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PASSIVE REJECTION OF HEAT FROM AN ISOTOPE HEAT SOURCE THROUGH AN OPEN DOOR

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16. Abstract

The isotope heat-source design for a Brayton power system being developed at the Lewis Research Center includes a door in the thermal insulation through which the heat can be passively rejected to space when the power system is not operating. The results of an analysis to predict the heat-source surface temperature and the heat-source heat-exchanger temperature during passive heat rejection as a function of insulation door opening angle are presented. They show that for a door opening angle greater than 20°, the temperatures are less than the steady-state temperatures during power system operation.

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PASSIVE REJECTION OF HEAT FROM AN ISOTOPE HEAT SOURCE THROUGH AN OPEN DOOR

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SUMMARY

An important element in the design of isotope heat sources for space power systems is the provision for safe, passive heat rejection when the power system is not in operation. In the case of the isotope Brayton power system currently being developed at the Lewis Research Center the primary method of heat rejection is through an open door in the insulation. The insulation door in the reference design is behind the heat exchanger, so the thermal energy must be radiated from the heat-source unit to the heat exchanger, transferred through the heat exchanger, and radiated from its back side through the insulation door opening to space.

A thermal analysis was performed for this design to predict the temperatures of the heat-source-unit surface and the heat-source heat exchanger during passive heat rejection. The results presented give the temperature profile across the rear surface of the heat exchanger and the temperature difference through the heat exchanger and between the heat exchanger and the heat-source unit as a function of the angle of the insulation door opening. For a fully opened door (>90°), the temperature difference between the hot and cold sides of the heat exchanger was predicted to be 133 K (240° F). The temperature difference between the heat-source-unit surface and the heat exchanger for this case was 119 K (215° F) and resulted in a heat-source-unit temperature of 1061 K (1450° F). The results also show that, if the insulation door opening angle is greater than 20°, the heat-source-unit surface hotspot temperature will be lower than the steady-state temperature (~1260 K; ~1800° F) of that surface during power system operation.

INTRODUCTION

A Brayton cycle power system for generation of auxiliary electric power in space is being developed at the NASA Lewis Research Center (refs. 1 and 2). The energy source for this system would be radioisotopes. In the reference design (see ref. 3) the isotope fuel is contained in capsules which are assembled in a planar array onto a support structure. The array and its support are referred to as the heat-source unit (HSU) and are mounted in a heat-source reentry vehicle (HSRV), as shown in figure 1. The reentry vehicle is designed to provide safe, intact reentry of the HSU should atmospheric entry occur. During power system operation the thermal energy of the HSU is radiated to the heat-source heat exchanger (HSHX), where it is transferred to the Brayton working gas. The HSU and HSHX are surrounded by thermal insulation, as shown in figure 1.

During periods in which the Brayton power system is not operating, all the thermal energy generated in the isotope fuel must be rejected. The Brayton energy source subsystem design must therefore include a means for rejecting this thermal energy while maintaining the temperatures of the heat sources, the support structure, the heat-source reentry vehicle, and the heat-source heat exchanger at acceptable levels. For safety reasons the means for rejecting the heat should be passive and capable of maintaining these temperatures at or below their steady-state operational levels. The primary means of passive heat rejection used in the reference design is to include a door in the insulation which surrounds the heat-source heat exchanger and heat-source unit (see fig. 1). The insulation door remains in the opened position when the power system is not operating. The thermal energy is transferred by radiation from the HSU to the HSHX and by combined conduction and radiation through the HSHX, and is then radiated from the back side of the HSHX through the opening in the insulation.

