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KIPS
KILOWATT ISOTOPE POWER SYSTEM

"CONTROL SYSTEM DYNAMIC ANALYSIS"

TOPICAL REPORT

Sundstrand Advanced Technology Operations

ROCKFORD, ILLINOIS 61101
unit of Sundstrand Corporation

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KIPS
KILOWATT ISOTOPE POWER SYSTEM

"CONTROL SYSTEM DYNAMIC ANALYSIS"

TOPICAL REPORT

Prepared for the
U.S. ENERGY RESEARCH AND
DEVELOPMENT ADMINISTRATION
UNDER CONTRACT EX-76-C-16-3071

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Prepared by: Sundstrand Advanced
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PREFACE

This report discusses the modeling techniques used to simulate the dynamic performance of the KIPS Power Conversion System fluid loops as presented at the third design review in December 1976. A companion topical report titled KIPS Control System Selection describes the overall control system requirements, selection and function of components, and a recently completed trade-off of the present hydromechanical valve against an electromechanical approach to the control valve package.

The KIPS utilizes two speed modes (1) essentially constant speed in orbital operation by utilization of parasitic loads to compensate for varying spacecraft loads and (2) "Speed Wild" for launch and other high vibration environments when a higher speed is desirable to provide added bearing load capability and the corresponding extra pump output pressure is available to provide higher jet velocity in the jet condenser. In this later mode the parasitic loads are non-operative and the speed will vary with the spacecraft load demand.

The analysis presented here shows that the KIPS is dynamically stable under all operating conditions. Further analysis and refinement is planned for the Phase II program.

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Introduction

This report discusses the modeling techniques used to simulate the dynamic performance of the KIPS*P.C.S. fluid loops.

Two types of analytic system dynamic models were set up to determine if the control system was stable and had sufficient response to keep the turbine inlet temperature within bounds under all transient conditions.

The first of these models is a hydraulic stability analysis of the fluid loop used to determine whether the loop amplifies or damps hydraulic perturbations in the control valve, regenerator, and boiler. Results of this analysis are covered in section 1.

The second of these models is a full system dynamic model in which the effects of step changes in load, radiator sink temperature, and other operating conditions can be investigated. A description of the model and results achieved is covered in Section 2.

The control system includes an electronic speed control which applies parasitic load as required to maintain constant **C.R.U. rotative speed, a dual function flow control valve to maintain constant turbine inlet temperature and to keep flow rate constant during speed transients, and a radiator bypass valve to maintain constant system heat rejection. A block diagram showing the location of various system components in the fluid loop is shown in Fig. 1.

* P.C.S. - Power Conversion System

** C.R.U. - Combined Rotating Unit

Fig 1 BASIC SYSTEM CONFIGURATION

The fluid loop control system is required to function during launch when the speed control is disabled and in a constant speed mode when deployed on a spacecraft.

During normal operation of the P.C.S. with the speed control circuit activated, the pulse width modulated speed control circuit applies electrical current to the dissipative load bank as required to maintain essentially constant speed. At constant speed and constant*H.S.A. heat input rate to the system, the power output of the system remains constant and the temperature control function of the flow control valve need only make slight trimming adjustments to accommodate minor changes in system heat rejection due to changing solar insolation and to accommodate the very slow rate of isotope decay. The pressure regulator function of the valve maintains constant mass flow to the turbine loop even if pressure fluctuations occur due to low frequency boiler instability or pump output ripple.

If the system operating point is not exactly thermodynamically balanced, turbine inlet temperature will begin to drift. The temperature control function of the flow control valve then corrects turbine inlet temperature by modulating flow. The liquid filled bulb of the temperature sensor has a 10 second time constant and provides an integrating effect due to its thermal inertia. As a result the temperature correction is slow and gradual compared to the response of the speed control and pressure regulator.

* H.S.A. _ Heat Source Assembly

During the large speed transient that occurs when the load bank is switched off to allow the system to run speed wild for launch or other high vibration environments, the pressure regulator continues to provide the flow controlling function required to maintain fluid loop stability. When load is dropped, turbine speed increases to seek a new power balance point. Due to the high inertia to load ratio of the C.R.U. the rate of change of turbine speed is very slow, taking at least 60 seconds to go from rated speed to maximum speed when electrical load is dropped. The pressure regulator portion of the valve is a low mass system with a fast response time so it can easily track the system speed and pressure changes that occur during frequency wild operation.

