

General Disclaimer

One or more of the Following Statements may affect this Document

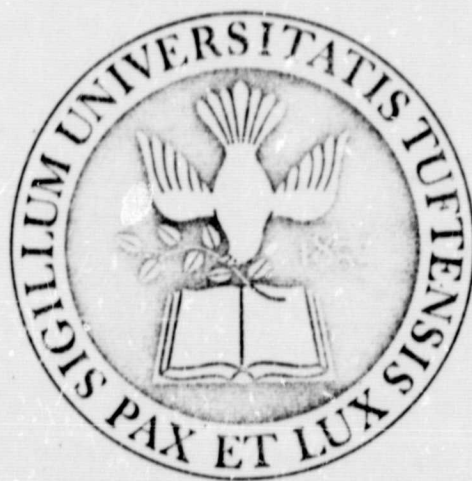
- This document has been reproduced from the best copy furnished by the organizational source. It is being released in the interest of making available as much information as possible.
- This document may contain data, which exceeds the sheet parameters. It was furnished in this condition by the organizational 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.

STUDY OF VARIABLE THERMAL

IMPEDANCE DEVICES

By JOHN H. SUNUNU

MAY 1970



PREPARED UNDER CONTRACT No. NAS 12-2128

BY

TUFTS UNIVERSITY

MEDFORD, MASSACHUSETTS

ELECTRONICS RESEARCH CENTER

NATIONAL AERONAUTICS AND SPACE ADMINISTRATION



FACILITY FORM 602

N70-41322	
(ACCESSION NUMBER)	(THRU)
32	1
(PAGES)	(CODE)
CR43873	09
(NASA CR OR TMX OR AD NUMBER)	(CATEGORY)

TABLE OF CONTENTS

	Page
Summary	1
Introduction	1
Basic Configurations of Potential Variable Thermal Impedances	2
The Design of the Roller VTI Under Investigation	3
Description of the Experimental Apparatus for the Roller VTI	6
Experimental Procedures for the Roller VTI	10
Experimental Set-up for Fluid Loop VTI	10
Experimental Procedure for the Fluid Loop VTI	11
Experimental Results	12
Conclusions	24
Appendix A - Comparison of Fixed Thermal Resistance Versus a Variable Thermal Resistance	25
Appendix B - Analysis of the Thermal Impedance of the Roller VTI	27
Appendix C - Analysis of the Thermal Impedance of a Fluid Loop VTI	29

STUDY OF VARIABLE THERMAL IMPEDANCE DEVICES

by John H. Sununu

TUFTS UNIVERSITY
Medford, Massachusetts

SUMMARY

This study was conducted under National Aeronautics and Space Administration Electronics Research Center Contract NAS 12-2128. The effort called for under this contract involved the study and experimental evaluation of a variable thermal impedance device (VTI) as applicable to inertial sensor subassemblies. In particular, it involved obtaining test data on an existing VTI and other representative devices.

The existing VTI had been built for NASA/ERC under Contract NAS 12-954 by Honeywell Aerospace, Florida. The basic concept involved the variation of the cross sectional area of the heat path between the source and sink. This experimental effort determined the extremes of the range of thermal impedances obtainable with this device, the effect of key parameters such as heat load and sink temperature on the thermal impedance, and the dynamic capability of the device to respond to rapid variations in environmental conditions.

In addition to evaluation of the roller VTI, tests were conducted to determine the extreme range of thermal impedances obtainable from a simple system consisting of a pair of liquid cold plates and a pump.

In both cases the data obtained was correlated to the analytical evaluation of system performance.

INTRODUCTION

The performance of the strapdown inertial sensing unit subassembly (ISUS) is extremely dependent on the thermal design of the system. The temperature of most inertial sensors must be kept virtually constant if accurate measurements are to be made. This constant temperature must be maintained even under extremely varying conditions of the environment. In order to accomplish this regulation, elaborate thermal control systems are often utilized.

In most applications, the temperature variations are eliminated by using heaters to maintain the temperature by varying the net power dissipation whenever the environmental conditions change. This approach often results in temperature control power requirements as high as, or even higher, than the actual power requirements of the inertial sensors themselves. The power requirement for ISUS space flight applications can be reduced considerably if, instead of varying the power dissipation from the block to the ambient, the thermal resistance between the load to ambient is varied. Some efforts have been made to incorporate a variable thermal impedance (VTI) into a few bread-board systems. To date, however, it appears that no flight strapdown system has made full use of this concept.

This approach is ideally suited to strapdown systems since in such a unit the inertial measuring unit can be designed so that virtually all the thermal paths from the critical sensors to the environment are conductive paths. Thus, all thermal performance dependence is shifted away from the relatively unpredictable convective and radiative modes of heat transfer. It is therefore quite simple to interpose within the conduction paths between source and sink a device whose impedance is controllable to achieve this variation of resistance from load to ambient.

The potential power savings that can be achieved by utilizing such a device can easily be calculated. The analysis of this power saving is shown in Appendix A where the performance of a system with a VTI is compared to the requirements of a system utilizing a fixed impedance.

The test program evaluated the VTI as a component under steady state and dynamic thermal environments. The heating loads, the simulated environmental temperature and the rate of change of temperatures and loads were representative of typical flight conditions.

The cold plate system examined provided some reference data for the potential extreme values of thermal impedances which might be achieved with a relatively simple system. Furthermore, the power requirements of such a system suggest that this basic configuration might be considered for design into a future flight system.

BASIC CONFIGURATIONS OF POTENTIAL VARIABLE THERMAL IMPEDANCES

Since the VTI concept is ideally suited to applications which have been designed to take advantage primarily of the conductive modes of heat transfer it is appropriate that we

examine mechanisms applicable to providing variation of the conductive impedance of a system to achieve the desired variation. In general, this can be accomplished by using either of three basic approaches.

In the first, a variation of the conductive path cross sectional area will result in an appreciable change in thermal resistance. Typical configurations taking advantage of this approach includes systems in which conductive members are introduced into the thermal path to augment a basic minimal path which is permanent to the device. This approach is equivalent to the introduction of a thermal shunt across a high resistance.

The second concept produces thermal impedance variation by providing a mechanism for changing the thermal resistance across an interface. This may be achieved by introduction of a liquid to fill a gap existing between primary conduction elements, or by allowing conductive members to snap into place sequentially against a primary member.

The third basic approach is to utilize a carrier element, such as a fluid loop, in order to provide the coupling between two segments of the system. In this manner it is quite easy to vary the rate of flow and thereby vary the intimacy of coupling between these two points. Configuration such as this can be provided simply by using two fluid cold plates and a fluid mover such as a pump. The pump can be actuated by means of a temperature controller and amplifier. For this system to have any significant advantage the power required for the pump must be considerably less than the power that would be required for thermal control heaters in a fixed impedance system.

It should also be noted here that a variation of thermal impedance can also be achieved by providing a variation in the radiation heat transfer coupling between two portions of the system. Of all the systems that have been used to date in actual flight applications virtually all of them have been based on changing the radiation heat transfer mechanism. A typical example of this is the venetian blind louvers utilized on some satellite applications to vary the thermal coupling from packages in the interior of the system to the environment. It appears however that utilization of such a concept defeats some of the principal advantages inherent in the strapdown system. Therefore, this investigation limited itself to those which could be grouped under the conductive mode.

THE DESIGN OF THE ROLLER VTI UNDER INVESTIGATION

The variable thermal impedance which was examined was an integral, sealed, fluid filled component. This VTI operated by changing the position of a set of truncated rollers contained

between two shaped aluminum plates. The gap between the rollers and the plates and the volume surrounding the rollers was filled with a moderate: (150 centipoise at 70°F) viscosity oil.

The rotation of the truncated rollers provided for a change in the conductive path between the upper and lower plates of the device. When the rollers, shown in Fig. 1, were rotated to a position which allowed for their orientation parallel to the plates, the system was in the least conductive mode. If the rollers were turned so that they would align themselves in the manner perpendicular to the plates, the system was in the most conductive mode. Another way of viewing the phenomena that takes place is to consider that the rollers are effectively varying the thickness of the fluid gap within the system and replacing the low conductive fluid with the higher conductivity material of the rollers themselves.

The principal components of the VTI are a temperature sensitive actuator, the driving rod, yoke and cam assembly, the linkage of the system, the rollers, a restoring bellows, and an expansion bellows.

The temperature sensitive actuator consists of a wax-copper substance which is enclosed in a sealed metal container. As the temperature increases this system changes phase and expands considerably and produces an appreciable force.

A driving rod with a diameter of approximately 1/8 of an inch and a length of two inches is connected to the small wax container. This rod transmits the force created by the expansion of the sensor to the yoke and cam assembly. The yoke and cam assembly receives the linear motion of the driving rod and converts this motion to a rotary motion to drive the rollers.

The VTI contained five truncated rollers which were made of aluminum and were approximately five inches long. The rollers were mechanically connected to the cam and were designed to rotate about a bearing assembly. The system is restored to its original position when the wax is cooled and allowed to relax, by means of a restoring bellows. This bellows also serves to provide a seal to contain the wax. In addition to the restoring bellows there is also a stainless steel expansion bellows located in the center of the component. This expansion bellows takes up any slack expansion of the fluid contained in the system.

Description of operation.--Under conditions in which the VTI is being used to control temperature, any increase in sensor temperature causes the wax to expand. This expansion exerts a force which is transmitted to the yoke and cam. This motion is then converted to a rotary motion and results in a rotation of the rollers through a linkage. This rotation drives the rollers away from the non-conductive position of Fig. 1 toward the conductive position. Thus the geometry of the small gap is varied considerably resulting in a change of the thermal conductance across the plate.