In a space application, there could be a limitation on maximum opening angle of the insulation door due to limitations of available space within the power system compartment of the vehicle. In such a situation, the thermal energy would be rejected from the back of the HSHX through the partially opened door, either to space or to the spacecraft compartment interior. As the angle of the door opening would be decreased, the temperature level of the HSHX and HSU and the temperature gradients induced across their surfaces would be increased. The analysis presented in this report is intended to predict these temperatures as a function of the insulation door opening angle. For simplification, the calculations are arbitrarily separated into two parts. First, the temperature distribution across the back side of the HSHX is calculated as a function of insulation door opening angle. This is done by assuming that the rejected heat flux is uniform across the HSHX. In the second part of the calculations, the temperature distribution in individual tubes of the HSHX core and on the HSU surface is calculated.

HEAT REJECTION FROM HEAT-SOURCE HEAT EXCHANGER

Thermal Model

If the distribution of heat transferred from the rear side of the HSHX is assumed, the temperature distribution across the rear surface can be calculated without considering the details of the heat transfer from the HSU to the HSHX and through the HSHX to its rear surface. The approach taken here is to assume that the total amount of heat generated in the HSU is radiated from the rear side of the HSHX (i.e., there are no losses through the insulation or HSRV structure) and that the heat flux rejected from the HSHX is uniform across the rear surface. This will yield a pessimistic (high) estimate of the hotspot temperature.

The thermal model is shown in figure 2. The cross section of the insulation enclosure (ABCD in fig. 2) is assumed to be square with a width equal to the diameter of the actual configuration. The insulation door (BEFC in fig. 2) is also assumed to be square. In the thermal model the surface representing the rear side of the HSHX (ABCD in fig. 2) is divided into a 10 by 10 matrix of nodes. The nodes shown with crosshatching approximate the 1.24-meter- (49-in. -) diameter core of the reference HSHX from which all the thermal energy is assumed to be rejected. The uncrosshatched nodes are assumed to be adiabatic. The surface representing the inside of the partially opened insulation door (BEFC) is also divided into nodes, all of which are assumed to be adiabatic. The surfaces ABE and DCF in figure 2 represent the space vehicle compartment walls and are also assumed to be adiabatic. Surface AEFD is divided into nodes with specified emissivity and temperature in order to represent the energy sink. In this analysis it is assumed that all nodes on surface AEFD are black and at 256 K (0° F). All surfaces in the thermal model are assumed to be diffuse and have wavelength-independent radiation properties.

A radiation exchange analysis of the enclosure was performed with this model to determine the temperatures of the nodes representing the HSHX core as a function of the insulation door opening angle φ. Reflection and reradiation of energy by the adiabatic surfaces were included. Since conduction exchange between nodes was neglected and since surfaces ABE, DCF, and BEFC were assumed to be adiabatic, this model should have yielded pessimistic (high) predictions of the HSHX hotspot temperature.

Results

In figure 3 the temperature distribution along the HSHX centerline is shown as a function of the insulation door opening angle. The calculations were made with the assumption that a total of 25 kilowatts-thermal is rejected with uniform flux from the nodes which represent the HSHX core shown in figure 2. The emissivity of these nodes was assumed to be 0.85, that is, it was assumed that an emissive coating is used on the HSHX core on both sides.

As expected, the temperature distribution is more even and the temperatures are lower for wide opening angles. As the angle of the door opening is decreased, the temperature of the HSHX surface near the door pivot increases rapidly for angles less than about 30°. The large temperature gradients shown for small door opening angles would in fact be somewhat reduced by conduction within the HSHX core, which was not included in this analysis.

These results were used as an input for the second part of the analysis, which is concerned with calculating the temperature gradient through the HSHX from the side facing the HSU to the rear side and the temperature difference between the HSU and the HSHX.

HEAT TRANSFER THROUGH HEAT-SOURCE HEAT EXCHANGER

Thermal Model

To predict the temperature distribution in the HSHX core, it was assumed that the core consists of closely spaced columbium - 1-percent-zirconium tubes, with 3.66-centimeter (1.44-in.) outside diameter and 0.114-centimeter (0.045-in.) wall thickness. The thermal model used, shown in figure 4, consists of a nodal network representing the cross section of one-half of one tube within the core. Thermal energy is radiated from the HSU surface to tube surface nodes 1 to 6. The heat is transferred by radiation across the inside of the tube and by conduction through the tube wall and is then radiated from nodes 7 to 12 to the sink. Radiative exchange and reflection of radiation between adjacent tubes and between the tubes and the HSU is included. The tube temperature is assumed to be symmetrical about the tube centerline, which bounds the nodal model, and the adjacent tube is assumed to have the same temperature distribution. Heat transfer in the longitudinal direction along the tube is neglected.

