Automatic Thermal Control Switches

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Two complementary types of flexible connection thermal control switches are described in detail. Both preflight and flight test results that demonstrate the successful operation of the switch that is presumed to have the more general application are presented. Finally, a more extensive postflight thermo-vacuum test that reveals the natural proportional control feature of the tested switch is included. The flight test of the switch was performed in the Get-Away Special (GAS) container which was flown on the third Shuttle flight.

Introduction

THE subject thermal switches have been designed and manufactured and the switch that is expected to have the more general application has been tested in a thermo-vacuum chamber and in the first flight of the GAS package on the third Shuttle flight (STS-3). Both switches are described, and the performance of the tested switch is given in detail. Incidental to the thermal switch development was the development of the flexible heat pipes which are used as heat carriers in the tested switch. Space limitations preclude a discussion of the heat pipe development in this paper.

Purpose of the Switches

The objective of the thermal switch development program was to provide an off-the-shelf thermal control device that will: 1) control heat flow between any two adjacent structural components that are otherwise isolated from each other; 2) accommodate limited motion between the two components; 3) require no electric power; 4) need no outside control; 5) provide the maximum amount of self-contained control logic; and 6) provide a thermal transport capacity of 5 to 50 W.

Description of the Switches

There are two types of switches (switch A and switch B), each of which is operated by a pair of phase change motors that exerts the required mechanical force when their temperatures exceed the phase change temperature of the working fluid within. The actuating motor is a special form (sealed for vacuum operation) of the phase change capsule used to control the coolant temperature in automobile engines. The switches are shown diagrammatica;ly in Figs. 1 and 2. These sketches are simplified to clarify the principles involved.

Switch B (Fig. 2) is "closed" at low (below phase change) temperature so that heat will flow from the source plate (to which the connector is attached) through the heat carriers to the payload item to be thermally controlled (to which the switch is attached). This switch opens when the payload item exceeds the motor fluid phase change temperature and heat promptly ceases to flow. A typical application would have the connector of switch B attached to a space radiator, and the switch proper attached to a piece of internal structure. The purpose would be to allow heat (e.g., solar energy) to flow into the internal structure only until that structure reached the phase change control temperature of the switch. This would provide a regulated heat input to the payload item.

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Switch A (Fig. 1) is the reverse of switch B; that is, heat flows through it when the plate to which the switch is attached is above the phase change temperature, and the heat flow is interrupted when the switch plate goes below the phase change temperature. Switch A would typically be attached to an internal heat collector plate, with the connector end of the heat carriers attached to a space radiator. In this manner, heat would flow to the radiator only when the internal structure was above the phase change temperature, so the payload would be protected against excessive heat loss when the radiator looks at deep space (with no sun or Earth view) for prolonged periods. Note that switch A is the one that is expected to have more general application and, therefore, was selected for the ground and flight testing.

Of course, an "AND" logic can be introduced by placing a type B switch on one plate and a type A switch on the other (in lieu of the normal connector attachment). In this manner, heat would flow only when the switch B plate (i.e., the space radiator) was below the phase change temperature (the radiator is not exposed to the sun), AND the switch A plate (heat collector plate within the payload) had a temperature in excess of the phase change temperature with the result that no excess inflow or outflow of heat will occur.

The operation of the switches may be seen in Figs. 1 and 2. These schematic sketches are designed to identify the important components as well as to clearly illustrate the principles upon which the switches operate. In both switches, the phase change power units when heated above the phase change temperature, initiate the desired switch action: to open or close the thermal circuit. This is accomplished by a lever arm that, actuated by the phase change motor, either makes or breaks the contact between the heat collector plate and the switch saddle. The only real difference between the two switches is the reversal of the mechanical action so that one switch would open when the other closed if they were both fixed to the same surface. One can easily envision the numerous practical applications of the switches for space use either singly or in combination. A typical application of switch A will be seen in the Get-Away Special (GAS) flight test described below.

Either switch will occupy a space approximately $4.0 \times 2.75 \times 2.0$ in. The switches alone weigh about 1.9 lb each. The aggregate weights are:

1) Switch A with five heat pipes and connector = 3.26 lb.
2) Switch B with five heat pipes and connector = 3.32 lb.

