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CONSTRUCTION AND TESTING OF A GAS-LOADED,

PASSIVE-CONTROL, VARIABLE-CONDUCTANCE HEAT PIPE

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CONSTRUCTION AND TESTING OF A GAS-LOADED, PASSIVE-CONTROL, VARIABLE-CONDUCTANCE HEAT PIPE

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ABSTRACT

A methanol heat pipe using nitrogen gas for temperature control has been constructed and tested. The system was run over a power ratio of 15 (2 to 30 watts) with the heat source near ambient temperature and with the heat sink at a nominal value of 32°F. Control was obtained with a metal bellows gas reservoir which was actuated by an internal liquid-filled bellows. The liquid bellows was pressurized by expanding liquid methanol which was contained in an auxiliary reservoir in the evaporator heater block. It was demonstrated that the temperature variation of the heat source was reduced from 36°F for the heat pipe with no control to 7°F with the actuated bellows control.

INTRODUCTION

The heat pipe as conceived by Grover [1]* and Trefethan [2] and developed by many other researchers has demonstrated its ability as a high thermal conductance device which has many useful applications. However, in order to become a fully operational thermal control device, the conductance

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* Numbers in brackets designate References at the end of the report.

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must be controllable. For if a heat pipe is closely coupled to a low temperature heat sink, and the maintenance of the temperature of the source depends only on the fixed impedance external to the pipe, the heat source (which in many applications may be electronic controls) can approach inoperable temperatures under low load conditions. It is desirable and necessary, in most cases, to design a heat transfer device which has a high thermal conductance at high power but one which reduces its conductance at low power dissipation levels. It is also desirable to require no electrical power to actuate the device to minimize the power drain and to increase the reliability.

Several control techniques have been advanced. Since the heat pipe operation depends on transport of the high energy vapor to the condenser and return of the liquid to the evaporator, perhaps the most obvious control mechanisms utilize control of the vapor or liquid flow rate. These mechanisms require a pressure drop in the vapor or a dry out condition in the evaporator [3]. They may have limited flexibility and reliability due to the inherent heat pipe principles and rewetting and restart features. Other techniques are based on the variation of thermal impedance with active condenser area. This variation can be accomplished by condenser flooding with excess liquid. The conductivity of most liquids is low enough such that significant impedance changes can be realized by altering the thickness of the liquid layer.

Another mechanism for altering the active condenser area is to impede the diffusion of vapor with a noncondensible gas. Any gas in the pipe is carried to the condenser where it accumulates and presents a blockage effect to vapor. The extent of the blockage depends on the mass of gas, the gas temperature, and the heat pipe total pressure. Since the gas is contained in the "shut-off" portion of the condenser, its temperature is nearly the same as that of the heat sink. This gaseous portion is a mixture which is in equilibrium with the liquid contained in the wicked wall and which contains a mole fraction equal to the liquid saturation pressure at the sink temperature divided by the heat pipe total pressure. For a fixed sink temperature, the saturation pressure is constant, and the gas partial pressure is directly related to the heat pipe total pressure. As the total pressure is governed by the saturation pressure of the pure working fluid in the pure vapor, the gas partial pressure is governed by the pure vapor temperature. And this vapor temperature is related to the source temperature through the source temperature and heat flux. It is apparent through the above described mechanism that the volume of noncondensible gas is inversely related to the source temperature. Thus the conventional heat pipe has inherent control aspects. As the source temperature increases, the gas plug decreases in volume and more condenser area is exposed to allow a higher heat load. Inversely, as the heat load decreases and as the source temperature is decreased, the gas expands to increase condenser blockage and to maintain a higher source temperature than would otherwise exist.

The control of the conventional gas loaded heat pipe can be enhanced by the addition of a gas reservoir which is to be attached to the end of the condenser to hold excess gas [4,5,6,7,8,9,10,11]. For the same relative total pressure change, $\Delta p/p$, the same relative volume change, $\Delta v/v$, occurs. Thus if the gas volume is larger, the volume change is larger, and the active condenser length is more significantly altered. Also the reservoir allows the heat pipe to be vacated of gas such that the entire condenser can become active. The fixed volume reservoir control can be further enhanced with the use of an electric heater element which controls the temperature (and the density) of the gas resident in the reservoir. Active control of the heater temperature requires auxiliary power. It, therefore, requires an external power supply, and it is subject to power failure. These two drawbacks are circumvented by the use of a self-actuated variable volume reservoir.

The self-actuated variable reservoir volume system as conceived by Dynatherm Corporation [12] utilizes two extensible metal bellows. The large bellows serves as the gas reservoir, and it is piped directly to the condenser end of the heat ipipe. A smaller bellows is mounted concentric to the large bellows. The small bellows contains a liquid which is called the auxiliary fluid and which is connected to a liquid volume in thermal contact with the heat source. The gas bellows is actuated by the small bellows which is pressurized by the expanding liquid as the source and auxiliary reservoir increase in temperature. Extension of the gas bellows displaces a portion of the gas from the heat pipe which causes the vapor-gas interface to move towards the end of the heat pipe exposing more active condenser. The enlarged condenser area attenuates the heat source temperature increase. Desired design features of this system are: large thermal expansion, small compressibility; large gas bellows area, small liquid bellows area. The fluid properties are limited by available compatible materials and the bellows areas are limited by manufacture, choice of spring constant, materials compatibility, and small bellows maximum allowable pressure. The objective of this study was to construct and test a self-actuated, variable-reservoir-volume, controlled heat pipe as conceived in the report by Dynatherm Corporation [12] and as described above.

THE EXPERIMENTAL APPARATUS

The experimental program consisted of two phases. Phase 1 includes construction and testing of the basic heat pipe and gas reservoir control system. This system was modeled with the TRW-GASPIPE computer program, and tests were performed to assess the performance in various modes of operation. " primary modification to the system for Phase 2 cons: ted of the insertion of a hollow plug into the condenser volume to increase the sensitivity. Since the screen wick which was used during Phase 1 was damaged during this modification, a new wick of slightly different characteristics was installed for Phase 2. The modified system was also tested in various modes, and comparison of the results for the two programs was made.

The Heat Pipe

The "Feedbac: Control Heat Pipe Assembly" is shown schematically in Figure 1, and the details of the basic heat pipe are shown in Figure 2. Specifications for the heat pipe are listed below in Table 1. The wick

Table 1 HEAT PIPE SPECIFICATIONS

material	stainless steel
length	36 in.
outside diameter	0.500 in.
wall thickness	0.016 in.
end flanges	stainless steel
flange seals	buna-n "O" rings
fluid	methyl alcohol (laboratory purity 99.9%)
wick	2 layers of 200 mesh stainless steel screen with an artery formed by 0.018 in. dia. wire as shown

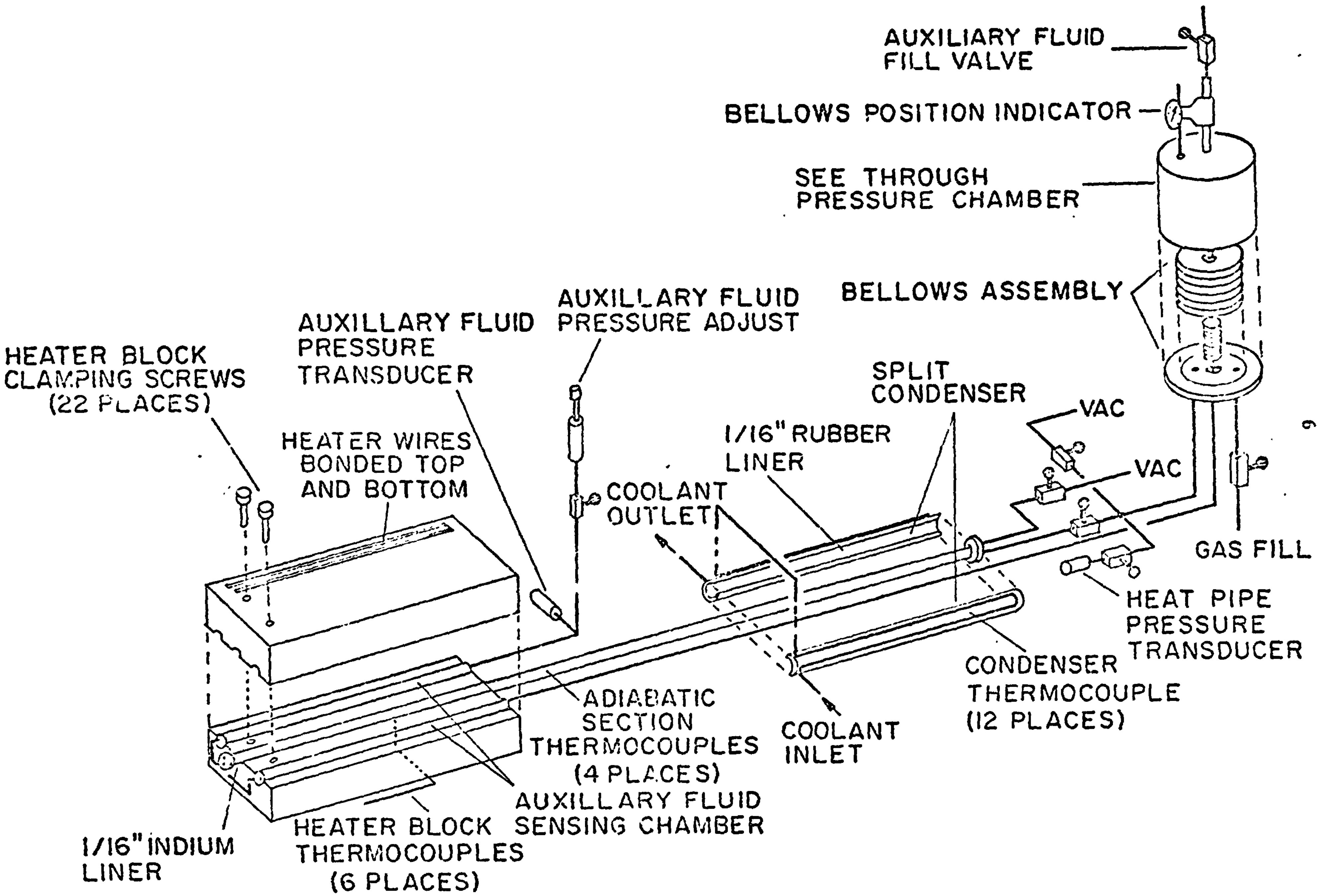


Figure 1. Feedback Control Heat Pipe Assembly

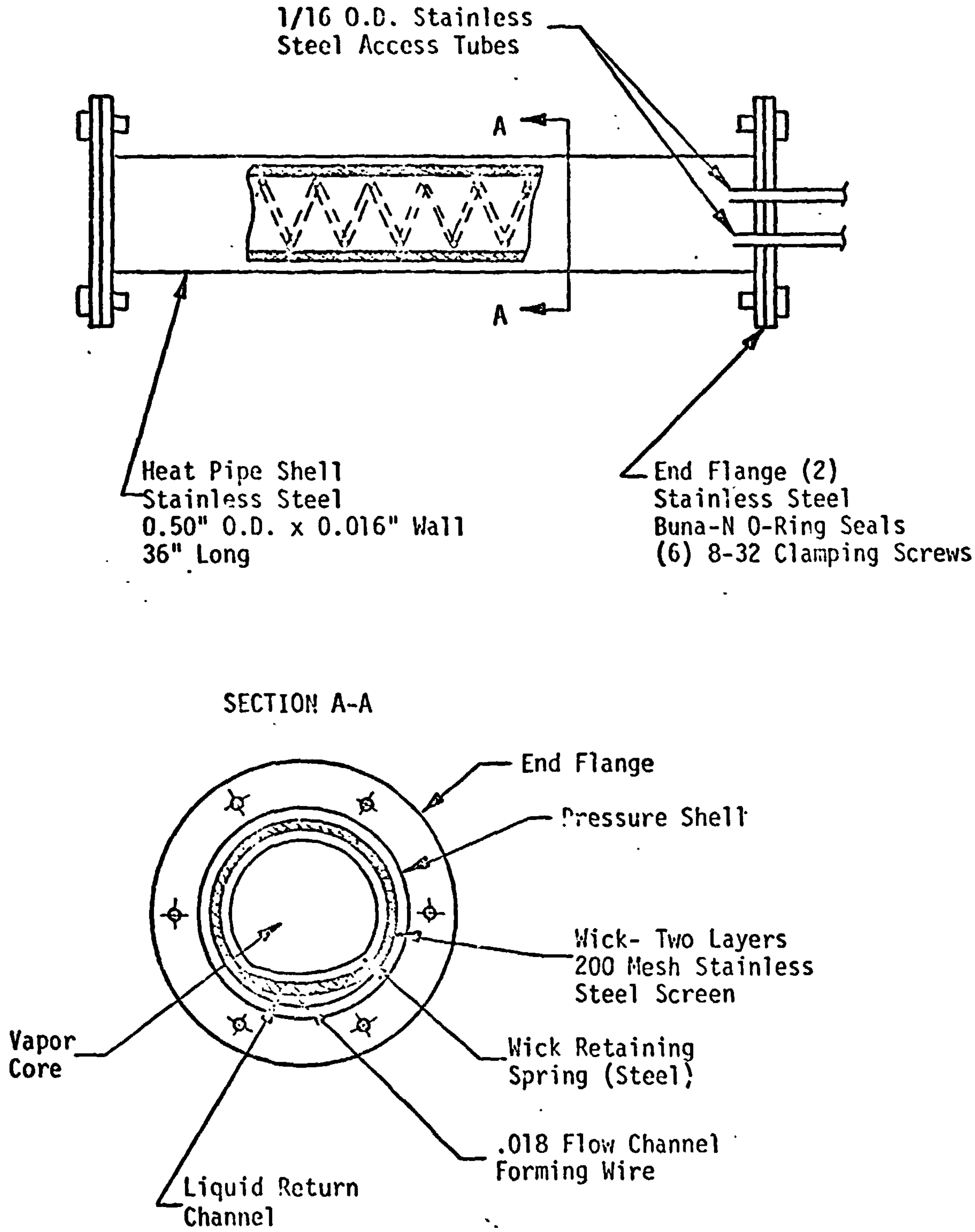


FIGURE 2 Basic Heat Pipe Design Features

consists of 2 layers of 200 mesh stainless steel screen which is held against the wall by an expansion spring having 4 coils per inch of 0.02 in. wire. A longitudinal wire between the wick and the wall forms a liquid return artery. The heat pipe was made with removable end flanges to facilitate modification, and the "O" ring seals were found to be trouble-free and adequate in all respects. The pipe was divided into thirds for evaporator, condenser, and adiabatic sections of equal size.

