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FINAL REPORT

DESIGN STUDY OF PASSIVE DETECTOR COOLING TECHNIQUES

Contract No. NAS 5-21009

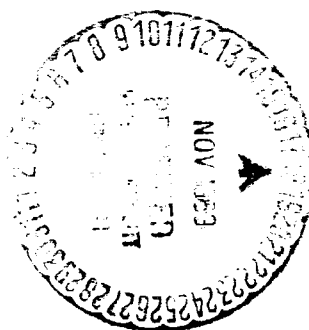
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Contract No. NAS5-21009

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ABSTRACT

This report documents the results of a design study dealing with the analysis of detector cooling techniques using passive radiator systems for the ATS-F Laser Communications Experiment.

The study begins with a review of existing concepts and designs of radiator systems. The advantages and disadvantages of various approaches are delineated for a system to be utilized on the ATS-F spacecraft configuration. Basic design concepts for passive coolers are then discussed with emphasis on the methodology of shielding and the implications on the surface thermal/optical properties.

A separate section deals with the heat inputs to a typical cooler. These include sun and earth heat inputs, ATS-F solar panel and antenna heat inputs, and heat loads associated with the detector subsystem.

Analyses are performed to determine the weights and volumes of LCE radiator systems for two spacecraft configurations. Detailed radiation cooler performance calculations are then made for one spacecraft configuration. Results are reported for the temperature-time history over an orbit for three different times of the year.

A brief study of test methods applicable to the developmental, qualification, or acceptance level testing for the experiment subsystem is also reported.

ACKNOWLEDGMENTS

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SUMMARY AND CONCLUSIONS

A study of requirements for detector cooling techniques by means of passive radiation coolers for the ATS-F Laser Communications Experiment (LCE) has been undertaken. The analysis was carried out for two typical ATS-F spacecraft configurations presently under consideration.

The basic thermal requirement for the LCE radiative cooler is to provide cooling in the range of 100-150K for a Hg Cd Te detector. The radiative cooler is to be mounted in a north or south viewing bay of the ATS-F spacecraft experiment module in geo-synchronous orbit. It is a further requirement that the detector be maintained within its design temperature range for periods exceeding one year.

The design study begins with a brief review of existing concepts and designs available in the literature. The general concepts of all designs are found to be similar in that the low-temperature radiator surfaces are shielded from direct solar heating and have a large view to space. Various staging and shielding arrangements are used, and the methods for supporting the low-temperature radiator to which the detector is attached differ.

A detailed study of the basic concepts for passive coolers is presented. The methodology of shielding by specularly reflecting surfaces is reviewed and the geometric construction of the cooler in relation to the spacecraft configuration is examined. It is concluded that a design analysis should take into consideration non-meridional rays when shielding surfaces are designed. Furthermore, a very high degree of specularity is needed to effectively shield a low-temperature stage.

A typical cooler design is introduced and operational principles discussed; requirements are specified for thermal/optical properties. Basic principles of external shielding and staging are reviewed. It is concluded that to be effective external shields must be specular and must not reflect incident energy onto a low-temperature stage.

The effect of external heat inputs on obtainable cold-stage temperature is reviewed. It is concluded that in most cases, the cooler design should ensure all external heat inputs are shielded from the cold stage.

A detailed tabulation of incident heat inputs to the radiation cooler is given. These include direct solar fluxes; reflected solar input from the spacecraft, spacecraft appendages and from the earth; and direct emission from the spacecraft, spacecraft appendages and from the earth. Heat inputs from conductive and I^2R losses in the detector lead wires are examined and a method is illustrated for determining heat inputs through the optical aperture.

Parametric studies are made to determine sizes and approximate weights of radiative coolers required to maintain specified low-stage temperatures. The analysis is carried out for both spacecraft configurations assuming operation at the summer solstice when heat inputs are highest.