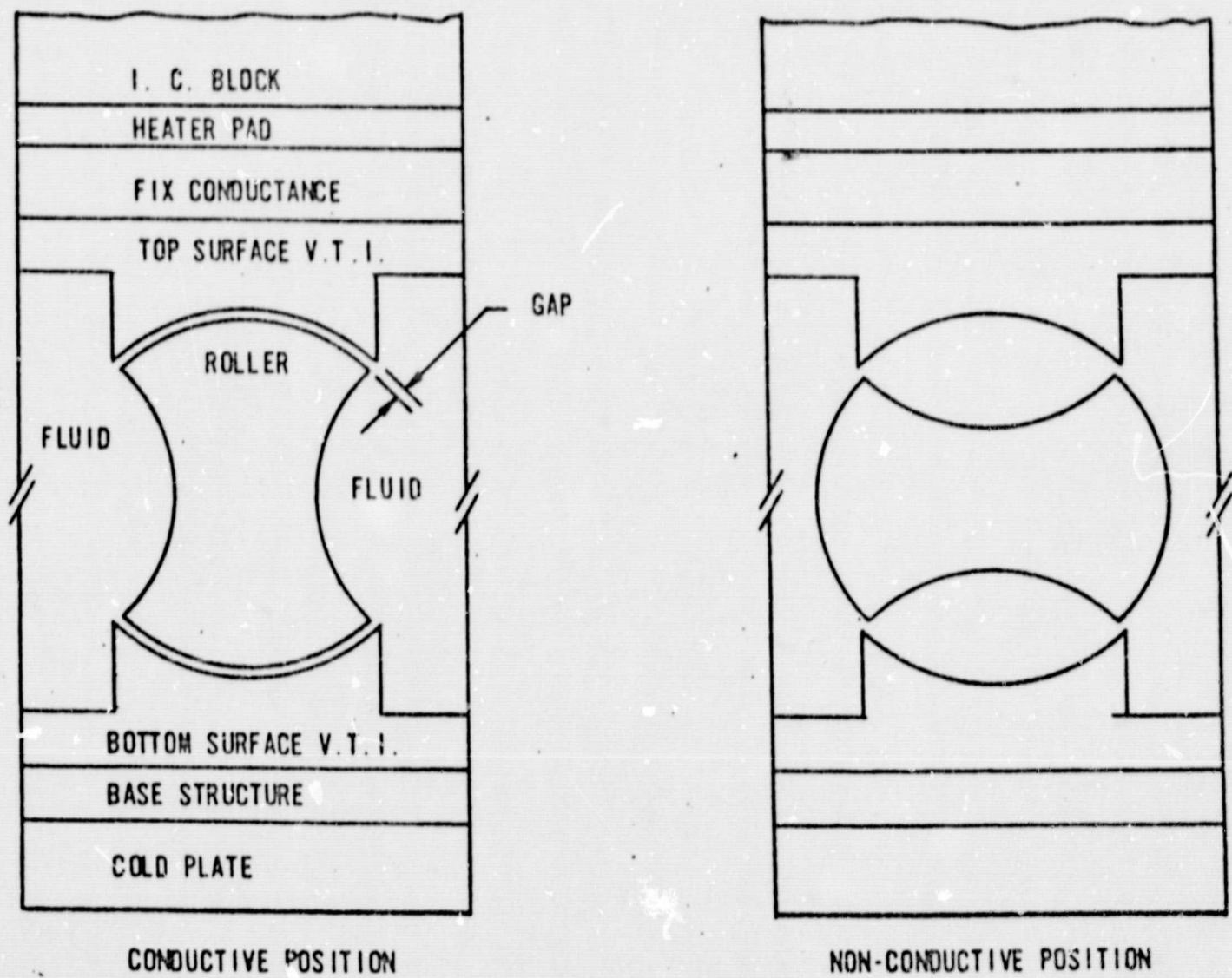


FIGURE 1. BASIC COMPONENTS OF THE VARIABLE THERMAL IMPEDANCE IN THE CONDUCTIVE AND NONCONDUCTIVE POSITION

DESCRIPTION OF THE EXPERIMENTAL APPARATUS FOR THE ROLLER VTI

The experimental setup used to evaluate the VTI is shown in the photographs of Figure 5. The component was mounted in intimate contact between a plate selected to represent the source, and a cold plate which was used to simulate the environment. This sandwich was then encased in styrofoam. This enclosure is the white object in the center of the photograph. A sufficiently large thickness of foam was utilized to insulate the system entirely from the environment.

A refrigeration cooler pump and heated reservoir system was used to circulate temperature controlled water to the cold plate. This system is shown in the lefthand portion of the photograph. The temperature of the heat load source plate, the sink cold plate and the VTI itself were monitored by means of thermocouples and the data was collected on the strip chart thermocouple recorder shown at the right of the photograph.

The heating of the dummy load plate was provided by means of three electrical resistance heating blankets which were bonded to the surface of the plate utilizing Eastman 910 adhesive. The plate served to distribute the heat generated by the heat element to provide a relatively uniform distribution across the VTI surface. This was a more satisfactory representation of the actual system than bonding the heat elements directly to the VTI surface. The gap between the dummy source plate and the VTI had an extremely light coating of silicone oil between the two surfaces. The surface representing the environment was a combination of both a fluid cold plate and a mating plate. The presence of the intermediary plate allowed for counterboring and channeling to permit use of thermocouple measurements in the center of the cold plate and VTI cold surface.

All temperatures were measured utilizing copper-constantan thermocouples with enamel and glass insulation (Leeds & Northrup type No. 30-555-1). Thermocouples were welded in an inert gas (Argon) utilizing a thermocouple welder (Dynatec Model 116). After welding, thermocouples were soldered to a 3/8" square 0.002" brass shim stock which in turn was bonded to the VTI surface using Eastman 910 adhesive. The temperatures sensed by the thermocouples were recorded on a Leeds & Northrup Speedomax W with 24 channels.

The thermal conductivity of the styrofoam enclosure was approximately .02 Btu/hr. per sq. ft. °f/ft. The wall thickness of the foam oven was at least 2" everywhere and in most cases as high as 4". The thermocouple and heater element wires were passed through the styrofoam. Holes were also carved into the styrofoam to allow for the inlet and outlet tubes of the cold

plate. The entire sandwich was bonded together utilizing an upper and lower wooden plate to compress the oven and VTI sandwich utilizing three 13" lengths of 1/4-20 threaded rod. This provided a compact assembly which was approximately equal to a 13" cube which could be oriented as desired to test the VTI at different orientations. A small removable peephole was carved into the enclosure to allow for visual observation of the roller motion.

The power to the heater elements was regulated by means of a variac and the voltage and currents were recorded. The cold plate consisted of a brass section with a series channel milled into it for water flow. A copper cover plate was provided and bolted to the milled section with a gasket sandwich between them. The final assembly was sealed utilizing RTV silicone rubber.

The refrigeration and circulation of the water was provided by a Cordley SR 30 cold water chiller and circulator. A large reservoir was added to the system to provide for a more stabilized water temperature. This system was able to contain and circulate approximately 3 gallons of water as coolant.

A safety float and limit switch was provided to allow for operation of the system around the clock with assurance that in the event of loss of water the entire system would automatically shut down.