The energy flux from the HSU is taken to be equal to the total thermal generation rate of all the fuel in the HSU divided by the projected surface area of the HSHX core.

The effective sink (node 38) is assumed to be black and at an effective sink temperature obtained from the analysis of the heat rejection through the partially opened insulation door which is given in the previous section. That analysis resulted in the prediction of the temperature of each node T_i of the thermal model in figure 2 as a function of insulation door angle. These temperatures are used to calculate an effective sink temperature $T_{s,i}$ for each of the nodes in that model as follows:

$$T_{s,i} = \left(T_i^4 - \frac{q_i}{\epsilon_{HX}\sigma}\right)^{1/4} \tag{1}$$

where q_i is the flux rejected from node i, ϵ_{HX} is the emissivity of the HSHX core (0.85), and σ is the Stephan-Boltzmann constant. These sink temperatures are then used as an input boundary condition (temperature of node 38) for the thermal model in figure 4.

Results

Temperature distributions in the HSHX and on the HSU surface for the thermal model shown in figure 4 were calculated as a function of the sink temperature by using the CINDA-3G computer code (ref. 4). The results given in figure 5 are for an emissivity of 0.80 for the HSU surface, 0.85 for all exterior surfaces of the tubes, and 0.30 for all interior surfaces of the tubes. The effective tube temperature T_E for the hot side (the side facing the HSU) and the cold side (the side facing the sink) of the tube were calculated as follows:

$$T_E = \left(\frac{1}{\sigma}\frac{\sum_i A_i F_{ij} \sigma T_i^4}{\sum_i A_i F_{ij}}\right)^{1/4} \tag{2}$$

where A_i is the area of node i, F is the total radiation exchange factor, the subscript j refers to either the HSU or sink node, and the summation is over all tube surface nodes involved in the radiative exchange with node j.

The results in figure 5 can be related to specific locations on the HSHX for various insulation door opening angles by using equation (1) and the results of the analysis employing the thermal model of figure 2. This has been done for the hottest point, that is, the point on the centerline nearest the insulation door pivot and represented by node 2 in figure 2. These results are given in figure 6 as a function of insulation door opening angle.

As the insulation door angle is decreased, the results in figure 6 show that the HSHX and HSU hotspot temperatures increase rapidly. At an angle of about 20° the HSU hotspot surface temperature is in the range of temperatures of that surface during power system operation. As has been discussed, the present calculations are conservative (high) in predicting this hotspot temperature. It can then be concluded that, when the insulation door opening angle is 20° or greater and the energy sink is space, the temperature of the HSU surface will reach a steady-state value less than its steady-state temperature during power system operation.

CONCLUDING REMARKS

Heat-source heat-exchanger and heat-source-unit surface temperatures were predicted for passive heat rejection from the rear surface of the heat exchanger to space through an opened door in the thermal insulation. The thermal model shown in figure 2 was first used to determine temperature distributions across the rear side of the heat exchanger for various angles of insulation door opening. These results were then used as input to the thermal model shown in figure 4 to determine the temperature gradient through the heat-exchanger tubes and the surface temperature of the heat-source unit.

The results show a significant temperature gradient through the heat-exchanger tubes and from the heat exchanger to the heat-source-unit surface. For a sink temperature of 256 K (0° F) corresponding to the heat-exchanger rear surface radiating to space with the insulation door fully open, the effective temperature difference between the hot and cold sides of the heat exchanger was predicted to be 133 K (240° F). The temperature drop from the heat-source unit to the heat-exchanger hot side for this case is 119 K (215° F). Emissivities of 0.80 for the heat-source unit and 0.85 for both sides of the heat exchanger were assumed.