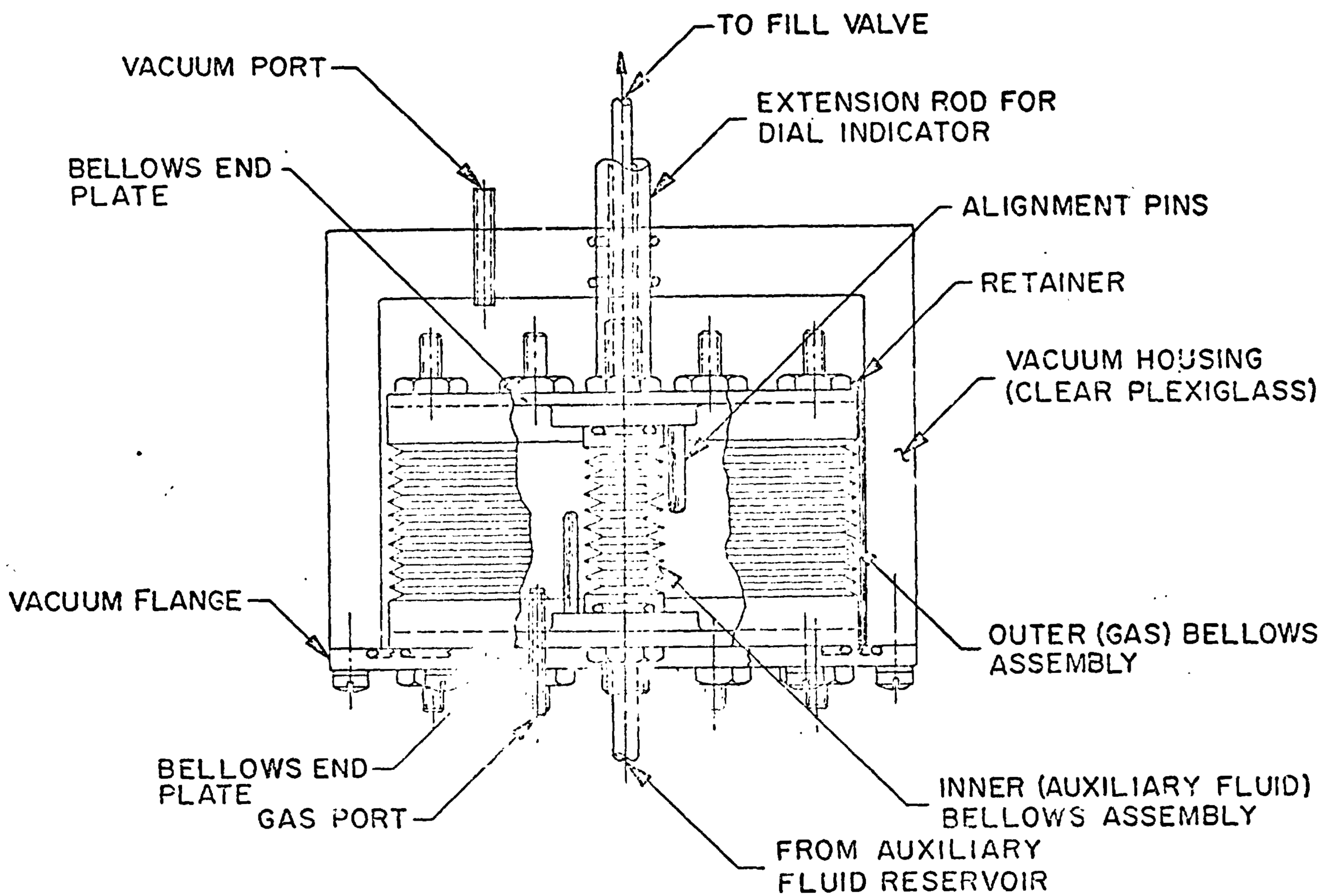
The aluminum heater block has overall dimensions of 3 x 1 1/2 x 12 inches. It is split into two halves and machined to receive the heat pipe and 2 auxiliary fluid reservoirs. When assembled it is fastened together with 22 bolts. The heaters were fabricated by laying nichrome wire in shallow channels with epoxy in the top and bottom faces of the block. Each of the 2 heaters has a resistance of about 50 ohms. The channel for the heat pipe was drilled oversize such that a 1/16 in. layer of indium metal could be installed. The indium metal was made slightly thicker than the gap such that it was squeezed upon assembly. Six thermocouples were installed at regular intervals between the indium metal and the heat pipe, and they were pressed into the indium in good contact with the wall.

The condenser section cooler was fabricated from 0.625 in. inside diameter x 0.125 in. thick wall brass tube with copper tubes for the coolant brazed with a full fillet to the outside. The split brass tube was oversize to accommodate filler material which was used to introduce a fixed thermal impedance. For this study, neoprene rubber was used as a thermal impedance, and the 2 halves of the brass tube heat sink sections were clamped tightly to the heat pipe. Thermocouple wires were inserted through the brass tube and rubber, and the individual junctions were pressed between the rubber and the pipe wall. The installation was such that the thermocouple junctions were mounted on a horizontal midplane along the pipe on 1 inch intervals. The coolant fluid was ethylene glycol which was circulated from a constant temperature bath controlled to 0.1°C according to the manufacturer's specifications. The flow rate was sufficient to maintain the entire brass tube in a nearly isothermal condition. The neoprene rubber liner served an important function in addition to acting as a fixed thermal impedance. Without the rubber, the heat pipe wall would have been maintained at a nearly uniform temperature close to the coolant temperature (always near 32°F in this study). Thus in both the active portion of the condenser where heat transfer takes place and in the blocked portion the temperature of the wall would have been nearly the same. However, the rubber thermal impedance caused the wall temperature to be at a significantly higher temperature in the active portion than in the blocked portion.

The Phase 1 heat pipe was modified for Phase 2 to increase the control sensitivity. The modification consists of the addition of a 5/16 in. dia. x .020 in. wall x 12" long stainless steel tube on the center line of the condenser section of the heat pipe. The tube was added to reduce the internal volume of the condenser. The reduced condenser volume yields a larger bellows to condenser volume ratio and consequently a greater movement of the gas-vapor interface per unit change in bellows position. The 5/16 in. dia. tube reduces the condenser volume to approximately 1/2 of its original value. The tube was added to the pipe by machining a new end flange for the condenser end of the heat pipe and soft soldering the new tube to the flange. The tube is sealed at the end in the pipe and open eneded where it protrudes through the end flange. Twelve chromel/constantan thermocouples were located on the top center-line of the tube at 1 in. inter The thermocouples were threaded through holes drilled in the e with e junctions formed in the soft solder puddled over each of the drilled holes. The lead wire was routed out of the heat pipe through the interior of the 5/16 in. dia. tube. Two 1/16 in. dia. stainless tubes were attached to the new tube in the same manner as the thermocouples. These tubes were located approximately 0.30 inches from the flange end of the condenser and protruded roughly 0.03 inches above the surface of the condenser plug tube. Their radial orientation was 15° on either side of the top center-line of the plug tube, and they were used to connect the condenser to the bellows system and to act as a fill or vacuum access.

The Gas Reservoir and Auxiliary Fluid Actuation System

The placement and orientation of the control components is shown in Figure 1, and the details of the bellows assembly are shown in Figure 3. Figure 4 is a photograph of the bellows assembly as used in Phase 1. Table 2 contains pertinent information regarding the control system. The bellows assembly as shown in Figures 3 and 4 was used in the Phase 1 study. It consists of the inner and outer bellows, their flanges and attachment fixtures, and a clear plexiglas housing. The vacuum housing allowed control of the external pressure on the gas bellows, and the extension rod was used for mounting a dial indicator for measuring the bellows displacement. The flanges were welded to the bellows by the manufacturer, and the inner bellows was supplied with 1/8 in. np.t. nipples which passed through the outer bellows end plates. This design eliminated seals between the inner and outer bellows. The "O" rings shown on the auxiliary belllows flanges were used to seal between the gas reservoir and the vacuum chamber. Alignment pins were located