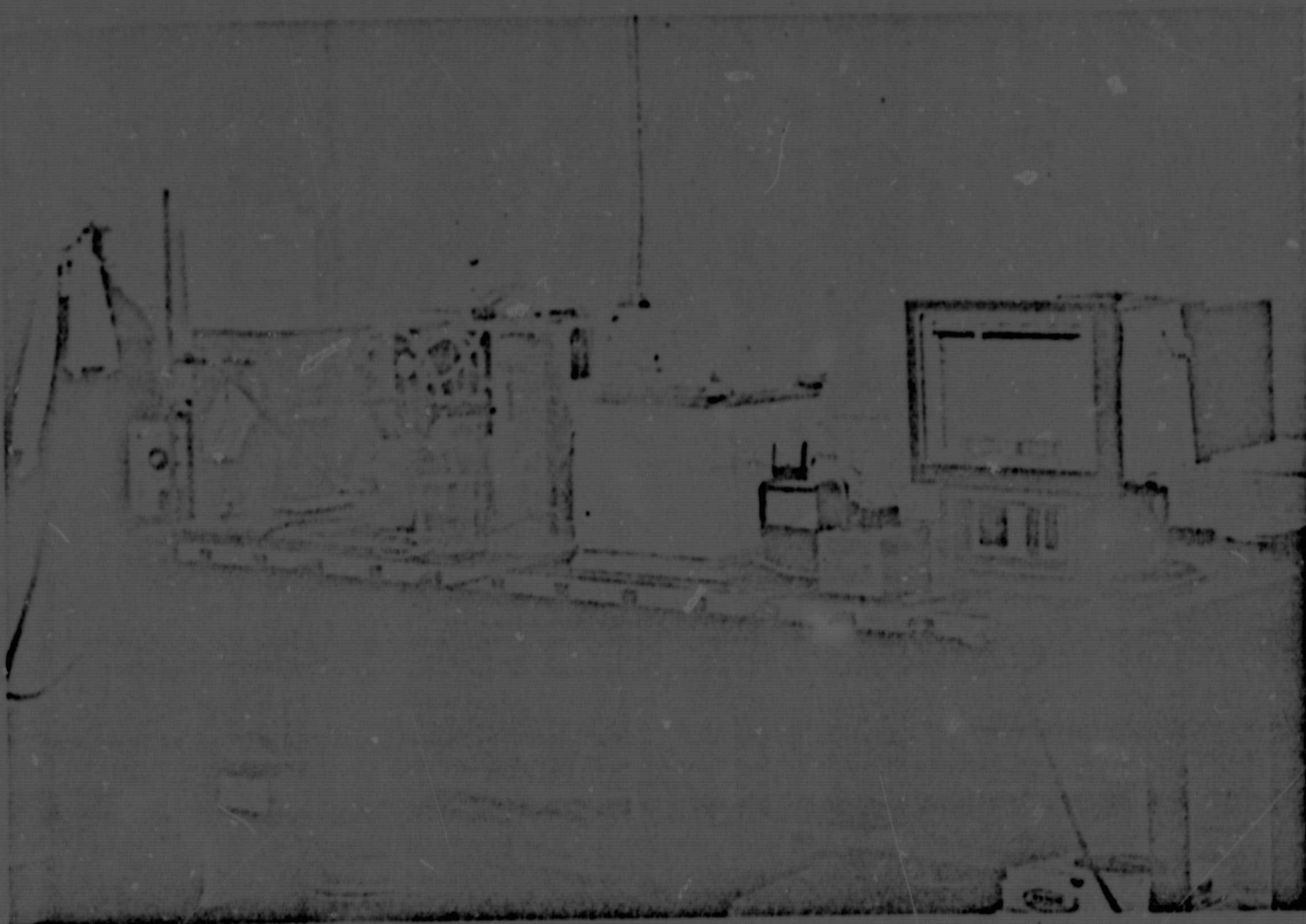


FIG. 2 Experimental Set Up for Testing Roller VTI



Fig. 3 VTI Mounted on Cold Plate with Thermocouples



FIG. 4 VTI Upper Surface and Control Surface Plate

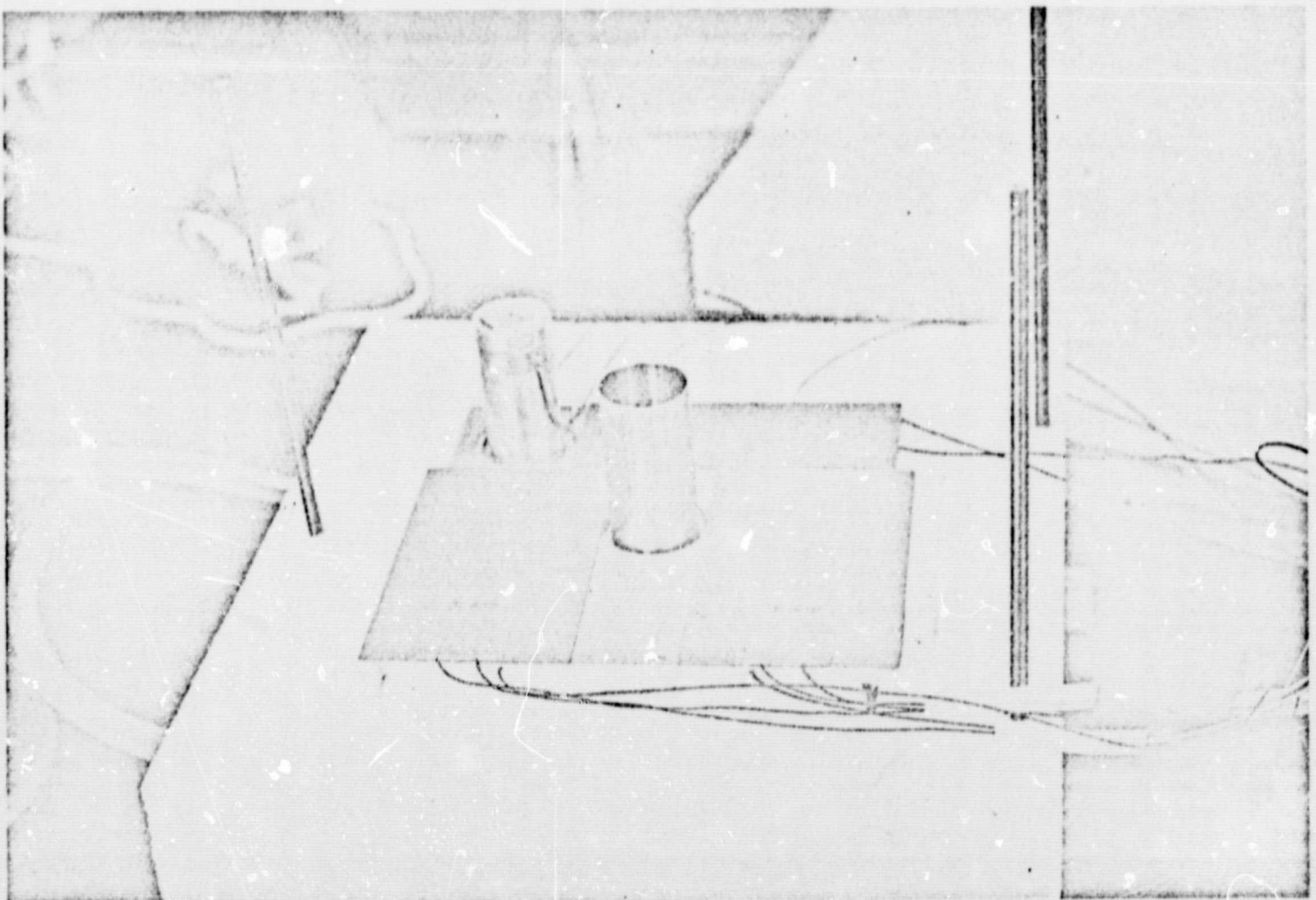


FIG. 5 Roller VTI in Styrofoam

EXPERIMENTAL PROCEDURES FOR THE ROLLER VTI

In order to obtain data on the extremes of the thermal impedance values of the system and to determine the relative sensitivity of the various orientations an essentially straightforward experimental procedure was used. For a given set point of input power, coolant temperature, and orientation, the system was allowed to come to equilibrium. Equilibrium in this case was defined as being that condition where no appreciable variation of temperature with time could be noted on the recordings as corrected by the L & N chart. For most situations of the system this occurred within approximately 45 minutes to an hour and a half.

The evaluation of the response of the VTI to the dynamic variation of conditions required a slightly different procedure. In fact, the response to three slightly different modes was examined.

The procedure for the first mode of operation consisted of allowing the VTI to come to equilibrium at a fixed value of thermal load, under fixed environmental conditions. The environment was then allowed to vary in temperature as a ramp function from the initial temperature to a second considerably different temperature. All the significant temperature of the system were continuously recorded.

The second mode of operation was somewhat similar to the first; however, the temperature change in the environment was not limited to a slow ramp but was varied as rapidly as possible to simulate a step function change in the environment. Once again all the critical parameters of this system was monitored throughout until the system came to equilibrium at the new conditions.

For the third mode of operation the environmental conditions were kept constant and the normal power to the VTI was varied. The variation of the power occurred as a step function. It was noted while making the changes in power that it was difficult to maintain constant environmental conditions due to the added heating load to the cooling system. However, for all reasonable variations of power, almost constant conditions could be maintained, and the data obtained is quite valid.

EXPERIMENTAL SET-UP FOR FLUID LOOP VTI

A very simple test set up was utilized to examine the performance of a fluid loop VTI. It consisted principally of two cold plates which had been fabricated from copper plate with copper tubing soldered to the plate. These two cold plates were separated by means of an insulating material one inch in thickness. One of the cold plates had a heater blanket similar to that used

in the initial VTI test bonded to its surface. The second cold plate was mated to a third cold plate which was connected to the coolant loop. Thus a heat source and heat sink was provided for the system.

The fluid loop between the first and second cold plate was provided by utilizing a small twelve watt pump in the system. The principal objective of these tests were to determine the extreme values of thermal resistance available under such conditions. A number of tests in intermediary points were also conducted in order to determine a smooth curve between the extreme values obtained.

In all cases experimentation of this system was confined to a steady state analysis at each individual point.

EXPERIMENTAL PROCEDURE FOR THE FLUID LOOP VTI

A thermal load was applied to the system with a fixed rate of fluid flow and the system was allowed to come to equilibrium. For each case the thermal dissipation of the heater blanket, the flow rate of coolant, the temperature coolant of the coolant and the flow rate of the master loop itself were constant. Flow control through the master loop was obtained by utilizing a by-pass loop, although in an actual application a variable speed or variable flow pump could be utilized. The flow rate through the master loop was measured by using a small rotometer flow meter.

The system was housed in an insulated enclosure, very similar to that used for the roller VTI testing.

EXPERIMENTAL RESULTS

Results for the Roller VTI, Steady State--The results for the steady state experiments on the Roller VTI can best be represented by tabulating the thermal resistance of a system both as a function of the nominal temperature of the source surface and as a function of the power dissipated.

The information shown in Figure 6, which plots the thermal resistance of the system as the function of a source temperature, shows that although there is some dependence on orientation of the performance at source temperatures which are away from the control point, the performance of all the orientations when the operation is within the control limits of the system is quite similar. In fact, it should be said that there is virtually no variation in performance of any of the orientations under this mode. An examination of Figure 7 which plots the thermal resistance of the system as a function power dissipation also leads to the same conclusion.