The boiler tubing has a large thermal storage effect which prevents rapid fluid temperature excursions with flow perturbations. The boiler vapor inventory also acts as an accumulator in the fluid loop to filter any feed pump or control valve hydraulic ripple.

The regenerator acts as an additional stabilizing influence during speed wild operation. When the turbine speed increases, turbine outlet temperature rises, causing the regenerator temperature to rise and effectively storing heat in the regenerator. When load is reapplied this stored heat is returned to the boiler inlet liquid flow, providing additional power capability to handle any high transient loads that may occur.

A radiator bypass valve is also included in the control system.

This valve is not essential to normal system operation and is designed to fail closed so that the system can continue to operate. The valve bypasses some liquid flow around the radiator to keep the mixed radiator outlet temperature at 165 degrees F in cases where the radiator sees a cold sink and can reject excessive heat. Without the bypass valve, system electrical output can drop by 40 watts (1300 watt system) under the worst case heat sink condition. The valve has a slow response time to insure that no rapid perturbations are caused in the radiator flow loop.

SECTION 1

CONTROL SYSTEM HYDRAULIC MODEL

A cross section of the KIPS flow control valve incorporating both the temperature control and pressure regulator functions is shown in Figure 2. Analysis of the hydromechanical performance of the valve when installed in the turbine loop was performed using the hydraulic circuit of Figure 3. For modeling purposes, the thermodynamic effects of the system were assumed to remain constant for any given steady state operating condition. Therefore, pump pressure, turbine nozzle flow, and the temperatures and densities at the nodes 1, 2, 3, and 5 were assumed constant for a given condition. The model encompasses the hydraulic and mechanical dynamics of the system from the pump to the turbine nozzle. The power bond graph, Figure 4, describes the nodes and methods used to model the system. A system block diagram and parameter list is shown in Figure 5 and 6 respectively. The model includes the hydraulic compressibility (nodes 1,2,3, and 5), the transmission line from the valve to the phase transition point, and mechanical dynamics of the valve. All performance is based on lumped parameter analysis techniques. The dynamic equations can be developed as follows:

KIPS GDS FLOW CONTROL VALVE

FIGURE 2

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FIGURE 3

KIPS FLOW CONTROL VALVE
HYDRAULIC CIRCUIT

$\longrightarrow$ Power flow
 SF - Flow source
 C - Capacitance
 I - Inductance
 R - Resistance

FIGURE 4
KIPS FLOW CONTROL VALVE
POWER BOND GRAPH

BY Attorney

FIGURE 5

KIPS FLOW CONTROL VALVE

BLOCK DIAGRAM

FIGURE 6

PARAMETER LIST

PARAMETER	DESCRIPTION	UNITS
a_{xy}	Orifice area (node x and y)	in^2
A_x	Area (node x)	in^2
C	Viscous friction coefficient	$\text{lb}\cdot\text{sec}/\text{in}$
F_N	Flapper nozzle force	lbf
d	Nozzle diameter	in
k_O	Flow coefficient	$\text{in}^2\text{lb}^{1/2}$
k	Spring rate, effective flapper-bellow	lb/in
l	Flapper length to nozzle	in
L_3	Line length to vapor fluid transition	in
m_e	Effective flapper-bellows mass	lbm
P_x	Pressure (node x)	psi
Q_{xy}	Flow (node x to node y)	in^3/sec
r	Length To Nozzle	in
S	Laplace operator	-
V_x	Node x volume	in^3
X	Flapper position	in
X_O	Flapper initial position	in
β	Working fluid effective bulk Modulus	psi
W	Fluid flow rate	in^3/sec

JOB KIPS FLOW
CONTROL VALUE
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Combining the system equations yields:

$$\dot{P}_2 = \frac{P_2}{V_2} \left[Q_{12}(x) k_0 \sqrt{P_1 - P_2} - Q_{23} k_0 \sqrt{P_2 - P_3} - A_2 \dot{X} \right]$$

$$\dot{P}_3 = \frac{P_3}{V_3} \left[Q_{23} k_0 \sqrt{P_2 - P_3} - Q_5 + A_4 \dot{X} \right]$$

$$\dot{P}_5 = \frac{P_5}{V_5} \left[Q_5 - \frac{P_5 + 6.8367}{2551.02} \right]$$

$$\ddot{X} = \frac{1}{m} \left[-k_2 X - F_N - A_3 P_3 - C \dot{X} + A_2 P_2 + F_{P_2} \right]$$