HEAT PIPE BELLOWS ASSEMBLY

Figure 3

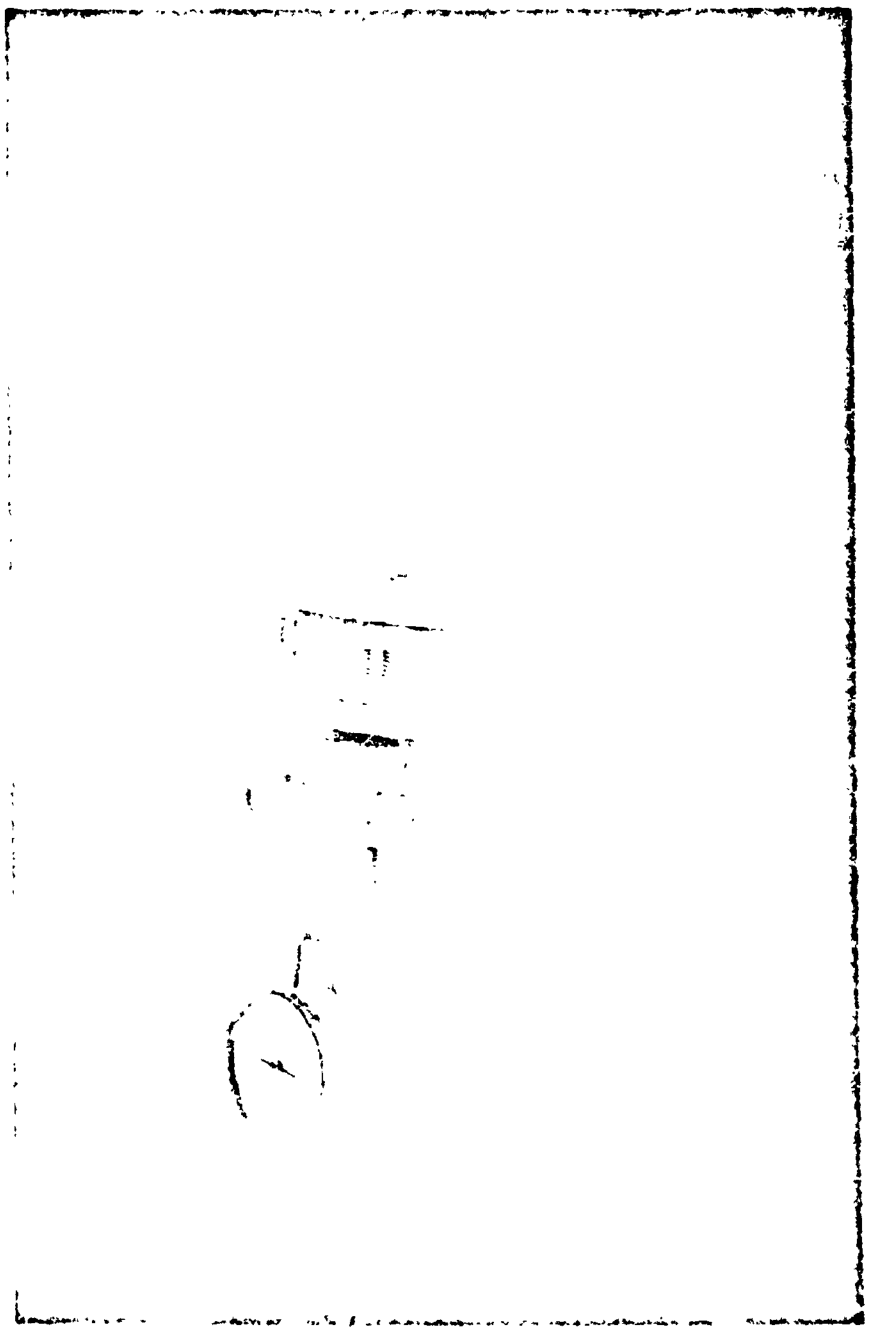


Figure 4 Photo of Bellows Assembly

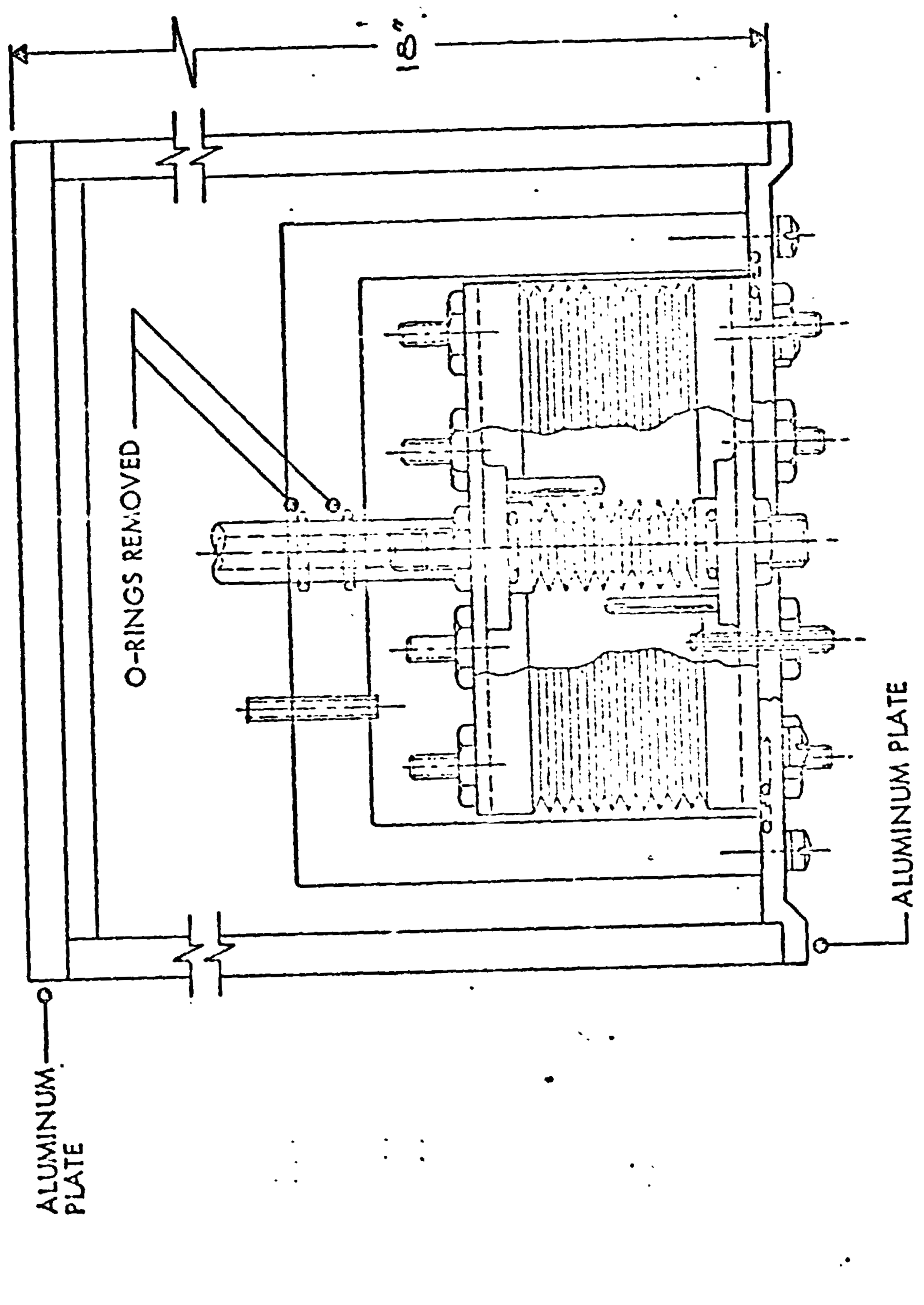
Table 2 CONTROL SYSTEM SPECIFICATIONS

noncondensible gas	nitrogen
auxiliary fluid	liquid methyl alcohol
auxiliary reservoir	0.50 in. I.D. x 12 in. long

bellows system*	gas		auxiliary
inside diameter, in.	2.00		0.20
outside diameter, in.	2.99		0.40
spring constant, lb./in.	16.		15.
max. press. differential, psi	--		200
length in compression, in.		0.35	
length in extension, in.		1.05	
stroke		0.70	

* Bellows procured from Metal Bellows Corp., Chatsworth, California.

at 4 locations to eliminate squirming of the inner bellows when under pressure. The bellows assembly was modified for Phase 2 of the study to reduce friction which caused hysteresis in the assembly motion. The modification consisted of a larger plexiglas cylinder with alumin end flanges, removal of the "O" rings for the extension rod, and removal of the retainers which had limited the bellows motion as shown in Figure 5.



MODIFIED BELLOWS ASSEMBLY

Figure 5

TEST CONDUCT AND RESULTS

The test program consisted of many preliminary tests leading to the final testing of the feedback controlled heat pipe. The procedures and results are presented chronologically to show how the program developed.

Simple Heat Pipe Checkout

The pipe was pumped down with the mechanical vacuum pump and charged with 10 cc methanol. Operation at low power levels (about 5 to 10 watts) was successful immediately. Residual noncondensible gas was evident at the condenser end, but this was easily eliminated by successive "burpings" to the vacuum system. At power levels greater than 10 watts, the temperature at the far end of the evaporator would rise above the other temperatures in the evaporator by an amount which indicated incipient wick dry-out. It was necessary to overcharge the heat pipe by about 150% to maintain near isothermal operation. While it was disappointing to rely on excess liquid for adequate performance of the heat pipe, it was not nessary to have a perfectly working heat pipe to demonstrate the variable conductance mechanisms.

Characteristics of the Feedback Loop

After successfully demonstrating the performance of the simple heat pipe, the next test was to check out the operating characteristics of the feedback system. The complete bellows assembly was fitted with a loading plate and placed in a compression testing machine in its operational attitude as shown in Figure 6. Figure 7 shows the results of this test. The unloading curve is not an accurate assessment of the hysteresis since it was observed at lower forces that the loading head was moving faster than the bellows. At some

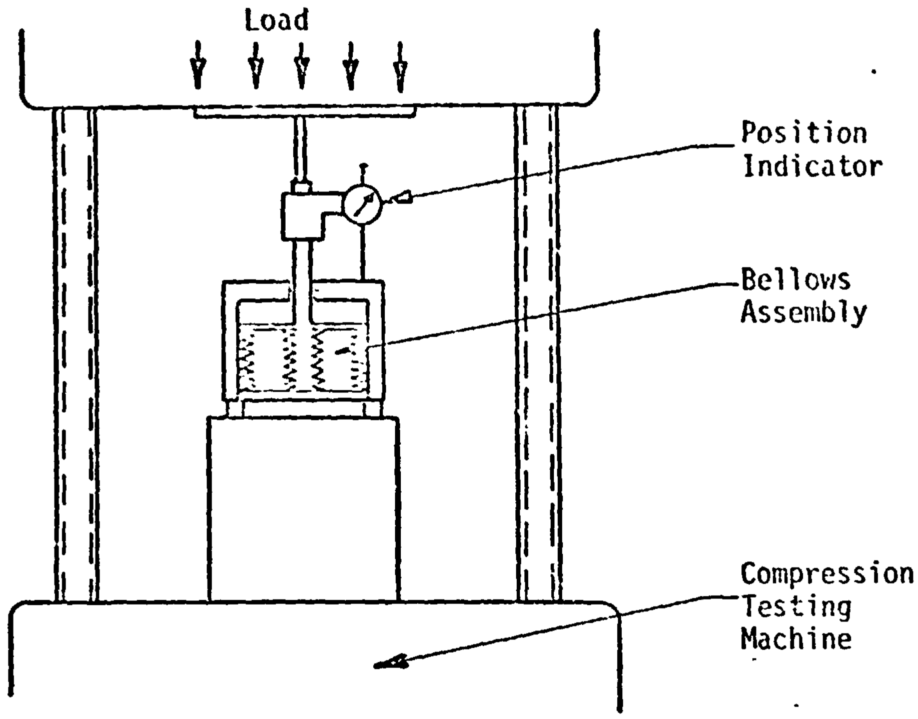


FIGURE 6 - Bellows Spring Constant - Test Set-Up

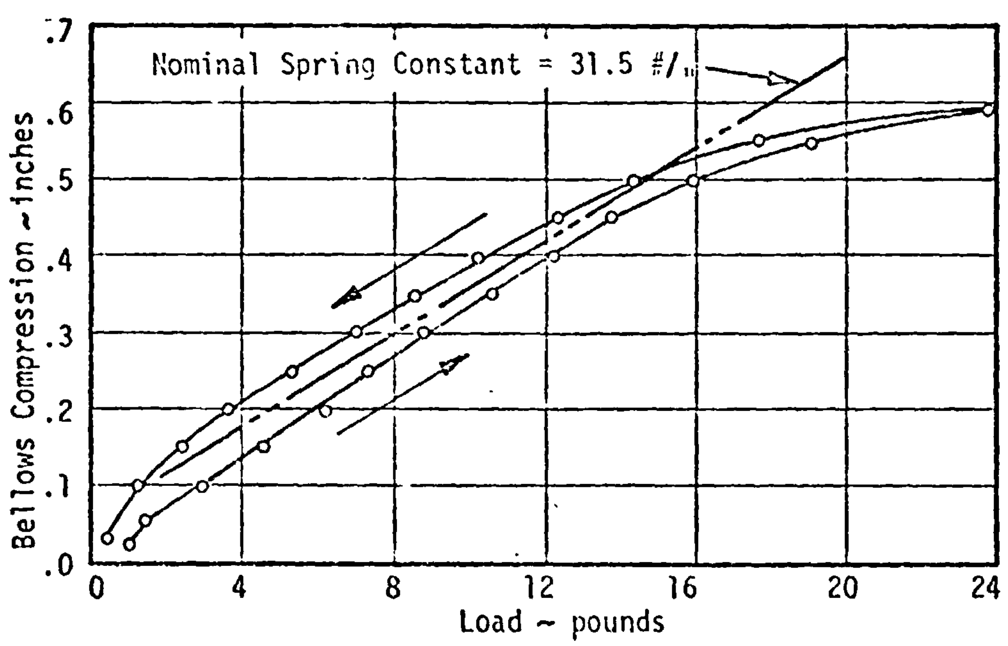


FIGURE 7 - Bellows Spring Constant - Test Results

points after 10 to 15 minutes, the bellows force was still changing. The loading curve is quite linear over the middle 0.4 inches of travel and was used to calculate a spring constant of 3.15 lb_f/in.

Following the test for spring constant, the bellows assembly was connected to a vacuum system as shown in Figure 8. The bellows was loaded by adjusting the manifold pressure causing it to travel from fully open to fully closed. Figure 9 shows the system hysteresis when going from the ope "equilibrium pressure" condition to fully compressed and then returning. The bellows movement is plotted against the pressure differential so that the effective pressure area, A_1, of the large bellows could be calculated.

This was done by using the calculated spring constant, and A_1 was found to average 4.46 in^2 for the center 0.4 inches of travel. The bellying of the slopes of the loading and unloading curves indicates the effective pressure area of the bellows does change as they are opened and that it is largest when the bellows are near a rest position.

It had been assumed that the effective area published in the manufacturer's catalog meant both effective "pressure" area and "volume displacement" area and, hence, the term A_1 in all the derivations was used for both purposes. The scheme shown in Figure 10 was used to measure the initial (closed bellows) volume and to determine the change in volume as the bellows opened. The results of this test are shown in Figure 11. effective area, for volume, is then the slope of the curve in Figure 11 or 5.08 in^2. This compares with the mean effective pressure area of 4.46 and the catalog value of 4.89 (which does not have the area of the small bellows of approximately 0.075 in^2 subtracted out). It appears that the published value could be used with only a small error introduced.