Detailed transient calculations are also carried out for a typical radiation cooler designed for one spacecraft configuration. The power dissipation on the low-temperature radiator for this illustrative example is chosen to be 40 milliwatts, comprising the detector bias power, laser local oscillator power dissipation, I^2R heating in the detector lead wires and the heat inputs via the detector optical aperture. Temperatures are predicted as a function of orbital position for three different times of the year--summer solstice, equinox and winter solstice. Over a year, a typical span of cold-stage temperatures is from 100°K to 125°K. During summer solstice, orbital temperature fluctuations near 20° are predicted. It is found that these fluctuations can be decreased somewhat by increasing the thermal mass of the cold stage. This radiator subsystem would occupy a volume of approximately 700-800 in³ and would weigh approximately 2-3 pounds.

A parametric analysis is also performed to determine the influence of uncertainties or changes in design parameters on cold-stage temperature. A method of evaluating the importance of these uncertainties is presented.

From this design study we conclude that LCE detector temperatures in the 100-125K region can be attained by use of a passive radiator mounted in the experiment module of the ATS-F spacecraft. The most important environmental heat inputs to the passive radiator system are direct sunlight and thermal emission from the ATS-F solar panel array. The detector bias dissipation, local oscillator power and the detector lead wire configuration significantly influence the size of the passive radiator and operating temperature level.

For the radiator designs considered, the surfaces used to shield the low-temperature radiator must be highly specular and have a low solar absorptance and infrared emittance.

From a brief study of the requirements for thermal-vacuum testing of the passive radiator, we conclude that liquid-nitrogen-cooled cryoshrouds will provide a sufficiently low background temperature for design qualification and flight-acceptance testing. To obtain a high degree of confidence in the thermal performance of the passive radiator system, it is recommended that the heat flux incident on the radiator from direct sunlight and the thermal emission from the solar panels be simulated as closely as possible during the thermal-vacuum test program. The design of the thermal-vacuum test setup will be critical since stray heat inputs and reflections of the simulated solar beam within the cryoshroud of the thermal-vacuum test chamber can have a significant effect on the accuracy of experimental measurements of detector temperature.

1.0 REVIEW OF EXISTING CONCEPTS

There are a number of NASA GSFC experiments which require passive cooling for infrared detectors, and work is currently underway on the development of passive radiators for these systems. These experiments are to be flown on the Nimbus, Tiros, and Applications Technology satellites. At present, a passive radiator for an infrared radiometer designed to cool a detector to the 200°K temperature level is being flown on the Nimbus III spacecraft.

The passive radiator concepts for these spacecraft will differ because of the spacecraft geometry and the nature of the environmental inputs. For example, the Nimbus and Tiros satellites are flown in near-polar, low-altitude earth orbits; whereas, the ATS-F is to be flown at geo-synchronous altitude. The primary environmental heat loads on the passive radiators used for low-altitude systems are thermal emission and reflection from the earth, and to a lesser degree direct sunlight (depending on the position of the orbit plane relative to the sun and the location of the cooler on the spacecraft). At geo-synchronous altitude, heat inputs from the sun dominate the thermal design; however, in the particular case of the ATS-F configuration, the heat inputs from the spacecraft appendages are also important. Thus, it can be seen that the thermal design of the passive radiator to shield the low-temperature radiator stage is strongly influenced by the orbital parameters and the location of the passive cooler relative to the parent spacecraft. For this reason, it is impossible to design a radiation cooler which is generally applicable to any mission or spacecraft. Furthermore, a detailed analysis of the applicability of particular design concepts to the LCE requirements cannot be made. To our knowledge, no published work exists on the design of radiative coolers directly applicable to the ATS-F mission. The following discussion of published work and patents will therefore deal with the general concepts of shielding and the features of the designs which may be applicable here.

