DATE 9-9-63
PAGE 24 OF 26

VIII. Combination of Equations

Each node has an unknown, its temperature, to be determined as a function of time. In addition, the vapor volume nodes have the gas density and condensation rate to be determined. One equation has been set up for each of these unknowns. The equations are interdependent, however, because the heat flow into each node depends on the temperature of the adjacent nodes. These 116 equations, therefore, must be solved simultaneously.

The differential equations are to be solved by an iteration process, with time the independent variable. The initial conditions will be probably assumed to be saturated vapor at the initial temperature in the vapor chambers and some given temperature, probably 0°F, throughout the unit.

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~~UNCLASSIFIED~~
NO. NAA-SR-MEMO 8963

DATE 9-9-63

PAGE 25 OF 26

IX. Conclusion

In order to solve the equations which have been set up, certain parameters must be numerically evaluated. This includes information on thermal paths, system geometry, initial conditions, and operational conditions.

Thermal conductivities, heat transfer coefficients, effective distances between adjacent nodes, cross-sectional and surface areas of nodes, surface emissivities and view factors, vapor flow cross-sectional areas, and node volumes and masses are among the system parameters that must be evaluated. Mr. George Asanovich had been carrying out the calculation of these quantities from CRU-5 working drawings and from thermal data of system materials.

In addition, values or expressions for turbine work removal, internal electrical heat generation, frictional heat generation in bearing, pump, and rotor surface, supply mercury gas conditions, and mercury gas conditions in the chamber to which the CRU-5 vapor flow discharges must be known as inputs for the problem. Estimates for these are to be obtained from those studying these particular aspects of the problem.

Furthermore, an assumption must be made with regard to the mode of heat transfer from the outer surface of the CRU-5 containment vessel, or can, to its surroundings.

The necessary thermodynamic, fluid dynamic, and heat transfer equations have been set up. Numerical evaluation of system geometrical and thermal parameters must be completed, though a considerable amount of this work is done.

The resulting equations must then be set up in an iterative scheme for solution, and be programmed for the digital computer. This important portion of the work was not started.

In setting up the mathematical model for this analysis, by necessity, a number of assumptions and simplifications were made which would affect the accuracy of the result. Probably more important, however, would be the degree of accuracy to which certain input quantities are known, as turbine transient operating characteristics, frictional energy released in the liquid pump and bearings, heat generation by electrical losses, and transient mercury vapor conditions in supply and exhaust lines. The accuracy of the results would depend in large measure on the accuracy with which the above mentioned items are known. Much of these data may only be obtainable experimentally.

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NO. NAA-SR-MEMO 8963
DATE 9-9-63
PAGE 26 OF 26

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