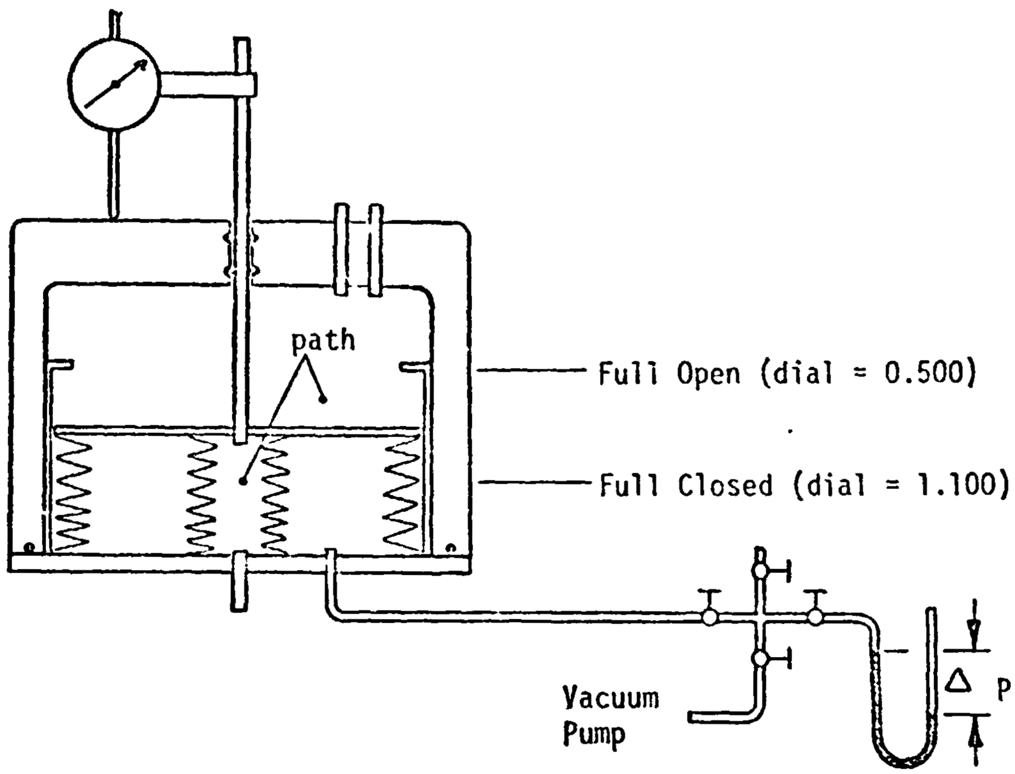


FIGURE 8 - Bellows System Hysteresis - Test Set-Up

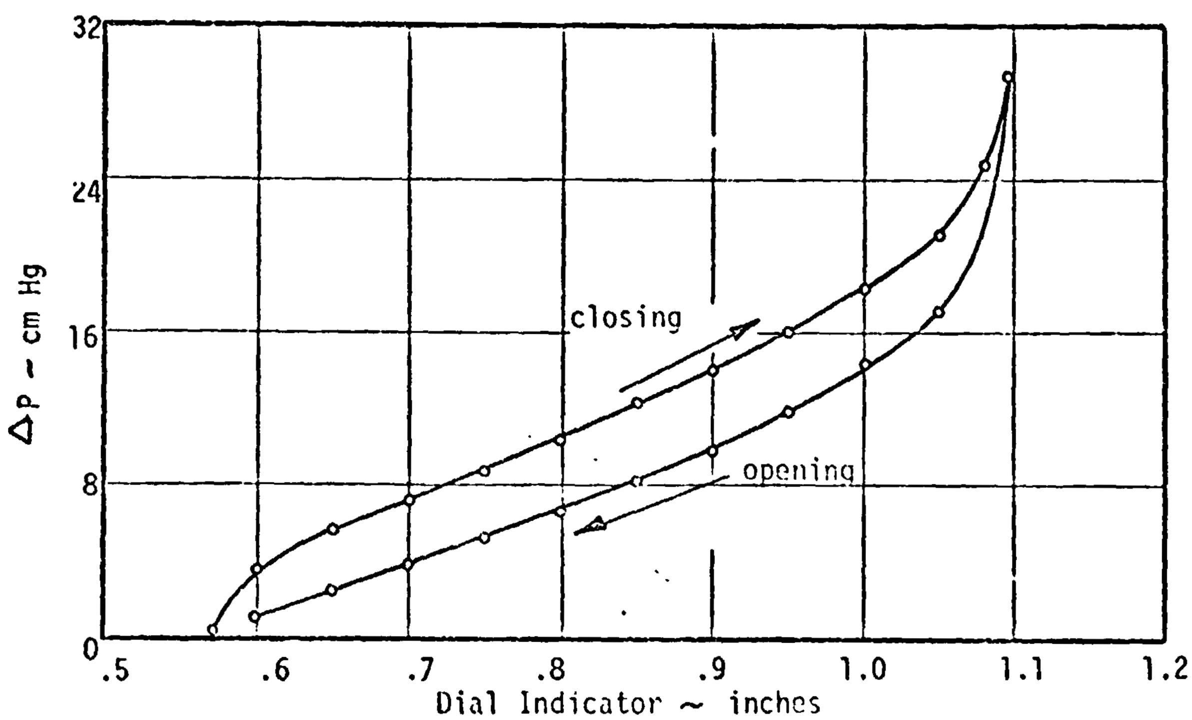


FIGURE 9 - Bellows System Hysteresis - Test Results

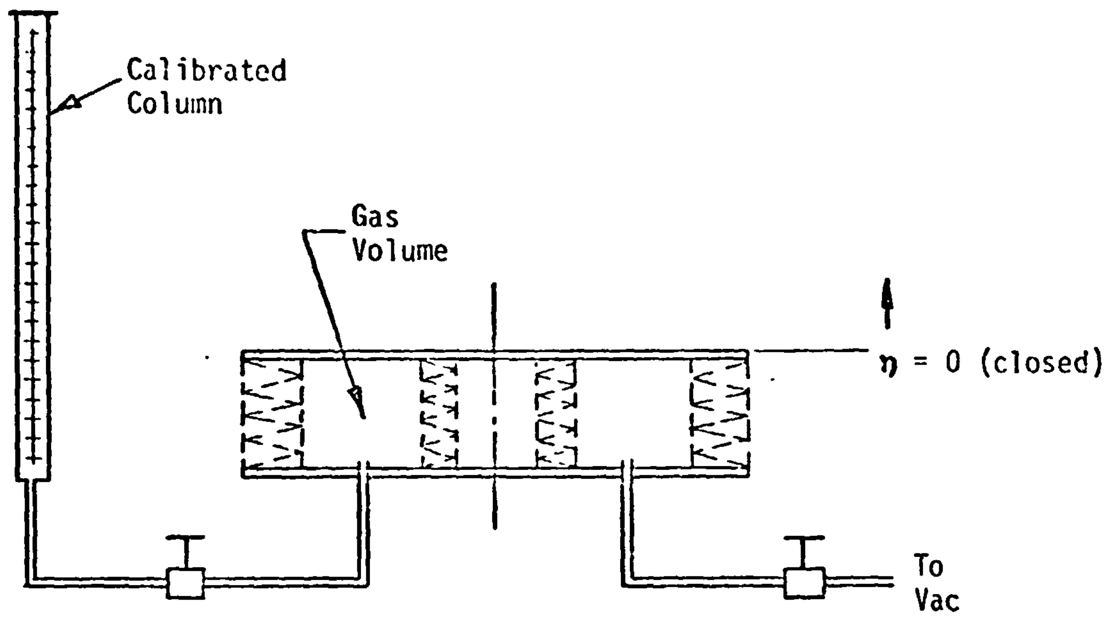


FIGURE 10 - Bellows Volume/Volume Displacement - Test Set-Up

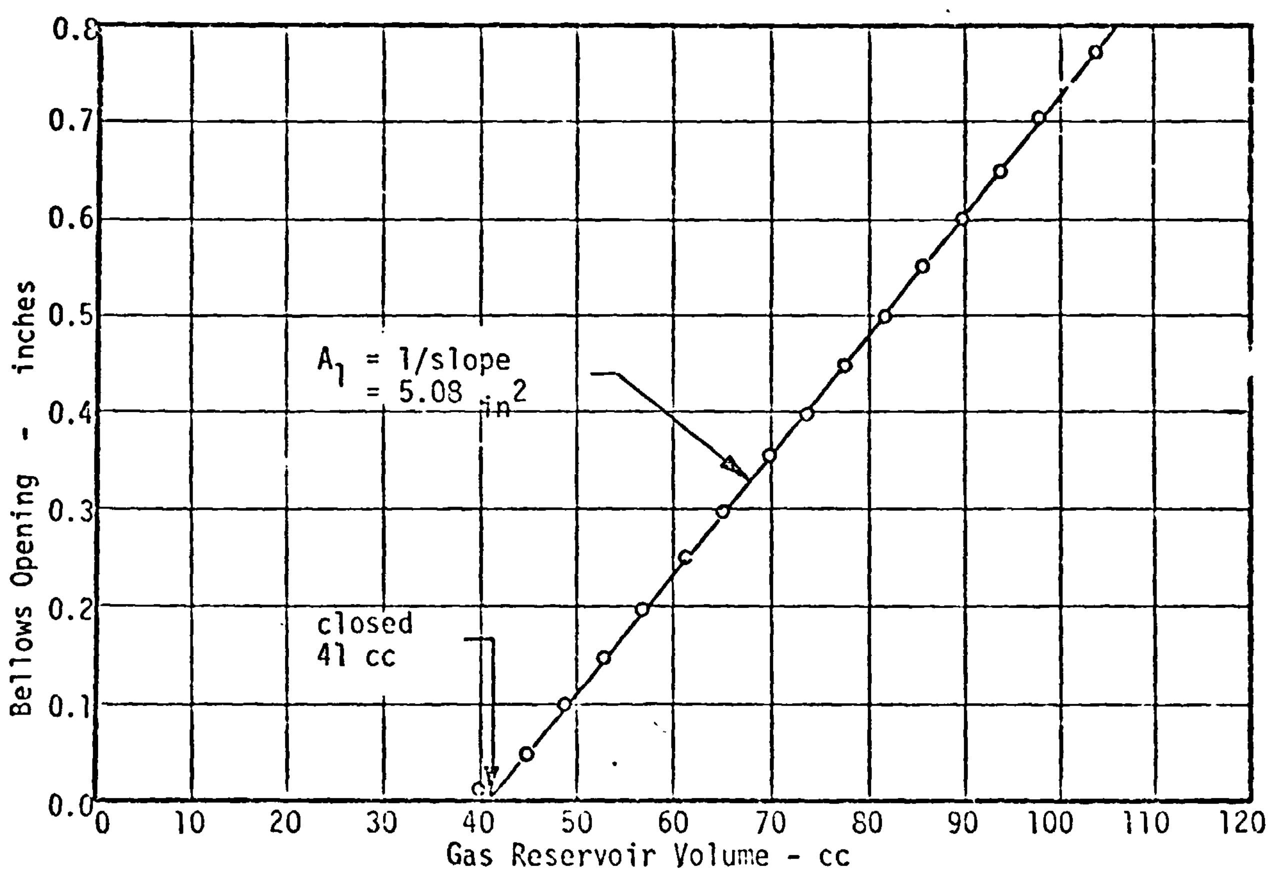


FIGURE 11 - Bellows Volume/Volume Displacement - Test Results

Gas Bellows/Auxiliary Fluid System Response

Analysis of the bellows system response appears in Appendix A. This analysis was used in the selection of the bellows sizes, although availability and cost were over-riding considerations. Although the expression given for bellows movement during actual operation is governed by Eq. (A-9), the modified Eq. (A-12) plotted in Figure A- 2 could be checked without operating the pipe. The auxiliary reservoir was connected to the bellows assembly and charged with liquid methanol. The heat pipe was then evacuated (no methanol in the pipe). The vacuum manifold used to maintain the bellows assembly back pressure was adjusted until the bellows just began to open. The auxiliary fluid reservoir fill valve was then secured, sealing the system. The heater block temperature was raised slowly but the bellows did not respond until the temperature had risen substantially, in some cases as much as 40°F. Once the motion started it continued linearly until the bellows was full open. Possibly the auxiliary system had some residual gas which had to be compressed before the bellows would function.

The reservoir charging technique had consisted of simply evacuating the reservoir with a vacuum system and subsequent back filling with methanol. The ultimate pressure of the vacuum pump was less than 1 torr which should have been adequate. Since the above results indicated residual gas in the system, the technique was changed to include bringing the reservoir above the normal boiling point and letting the vapor discharge through a liquid methanol filled flask. It was hoped that any trapped gas would be carried out with the methanol vapor and when the reservoir cooled, only methanol would re-enter it. Subsequent tests showed that this procedure was also apparently not adequate for charging the auxiliary reservoir. However, in the course of operating the system, it was found that the auxiliary liquid could be prepressurized using a simple scheme. The back pressure on the bellows was adjusted to open the bellows approximately 0.10 inch. Then after sealing the auxiliary system, the back pressure was increased causing the bellows to prepressurize the auxiliary fluid to approximately 30 psig.