A comprehensive study on a two-stage radiant cooler designed to operate below 100°K in near-polar, low-altitude (500 n. mi.) Nimbus-type

orbits has been made by Annable, et al. (1967).¹ The shielding system for the low-temperature patch to which the detector is attached is a frustum of a pyramid with the second-stage patch located at the base of the pyramid. The opening of the pyramid is made rectangular to provide different "view angles" in two directions. The interior surfaces of the pyramid are coated to be efficient specular reflectors. The thermal analysis of the system is based on meridional ray traces within the specular surfaces. A retractable caging mechanism is used to support the low-temperature stage during the launch environment. Thermal-vacuum tests of the cooler were made to establish the thermal performance. The absorbed environmental heat inputs were simulated indirectly by maintaining the first-stage (high-temperature) pyramid shield near the calculated orbital temperature and measuring the low-temperature, second-stage patch. Because the low-temperature radiator was designed to operate in the 80-90K region, a helium cooled, "black" target operated near 30K was used in testing the thermal performance.

The thermal and mechanical design of a similar design concept has been reported by Wallschlaeger (1968)² for a TOS radiometer. For this spacecraft, sunlight is not incident on the cooler mouth and the two-stage shielding arranged in the form of an irregular pyramid is designed to preclude direct or earth-reflected radiation from striking the second-stage patch. The two-stage cooler weighs approximately 4 pounds, and the second-stage has a power dissipating capability of approximately 1 milliwatt at 77K. To minimize thermal loads during orbital operation, the radiator is uncaged after launch by means of a pneumatic system.

1. Annable, R. V., et al., Day-Night High Resolution Infrared Radiometer Employing Two-Stage Radiant Cooling, Report by ITT Industrial Laboratories to NASA GSFC, Contract NAS 5-10113, December 1967.
2. Wallschlaeger, W., Design of the VHRR for the Improved Tiros Operational Satellite Spacecraft, Report by ITT Industrial Laboratories to NASA GSFC, Contract NAS 5-10491.

Fuschillo, et al. (1969)³ describe a radiation-cooled cryostat designed to operate in an earth orbit in the ecliptic plane. The invention deals with a pill-box radiator surrounded by annular shields to minimize heat inputs from the earth and sun. Calculations presented in the patent disclosure indicate that temperatures of 10.5K can be reached in a high-altitude orbit in the plane of the ecliptic. The results are, in our view, subject to question; and the predicted temperature levels are not applicable to an orbit in the equatorial plane, e.g., a geo-synchronous orbit.

Buller, J. S. (1969)⁴ describes a radiation cooler for use in space comprising a plurality of radiation shields surrounding the detector to be cooled. The cooler invention embodies a multi-stage concept for limiting the heat flow to the lowest temperature stage to which the detector is attached. One of the configurations described consists of a multiplicity of conical radiation shields. An application cited for the invention is cooling a detector in an orbiting synchronous satellite to 89K with the active radiating surfaces shielded from direct sunlight.

The above discussion of passive cooler design concepts is limited in scope presumably due to proprietary data or to security classification. There are, perhaps, other references available in the literature; however, an exhaustive literature search was not within the scope of work. Based on the brief survey conducted, it can be seen that the designs for passive radiator systems are similar in concept if not in detail. The shielding of environmental heat inputs is of extreme importance in design. Careful and detailed thermal analyses must be a part of the design process to adapt passive cooling concepts to a particular spacecraft configuration. The design must be based on a complete understanding of the subtle problems of controlling heat flows which are extremely small in magnitude if it is desired to cool detectors to cryogenic temperatures.

3. Fuschillo, N., et al., Radiation Cooled Cryostat, U. S. Patent 3,422,636, 21 January 1969.

4. Buller, J. S., Radiation Cooler for Use in Space, U. S. Patent 3,422,886, 21 January 1969.

In this case, the integration of the cooler with the spacecraft is extremely important. The knowledge of the thermal/optical properties of surfaces used in shielding the low-temperature stage and the behavior of these surfaces in the space environment may determine the ultimate capability of radiative coolers.

2.0 BASIC DESIGN CONCEPTS FOR PASSIVE COOLERS

A review of the general concepts of detector cooling by passive radiation coolers has been presented in the preceding section. While the various radiators were designed for different missions and spacecraft configurations, the basic principle of shielding the low-temperature surface from incident-radiant energy by reflecting cone or cone-like structures is generally applicable. On the other hand, the methodology of shielding varies from design to design.