$$\dot{Q}_5 = \frac{A_3}{P_3 L_3} (P_3 - P_5)$$

where

$$Q_{12}(x) = \pi d \left(x_0 - \frac{x_L}{\tau} \right)$$

These equations can be linearized to yield:

$$\dot{P}_2 = \frac{P_2}{V_2} \left[\frac{\partial Q_{12}}{\partial X} X + \frac{\partial Q_{12}}{\partial P_1} P_1 + \left(\frac{\partial Q_{12}}{\partial P_2} - \frac{\partial Q_{23}}{\partial P_2} \right) P_2 - \frac{\partial Q_{23}}{\partial P_3} P_3 - A_2 \dot{X} \right]$$

$$\dot{P}_3 = \frac{P_3}{V_3} \left[\frac{\partial Q_{23}}{\partial P_2} P_2 + \frac{\partial Q_{23}}{\partial P_3} P_3 - Q_5 + A_4 \dot{X} \right]$$

$$\dot{Q}_5 = \frac{A_3}{P_3 L_3} [P_3 - P_5]$$

$$\ddot{X} = \frac{1}{m} \left[-kX - \frac{\partial F_N}{\partial X} X - \frac{\partial F_N}{\partial P_1} P_1 - \frac{\partial F_N}{\partial P_2} P_2 - A_3 P_3 + A_2 P_2 - C \dot{X} \right]$$

The partial derivatives are: where (°) indicates steady state conditions.

$$\frac{\partial F_H}{\partial X} = - \frac{8 L^2 C_D^2 (X_0 - \frac{X L}{r}) P_1^{\circ}}{r^2}$$

$$\frac{\partial F_H}{\partial P_1} = \frac{L \pi d^2}{4 r}$$

$$\frac{\partial F_H}{\partial P_2} = - \frac{L \pi d^2}{4 r}$$

$$\frac{\partial Q_{12}}{\partial X} = - \frac{\pi d k_0 (P_1^{\circ} - P_2^{\circ})^{\frac{1}{2}}}{r}$$

$$\frac{\partial Q_{12}}{\partial P_1} = \frac{\pi d (X_0 - \frac{X L}{r}) k_0 (P_1^{\circ} - P_2^{\circ})^{\frac{1}{2}}}{(P_1^{\circ} - P_2^{\circ})}$$

$$\frac{\partial Q_{12}}{\partial P_2} = - \frac{\pi d (X_0 - \frac{X L}{r}) k_0 (P_1^{\circ} - P_2^{\circ})^{\frac{1}{2}}}{(P_1^{\circ} - P_2^{\circ})} = - \frac{\partial Q_{12}}{\partial P_1}$$

$$\frac{\partial Q_{23}}{\partial P_2} = \frac{a_{23} k_0 (P_2^{\circ} - P_3^{\circ})^{\frac{1}{2}}}{(P_2^{\circ} - P_3^{\circ})}$$

$$\frac{\partial Q_{23}}{\partial P_3} = - \frac{a_{23} k_0 (P_2^{\circ} - P_3^{\circ})^{\frac{1}{2}}}{(P_2^{\circ} - P_3^{\circ})} = - \frac{\partial Q_{23}}{\partial P_2}$$

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The state equations are: $x_1 = P_2$, $x_2 = P_3$, $x_3 = Q_5$, $x_4 = X$, $x_5 = \dot{X}$, $x_6 = P_5$

$$\dot{x}_1 = \frac{\beta_2}{V_2} \left[\frac{\partial Q_{12}}{\partial X} x_4 + \left(\frac{\partial Q_{12}}{\partial P_2} - \frac{\partial Q_{23}}{\partial P_2} \right) x_1 - \frac{\partial Q_{23}}{\partial P_3} x_2 - A_2 x_5 \right]$$

$$\dot{x}_2 = \frac{\beta_3}{V_3} \left[\frac{\partial Q_{23}}{\partial P_2} x_1 + \frac{\partial Q_{23}}{\partial P_3} x_2 - x_3 + A_3 x_5 \right]$$

$$\dot{x}_3 = \frac{A_3}{P_3 L_3} [x_2 - x_6]$$

$$\dot{x}_4 = x_5$$

$$\dot{x}_5 = \frac{1}{m_e} \left[-k_e x_4 - \frac{\partial F_N}{\partial X} x_4 + \left(\frac{\partial F_N}{\partial P_2} + A_2 \right) x_1 - A_2 x_2 - C x_5 \right]$$

$$\dot{x}_6 = \frac{\beta_5}{V_5} \left[x_3 - x_6 / 2557.02 \right]$$

Stability analysis was performed using a linearized model as shown schematically in the block diagram of Figure 5. Six conditions were analyzed. These are listed below.