Now, when the source temperature was increased, the bellows responded immediately. The results of this system checkout both before and after the prepressuring technique was employed are shown in Figure 12. This test pointed to a serious deficiency in the setup; that of not being able to independently adjust the auxiliary liquid pressure. Therefore, at this point an auxiliary fluid pressure adjusting chamber was added to the system. It consisted of a thick walled brass cylinder with a screw piston which altered the volume (pressure) of the reservoir.

The bellows sensitivity, that is, displacement due only to a heater block temperature change, was approximately .021-.023 in/°F. This was considerably lower than the design prediction of .038 in/°F from Figure A-2. The repeatability of the slope of the test results indicated that the system response was as measured and that the error lies in the calculated predictions. Therefore, the final phase of the test program was begun.

Variable-Conductance Operation

The entire assembly, as shown in Figure 1, was next operated as a unit. The system was charged in the following manner. First, the auxiliary system was charged with the methods outlined above (evacuate, backfill, boil and backfill), with the fill ports left open. Next, the heat pipe and gas bellows

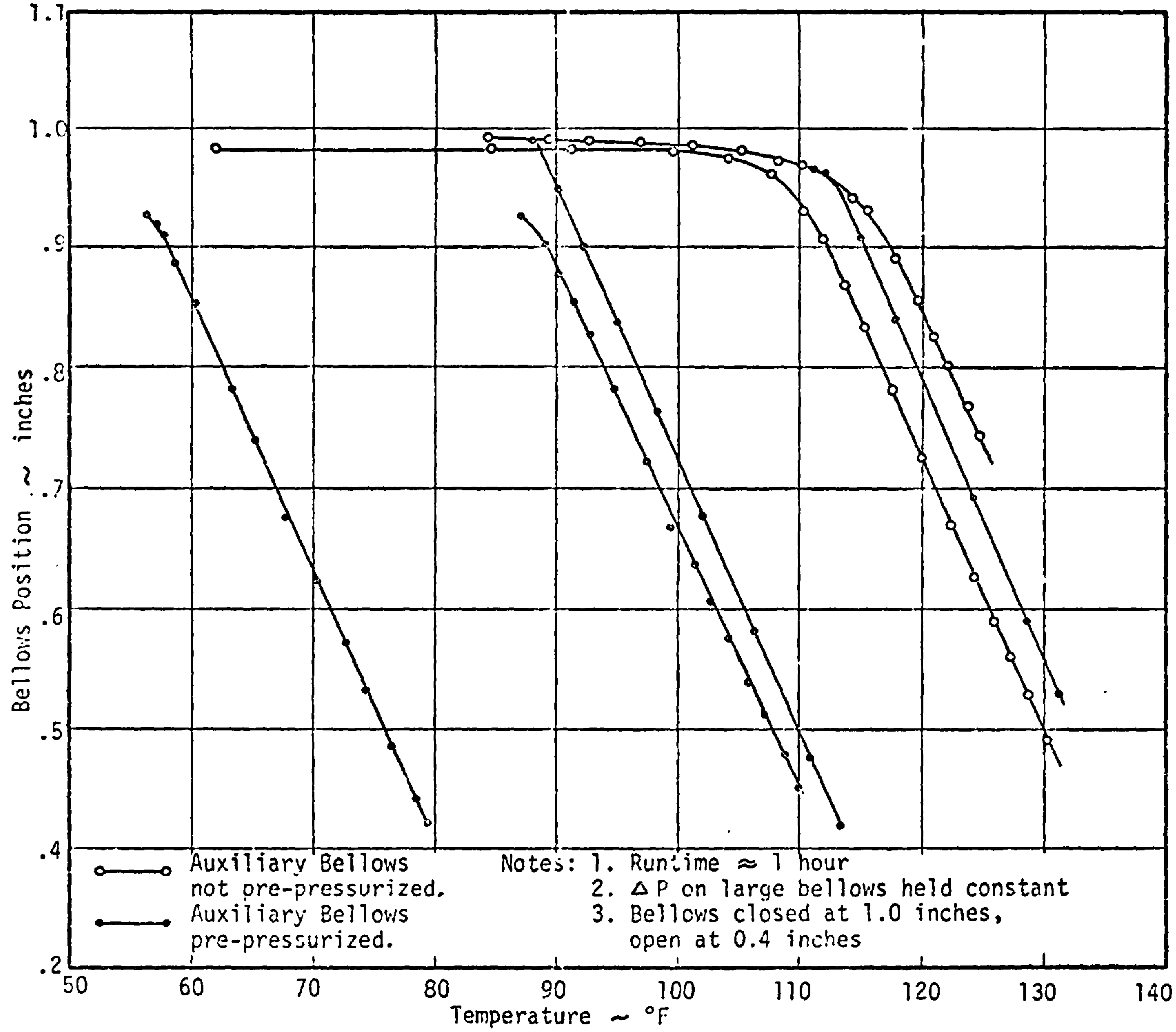


FIGURE 12 - Bellows Response to Source Temperature Variations

system were evacuated by pumping through the manifold for approximately 24 hours. The valve between the heat pipe and bellows was then closed. Note that with the valve arrangement shown, either the heat pipe or the gas bellows could be evacuated independently. Next the heat pipe was back filled with approximately 20 cc of methanol. The source was then turned on so that the condenser thermocouples would be driven above the sink temperature. The vacuum was then opened to the heat pipe interior and the condenser temperature monitored. When the couple nearest the end came up to the rest of the condenser, indicating a fully active heat pipe, the vacuum valve was closed and secured and the power and cooling loop turned off. The bellows back pressure was then adjusted to bring the bellows full open.

When the pipe came back to equilibrium at room temperature, the noncondensible gas (nitrogen) was introduced into the bellows until the total pressure was approximately 1 psi above the heat pipe total pressure. This over pressuring prevented methanol from entering the bellows when the valve between the heat pipe and bellows was opened. Next the cooling loop was activated which caused the heat pipe to start operating in its normal mode. After a few minutes the main valve was opened and the nitrogen gas entered the heat pipe. Because of the mass of the heater block, it had not yet started to drop in temperature. The over filling with nitrogen caused the vapor/gas interface to move far into the adiabatic section. Carefully the vacuum valve to the heat pipe was opened and as the nitrogen was slowly pumped out, the movement of the vapor/gas interface could be seen along the condenser by watching the thermocouples rise in turn to near the vapor temperature. The bellows back pressure was then increased until the bellows was within about 0.050 in. of being closed. The nitrogen gas was then pumped out until the vapor/gas interface resided at approximately the number twelve thermocouple (0.5 in. into the condenser from the adiabatic section). Finally, the fill ports in the auxiliary reservoir were secured and the pre-pressuring screw piston was adjusted until the bellows just began to move. The variable-conductance heat pipe was ready for operation. After a number of days of taking data and flexing the system, it was noticed that the gas/vapor interface position was not repeatable with heat load. It was thought at first that gas was once again leaking into the system. However, it became noticeable that the auxiliary fluid pressure would decay slightly when the temperature was held constant. If a single failure was the cause of both problems it had to be that either the o-ring seal between the gas and auxiliary systems was leaking or the small bellows had become leaky allowing the auxiliary methanol to leak into the large bellows gas chamber. The system was again disassembled. The small bellows was fitted in a test jig so it could be pressurized with laboratory high pressure air when immersed in a water tank. Three tiny bubbles formed on the leaves of the bellows growing at a rate of about 1/16" in diameter in fifteen minutes at approximately 50 psi. Attempts to solder the holes closed caused the bellows to leak like a sieve. A new bellows was ordered to the same specifications. The manufacturer reminded us that the maximum pressure the bellows was good for was 200 psid. Looking back through the preliminary tests, there were times when this pressure was driven as high as 300 psid and quite possibly higher by inadvertently driving the bellows back pressure too high. The set-up was already quite cumbersome with too many possible failure points and unknown valve, fitting and associated tubing volumes. Therefore, it was decided that the auxiliary pre-pressuring chamber could be used to keep the system pressure within limits, and no pressure relief valve was installed.

After the bellows arrived, it was immediately installed in the system and operation started again. Finally, the system performed under all operating conditions. It was run in a total of five modes. (1) Normal heat pipe with no gas; (2) Conventional variab nductance (c-v-c) with just the condenser volume; (3) c-v-c with a fix osed bellows volume; (4) c-v-c with a fixed open bellows gas volume, and (5) the feedback-controlled mode with variable gas bellows volume. The tests were run over a power range of 2-30 watts, nominal. The results of these tests are shown in Figure 13 for variation of source temperature and in Figure 14 for variations of working fluid vapor temperature.