Two types of heat carriers are available. The primary carriers are (up to) five flexible heat pipes. These are used when maximum heat-transfer rate is required. The second type is a thermal-insulation-wrapped, flexible, copper cable (i.e., welding cable). The cables are used when the total required heat-carrying capacity is low or when the heat can be dissipated over a longer time period—hence at a slower rate. The cable carriers are more rugged and less costly by far than the heat pipes so they should always be used when the heat carrying capacity is sufficient.

A test was run to determine the heat-carrying capacity of the thermally insulated copper cable. A 0.25-in. copper cable (composed of 49 individual 0.032-in. strands) 6.75 in. long was tested. The results showed that the heat-carrying capability of the cable was within 6% of the value calculated on a purely theoretical basis (assuming no radiative or convective losses), considering the sectional area to be the sum of the areas of the 49 strands and the length to have its true value. The measured conductance was 0.0306 W/°F. The theoretical value was 0.0325 W/°F. Thus, for the range of the test, the five copper cables of one switch would carry 0.153 W/°F temperature difference across the cable length.

Preflight and Flight Tests

A sketch of the thermal switch experiment in the GAS preflight and flight verification tests is given in Fig. 3. Briefly, it consists of a $3 \times 7 \times 0.25$-in. aluminum plate with the thermal switch attached to one side and a 40-W resistance heater to the other. This package is wrapped in Dacron felt insulation with multilayer insulation (MLI) on the outside and is attached through fiberglass phenolic thermal isolators to the support structure in the GAS container. The connector end of the switch heat-carrying system (heat pipes) is connected to the GAS container lid. Originally, it was intended that the GAS lid be uninsulated to form an effective space radiator. However, a shortage of battery power dictated the necessity for flying with the GAS lid thermal insulation cap in place. This configuration was used in both the preflight thermo-vacuum test and the flight test. The pressure in the GAS container for both tests was 1 atm.

Thermo-Vacuum Test

The thermo-vacuum test results are shown in Fig. 4. The switch plate heater was turned on at about 8.75 h and the switch plate temperature rapidly rose to 45°C. The switch phase change motor actuates at about 36°C, and this actuation is apparent at 8.90 h. Thus there is an inferred gradient between the switch plate and switch body at the motor capsule location of 10°C. Once closed, the switch channels heat to the GAS container lid. The lower curve of Fig. 4 shows the resulting rise in the lid temperature. Note that if the lid were free to radiate to the cold chamber walls (−170°C), the flat portion of the switch plate temperature would reach an equilibrium (horizontal) state. The presence of the container lid insulation cap results in the substitution of the entire heat mass of the container and payload as a heat sink in lieu of the desired radiation. At about 15 h, the 40-W heater is turned off and the switch plate temperature (switch is closed so heat continues to flow from the switch plate to the GAS container lid) rapidly drops. At about 15.45 h, the switch starts to open so the rate of cooling of the switch plate decreases. The fact that this slope change is gradual rather than sudden offers the first hint of the proportional control characteristic of the switch that is discussed below in the postflight switch thermo-vacuum test. There is, of course, a concurrent decrease in the container top temperature resulting from the radiation of the entire container to the cold chamber walls combined with the cessation of the heater input power.

STS-3 Flight Test

The results of the flight test of the thermal switch are recorded in Figs. 5 and 6. The same restrictions on instrumentation that were mentioned for the preflight test apply to the flight package. The most significant difference between the preflight and flight tests is the generally higher temperature levels reached in flight. This is believed to lead directly to the anomalous "humps" in the thermal switch plate temperature curves of Figs. 5 and 6. These "humps" start at 65-70°C switch plate temperature. Since these temperatures were never reached in the preflight test, no "humps" were observed.