CONDITION	P ₁	P ₂	P ₃	T ₁	T ₅
1	68.6	63.62	58.46	173	645
2	139.71	69.02	63.06	200	645
3	221.07	70.51	64.33	220	645
4	84.25	79.05	74.03	212	660
5	161.6	86.83	80.91	230	660
6	233.24	88.56	82.44	240	660

Figures 7 through 18 show the frequency response of the turbine nozzle pressure to disturbances in pump discharge pressure. For this analysis, the mechanical damping of the flapper and bellows was assumed zero. As a result, gain peaks are evident at some conditions near the resonant mechanical frequency of the valve. In actuality, the damping of the trapped fluid in the bellows in conjunction with viscous friction effects would eliminate any mechanical resonant peaks.

This analysis demonstrates that the hydro-mechanical stability of the flow control valve is acceptable in that input disturbances are attenuated at all given conditions. Predominant time constants of the hydro-mechanical system range from 6 - 36 seconds depending upon the performance conditions.

FIGURE 8
CONDITION 1

FIGURE 13
CONDITION 4

SECTION 2

Control System Dynamic Model

A Computer simulation model of the Sundstrand organic rankine cycle energy conversion system has been developed to permit investigation of the system dynamic performance characteristics. This model is capable of simulating the important parameter interrelationships as first order dynamic responses of system temperatures, flowrates, and control signals.

The energy conversion system consists of the heater, turbine, regenerator, jet condenser, pump, radiator and the necessary controls. Schematically shown in Fig. 1 this system performs the conversion of the energy in a hot fluid to an output of electrical power. The analysis supporting each component model will be described, followed by a consideration of the integration of the complete energy conversion system.

Heater

A wide range of possibilities are available in modeling boiler systems, such as the heater being discussed here. The level of complexity increases very rapidly as small increments of detail are added to the model. The problem was to design a model which accurately allowed simulation of first order dynamic effects while at the same time remaining computationally efficient. The schematic shown in Fig. 19 illustrates the general structure of the model which has been used.

The heat model consists of three nodes lumping one half of the tube mass into each of two and the total liquid mass into the third. The isotope heat source is represented by a fixed heat flow into each of the wall nodes. Lumping the total

FIG 19 KIPS BOILER MODEL SCHEMATIC

liquid mass into one node allowed considerable simplification of the analysis by enabling a simple enthalpy balance to represent a process including the liquid-vapor phase change. This method omits detailed consideration of the variety of heat transfer processes existing, but for the present application allows the heater to interact in a useful way as part of the system model.

The differential equations governing heater dynamics are:

Liquid node (1)

$$m_1 \bar{C}_{p1} \frac{dT_{out}}{dt} = \bar{h}_1 A_1 \Delta T_{w1} - (P_{out} - H_{in}) \dot{m}_1$$

Wall nodes (2)

$$M_w C_{pw} \frac{dT_{wo}}{dt} = HTIN - h_{lo} A_1 (T_{wo} - T_{out})$$

$$M_w C_{pw} \frac{dT_{wi}}{dt} = HTIN - h_{li} A_1 (T_{wi} - T_{in})$$

Where

T_{in} = liquid inlet temperature (°F)

T_{out} = liquid outlet temperature (°F)

T_{wo} = wall temperature at outlet (°F)

T_{wi} = wall temperature at inlet (°F)

$HTIN$ = Isotope heat input (Btu/sec)

h_{1o} = heat transfer coefficient at outlet (Btu/ft²/sec/°F)

h_{1i} = heat transfer coefficient at inlet (Btu/ft²/sec/°F)

$h_1 = (h_{1o} + h_{1i})/2$

A_1 = liquid-wall surface area (ft²)

ΔT_{wl} = log mean of $(T_{wo}-T_{out})$ & $(T_{wi}-T_{in})$

H_{out} = liquid enthalpy at outlet (Btu/lb)