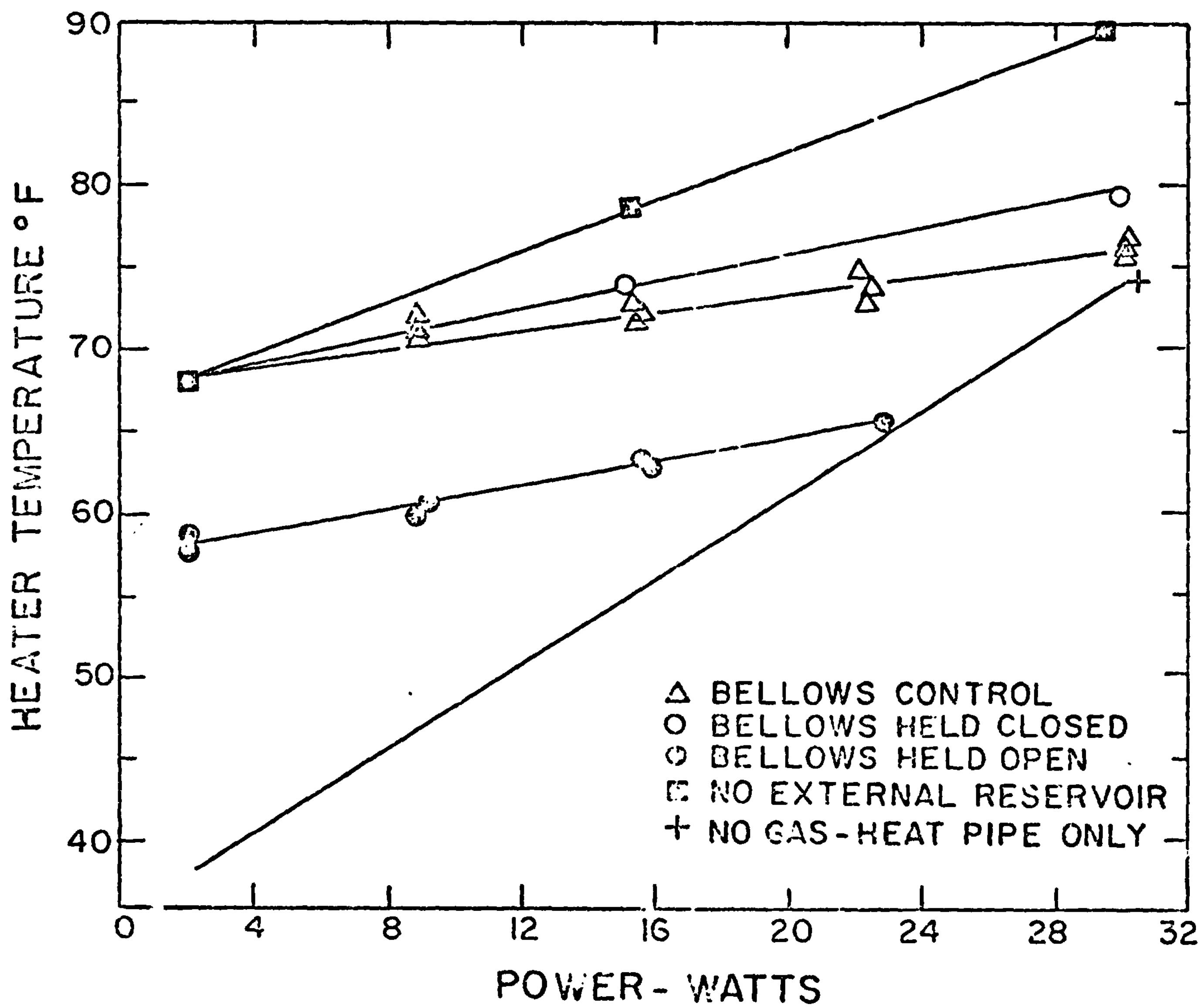


Figure 13. Heater Block Temperature versus Source Power

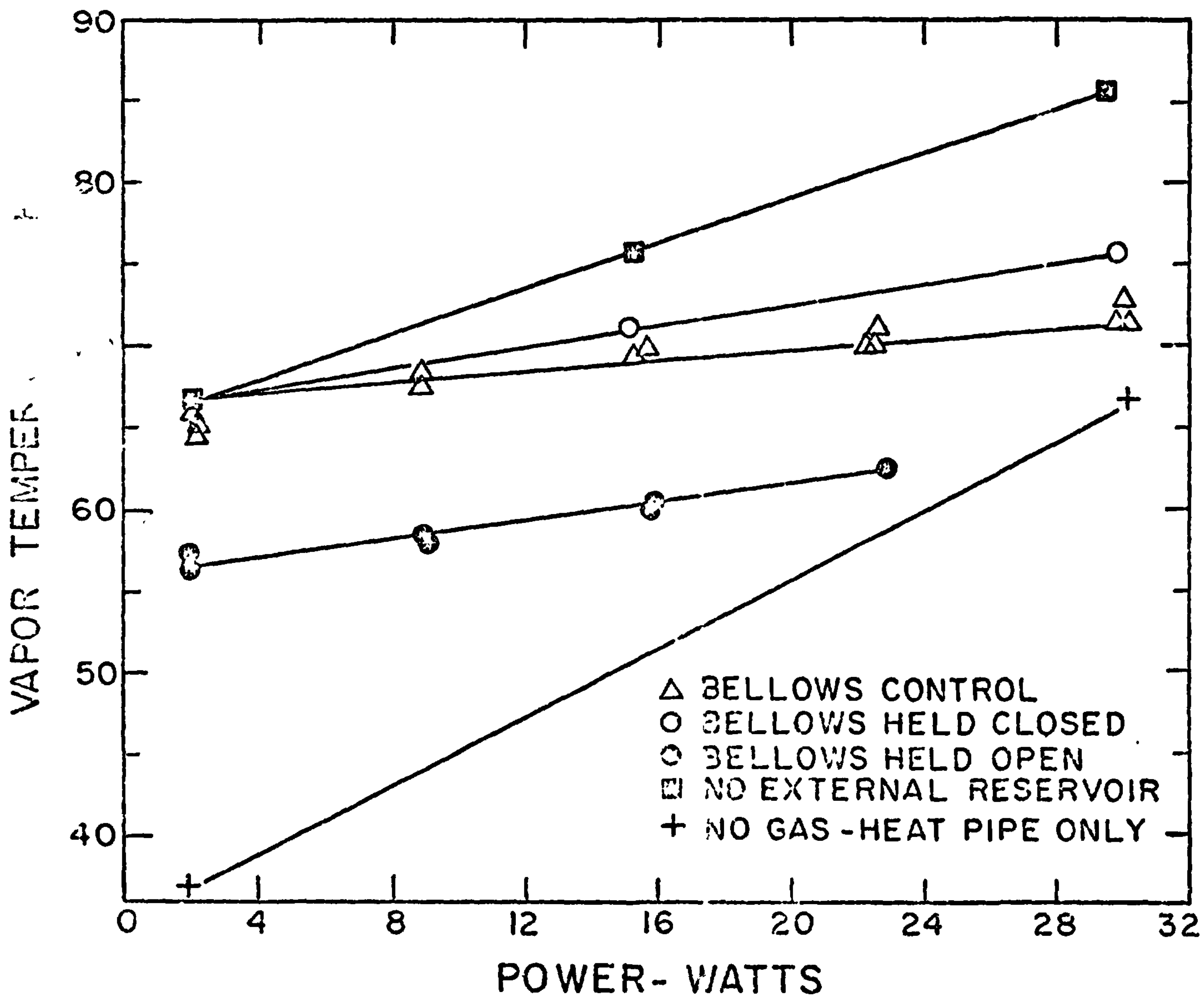


Figure 14. Vapor Temperature versus Source Power

EXPERIMENTAL RESULTS AND DISCUSSION OF SYSTEM PERFORMANCE

Unmodified Pipe Results

The data shown in Figures 13 and 14 constitute the primary results of the program. Figure 13 shows the heater block temperature versus heater power, and Figure 14 shows the adiabatic section temperature versus heater power. The adiabatic section temperature was shown to be essentially the same as the pure vapor temperature. Data are shown for the 5 modes of operation, and they show the improvement in temperature control with the use of a noncondensible gas and further with the feedback control feature. Baseline operation of the basic pipe without gas is indicated by the steepest line which shows that the source temperature drops to about 38°F from 74°F when the power is reduced. The performance is improved considerably when gas is added to the pipe but with no reservoir at all. In this mode the temperature increased from about 68°F to 90°F when the power was increased. Note that it was possible to adjust the source temperature to some value (68°F) at low power in this mode; whereas with no gas present, no such regulation was possible. The advantage of a gas reservoir is demonstrated by the points for "bellows held closed" and "bellows held open" operation. These volumes are 2.9 cu.in. and 6.0 cu.in. respectively. The total amount of nitrogen in the pipe reservoir was the same for both of these modes, and the lower temperature of the open mode as compared to the closed bellows mode were due to the presence of less gas in the pipe and hence more active condenser. Note that no data was acquired beyond 22 watts with the reservoir in the fixed-open position. This was due to the fact that the vapor/gas interface had moved almost to the end of the condenser, and operation at higher power levels would have resulted in complete elimination of gas from the pipe.

Continued operation in this state would have caused diffusion of methanol vapor into the reservoir with subsequent long time intervals to steady operation. It is anticipated that the source temperature for 30 watt operation would have been the same as for the basic pipe, i.e., 74°F. The best control sensitivity was experienced with bellows actuation by the auxiliary fluid pressure. In this mode, temperature variation of the source was limited to about 9°F. Operation in the feedback mode produced a vapor/gas front movement of approximately 10 inches for the full power change. This displacement represents almost the entire length of the condenser section. The vapor temperature was limited to a 3°F rise for the same conditions as shown in Figure 14, and it was less than the source temperature change for all modes of operation. Table 3 summarizes the results to the nearest degree, and it is clear that control is about the same with a reservoir of fixed size as it is with a variable size in the range tested.

TABLE 3. HEAT PIPE CONTROL COMPARISONS (UNMODIFIED PIPE)

	Temperature Change (°F) for Power Change from 2 to 30 w	
	vapor	source
Basic heat pipe (no gas)	30	36
No external reservoir (gas in pipe)	18	22
2.9 cu.in. reservoir (bellows closed)	9	13
Variable volume reservoir - feedback control	3	10

The reported data are from runs which occurred over a period of approximately one month. In that time the heat pipe charge and system gas content were not altered. The temperature reproducibility at both high and low powers indicate that the system was essentially leak-tight. Approximately forty-five minutes were required for each steady-state condition. This did not allow enough time for the equilibrium to exist between the molal content of the vapor-gas mixture in the reservoir and the condenser. However, because the temperature of the inactive portion of the condenser was nearly constant, the composition of the mixture at the condenser end did not vary appreciably. For cases where a test condition was left overnight the temperature changes were not discernible.

Modified Pipe Results

As earlier discussed, the system was modified for the second phase of the program. These modifications were threefold: (1) A plug was inserted in the condenser gas/vapor volume, (2) the vacuum housing around the bellows assembly was enlarged to include the dial indicator, and (3) the 3 retainer clips were removed from around the large bellows. The effect of (1) was to increase the control sensitivity to creating a larger change in the active condenser length for the same bellows displacement, and modifications (2) and (3) were to reduce friction in the bellows motion. Removal of the retainer clips also allowed the bellows to expand until it was stopped by the plexiglas housing. Therefore the "full open" bellows position represents a gas reservoir volume that is 14% larger than in Phase 1.

The results of the tests with the modified system are tabulated in Table 4. Comparison of results for equivalent conditions in Tables 3 and 4 shows that the modified pipe had improved control capability. Whereas the temperature rise of the source was 10°F with the full condenser volume accessible to the gas, the rise was limited to 7°F with the plug installed.

The total gas content was greater for the Phase 2 operation allowing operation of the pipe with the bellows full open and at 30 watts power The larger gas content caused the temperature levels to be somewhat higher in Phase 2 than in Phase 1. Comparison of all modes is possible on the basis of temperature rise per unit power change as is shown in Table 5.

TABLE 4. HEAT PIPE CONTROL COMPARISONS (MODIFIED PIPE)

	Temperature Change (°F) for Power Change from 2 to 30 watts	
	Vapor	Source
2.9 cu.in. reservoir (bellows closed)	8	12
6.8 cu.in. reservoir (bellows open)	5	10
Variable volume reservoir - feedback control	2	7

TABLE 5. COMPARISON OF MODIFIED AND UNMODIFIED SYSTEM CHARACTERISTICS

	Change in Heater Temperature Per Unit Power, °F/watt	
	Unmodified system	Modified system
Minimum reservoir vol.	0.46	0.43
Maximum reservoir vol.	0.46	0.36
Variable reservoir vol.	0.36	0.25

The significantly improved temperature stability offered by gas control techniques in general is obvious when any of the values of Table 5 are compared to the uncontrolled systems response of $\frac{36}{28}$ = 1.29 °F/watt. It is also apparent that the moidfied system had approximately 30% improvement in the attenuation of the temperature change of the source.

COMPUTER MODELING OF THE CONTROLLED HEAT PIPE

The TRW Systems, Inc. computer program called CASPIPE was used to model the variable conductance heat pipe. This program will predict the total heat transfer rate given the amount of gas or the total amount of gas present in the reservoir and the pipe if the heat load is specified. In either case, a fixed size gas reservoir is assumed. The CASPIPE program was used to calculate the amount of noncondensible gas since no experimental determination of this quantity was made, and the resulting temperature profiles were compared to the experimental temperatures. The program requires as input data specification thermal conductance values for the wick-liquid matrix and for the conduction path from the pipe wall to the coolant. In this experiment the thermal path from the wall to the coolant is complicated by contact resistances at both faces of the neoprene rubber and by the compressibility of the neoprene itself. A series of 30 runs was made at 5 power settings while changing the active condenser length by displacing the gas bellows through adjustment of the back pressure in the plexiglas chamber. The results of these tests are shown in Figure 15 as adiabatic section temperature (vapor temperature) versus active condenser length for the 5 power settings. The average conductance was calculated to be 7.48 watts/sq.ft.°F ($\pm$ 6%). Calculated performance curves are shown for comparison and the value specified is considered to represent the data well. The agreement is adequate for the resolution of the active condenser length which is limited to about 1/2 in. due to the thermocouple spacing. The condenser conductance and the other parameters were used to calculate the gas content for 10 runs over the power range from 2 to 30 watts and for a range of positions. The average