A possible explanation, although data frequency is insufficient to confirm these suppositions beyond reasonable doubt, assumes that the steepening slope of the switch plate temperature curve (start of "hump") results from a partial drying-out of the heat pipes as the temperature approaches 70°C. Thus the heat input being constant (40 W) and the effectiveness of the heat pipes in transmitting this heat decreasing, the switch plate temperature necessarily rises at

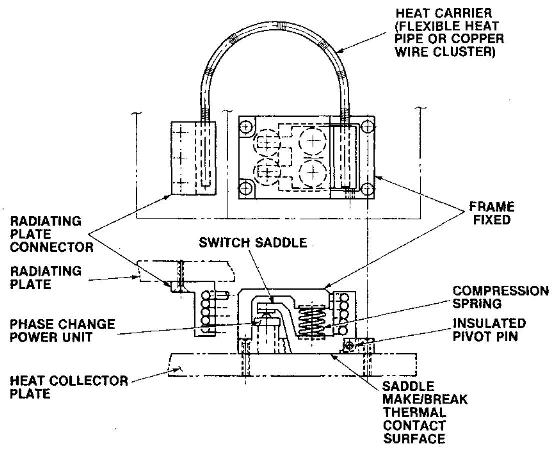

Fig. 2 Type B switch (open when collector plate is hot).

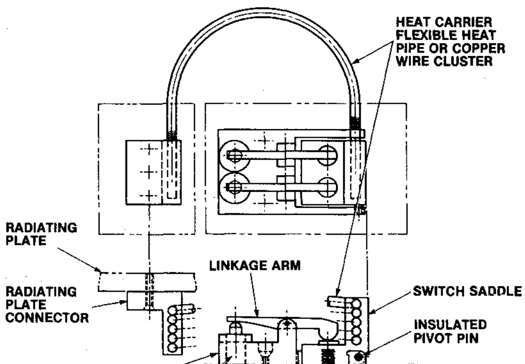

Fig. 1 Type A switch (closed when collector plate is hot).

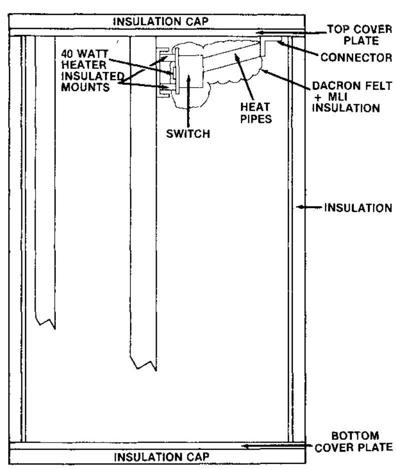

Fig. 3 Schematic of GAS container showing thermal test rig.

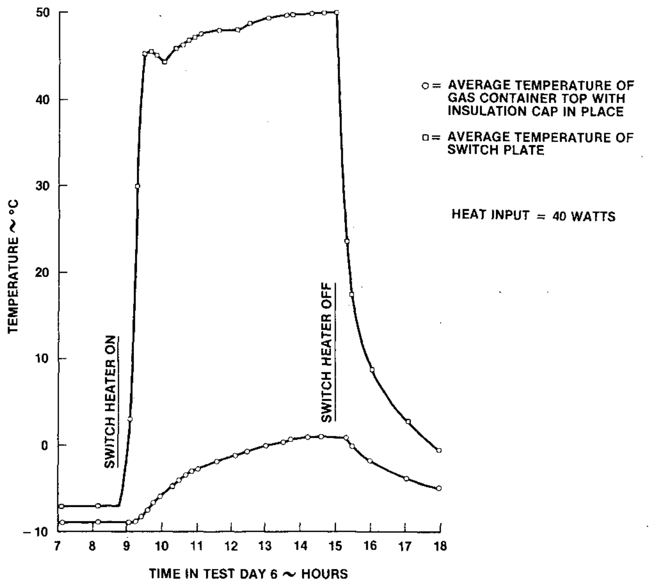

Fig. 4 Thermal switch GAS preflight thermo-vacuum test.