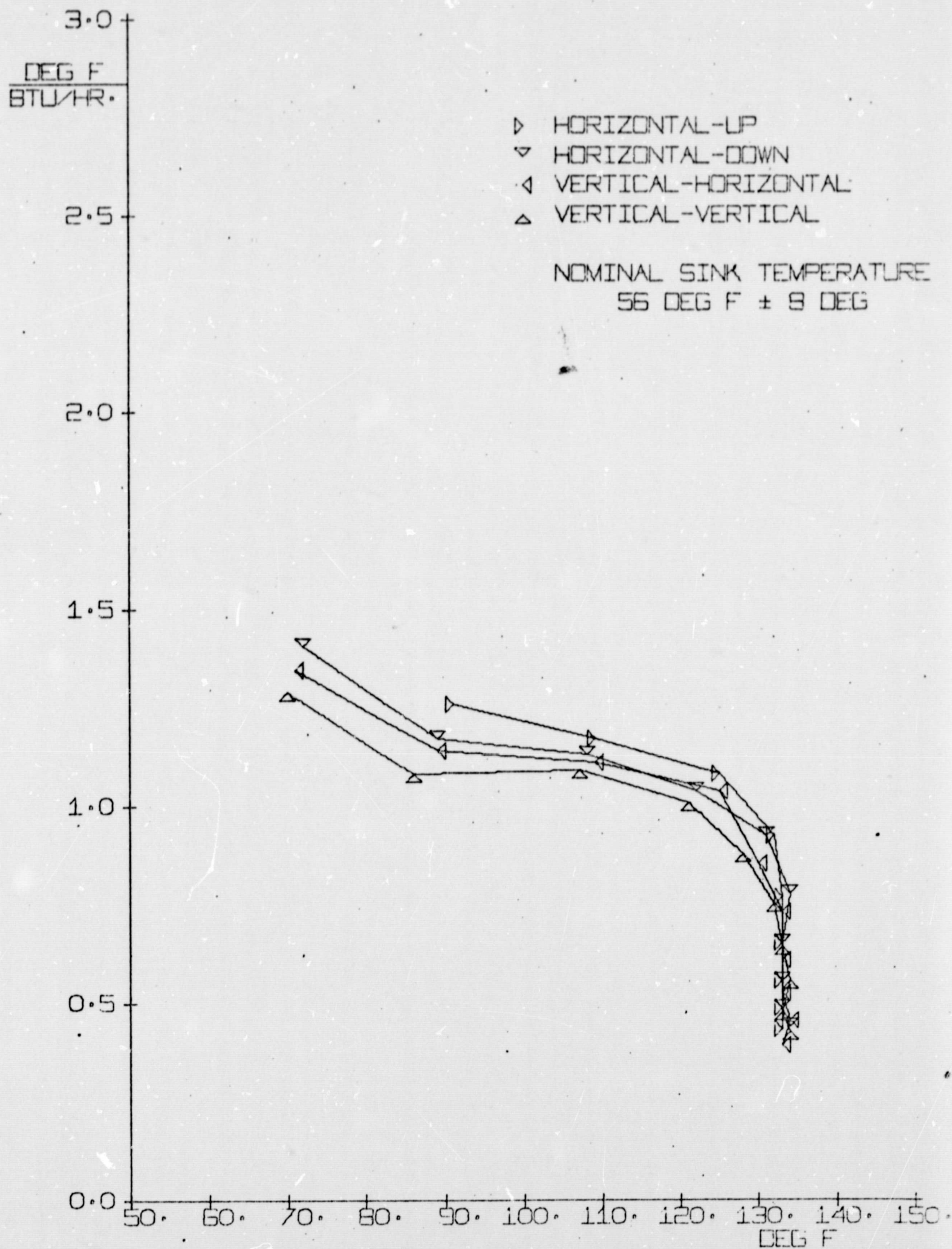
The steady state data shown in Figures 6 and 7 indicate that for an ambient temperature of approximately 60°F the rollers begin to provide control at a power level of just under 25 watts. The range of control of the source surface temperature is approximately 131 to 135°F.

A comparison of the maximum and minimum thermal resistances achieved in this experiment compare very favorably with the theoretical range predicted in Appendix C. The experimentally maximum conductance was approximately 1.5 Btu/hr. per °F and a minimum value was observed at .5 Btu/hr per °F. This compares well with the range of .69 to 1.9 predicted analytically.

Results for the Roller VTI, Transient Case--A presentation of the data for the dynamic testing is slightly more difficult to summarize. For this reason, the data, in a somewhat condensed form, has been presented in Tables 1 through 2 and plotted in Figures 8 through 11.

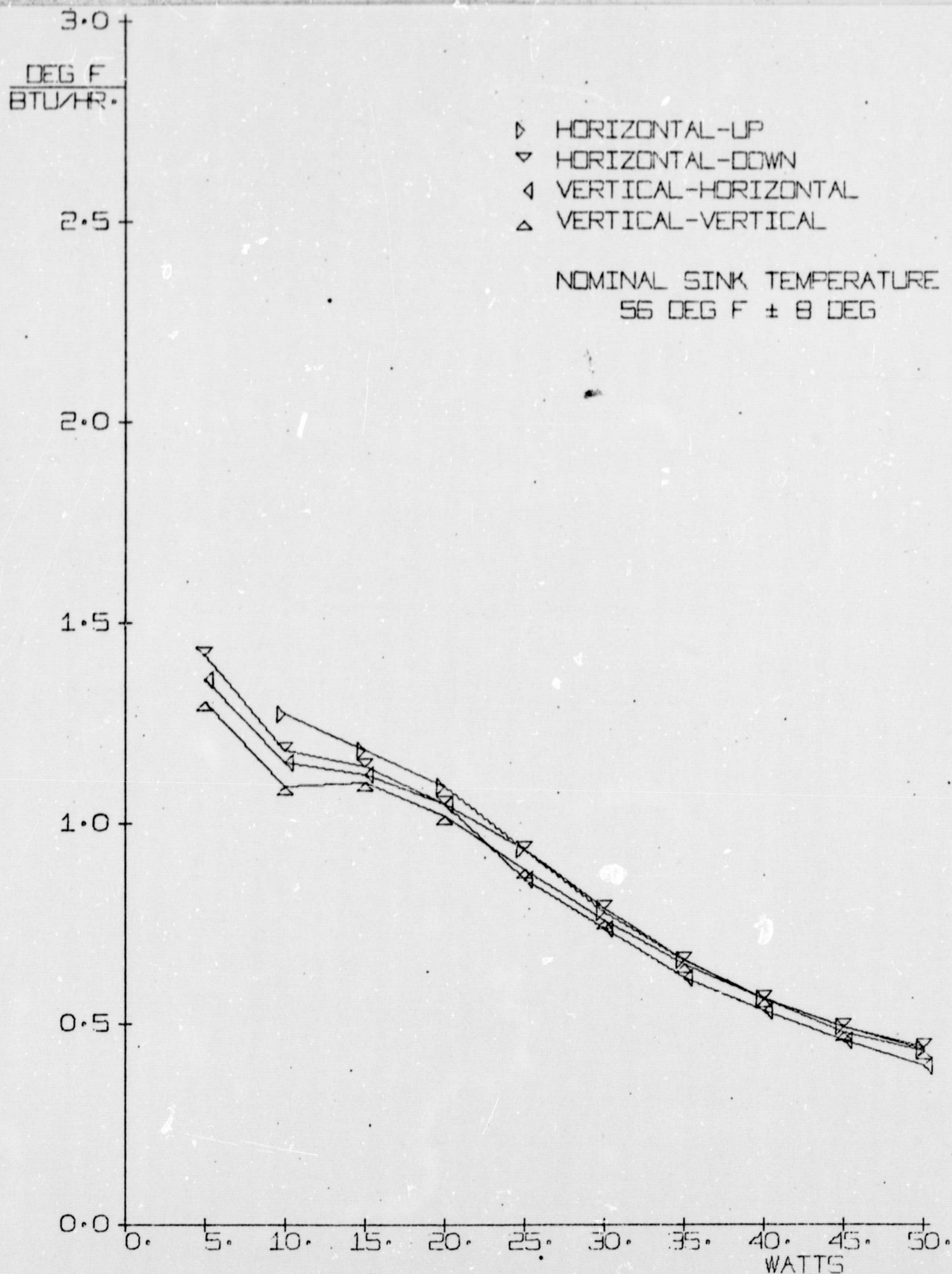
The results of the dynamic testing can best be described by discussing the performance of the system during some of the representative cases. The data as shown is tabulated in terms of the events which define the variation of the parameter values as a function of time.

For each individual event the initial temperature of the thermal sink and the final temperatures of the sink is given, the initial temperature of the control surface and the final temperature of the control surface is given, and the power dissipation is given. The elapsed time is also given. In most cases the time representative of the temperature change of the ambient and of the control surface are identical. However, in a few cases where