Considering the hotspot temperature shown in figure 6, it can be concluded that, when the insulation door is opened to space at angles greater than about 20°, the heat-source-unit surface temperature will be lower than the steady-state range of temperatures of that surface during power system operation with the insulation door closed (i.e., <1260 K; $<1800^{\circ}$ F).

Lewis Research Center,
National Aeronautics and Space Administration,
Cleveland, Ohio, October 8, 1971,
112-27.

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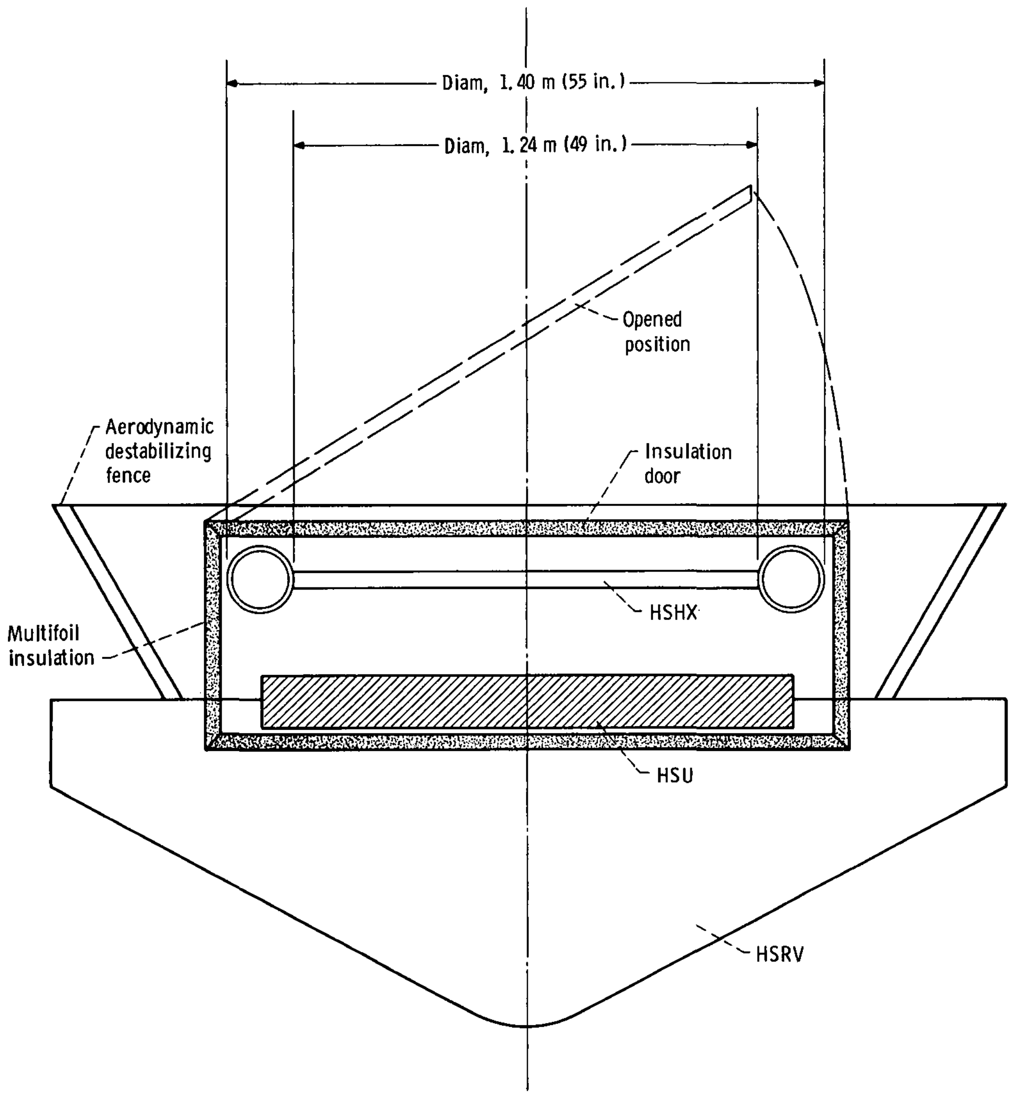
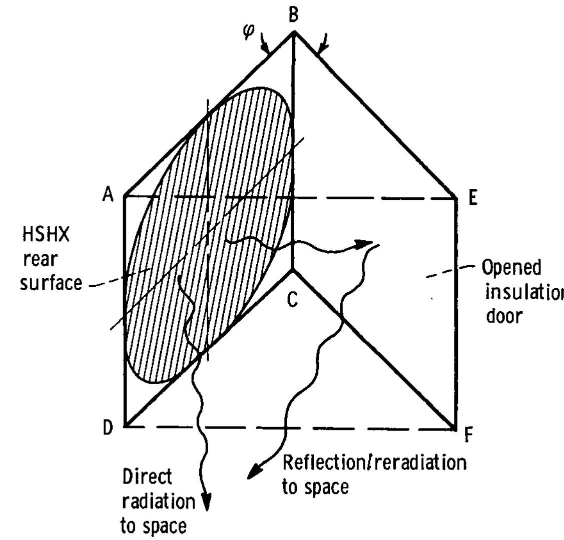
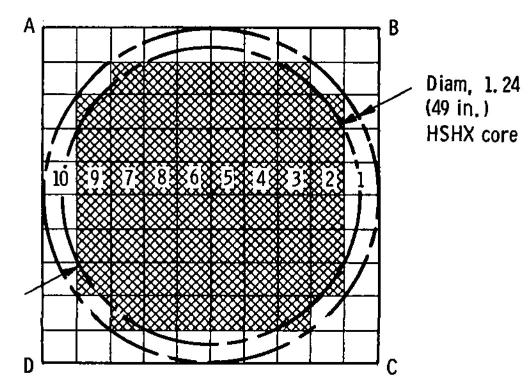