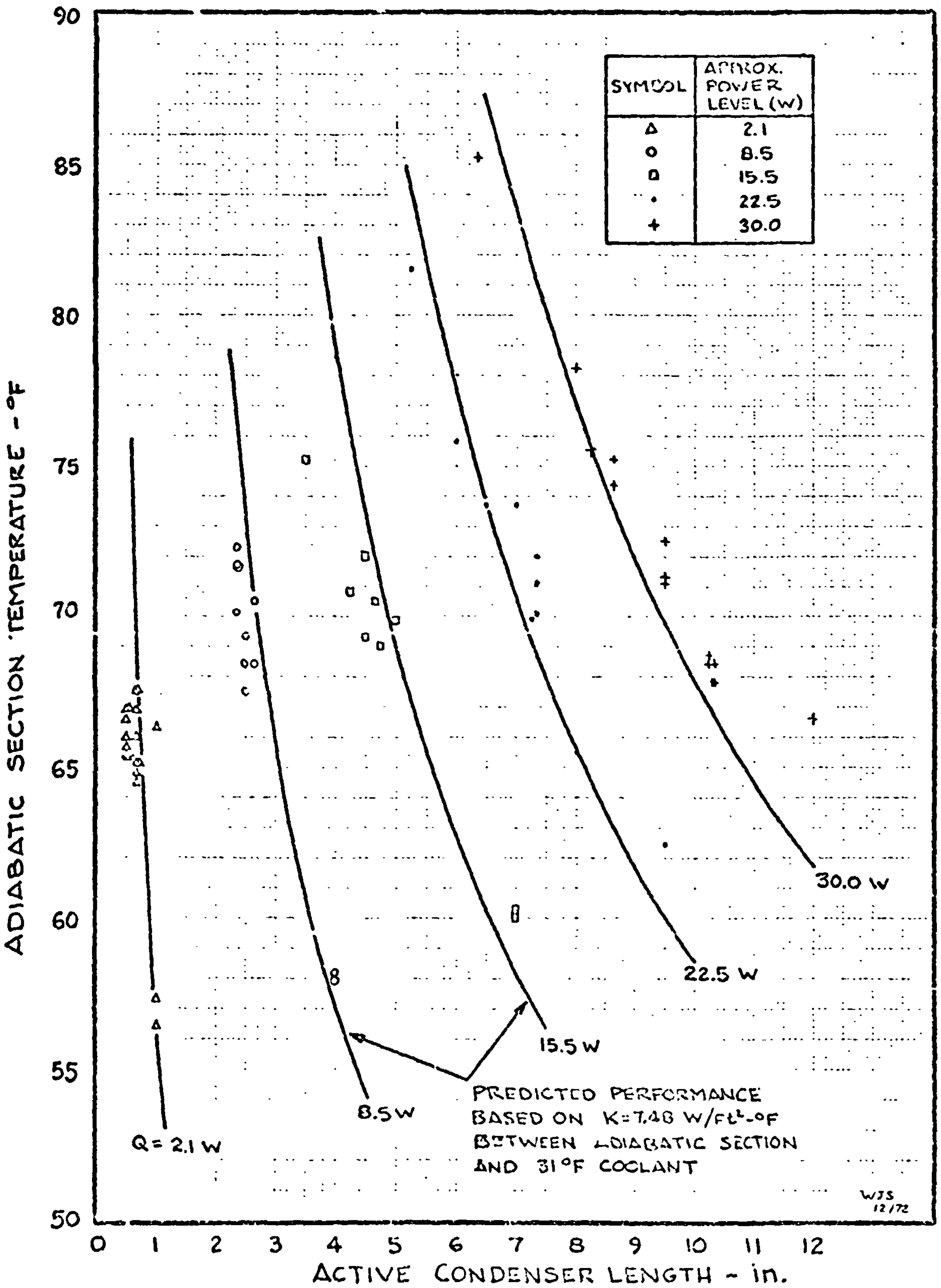


FIGURE 15 - Gas Front Position Effects on Operating Temperature

value was calculated to be 6.466×10^{-7} lb-mole + 6%-0% for the extremes. The 10 runs were selected for their range of conditions and for the closeness of their condenser conductance values to the average. The calculated gas amount was us . with GASPIPE to predict the heat load for 5 out of the above 10 runs. Table 4 summarizes the results for the 5 runs.

TABLE 6. COMPUTER MODELING RESULTS

Heat Load Experiment watts Q_e	Control Mode	Evaporator Temperature °F	Heat Load Calculated watts Q_c	Ratio $\frac{Q_e}{Q_c}$
2.1	control	67	0.3	6.8
9.0	control	69	9.2	.97
15.3	control	69	18.3	1.1
22.6	bellows closed	76	25.1	.9
29.9	control	73	29.9	1.0

The table shows fair agreement with the exception of the low power case where the fraction of active condenser is extremely sensitive to the gas front location. At this power setting, there is less than 1/2 in. of active condenser, and small errors in the extent of the gas blockage can significantly alter the heat load.

CONCLUSIONS

This program has demonstrated the value of a variable-conductance heat pipe in the control of the temperature of a power source which ranged from 2 to 30 watts. With the heat sink at 32°F, the power source temperature ranged from 37°F to 67°F, but with a variable-conductance passive feedback control system, the source temperature was maintained at 70°F ± 3 1/2. The performance of the feedback bellows volume actuation system is less impressive when compared to the thermal control which is obtainable with a fixed volume reservoir. The best control in this mode was found with a volume of 6.8 cu.in. when the source was maintained at 65°F ± 5. Considering the absolute improvement in the source temperature variation from 10°F to 7°F (for comparable fixed and variable volume reservoirs), the improvement is not dramatic. In fact, the size and weight of the bellows and the susceptibility to failure of the bellows system by overpressurization or fatigue could make the feedback actuation loop impractical.

The addition of a plug to occupy approximately one-half of the condenser volume significantly improved the performance. For example, in the variable reservoir volume mode the temperature rise was reduced from 10°F to 7°F for the power range of 2 to 30 watts.

Computer modeling of the gas-controlled heat pipe was done with the TRW-GASPIPE program. This study was an inadequate test of that program since the amount of gas in the system was not independently determined; however, the limited application demonstrated that the program is a useful tool for the prediction of performance of such systems.

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NOMENCLATURE

A	area
C	constant defined in text
f( )	function
k	bellows combined spring constant
L	length
P	pressure
Q	heat transfer rate
R	thermal resistance
T	temperature
V	volume
β	isobaric thermal expansion coefficient
η	bellows position
κ	isothermal compressibility
λ	enthalpy of vaporization

Subscripts

1	effective area of auxiliary fluid bellows
2	effective area of gas bellows
a	auxiliary fluid
c	condenser
ext	external
g	gas
i	initial condition (for bellows i represents the unstressed position)

Subscripts (continued)

o	sink
p	heat pipe
r	reservoir
s	source
x	cross-section
v	vapor
x	cross-sectional

Appendix A

This section contains a simplified analysis of the bellows system with auxiliary fluid actuation. The analysis was used in the design of the system and the specification of the bellows characteristics.

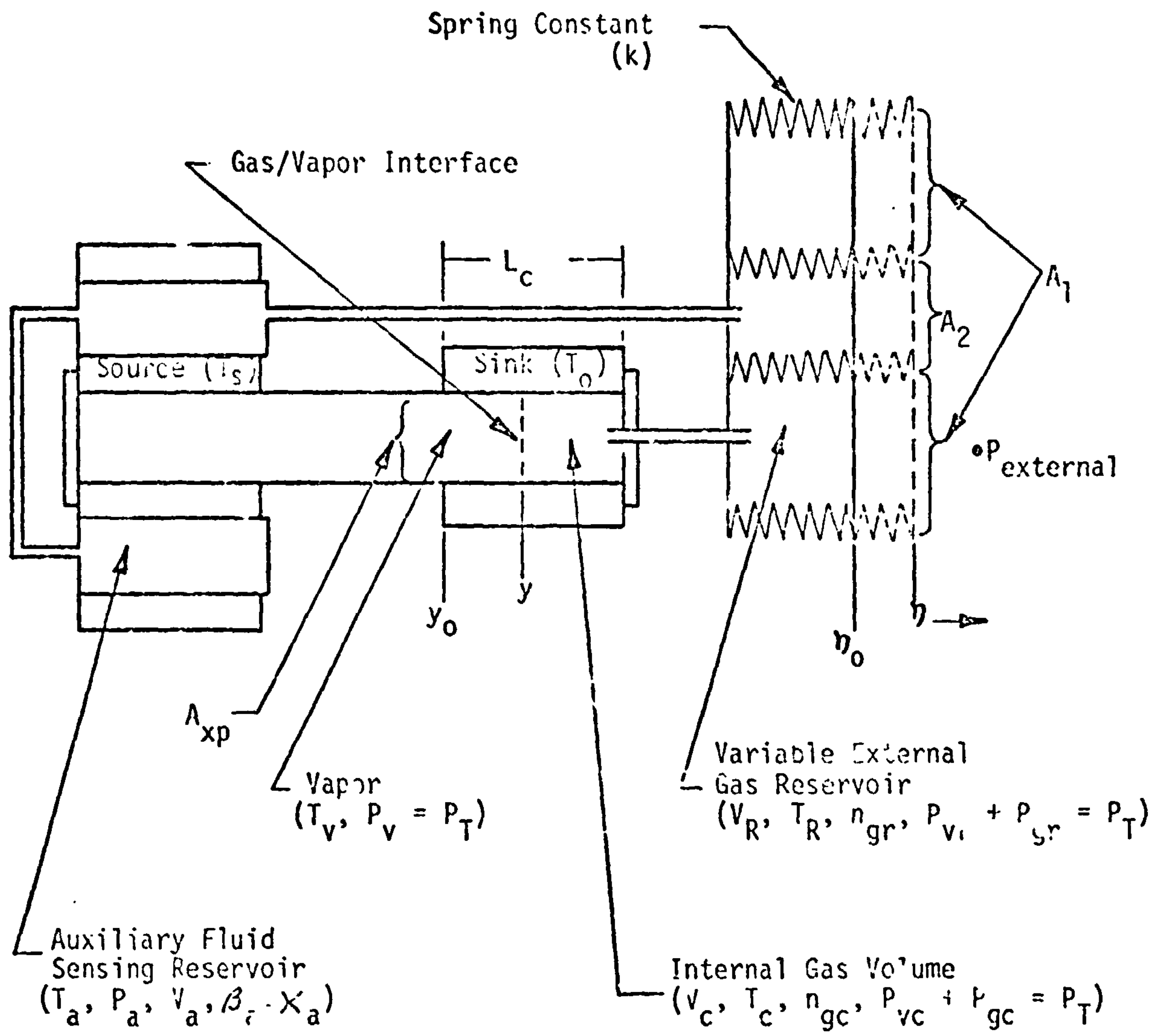

FIGURE A-1 - Feedback Variable Conductance Parameters system. In some cases, sizes were adjusted slightly to accommodate design integration, such as the size of the auxiliary bellows. The cost of a bellows tailored to the exact sizes recommended was exorbitant. Therefore, some off-the-shelf sizes were substituted after analysis to demonstrate that they would be suitable. The first step in the design was to choose an effective area for the large (gas) bellows. An off-the-shelf size of 4.89 sq. in. effective area was almost identical to the original design. The remainder of the bellow dimensions were calculated after assuming that only the auxiliary fluid would be used to actuate the bellows. In actual practice, the rising total pressure in the heat pipe would augment the auxiliary fluid. Also assumed in this sizing analysis was that compression of the noncondensible gas over the operating range would be negligible. These assumptions are conservative but they do place upper bounds on the optimizing process.

After choosing A_1 it is easily calculated ($\Delta\eta = L_c(A_{xp})/A_1$) that a displacement of the bellows by 0.381" is required to move the gas/vapor interface the length of the condenser.

The actual sizing of the small actuating bellows is an iterative process requiring trades between spring constant and operating pressure. By ignoring the effect of the heat pipe total pressure on the bellows position, Eq. (A-9) reduces to

$$\frac{d\eta}{dT_s} = \frac{A_2 \beta_a V_a}{A_2^{\,2} + \kappa_a V_a k} \tag{A-12}$$

Figure A-2 shows values of the measure of control sensitivity, $d\eta/dT_s$, as a

1. $\ddot{\eta} = 0$; $\dot{\eta} = 0$ (acceleration and velocity damping are negligible
2. The external pressure on the bellows system is constant.
3. The entire volume of auxiliary fluid is at T_s.
4. Hysteresis in the bellows is neglected.
5. The bellows spring constant is constant.
6. The cubic expansion and compressibility factors for the auxiliary fluid are constant over the operating range.
7. The expansion of the auxiliary chamber walls is negligible.

With these assumptions, the following performance equations are derived:

A force balance on the bellows position gives

$$P_{ext} (A_1+A_2) + k (\eta-\eta_i) = P_a A_2 + P_v A_1 \tag{A-1}$$

differentiating with respect to P_v and noting that $P_v = f (T_v)$

for a two phase system

$$A_1 f' (T_v) + A_2 \frac{dP_a}{dT_v} - k \frac{d\eta}{dT_v} = 0 \tag{A-2}$$

Next, using assumption 3 and noting V_a = function (T_a, P_a)

$$dV_a = \left(\frac{\partial V_a}{\partial T_a}\right)_p dT_a + \left(\frac{\partial V_a}{\partial P_a}\right)_T dP_a \quad \text{or} \tag{A-3}$$

$$dV_a = \beta_a V_a \, dT_a - \kappa_a V_a \, dP_a$$

where $\beta_a \equiv \frac{1}{V_a}\left(\frac{\partial V_a}{\partial T_a}\right)_p$ and $\kappa_a \equiv \frac{1}{V_a}\left(\frac{\partial V_a}{\partial P_a}\right)_T$

dividing (A-3) by $d\eta$, using the chain rule to define

$$\frac{dP_a}{d\eta} = \frac{dP_a}{dT_v} \cdot \frac{dT_v}{d\eta} \quad \text{and noting} \quad \frac{dV_a}{d\eta} = A_2$$

$$A_2 = \beta_a V_a \frac{dT_a}{d\eta} - \kappa_a V_a \frac{dP_a}{dT_v} \cdot \frac{dT_v}{d\eta} \tag{A-4}$$

Combining (A-2) and (A-4) gives

$$A_2 = \beta_a V_a \frac{dT_a}{d\eta} - \frac{\kappa_a V_a k}{A_2} + \frac{A_1}{A_2} \kappa_a V_a f'(T_v) \frac{dT_v}{d\eta} \tag{A-5}$$

By definition of R_s,

$$Q = \frac{1}{R_s}(T_s - T_v) = \frac{1}{R_s}(T_a - T_v) \tag{A-6}$$

differentiating with respect to η

$$\frac{dT_a}{d\eta} = \frac{dT_v}{d\eta} + R_s \frac{dQ}{d\eta}$$

combining this with (A-5) and noting $\frac{dQ}{d\eta} = \frac{dQ}{dT_v} \cdot \frac{dT_v}{d\eta}$

$$\frac{d\eta}{dT_v} = \frac{A_2\beta_a V_a + A_1 \kappa_a V_a f'(T_v) + A_2 \beta_a V_a R_s \frac{dQ}{dT_v}}{A_2^2 + \kappa_a V_a k} \tag{A-7}$$

This expression can also be written in terms of T_s

$$\frac{d\eta}{dT_s} = \frac{A_2\beta_a V_a + A_1 \kappa_a V_a f'(T_v)\left[(1-R_s)\frac{dQ}{dT_v}\right]^{-1}}{A_2^2 + \kappa_a V_a k} \quad \text{or} \tag{A-8}$$

$$\frac{d\eta}{dT_s} = \frac{A_2\beta_a V_a + A_1 \kappa_a V_a f'(T_v) \frac{dT_v}{dT_s}}{A_2^2 + \kappa_a V_a k} \tag{A-9}$$

An examination of the terms in Eq. (A-8) and (A-9) gives some insight into maximizing performance, even though an expression for dQ/dT_s has not yet been developed. The parameter $d\eta$ represents the motion of the bellows (positive $d\eta$ for an opening bellows). Therefore, it is desirable to maximize the right hand side of Eq. (A-8) and (A-3).