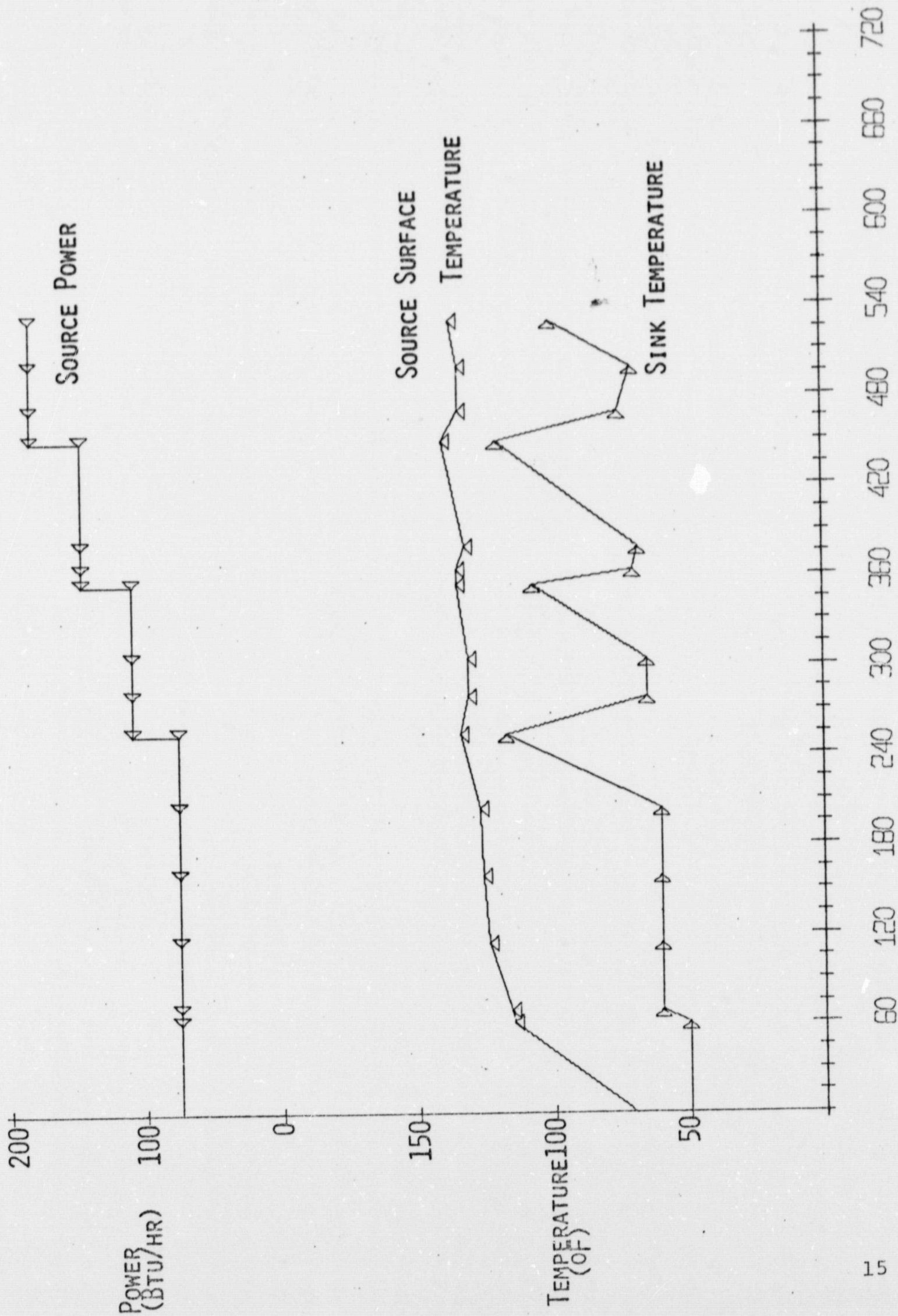
THERMAL IMPEDANCE AS A FUNCTION
OF SOURCE TEMPERATURE

FIGURE 6



THERMAL IMPEDANCE AS A FUNCTION
OF POWER DISSIPATION

FIGURE 7



ELAPSED TIME (MINUTES)