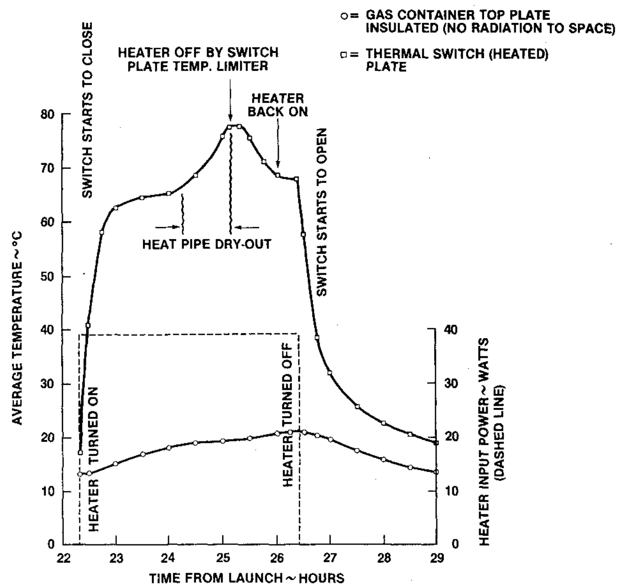

Fig. 5 STS-3 flight test of thermal switch in GAS container (first turnon).

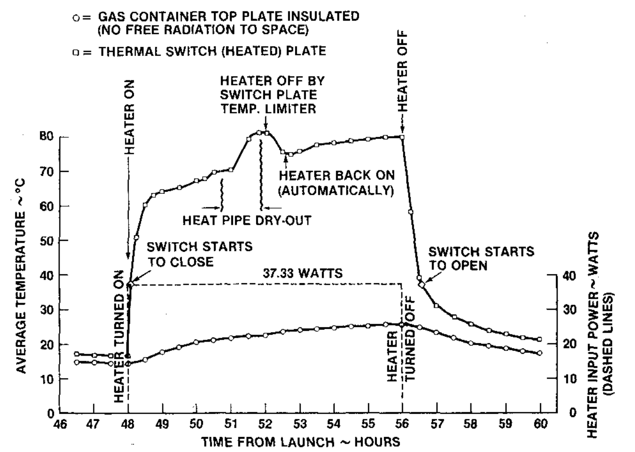

Fig. 6 STS-3 flight test of thermal switch in GAS container (second turnon).

the faster rate. At about 80°C, the switch plate temperature limiter shuts off the power. The heat pipes, recovering from partial dry-out, now carry the heat off at a normal rate so the negative slope or the down side of the hump is very similar to the positive slope which precedes it. Somewhere near the slope change point on the down portion of the hump, the heat pipe temperature limiter turns the heater power back on. Therefore (see the second turnon case in Fig. 6), the temperature of the switch plate starts up rather steeply (52.75 to 53.5 h) and then changes to a noticeably less steep slope after 53.5 h. This is assumed to result from a slow recovery of the liquid/vapor balance (hence, increased heat transmitting efficiency) of the heat pipes. This satisfactorily explains why the slope after turnoff (at 26.3 h in Fig. 5 or 56 h in Fig. 6) is much steeper than it appears to be when the temperature limiter turns the power off. One must bear in mind that it is also possible to explain this slope difference by assuming that the temperature limiter cycled on and off during the periods shown (in Figs. 5 and 6) as "heater off by limiter."

Although the lack of capacity for more detailed instrumentation in both the preflight and flight thermal switch tests prevented the obtaining of detailed behavior of the switch, the results do clearly indicate that the switch performed correctly.

Thermo-Vacuum Test of Switch A

The type A automatic thermal switch (passes heat when switch base temperature exceeds phase change temperature of actuation capsules) was tested in a thermo-vacuum chamber. The test package, shown in Fig. 7, consisted of an upper aluminum plate with resistance heaters producing up to 40 W on the upper surface and the thermal switch fixed to the under surface, a "bottom" aluminum plate to which the switch system connector is fixed, and an LN-cooled plate to which the bottom plate radiates. The lower bottom plate and upper cooled plate surfaces are painted black for maximum radiative heat transfer. The switch heat pipes and connector are the identical unit that was used in the two tests discussed above.

The test package was insulated with 1/8-in. Dacron felt and MLI as shown by the dashed lines in Fig. 7. Not shown is the MLI wrap around the five heat pipes. Figure 8 shows the test rig in the test chamber, the walls of which were maintained at room temperature throughout the test. Each of the two aluminum plates was instrumented with six thermocouples. Each of the five heat pipes had a thermocouple mounted on each end and one in the middle. There were four thermocouples on the LN-cooled plate, one on the switch saddle and one on the connector, bringing the total number of thermocouples to 33.