Figure 1. - Reference Brayton energy source subsystem.



(a) Thermal model enclosure.



(b) HSHX nodes.

Figure 2. - Thermal model for heat rejection from HSHX through insulation door opening.

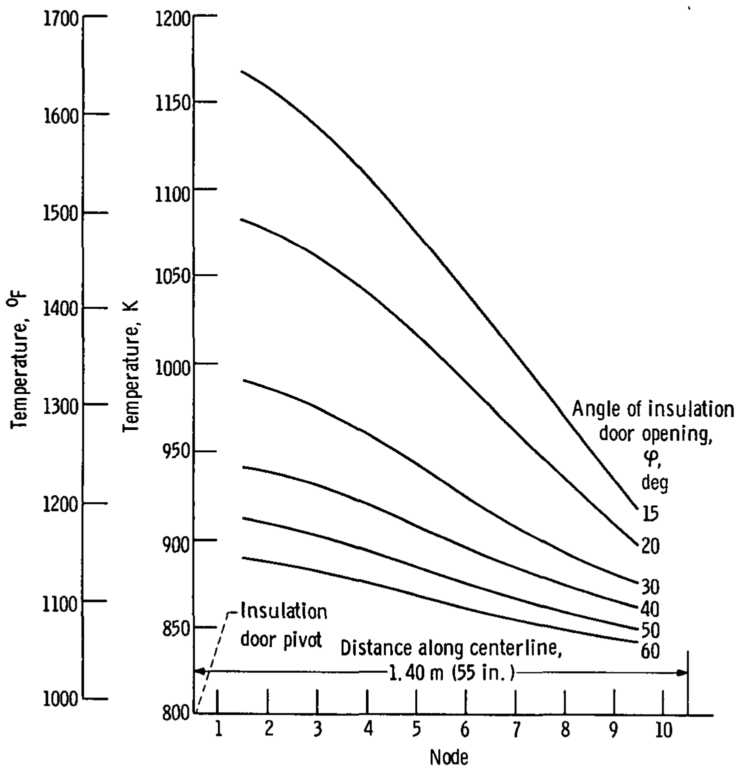


Figure 3. - Temperature of rear surface of HSHX. Heat rejection, 25 kilowatts-thermal; HSHX emissivity, 0.85.