H_{in} = liquid enthalpy at inlet (Btu/lb)

$\dot{m}_1$ = liquid flowrate (lb/sec)

C_{pl} = mean specific heat of liquid (Btu/lb/°F)

C_{pw} = specific heat of wall (Btu/lb/°F)

M_w = tube mass total (lb)

$M_1 = V_1 \rho_1$ (lb)

V_1 = tube internal volume (ft³)

ρ_1 = mean liquid density (lb/ft³)

The heat transfer coefficients are the result of the following calculation

$$h_1 = J(RE) \dot{m} C_{pl} / A_{f1} / PR^{2/3}$$

Where

$J(RE)$ = empirical factor based on heat transfer configuration
(ref Kays & London fig ST-1)

$\dot{m}$ = liquid massflow (lb/sec)

C_{pl} = liquid specific heat (Btu/lb/°F)

A_{f1} = flow area liquid path (ft²)

Re = Reynolds number = $d_h \dot{m} / A_{f1} \mu_1$

Pr = Prandtl number = $C_{pl} \mu_1 / K_1$

d_h = hydraulic diameter of liquid flow path (ft)

μ = liquid viscosity lb/ft/hr

K_1 = liquid thermal conductivity Btu/ft/hr/°F

The heater flow dynamics can introduce considerable complexity into the model and have been simplified to a single dynamic equation:

$$\frac{d p_{tin}}{dt} = \tau_p (P_{pump} - \sum \Delta P - P_{tin})$$

where

P_{tin} = turbine inlet pressure (psia)

τ_p = pressure time constant (1/sec)

$\sum \Delta P$ = total pressure drops (psi)

P_{pump} = pump outlet pressure (psia)

This process represents the inertial and accumulator effects present between the pump and heater outlet. The value of the time constant employed here results from a detailed hydraulic model described in Section 1.

Turbine

The turbine is represented by a table of data derived from a complex turbine off-design computer analysis developed by Sundstrand. The data carries the effects of varying speed, mass flow, inlet pressure and temperature, and exhaust pressure, so that the model can respond to changes in these conditions.

To explain the turbine model, start with these definitions of input information:

RPM - turbine speed (rpm)

Pin - inlet pressure (psia)

Tin - inlet temperature (°F)

Pe - exhaust pressure (psia)

XM - turbine mass flow (lb/sec)

R - gas constant (16.75 ft/°F)

γ - ratio of specific heats 1.0436

The adiabatic head supplied to the turbine is:

$$H = R(T_{in}) \frac{\gamma}{\gamma-1} \left(1 - (P_r)^{\frac{1-\gamma}{\gamma}} \right) \quad \text{ft}$$

where

$$P_r = P_{in}/P_{ex}$$

With H known the spouting velocity is found:

$$C_o = \sqrt{2g_c H} \quad \text{fps.}$$

The turbine pitchline velocity U is calculated by

$$U = \text{RPM} (D) \pi / 720 \quad \text{fps}$$

where

$$D = \text{turbine pitch diameter (in)}$$

This allows calculation of the U/Co ratio

Knowing these nondimensional numbers, the torque coefficient can be obtained by interpolating the data shown in Fig. 3

with the current P_r and U/Co values. The torque coefficient τ is defined as the ratio of actual to theoretical torque according to the following relation.

$$\tau = \sqrt{72g_c} T / (XM \sqrt{H} D)$$

where

T = turbine torque(ft-lb)

XM = turbine massflow

This relation may be solved for turbine torque which is the desired output.

$$T = \tau XM \sqrt{H} D / \sqrt{72g_c} \quad (\text{ft.lb})$$

Regenerator

The regenerator is a critical element in the system model because it provides a coupling or feedback effect on both hot and cold flow paths. A simple two node model will reasonably represent the regenerator thermal characteristics. One node lumps the metal mass of the regenerator core and allows a heat flow balance to be made. The other node represents the liquid contained within the core and allows a second heat balance.

These two balances are expressed as differential equations relating heat flows and thermal capacity of the two nodes.