The results of the test are summarized in Fig. 9. Average temperatures on the upper plate, lower plate, saddle, connector, and the LN-cooled plate are plotted as they approach equilibrium. Four power levels in the upper plate heaters are run with the chamber evacuated (10, 20, 25, and 30 W) and four (10, 20, 30, and 40 W) with the chamber filled with dry nitrogen (1 atm pressure). Figures 10 and 11 record the average equilibrium temperatures at the five locations covered in Fig. 9 as well as the average temperatures of the evaporator, midpoint, and condenser ends of the heat pipes for, respectively, a vacuum and 1 atm of dry nitrogen.

There are two specific limitations inherent in the test. The first, the presence of gravity, was unavoidable and was offset as much as possible by "tipping" the experiment to assure that the condenser ends of the heat pipes were slightly above the evaporator ends. The second, the physical restriction on the cold plate size (sic, radiation area), was a simple cost factor. The 12 × 14-in. LN-cooled plate was available at no cost so the resulting 8 × 12-in. experiment package limitation restricted the area of the bottom (radiating) plate to only two-thirds of a square foot. This "choked" the heat dissipation

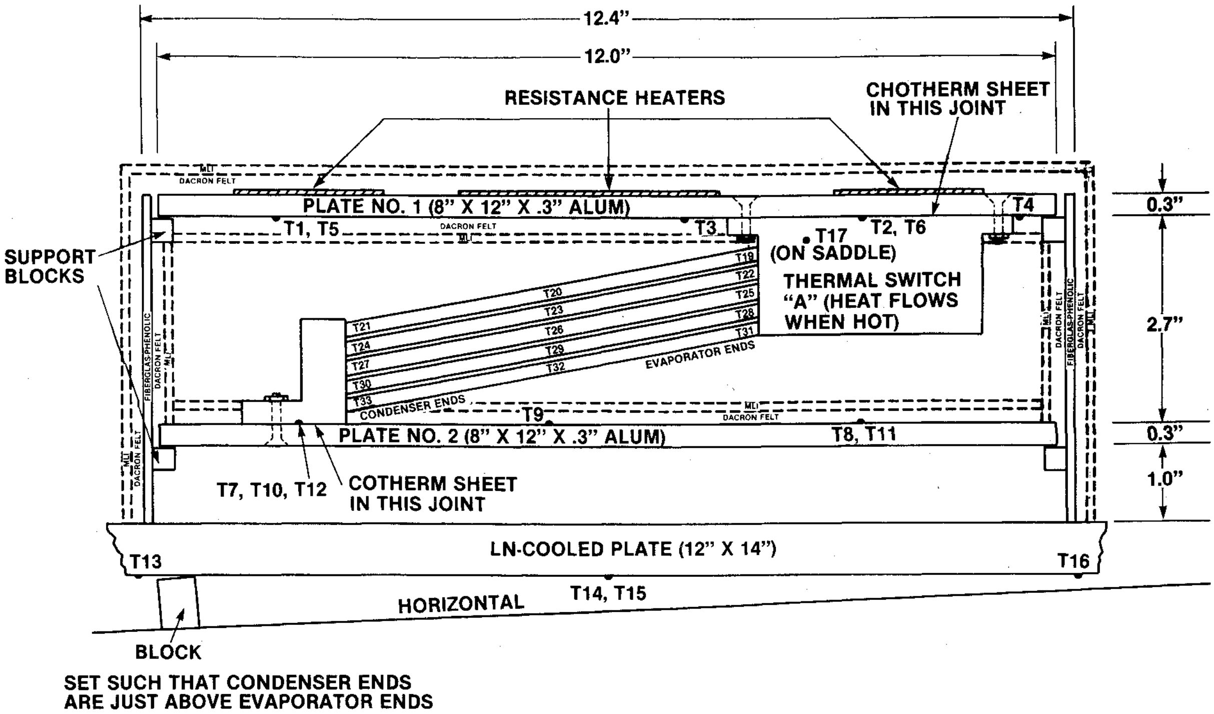

Fig. 7 Sketch of the test rig.