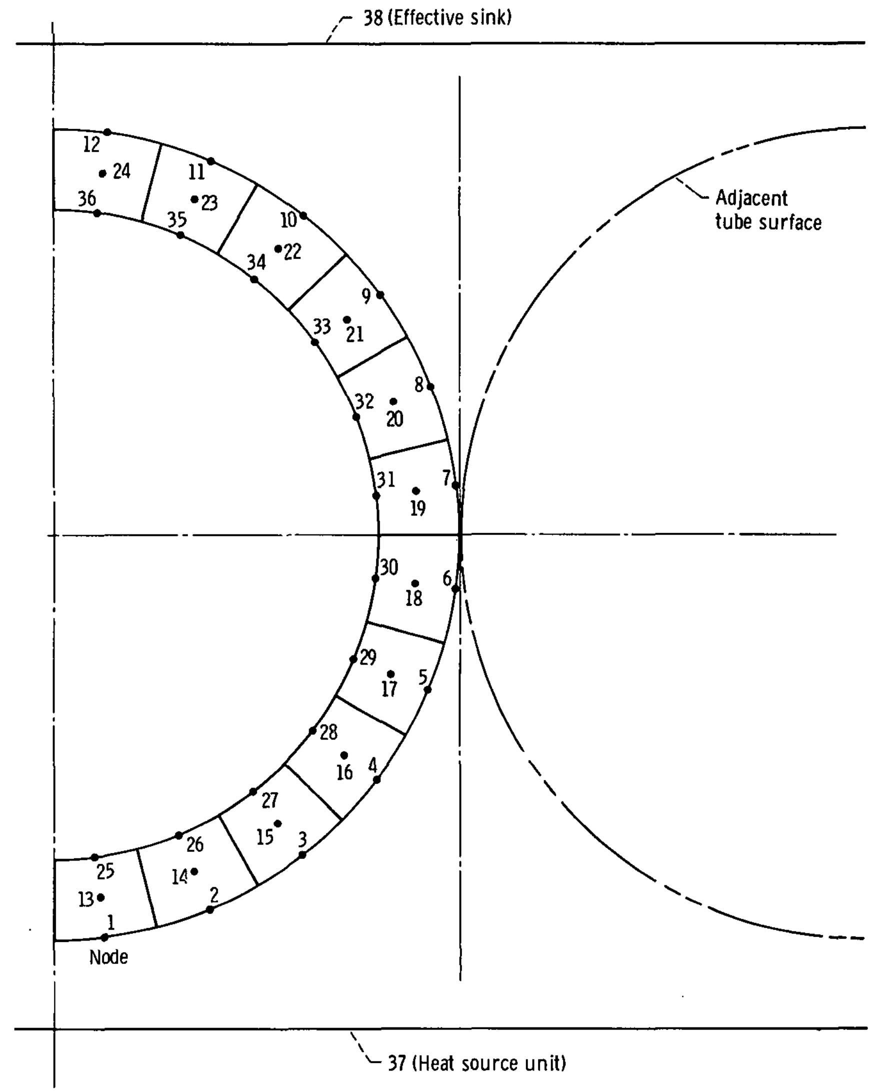


Figure 4. - Thermal model of HSHX tube.