FIGURE 8 TIME HISTORY OF TRANSIENT SEQUENCE ONE

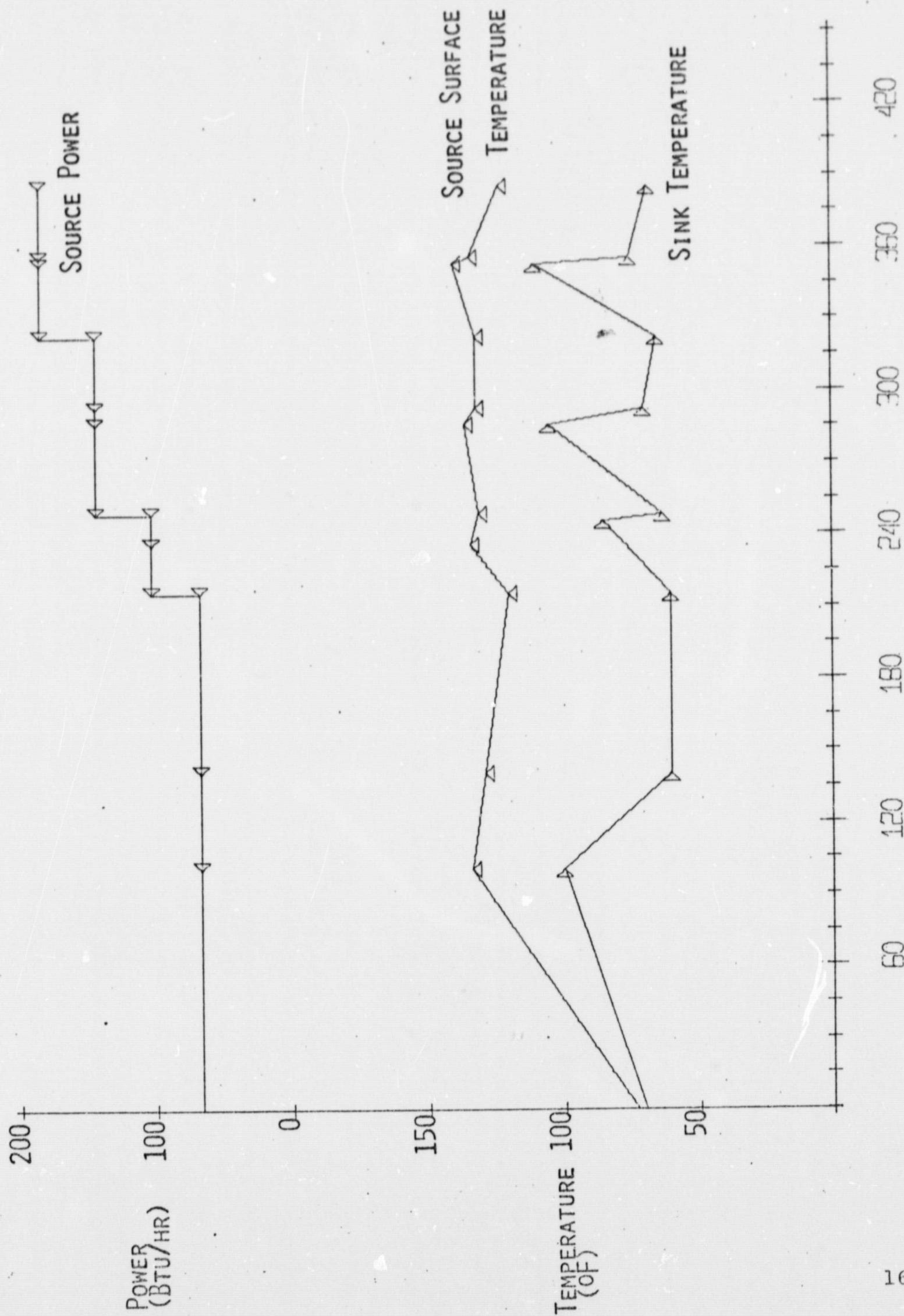


FIGURE 9 TIME HISTORY OF TRANSIENT SEQUENCE TWO

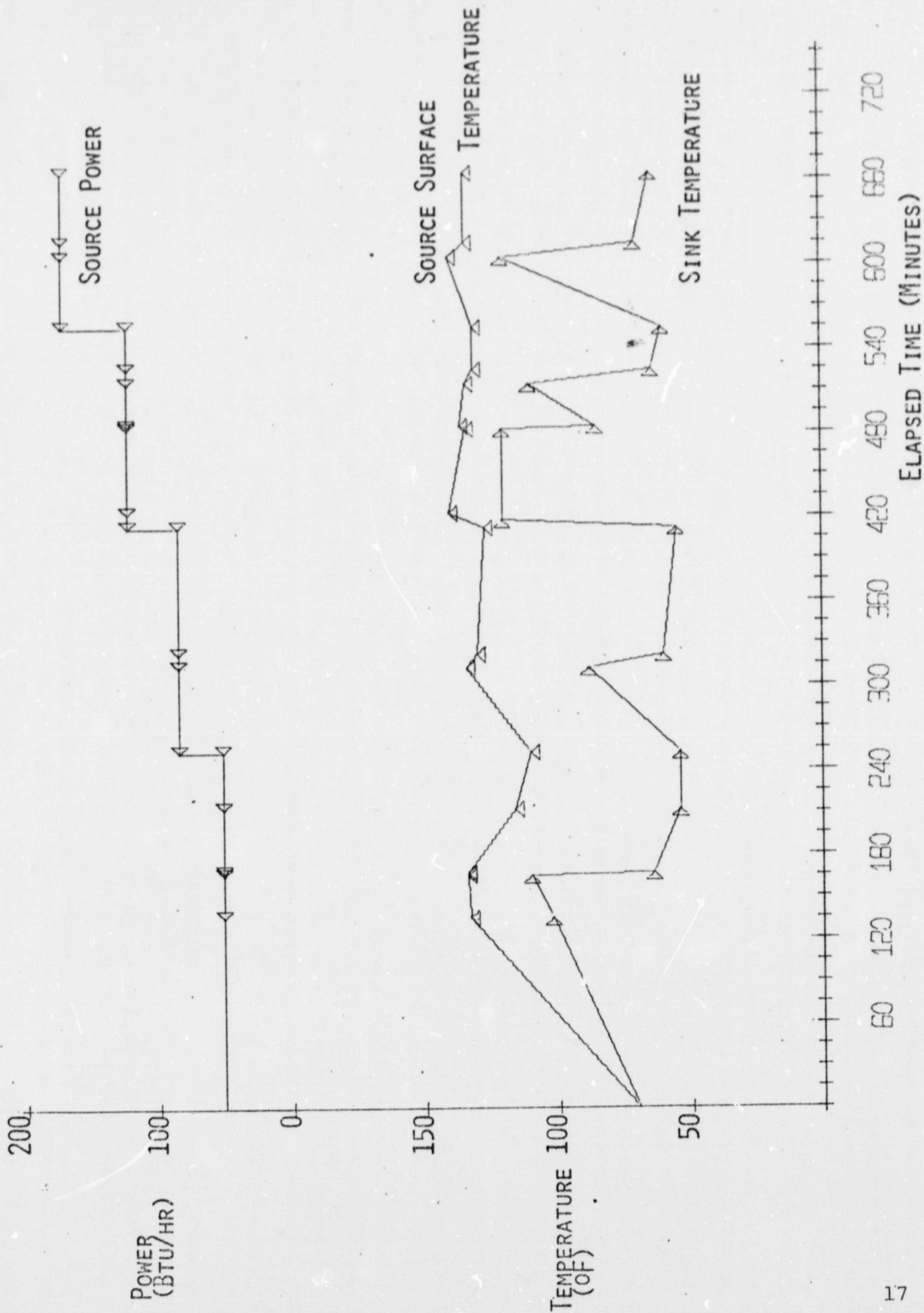


FIGURE 10 TIME HISTORY OF TRANSIENT SEQUENCE THREE

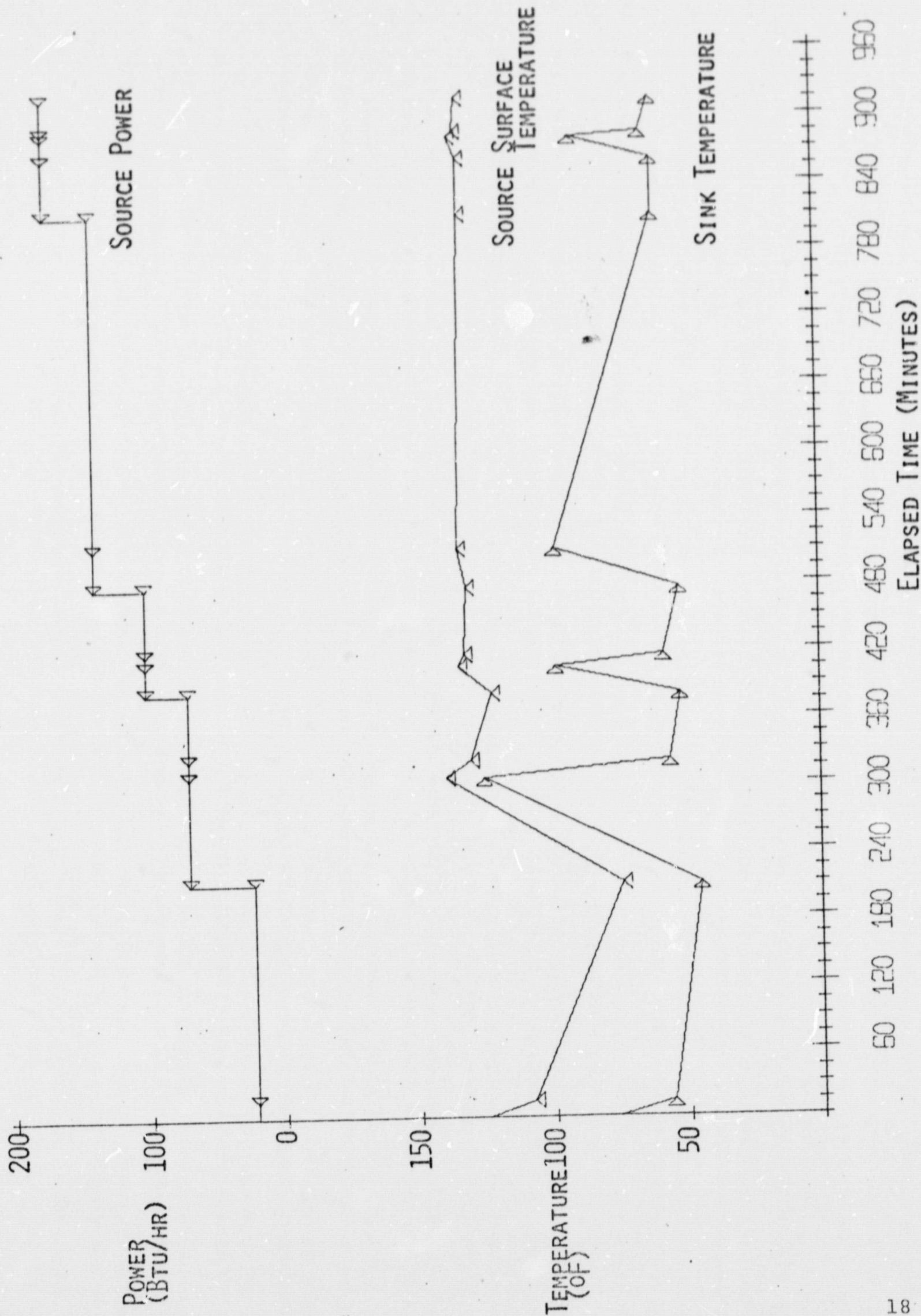


FIGURE 11 TIME HISTORY OF TRANSIENT SEQUENCE FOUR

TABLE 1 TRANSIENT SEQUENCE DATA

SEQUENCE ONE

EVENT	INITIAL AMB TEMP	FINAL AMB TEMP	ELAPSED TIME AMB	INITIAL SURF TEMP	FINAL SURF TEMP	ELAPSED TIME SURF	POWER BTU/HR	AMBIENT DT/D TIME	SURFACE DT/D TIME
1	50.00	50.00	60.00	70.00	115.00	60.00	75.36	0.000	0.750
2	50.00	60.00	8.00	115.00	116.00	8.00	75.36	1.250	0.125
3	60.00	60.00	45.00	116.00	124.00	45.00	75.36	0.000	0.177
4	60.00	60.00	45.00	124.00	126.00	45.00	75.36	0.000	0.044
5	60.00	60.00	45.00	126.00	127.00	45.00	75.36	0.000	0.022
6	60.00	117.00	50.00	127.00	134.00	50.00	75.36	1.140	0.140
7	117.00	65.00	25.00	134.00	131.00	25.00	108.09	-2.080	-0.120
8	65.00	65.00	25.00	131.00	131.00	25.00	108.09	0.000	0.000
9	62.00	107.00	50.00	131.00	135.00	50.00	108.09	0.900	0.080
10	107.00	70.00	10.00	135.00	135.00	10.00	144.92	-3.700	0.000
11	68.00	68.00	15.00	132.00	132.00	15.00	144.92	0.000	0.000
12	65.00	120.00	70.00	132.00	140.00	70.00	144.92	0.771	0.114
13	120.00	75.00	20.00	146.00	134.00	20.00	182.43	-2.250	-0.600
14	70.00	70.00	30.00	134.00	134.00	30.00	182.43	0.000	0.000
15	70.00	100.00	30.00	133.00	137.00	30.00	182.43	1.000	0.133

SEQUENCE TWO

16	70.00	100.00	100.00	73.00	134.00	100.00	67.38	0.300	-0.610
17	100.00	60.00	40.00	134.00	129.00	40.00	67.38	-1.000	-0.125
18	60.00	60.00	75.00	129.00	120.00	75.00	67.38	0.000	-0.120
19	60.00	85.00	30.00	120.00	134.00	20.00	101.92	0.833	0.700
20	85.00	63.00	3.00	133.00	131.00	13.00	101.92	-7.333	-0.153
21	60.00	105.00	37.00	131.00	136.00	37.00	142.53	1.216	0.135
22	106.00	70.00	7.00	136.00	132.00	7.00	142.53	-5.142	-0.571
23	70.00	65.00	30.00	132.00	132.00	30.00	142.53	-0.166	0.000
24	65.00	110.00	30.00	131.00	140.00	30.00	183.28	1.500	0.300
25	110.00	75.00	3.00	140.00	134.00	3.00	183.28	-11.666	-2.000
26	75.00	68.00	30.00	134.00	123.00	30.00	183.28	-0.233	-0.366

TABLE 2 TRANSIENT SEQUENCE DATA

SEQUENCE THREE

EVENT	INITIAL AMB TEMP	FINAL AMB TEMP	ELAPSED TIME AMB	INITIAL SURF TEMP	FINAL SURF TEMP	ELAPSED TIME SURF	POWER BTU/HR	AMBIENT DT/D TIME	SURFACE DT/D TIME
27	70.00	102.00	135.00	70.00	133.00	135.00	52.37	0.237	0.466
28	102.00	110.00	30.00	133.00	134.00	30.00	52.37	0.266	0.033
29	110.00	64.00	2.00	134.00	134.00	2.00	52.37	-23.000	0.000
30	64.00	54.00	45.00	132.00	116.00	45.00	52.37	-0.222	-0.355
31	54.00	54.00	40.00	116.00	110.00	40.00	52.37	0.000	-0.150
32	54.00	88.00	60.00	112.00	134.00	60.00	84.22	0.566	0.366
33	90.00	60.00	10.00	134.00	130.00	10.00	84.22	-3.000	-0.400
34	60.00	55.00	90.00	130.00	127.00	90.00	84.22	-0.055	-0.033
35	120.00	120.00	5.00	134.00	140.00	10.00	122.21	0.000	0.600
36	60.00	120.00	65.00	128.00	134.00	60.00	122.21	0.923	0.100
37	120.00	85.00	2.00	140.00	136.00	2.00	122.21	-17.500	-2.000
38	85.00	110.00	30.00	136.00	134.00	30.00	122.21	0.833	-0.066
39	112.00	64.00	10.00	136.00	131.00	10.00	122.21	-4.800	-0.500
40	64.00	60.00	30.00	131.00	131.00	30.00	122.21	-0.133	0.000
41	60.00	120.00	50.00	132.00	140.00	50.00	170.56	1.200	0.160
42	120.00	70.00	10.00	144.00	134.00	10.00	170.56	-5.000	-1.000
43	70.00	64.00	50.00	134.00	134.00	50.00	170.56	-0.120	0.000

SEQUENCE FOUR

44	76.00	56.00	15.00	125.00	108.00	15.00	22.19	-1.333	-1.133
45	56.00	45.00	195.00	108.00	74.00	195.00	22.19	-0.056	-0.174
46	45.00	125.00	95.00	76.00	139.00	95.00	69.70	0.842	0.663
47	125.00	56.00	15.00	139.00	130.00	15.00	69.70	-4.600	-0.600
48	56.00	52.00	60.00	129.00	122.00	60.00	69.70	-0.066	-0.116
49	52.00	98.00	25.00	120.00	134.00	25.00	101.24	1.840	0.560
50	98.00	58.00	10.00	134.00	132.00	10.00	101.24	-4.000	-0.200
51	58.00	52.00	60.00	132.00	131.00	60.00	101.24	-0.100	-0.016
52	52.00	98.00	35.00	131.00	134.00	35.00	137.76	1.314	0.085
53	100.00	60.00	300.00	131.00	132.00	300.00	137.76	-0.133	0.003
54	60.00	60.00	50.00	132.00	132.00	50.00	170.56	0.000	0.000
55	60.00	90.00	20.00	132.00	135.00	20.00	170.56	1.500	0.150
56	90.00	64.00	5.00	135.00	133.00	5.00	170.56	-5.200	-0.400
57	64.00	60.00	30.00	133.00	132.00	30.00	170.56	-0.133	-0.033

either the environment or the control surface achieved its temperature change in a significantly different time period than the other, two different times are indicated.