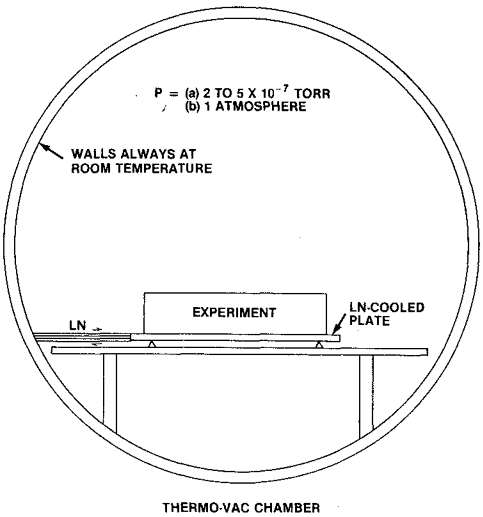

Fig. 8 Experiment in test facility.

and raised the temperature of both the upper and lower plates, causing the heat pipes to work in a temperature region above the design values. As expected, their efficiency was adversely affected.

It is believed that, if the radiator area was sufficient to discharge 50 W at an acceptable (50°C) switch and heat pipe temperature, the pipes could handle this load. The pipes were designed for operating temperatures of 10 to 50°C, with optimum performance at 20°C. Each pipe has carried 10 W with ΔT (evaporator to condenser ends) of 7.5 to 15°C.

One of the concerns in the design of the thermal switch was the possibility of switch cycling. That is, if the system is capable of transporting through the heat pipes 40 W, and only 10 W are input to the switch plate, it was feared that the switch would remain open (no heat flow) until its temperature reached the required 36 to 40°C that would close it. Heat would then flow at a rate of 40 W or 30 W in excess of the replacement rate until the switch cooled down to the point where it would open to stop the heat flow. This cycle could repeat indefinitely, doubt lying only in the extent of the period. There was concern that periodic cycling might prove to be a life-limiting factor for the switch or its phase change motors.

The test results were fortuitous in that the switch, subjected to lower-than-capacity heat rates, does not cycle at all. Rather, it becomes a proportional control device. The difference between the input heat rate and the system dissipation capacity is controlled by the force exerted on the switch saddle by the phase change motors. Thus, when only 10 W is imposed through the heaters, the switch temperature rises until the material in the actuating capsules starts to change phase. The continuing increase in temperature exerts a continuing increase in pressure on the switch saddle (the thermal contact joint) which, in turn, increases the joint efficiency of the saddle (hence, heat-transfer rate). When the pressure reaches the point at which exactly the input heat can pass through the switch, the process freezes or remains in dynamic equilibrium. This conclusion was suspected from the results of the GAS preflight and flight tests. It was confirmed, however, in the switch thermo-vacuum test in which a series of temperature readings 30 s apart was taken at three different heat load levels. In all cases, no indication of cycling was apparent.

In summary, this test resulted in the following conclusions:

1) The thermal switch performed well throughout the range of power inputs, limited only by the radiating capacity (surface area) of the switch plate.

2) The thermal switch does not cycle when subjected to heat loads below the heat-carrier and/or radiator capacity but acts as a porportional control.

3) The heat pipes tended to act as "boilers" at the lowest power input levels, but as effective heat pipes at the higher levels.

4) The maximum capacity of the heat pipes was probably not reached in this test as a result of the restricted radiation area.

5) The same switch and heat carriers (without having been dismantled at all) were used in the GAS flight verification payload thermo-vacuum test, the STS-3 flight of the GAS package, and the switch system thermo-vacuum test discussed here.