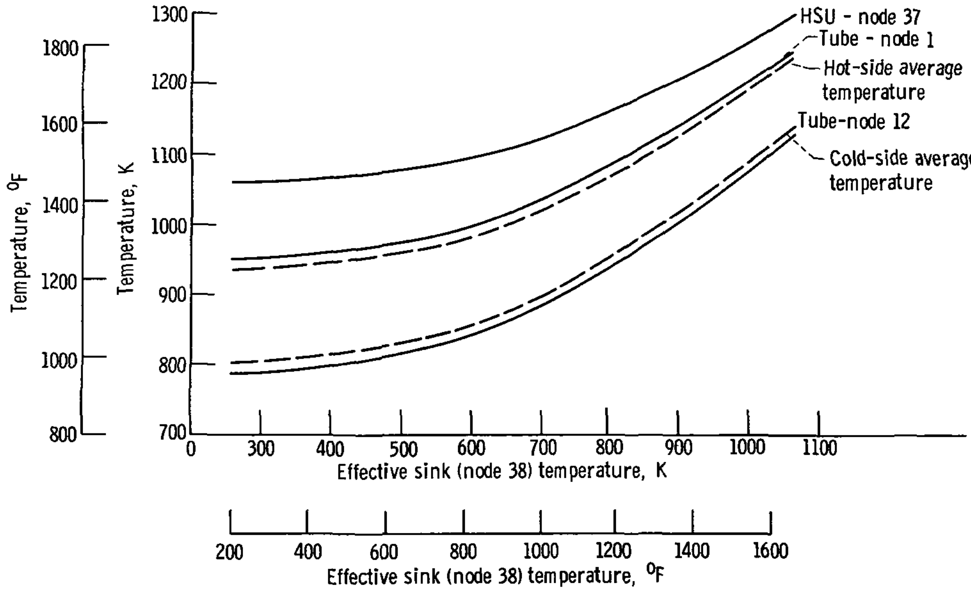


Figure 5. - HSHX and HSU temperatures as function of sink temperature for passive heat rejection.

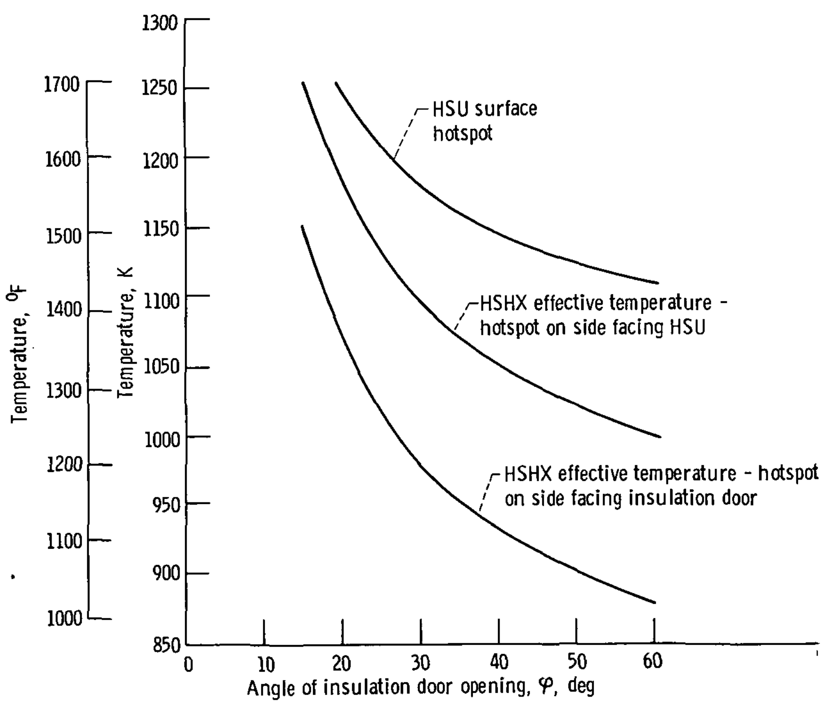


Figure 6. - HSHX and HSU hotspot temperatures for passive heat rejection.

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