KIPS TURBINE

FIG. 20 KIPS TURBINE PERFORMANCE

$$\text{Wall} \quad M_w C_{pw} \frac{dT_w}{dt} = \dot{m} (H_{vin} - H_{vout}) - h_1 A_{1\sigma} LM (T_v - T_l)$$

$$\text{Liquid} \quad M_l C_{pl} \frac{dT_{lout}}{dt} = h_1 A_{1\sigma} LM (T_v - T_l) - \dot{m} (H_{lout} - H_{lin})$$

where T_w = temperature of wall ($^{\circ}\text{F}$)

T_{lout} = temperature Dowtherm liquid outlet ($^{\circ}\text{F}$)

M_w = Wall mass (lb)

M_l = liquid mass = $V_l \times \rho_l$ (lb)

V_l = liquid volume of core (ft^3)

ρ_l = mean liquid density (lb/ft^3)

C_{pw} = specific heat of wall ($\text{btu}/\text{lb}/^{\circ}\text{F}$)

C_{pl} = specific heat of liquid ($\text{btu}/\text{lb}/^{\circ}\text{F}$)

$= (H_{lout} - H_{lin}) / (T_{lout} - T_{lin})$

CPV = specific heat of vapor ($\text{btu}/\text{lb}/^{\circ}\text{F}$)

$= (H_{vin} - H_{vout}) / (T_{vin} - T_{vout})$

H_{vin} = vapor enthalpy inlet (btu/lb)

H_{vout} = vapor enthalpy inlet (btu/lb)

H_{lin} = liquid enthalpy inlet (btu/lb)

H_{lout} = liquid enthalpy outlet (btu/lb)

$$LM(T_v - T_l) = \frac{(T_{vin} - T_{lout}) - (T_{vout} - T_{lin})}{\log_n \frac{(T_{vin} - T_{lout})}{(T_{vout} - T_{lin})}}$$

σ = proportional factor vapor to liquid $\Delta T = \frac{A_l h_l}{A_l h_l + A_v h_v}$

h_l = liquid heat transfer coefficient ($\text{btu}/\text{ft}^2/\text{hr}/^{\circ}\text{F}$)

$= h_{ldgn} (\dot{m}/\dot{m}_{dgn})^\gamma$

h_v = vapor heat transfer coefficient ($\text{btu}/\text{ft}^2/\text{hr}/^{\circ}\text{F}$)

$\dot{m}$ = Dowtherm massflow (lb/sec)

A node is not provided for the vapor flow through the regen-

erator because of the very low mass and hence thermal inertia involved. The low inertia allows the assumption of a continuous steady state equilibrium of the vapor heat input and heat loss to the regenerator core. This condition is represented by an iteration loop calculating the value of vapor outlet temperature required to balance the in and out flows. Mathematically this is expressed as:

$$\Delta T_{vw1} = \sigma Lm(T - T_1)$$

$$\Delta T_{vw2} = (T_{vin} - T_{vout}) \dot{m} C_{pv} / (h_v A_v),$$

iterate T_{vout} until $\Delta T_{vw1} = \Delta T_{vw2}$ with acceptable accuracy.

Regenerator performance may be expressed in terms of effectiveness defined by

$$\epsilon = \left[(T_{vin} - T_{vout}) / (T_{vin} - T_{lin}) \right] \times \text{CPV/min (CPV, CPL)}$$

This parameter is output by the regenerator model.

Jet Condenser

In the jet condenser the important dynamic effect is the mixed flow temperature variation and its dependent variable, condenser pressure. A simple heat balance is used to find the dynamic response of condenser temperature.

$$M_C C_{pl} \frac{dT_C}{dt} = m (H_{mix} - H(T_C))$$

where T_C = condenser accumulator temperature ($^{\circ}\text{F}$)

C_{pl} = condenser specific heat (btu/lb/ $^{\circ}\text{F}$)

M_C = accumulator liquid mass (lb)

m = Dowtherm mass flow (lb/sec)

H_{mix} = mixed enthalpy at inlet (btu/lb)

$H(T_C)$ = liquid enthalpy @ T_C (btu/lb)

Condenser pressure is found by determining the saturation pressure at T_C . The assumption is made that equilibrium is maintained at the liquid-vapor interface.

Flow Control

The flow control valve has been included in the system to maintain a relatively constant pressure at the turbine inlet in response to variations in pump pressure. Such control is obtained by placing a variable orifice in the pump outlet flow to generate a pressure drop. The position of the orifice opening and therefore pressure drop is a function of the turbine inlet conditions and the flowrate through the control.