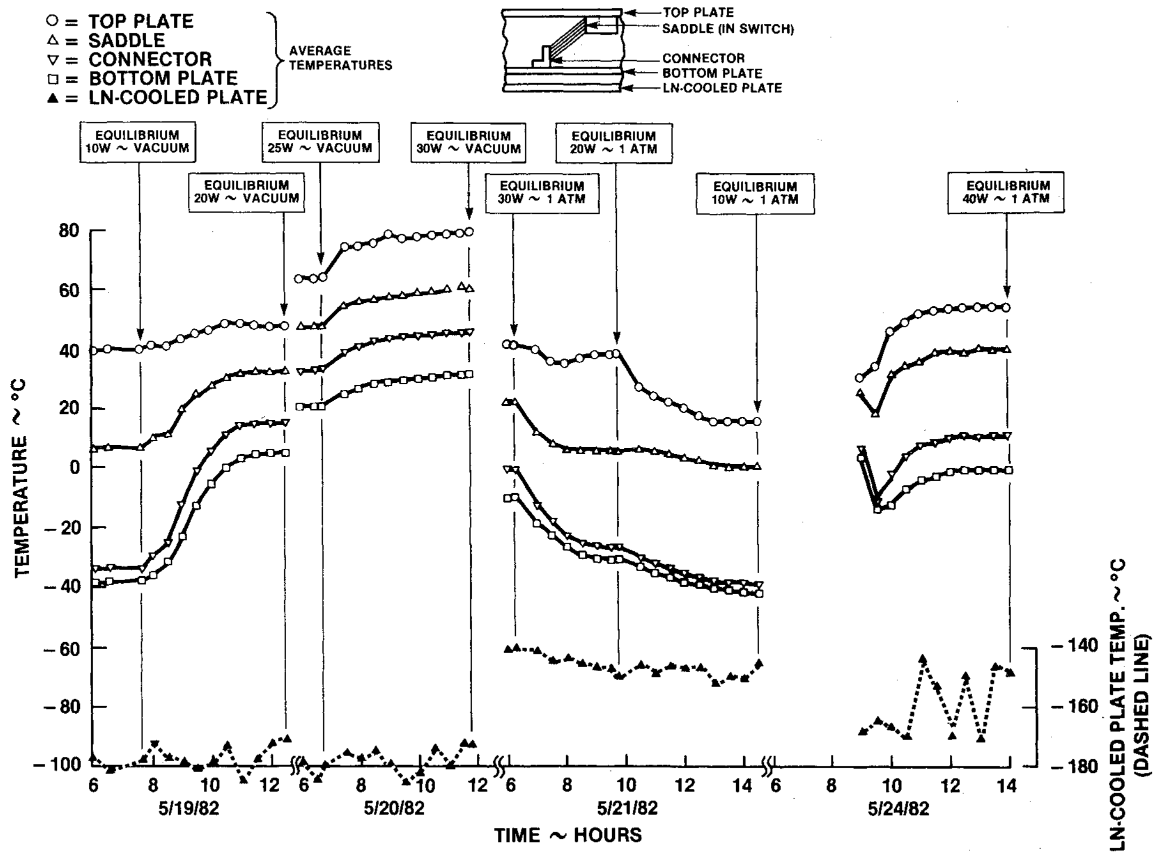

Fig. 9 Summary of thermal switch thermo-vacuum test.

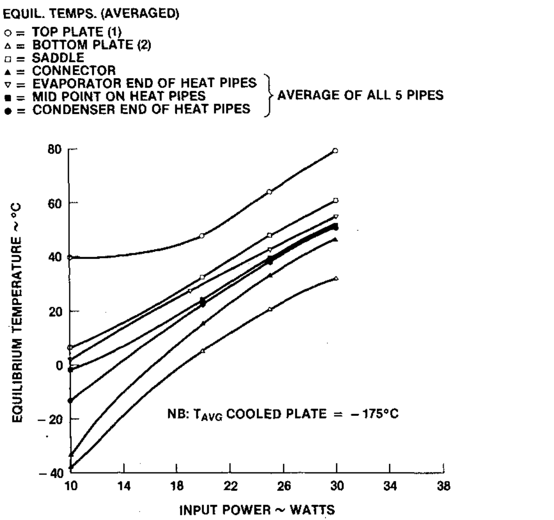

Fig. 10 Equilibrium temperatures vs heater input power for vacuum test.

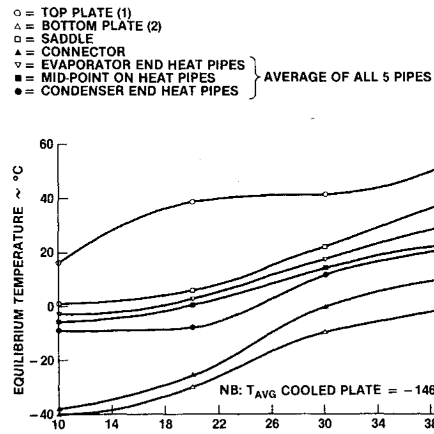

Fig. 11 Equilibrium temperatures vs heater input power for 1-atm test.