1. Minimize Auxiliary Fluid Compressibility - Although this fact is somewhat masked by Eq. (A-9), a look at Eq. (A-3) shows the change in dV_a/dT_a is degraded by increasing compressibility.

2. Maximize V_a - Again, the effect of V_a is not clear in Eq. (A-9) but Eq. (A-3) can be rearranged to show $dV_a/dT_a = V_a\ (\beta_a - \kappa_a \frac{dP_a}{dT_a})$ and as long as $(\beta - \kappa\ dP_a/dT_a)$ is positive, which it must be for positive feedback, increasing V_a will always improve performance.

Increasing A_1 - The extent of improved performance with increasing A_1 is dependent on the relative value of the terms in the numerator. The term including A_1 comes about from the pressure the gas/vapor mixture exerts on the large bellows and is really not part of the feedback loop.

4. Increase $f'\ (T_v)$ - This term also is not part of the design feedback loop, but it is a plus for the system. The magnitude, of course, depends on the working fluid and temperature. Note that this term is positive as shown by the Clapeyron relation

$$\frac{dP}{dT} = f'\ (T) = \frac{\lambda}{T(v_V - v_f)} = \frac{\lambda}{T v_{Vf}}$$

5. Increase dT_v/dT_s - By Eq. (A-8) this term is equal to $(1 - R_s\ dQ/dT_s)$ and is always positive with a stable system. Therefore, to maximize dT_v/dT_s requires that R_s be a minimum. (Note that all parameters in the term $A_1\ {}_aV_a\ f'\ (T_v)\ dT_v/dT_s$ are positive.)

6. Maximize the cubic expansion coefficient, β_a (see 2 above). This property, along with the compressibility, is set by the choice of working fluid and Eq. (A-9) shows its effect directly.

7. Optimize A_2 - For small changes in vapor temperature the last term in the numerator of Eq. (A-9) approaches 0, and it takes the form

$$\frac{d\eta}{dT_s} = \frac{A_2\ C_1}{A_2^2 + C_2} \qquad \text{where } C_1 \text{ and } C_2 \text{ are constants} \qquad \text{(A-10)}$$

A_2 has a maximum at $\sim \sqrt{C_2}$

However, Eq.(A-1) can be altered to show

$$\Delta P_a = \frac{1}{A_2}\left(k\ \Delta\eta - A_1\ \Delta P_v\right) \text{ where } \Delta P_a = P_a - P_{ext} \qquad \text{(A-11)}$$

Eq. (A-11) is always positive with increasing source temperature if the bellows feedback system is working at all. Therefore, very small values of A_2 may cause an exceedingly high pressure in the auxiliary fluid bellows. For a typical metal bellows this forces a trade off between the spring constant and the bellows allowable pressure.

8. Minimize the spring constant, k - While this obviously improves performance the preceding argument should be heeded before choosing a minimum wall thickness bellows to minimize the spring constant.

The design of the feedback control system requires sizing of the bellows and the auxiliary fluid reservoirs. Since the purpose of this program was to test a design based on the work done in [12] the sizes used in that study were used for guidelines even though no attempt had been made to optimize the function of A_2 for various values of bellows spring constants and for values of V_a, β_a and κ_a used in the design.

With the assumption that the vapor temperature undergoes a negligible change, the small bellows operating pressures can be calculated using Figure A-2 and the following modified form of Eq. (A-11)

$$\Delta P_a \leq \frac{1}{A_2} k \Delta \eta \qquad \text{(A-13)}$$

The required stroke is 0.381" and assuming an initial pressure of 1 atm.

$$\text{then } \Delta P_a \leq 14.7 + .381 \frac{k}{A_2}$$

The equality is plotted in Figure A-3.

The general trend of the curves indicates a desire to choose a bellows with a low spring constant. This dictated looking for a bellows combination whose spring constant was low but one where the allowable pressure was high enough.

Typically, the convoluted type bellows have very high spring constants (> 100 lb/in). A bellows with a spring constant of 100 lb/in would operate at over 700 psi to get maximum sensitivity and the required stroke. A bellows type called the "nesting ripple" has low spring constants. However, the thin wall construction limits the maximum working pressure on small diameter bellows to about 200 psid. Because of this pressure limitation the optimum size (A_2) could not be used. Consequently, an off-the-shelf size was picked with an effective area of .075 in and a spring constant of approximately 15. This combined with the outer bellows spring constant of 16 gave a total of 31 lbs/in. These values set the minimum design sensitivity at $d\eta/dT_s$ = 0.039 in/°F and the maximum required working pressure at 175 psid.

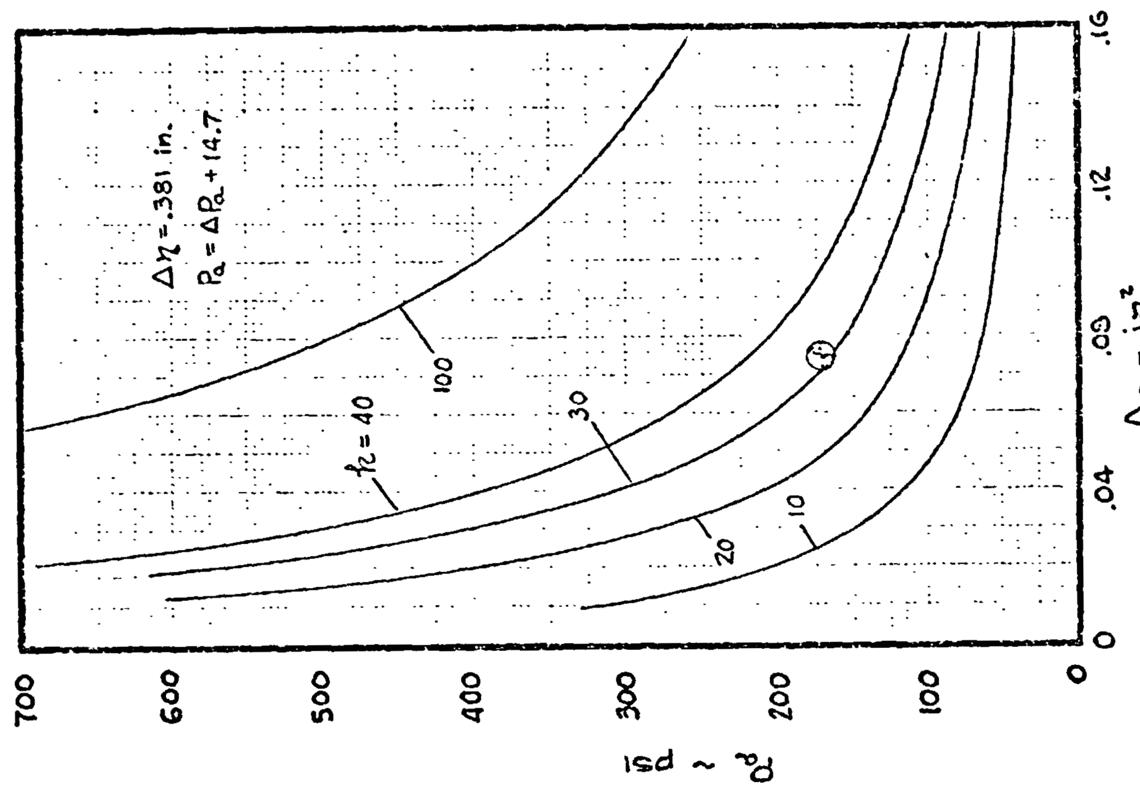

FIGURE A3 - Small Bellows Effect on Auxiliary Pressure

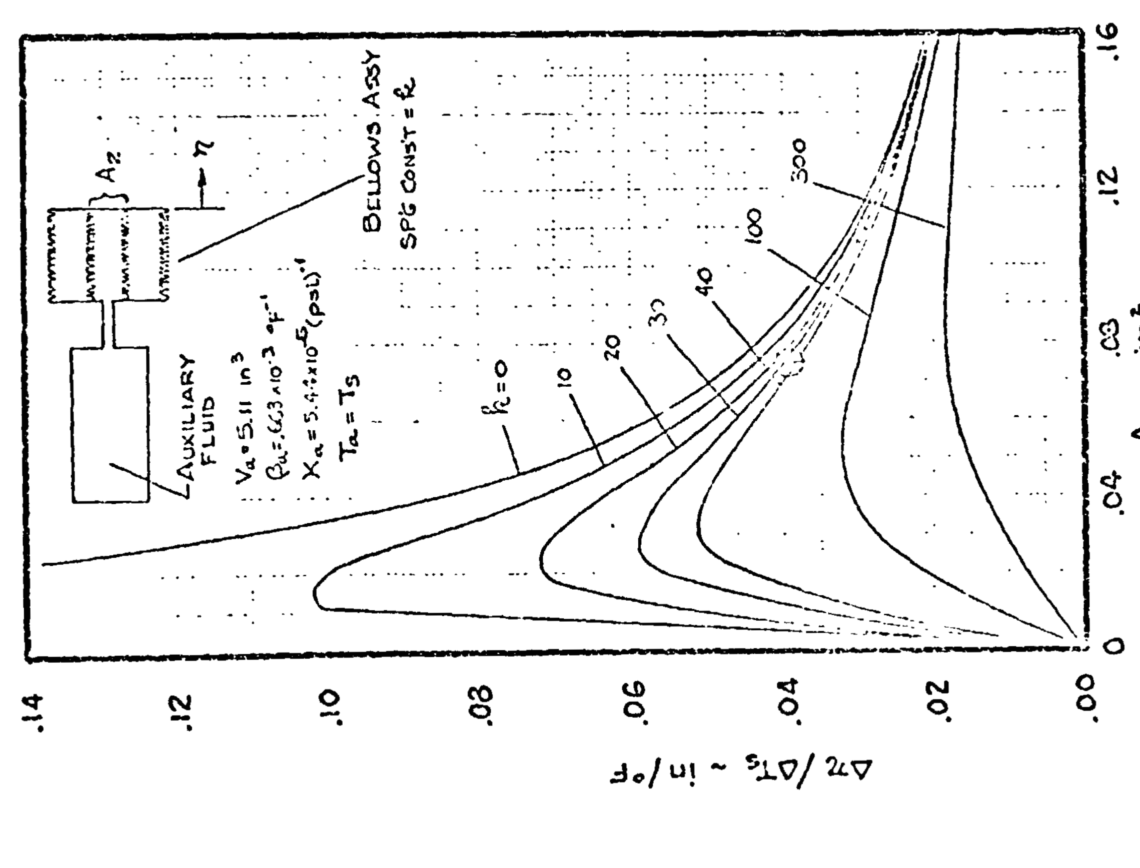

FIGURE A2- Small Bellows Effect on Control Sensitivity

One of the prime considerations of the bellows/heat pipe system is the pressure differential across the large bellows face. A heat pipe typically operates in an internal pressure range that is different than ambient. The pipe in this program was designed to operate between approximately 30-100°F corresponding to a total internal heat pipe pressure of about 0.2-4.0 psia. This means the external force on the bellows could be as high as 14.5 x 4.89 = 71 lbs. requiring an auxiliary pressure of 71/.075 = 950 psi just to overcome the atmoshperic pressure. For a vacuum application the internal pressure of the working fluid would drive the bellows wide open rendering it useless. This problem was overcome by encasing the bellows assembly in a sealed chamber. The pressure in the chamber could be adjusted so that at the desired initial conditions the bellows could be set at any position. The chamber was made of plexiglas so the bellows motion could be observed. An undesirable feature of the original system was the inability to externally adjust the auxiliary bellows fluid pressure once the system was sealed. This problem was alleviated by installing a screw piston in the auxiliary line. These two additions to the original design gave a good deal of versatility during testing.