As noted in the steady state tests the system did not begin to control until a surface temperature range of 130°F to 135°F was achieved. For ambient temperatures in the range of 40 to 60°F this usually corresponded to load of approximately 25 watts, or equivalently 85 Btu/hr.

The data tabulated represents information from four representative test sequences. For each sequence a number of common effects can be noted. Initially, the system was generally started from a relatively cooled condition (50°F to 76°F). This was followed by a warm up period in which the temperature of the control surface would be raised by the load being dissipated. In the cases where the ambient was kept relatively cool (i.e. approximately 50-60°F) power levels below 25 watts were insufficient to bring the system within the control range. This then had to be achieved by either raising the power above the 25 watt range or allowing the simulated ambient temperature to rise.

The next series of events of the test sequence indicated that system was capable of maintaining control for a wide range of power dissipation and changes in environmental temperatures. For the rate of temperature changes available (up to 10-20°F/min.) the system showed a remarkable ability to maintain control. In fact, as long as the parameter values stayed within the limits determinable for the steady state evaluation as being required for control, the system was able to maintain the control surface within the range of 130°F to 136°F.

The first representative sequence shown in Figure 8 will be examined in detail. This sequence consists of events from number 1 through number 15. The initial temperature for the environment was 50°F and as can be seen from the first five events, was initially maintained in the range of 50 to 60°F by means of the cooling system. The initial temperature of the control surface was 70°F and an initial power of 22.1 watts was applied to the VTI. The control surface temperature rose from 70°F to 115°F in approximately one hour. It then rose at slightly different rates to approximately 127°F after a total elapsed time of about three and a half hours. The general overall shape of the curve would be well described as being exponential. Throughout this time the power was maintained at 22.1 watts.

The steady state temperature at this load eventually rose to approximately 127°F and leveled off there. Thus, with an ambient temperature of 60°F and only 22.1 watts being dissipated there was insufficient power to drive the system into the control range. Therefore, at event number 6 the cooling system was turned off and the ambient temperature allowed to rise to 117°F. As

Thermocouple Number

Location

1	Expansion Bellows
2	Cold Plate, First Quadrant
3	Cold Plate, Second Quadrant
4	Cold Plate, Third Quadrant
5	Cold Plate, Fourth Quadrant
6	Control Surface, Fourth Quadrant
7	Control Surface, First Quadrant
8	Control Surface, Second Quadrant
9	Control Surface, Third Quadrant
10	Wax Sensor, Top
11	Sensor Bellows
12	Room Temperature

Projection of Thermocouple Locations

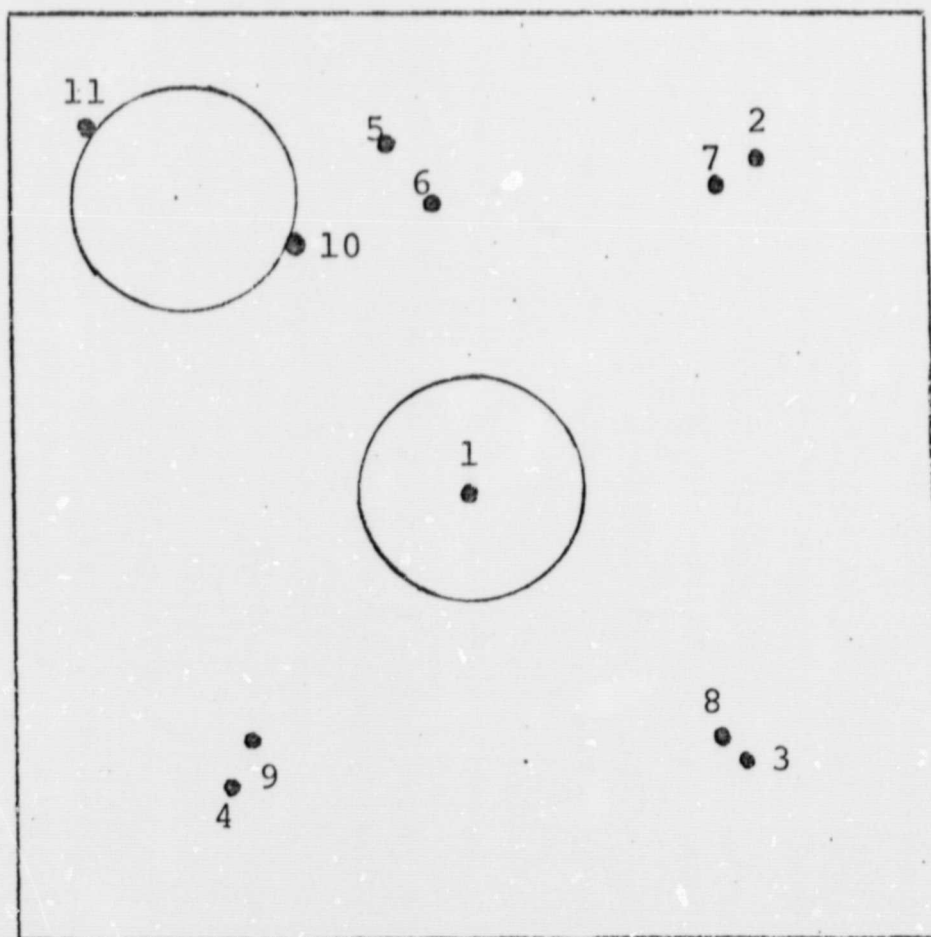


Figure 12. Thermocouple Locations on Roller VTI

shown in event number 6 in this process with the power level maintained at 22.1 watts, the control surface rose to 134°F; i.e. into the control range. It should be noted that the temperature crossed the 134°F just as the ambient was crossing 100°F and maintained that temperature up to an ambient temperature of 117°F. At this point (event number 7) the cooling was turned on again and the representative ambient allowed to drop to 65°F in a period of only 25 minutes. The VTI was able to maintain control even though the power level was simultaneously raised from 22.1 watts to 31.7 watts.

After allowing a period (event number 8) in which the ambient was maintained at 65° and the VTI allowed to come to equilibrium at 131°F, the cooling system was again turned off and the ambient allowed to rise to 107°F. Once again the VTI maintained control of the system and held it to the range of 131 to 135°F. The cooling system was turned on (event number 10) and the power simultaneously raised. In this case the VTI held a constant 135°F temperature for a period of ten minutes and then went into a gradual drift to a temperature of about 132°F for a period of 15 minutes (event number 11). The cooling power was then turned off with the power held at the 42.5 watt level and the ambient temperature allowed to rise to 120°F. In this case the combination of ambient and power loading was sufficiently high to force the rollers out of the control range as noted by the final control surface temperature of 140°F. In fact, for this condition the temperature eventually rose to 146°F. Event number 13 was represented by a turning on of the cooling system and a simultaneous raising of the power level to a 53.5 watt level. The reduction of ambient temperature to about 70°F (event number 14) was sufficiently low to allow this system to return to the control range. The temperature was then allowed to rise to 100°F. At this point the system just crossed out of control as noted by the 137°F final temperature of the control system. In each of the cases above the rate of temperature change is shown on the data sheet. These rates of changes were in the cooling mode, as high as 3.7°F per minute, and in the heating mode, rates of rise of up to 1-1/4° per minute.

The characteristics of the other three sequences are very similar to the first one as described above. In each case the system was started from a relatively cool temperature until the control mode was achieved. A combination of changing the ambient temperature and the heating load was used to exercise the system. As in the first sequence, it was noted that rates of change of these parameters were not significant in affecting the capability of the VTI to control temperature. In all cases the system performance was principally determined by the level of the power dissipation or the ambient temperature.

It should be noted that in two cases during the dynamic testing it was noted that the system tended to overshoot approximately one half to one degree when the system was within the control mode of operation. In both these cases the time involved

in this jump was in the order of three minutes, and it appeared that this might be due to a momentary sticking of the rollers.

Results for the Fluid Loop VTI--Although the amount of data obtained with the fluid loop VTI was not very extensive it is apparent from the results that a considerably large ratio of maximum conductance to minimum conductance could be achievable with this system. With the configuration evaluated a maximum conductance value was 5.8 Btu/hr.^{°F} and the minimum conductance value was .5 Btu/hr.^{°F}. This ratio of 11.6 to 1 is considerably larger than that obtained with the roller VTI.

CONCLUSIONS

The roller VTI with a 150 centipose viscosity oil performed very well both in steady state conditions and in the transient operation. Under a steady state mode a good range of thermal conductances was achieved. This range varied from .5 to 2.0 Btu/hr.^{°F}, with a resulting conductance ratio of about 4 to 1. The dynamic test indicated that the roller VTI is capable of operating very well in the transient mode if the variations of either the environment or the thermal loading to the system are kept within limits that do not drive the components outside of its controllable range. As long as the extremes of parameter values at the initiation and completion of the transients of the cycle is such that the system can remain within the control mode the rollers will be able to vary the resistance within the required values sufficiently quickly to allow the system to remain in control.

The fluid loop VTI showed itself to be capable of providing a very large ratio of conductance values. One of the restrictions, however, in any application will certainly be the pumping power required. In this particular case 12 watts were used. If the resulting conductance ratio is such that savings of heater power considerably above 12 watts can be achieved, then the fluid loop would provide a major improvement to the system design.

APPENDIX A

COMPARISON OF FIXED THERMAL RESISTANCE VERSUS A VARIABLE THERMAL RESISTANCE

In order to minimize the required thermal control power of a system which can encounter extremely different external temperatures it is advantageous to be able to vary the net thermal resistance of the system between the critical components and ambient. The magnitude of the power reduction can easily be determined.