Discussion

Unfortunately, limitations were placed on the thermal switch tests for both the flight and ground test cases. In the case of the GAS container tests (both preflight in the thermo-vacuum chamber and on the STS-3 flight), the limitation arose from the shortage of available power for heating the GAS payload. This shortage made it necessary to alter the original plan to fly with an uninsulated GAS top cover plate that would have provided better than 2.5 ft^2 of radiating surface and to fly, instead, with the cover insulation cap. Thence, for both GAS tests, the switch power had to be accommodated almost entirely by the heat mass of the GAS container and payload. The result was an appreciable rise in the operating temperature of the system, so the heat pipes were working outside their design limits and a loss of efficiency was experienced.

The ground thermol-vacuum test was similarly limited by the need (cost factor) to use an existing 12×14-in. LN-cooled plate that limited the radiating plate area to about 0.667 ft^2. This area limitation also resulted in an appreciable rise in the heat pipe operating temperature with an attendant loss of efficiency. Figure 12 is included to provide an estimate of the power-dissipating capacity of a 2.5-ft^2 radiator, the approximate area available for radiation to space on the GAS container when the top is uninsulated.

The upper curve in Fig. 12 is the calculated flight case with the 2.5-ft^2 radiator looking at deep space (no sun, no Earth).

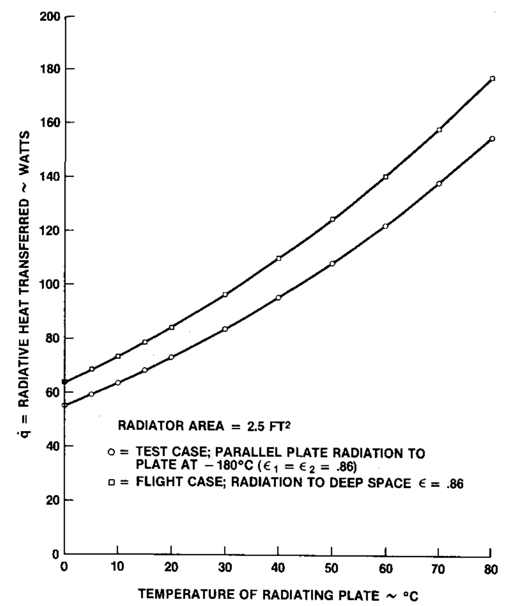

Fig. 12 Heat dissipation vs radiator plate temperature (theoretical).

The lower curve represents a simulated test case in which the radiator and LN-cooled plate areas (for parallel plate radiative transfer) are both 2.5 ft^2, and the cooled plate temperature is −180°C. In both cases the surface emissivity is 0.86. These data demonstrate that the switch system power dissipating capacity would be comfortably in excess of the 50-W (estimated) capability of the heat pipes even at a radiating plate temperature as low as 0°C. Moreover, the implication is quite clear that the thermal switch system will control a heat collector plate temperature to within 5 to 10°C of the motor control (phase change) temperature provided only that sufficient radiation area be available to avoid "choking" of the heat dissipation by the radiator.

Note (Fig. 10) that, for the abbreviated radiator area of the thermo-vacuum test (0.667 ft^2), this criterion can only be met for heater input powers up to about 20 W. Thereafter, the radiating plate "choking" effect requires that the system run at higher temperatures. In the 1-atm case (Fig. 11), the same criterion can be met for power levels up to about 35 W.

Conclusions

From the work reported here it is concluded that:

1) The type A thermal switch will operate effectively in either Earthbound or space applications.

2) This switch behaved as a proportional control device and was not subject to the undesirable rapid thermal cycling that could adversely effect its life span as well as its efficiency.

3) The switch transmitted 30 W of power in a vacuum and 40 W in a 1-atm environment under adverse test conditions (inadequate radiating area).

4) The close similarity in the working principles of the two types of switches leads to the presumption that the untested switch (type B) will perform in the same manner as type A, though the validity of this presumption has not been demonstrated.

Acknowledgment

Much of the credit for the successful performance of the thermal switches described here must go to the co-inventor, Joseph Cunningham (NASA/GSFC).

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