Two dynamic effects are related to the flow control. The first is a first order lag caused by the thermal response of the turbine inlet temperature sensor. This is represented as:

$$\frac{dT_{sen}}{dT} = \tau_{sen} (T_{in} - T_{sen})$$

τ_{sen} = Sensor time constant (1/sec)

T_{sen} = sensor indicated temperature ($^{\circ}F$)

T_{in} = Turbine inlet temperatures ($^{\circ}F$)

With knowledge of the sensor temperature and mass flow rate it is possible to determine valve pressure drop from a map of valve characteristics.

This map, fig. 21 yields the resultant delta pressure at all operating conditions.

$THROT = f(T_{sen}, X_m) \text{ psi}$

$THROT = \text{commanded } \Delta P @ T_{sen} \text{ and } X_m$

KIPS HIGH REL. VALVE CASE 10

fig 21 Flow Control Characteristics

The second dynamic effect is caused by the pressure propagation delay between the flow control and turbine inlet. This is approximated by a first order system represented by

$$\frac{dP_{tin}}{dt} = \tau_p (P_{pump} - THROT - \Delta P_{htr} - \Delta P_{deg} - P_{tin})$$

P_{pump} = pump outlet pressure (psia)

ΔP_{htr} = heater pressure drop (psi)

ΔP_{deg} = regenerator liquid side ΔP (psi)

τ_p = equivalent time constant (1/sec)

P_{tin} = turbine inlet pressure (psia)

Pump

The pump model employed does not simulate internal dynamics of the centrifugal pump. It rather contains empirical data for the pump in use on the system. This allows evaluation of pump performance as it is affected by changing speed and flow conditions based on test results. (fig.22)

Alternator

Similar test data was used to find the alternator performance. Data is entered as a function of alternator load to find the efficiency. (fig.23)

Radiator

The radiator exhibits very slow dynamics due to its large mass. An existing off-design performance calculation has been combined with a first order heat balance to represent radiator dynamics

$$Q_{rad} = f(T_{in}, AM, TS)$$

PRELIMINARY EST. PERFORMANCE WITH MILLED VANE FABRICATED IMPELLER, DOUBLE
SHROUDED, BACKWARD CURVED VANES, $\beta_2 = 22\frac{1}{2}^\circ$ TO 25° , 33,688 RPM.
FOR "DOWTHERM A" AT APPROX. 210°F, 62.17 LB/FT³ = 8.31 LB/GAL., 2.316 FT/PSI.
DESIGN POINT = 0.282 LB/SEC = 2.036 GPM AT $[\Delta P = 47.6 \text{ PSI} = 110.3 \text{ FT } \Delta H \text{ STATIC}]$
+ ESTIM 3.75 FT $\Delta V^2/2g = 114.0 \text{ FT} = \Delta H \text{ TOTAL}$

Fig. 22 Pump Performance

plt 5/4/7

Q_{rad} = radiator heat rejection (Btu/sec)

T_{in} = radiator inlet temperature ($^{\circ}\text{F}$)

AM = radiator flow (lb/sec)

TS = sink temperature ($^{\circ}\text{R}$)

$$\frac{dT_{\text{rad}}}{dt} = (AM * (H(T_{\text{rin}}) - H(T_{\text{rout}})) - Q_{\text{rad}}) / (M_e C_{\text{pr}})$$

$H(T_{\text{rin}})$ = enthalpy @ T_{rin} (Btu/lb)

$T_{\text{rout}} = 2 * T_{\text{rad}} - T_{\text{rin}}$ ($^{\circ}\text{F}$)

$H(T_{\text{rout}})$ = enthalpy @ T_{rout} (Btu/lb)

$M_r C_{\text{pr}}$ = mass x specific heat of radiator ($\frac{\text{Btu}}{^{\circ}\text{F}}$)

Radiator Bypass

The radiator is sized to be sufficient to reject the required heat at a sink temperature of 422°R at beginning of mission. At other sink temperatures and later in the mission this results in the radiator being able to excessively cool the liquid flowing through it. This excess cooling lowers the system power capacity and therefore a radiator bypass has been included to improve operation.

The bypass is represented by a simple temperature error sensing control defined by

$$\frac{dB}{dt} = \tau_B K_B (T_{\text{rout}} - T_{\text{rset}}) - B$$

B = bypass fraction

T_{rset} = temperature set point ($^{\circ}\text{F}$)

τ_B = valve time constant (1/sec)

K_B = temperature gain (1/ $^{\circ}\text{F}$)

Organic Rankine Cycle System and Controls

Having defined the component models we must turn to the more complex problem of interconnecting the parts and providing the necessary logic to control the system in a desirable mode.