Let

$$Q \text{ (total)} = Q \text{ (Sensors)} + Q \text{ (Electronics on Block)} \\ + Q \text{ (Control)}$$

or

$$Q_T = Q_O + Q_C$$

From the definition of the thermal resistance R ,

$$Q_T = \frac{1}{2} (T_{\text{sensor}} - T_{\text{ambient}}) \\ = \frac{1}{2} \Delta T$$

If R is fixed at R_F the minimum control power can be determined.

$$Q_{C,\min} + Q_O = \frac{\Delta T(\min)}{R_F}$$

where R_F is usually chosen so that $Q_{C,\min}$ is approximately zero.

At $\Delta T(\max)$

$$Q_O + Q_{C,\min} + Q_C' = \frac{\Delta T(\max)}{R_F}$$

$$Q_C' = \frac{\Delta T(\max)}{R_F} - (Q_O + Q_{C,\min})$$

or

$$Q_c' = \frac{\Delta T(\max) - \Delta T(\min)}{R_F}$$

and thus, for a fixed thermal resistance

$$\frac{Q_c'}{Q_o + Q_{c,\min}} = \frac{\Delta T(\max) - \Delta T(\min)}{\Delta T(\min)}$$

For the total possible range of external temperatures of 30°F to 120°F, and for sensor temperatures of 150°F

$$\frac{Q_c'}{Q_c + Q_{c,\min}} = \frac{(150-30) - (150-120)}{(150-120)} = 3.0$$

If however, R can be varied as ΔT varies, then $\frac{\Delta T}{R}$ can be held constant and Q_c can always be held essentially equal to $Q_{c,\min}$, with a net power savings equal to Q_c' calculated above.

Appendix B

ANALYSIS OF THE THERMAL IMPEDANCE OF THE ROLLER VTI

The model for the thermal paths of the roller VTI is shown in Figure B-1.

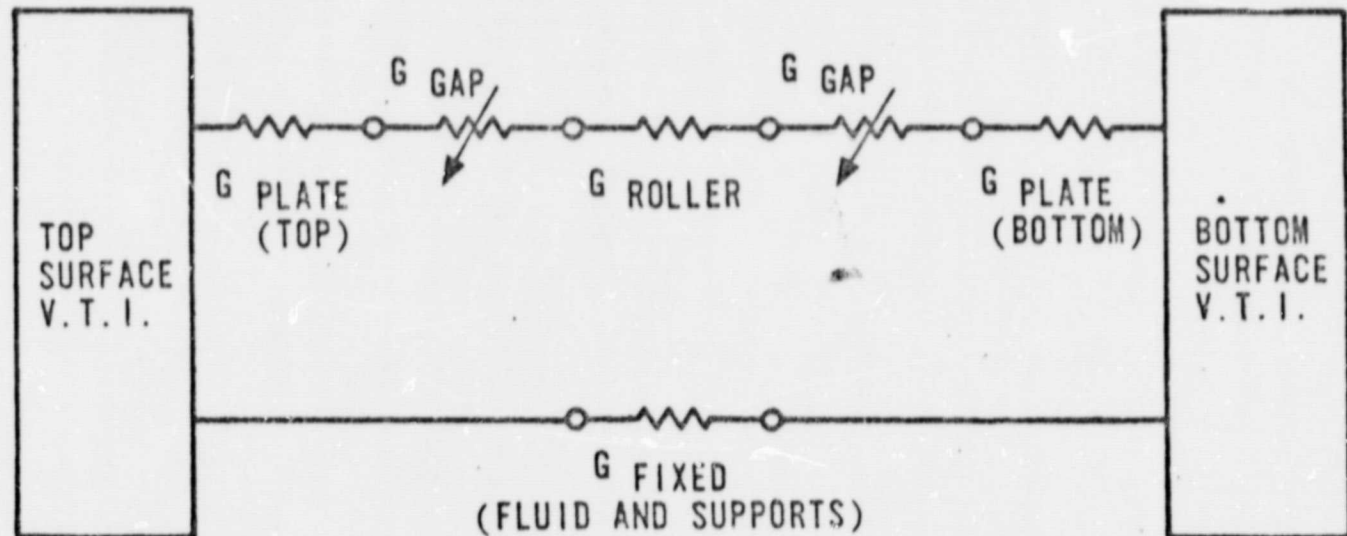


Figure B-1

For: $L_{\text{Roller}} = 25 \text{ in.}$

Diameter = 1.125 in.

Gap = 0.010 in.

$K_{\text{oil}} = 0.0755 \text{ Btu/hr-ft-}^{\circ}\text{F}$

These thermal resistances can be calculated as follows:

$$G_{\text{Plates}} = \frac{KA}{L} = \frac{(100)(6.9 \times 6.9)}{(12)(0.125)} = 3170 \text{ Btu/hr-}^{\circ}\text{F}$$

$$G_{\text{gap,conductive}} = \frac{KA}{L} = \frac{(.0755)(3.14)(25)(1.125/4.)}{(12)(2)(.010)} = 6.9 \text{ Btu/hr-}^{\circ}\text{F}$$

$$G_{\text{gap,nonconductive}} = \frac{KA}{L} = \frac{(.0755)(3.14)(25)(1.125/4.)}{(12)(2)(16)(.010)} = 0.435 \text{ Btu/hr-}^{\circ}\text{F}$$

$$\begin{aligned} G_{\text{Roller}} &= 0.584 \text{ KL} \\ &= (.584)(100)(25)/12 \\ &= 121.5 \text{ Btu/hr-}^{\circ}\text{F} \end{aligned}$$

$$\begin{aligned} G_{\text{Fluid}} &= \frac{KA}{L} \\ &= \frac{(.0755)(14.8)}{(12)(1.125)} \\ &= 0.0827 \text{ Btu/hr-}^{\circ}\text{F} \end{aligned}$$

$$\begin{aligned}
 G_{\text{Supports}} &= \frac{KA}{L} \\
 &= \frac{(.25)(27.6)(.25)}{(12)(1.125)} \\
 &= 0.128 \text{ Btu/hr-}^{\circ}\text{F}
 \end{aligned}$$

Thus the total conductances are

$$G_{\text{Fixed}} = 0.128 + .083 = .211 \text{ Btu/hr-}^{\circ}\text{F}$$

$$G_{\text{Variable Path}} = G_{\text{gap}} + G_{\text{Rollers}}$$

But

$$G_{\text{gap}} = 0.435 \text{ to } 6.9 \text{ Btu/hr-}^{\circ}\text{F}$$

$$G_{\text{Rollers}} = 121.5 \text{ Btu/hr-}^{\circ}\text{F}$$

Therefore

$$G_{\text{Variable Path}} = \frac{(.435)(121.5)}{121.93} \text{ to } \frac{(6.9)(121.5)}{128.4}$$

and

$$\begin{aligned}
 G_{\text{Tot}} &= (.434 \text{ to } 6.3) + .211 \\
 &= 0.645 \text{ to } 6.5 \text{ Btu/hr-}^{\circ}\text{F}
 \end{aligned}$$

or

$$R_{\text{Tot}} = 1.56 \text{ to } .152^{\circ}\text{F/Btu/hr.}$$

Appendix C

ANALYSIS OF THE THERMAL IMPEDANCE OF A FLUID LOOP VTI

Theoretically, if the parasitic losses in the system can be reduced to an absolute minimum the maximum resistance obtained with a fluid loop VTI is virtually infinite. In real practice, however, the requirement for structural integrity involves members with non-infinite resistance and therefore the actual value obtained is somewhat lower than this. In fact, in the system investigated experimentally the maximum resistance that could be achieved would be equivalent to the resistance of 1" insulation.

The maximum conductance available through the system depends on the contact area within the fluid packages of each of the cold plates, the velocity of the fluid being passed through the loop, the properties of the fluid being utilized in the main loop, and the contact resistance between the cold plate at the source and sink.

The net thermal impedance for a fluid loop VTI can best be analyzed in terms of the total temperature rise from the source surface to the sink surface at a nominal heat load of 1 Btu/hr. The individual components of the overall temperature rise can be broken down as follows:

a) Non-conductive mode

ΔT across top plate ($\Delta T_{T.P.}$)

ΔT through insulating space (ΔT_1)

ΔT across bottom plate ($\Delta T_{B.P.}$)

$$\begin{aligned}\Delta T_{T.P.} &= \frac{QL}{KA} = \frac{(1)(.25)(12)}{(200)(36)} \\ &= 0.00042^{\circ}\text{F}\end{aligned}$$

$$\begin{aligned}\Delta T_1 &= \frac{QL}{KA} = \frac{(1)(1)(12)}{(.02)(36)} \\ &= 16^{\circ}\text{F}\end{aligned}$$

$$\Delta T_{B.P.} = 0.00042^{\circ}\text{F}$$

Thus, the non-conductive thermal impedance is theoretically 16°F/Btu/hr.

b) Conductive mode

Tubing L = 24 in.
Tubing d = 1/4 in.

ΔT top plate to H_2O ($\Delta T_{T,P}$)

ΔT rise of H_2O (ΔT_{H_2O})

ΔT bottom plate to H_2O ($\Delta T_{B,P}$)

$$\Delta T_{T,P} = Q/hA = \frac{(1)(144)}{(100)(24)(3.14)(.25)} \\ = .12^{\circ}F$$

$$\Delta T_{H_2O} = \frac{Q}{\dot{m}C_p} = \frac{(1)}{(120)(1)} \\ = 0.008$$

$$\Delta T_{B,P} = 0.12^{\circ}F$$

Thus, the conductive thermal resistance is $0.248^{\circ}F/Btu/hr$.

The theoretical ratio is therefore approximately 64 to 1. In practice it would be reasonable to expect a value of approximately one quarter of this ratio or about 16 to 1.