The first step is to identify cause and effect sequences traveling from component to component by studying the system schematic (fig 1). At each interface flows of fluid and heat are identified and defined symbolically. The inputs and outputs of each component are determined and their sources or destinations found.

Those variables for which differential equations have been written, the state variables, are initialized and are available at the simulation start-up. They are:

- 1 Turbine speed (RPM)
- 2 Flow control sensor temperature (°F)
- 3 Radiator bypass fraction
- 4 Regenerator mean wall temp (°F)
- 5 Regenerator outlet liquid temp (°F)
- 6 Condenser temp (°F)
- 7 Heater inlet wall temp (°F)
- 8 Heater outlet wall temp (°F)
- 9 Heater fluid outlet temp (°F)
- 10 Heater outlet pressure (psia)
- 11 Radiator mean temperature (°F)

The Dowtherm flowrate is the basic variable within the system and its value is set by the flow control in relation to the turbine nozzles. With flowrate known several other parameters may be determined. Pump performance data gives pump pressure and

efficiency as a function of flowrate and turbine RPM. Pump pressure less flow control pressure drop, regenerator line loss and heater line loss gives heater out pressure necessary in the turbine flow calculation.

Knowing flowrate and heater outlet temperature and pressure, turbine torque can be calculated. The energy extracted from the Dowtherm flow by the turbine must be subtracted from the heater outlet energy flow to yield the regenerator vapor inlet conditions. This requires an iteration to find turbine outlet temperature by matching fluid enthalpy at outlet conditions with enthalpy at inlet conditions less power generated.

In combination with the turbine torque calculation are the effects of load transients on alternator and pump losses. Torque remaining after the load and losses are subtracted from the turbine torque yields the turbine acceleration torque. This torque divided by the sum of turbine and alternator inertias gives the turbine speed derivative.

$$\frac{d\omega}{dt} = (T_t - T_l - T_p - T_{alt}) \frac{1}{J_t} \frac{\text{rad}}{\text{SEC}^2}$$

ω = turbine speed (1/sec) (x30/ π yields RPM)

T_t = turbine torque (ft-lb)

T_l = load torque (ft-lb)

T_p = pump loss torque (ft-lb)

T_{alt} = alternator loss torque (ft-lb)

J_t = moment of turbine shaft (ft.lb.sec²)

Having defined the regenerator vapor inlet conditions, all necessary inputs to the regenerator model are available. This model determines the two derivatives of regenerator temperatures as well as providing the value for regenerator outlet vapor temperature. This temperature becomes the input to the condenser model.

We have now returned to our starting point. This circular logic path is exactly the simulation sequence. As the loop is repeatedly traversed all of the time varying system parameters are updated in response to changes in dynamic conditions.

2. Applications of Dynamic Model

The model has been structured so that all modes of system operation may be investigated. Such areas might be:

1. Load transients
2. Environmental effects
 - a. Sink temperature variation
 - b. Heat input changes
3. Parameter Variations
 - a. Effects of component performance variations
 - b. Control logic options
 - c. System operation as mission progresses

Some examples of the results obtained from simulation studies are shown in fig. 24-26 the first two figures show the impact of net load changes on system operation. The conditions studied are identical in both figures except the time constants in the flow control and pressure response differential equations are respectively the longest and shortest expected.

The stable response indicates that a surging instability between the flow control and heater is unlikely. In fig.26 the effects of a step change in radiator sink temperature is shown. As can be seen the dynamics in this area are very slow and the response is stable.

3. Conclusions

As shown by the figures included here plus additional studies of a similar nature, the KIPS operates in a stable fashion. The response to changes in operating conditions are accomodated by the control devices provided and no undesirable modes of interaction have been discovered.

Fig. 24

KIPS DYNAMIC SIMULATION

figure shows effects of net load changes at BOM,
 TSINK = 422° R, and longest expected time constants

Fig. 25 KIPS DYNAMIC SIMULATION

figure shows effects of net load changes at BOM,
 TSINK = 422° R, and shortest expected time constants,

Fig. 26

KIPS DYNAMIC SIMULATION

figure shows effects of sink temperature changes at BOM,
NET LOAD = 1.3 kw, and longest expected time constants.