Because it is not possible to consider here in detail all the design concepts discussed in the preceding section, we will base much of the discussion in the following sections on the radiation cooler designs presently employed at Arthur D. Little, Inc. However, the emphasis will be on the general principles involved and not on a specific design.

2.1 Methodology of Shielding

To obtain low detector temperatures, the detector is normally mounted on a high-emittance surface which dissipates power by radiating to space. The net power from external sources which must be dissipated by the low-temperature surface is reduced by surrounding the surface by a shield. These external heat sources may include the spacecraft, spacecraft appendages, and the earth; the incident energy may be emitted infrared radiant energy or reflected sunlight. In addition, direct solar heating may be important.

Shielding reduces external heat inputs by blockage or by reflecting the energy away from the cold stage. Since the shield will, itself, partially absorb this incident energy, the shield surfaces are generally made highly reflecting in both the solar and infrared spectral regions. Additionally, there is ordinarily a requirement that the surfaces be specular in order that the incident radiant energy be reflected away from the low-temperature stage. The implication that this requirement has on surface properties will be discussed in Section 2.2.3.

The basic method of screening the low-temperature surface by highly reflecting surfaces is illustrated in Figure 1. Here, the

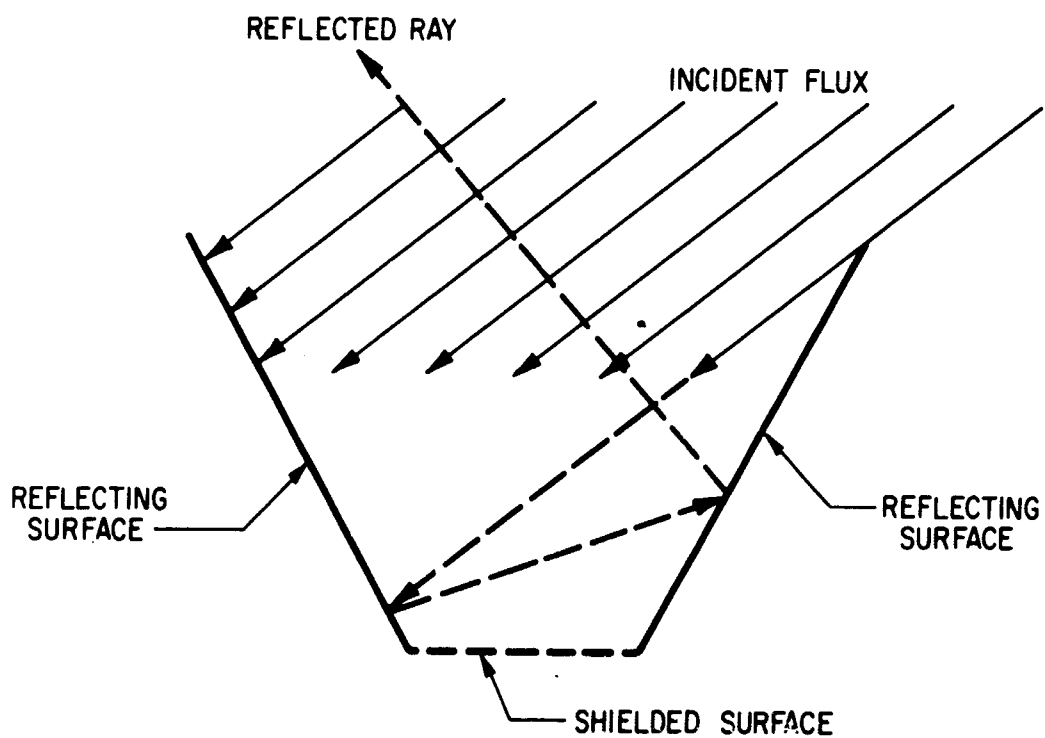


**FIGURE 1 SHIELDING OF INCIDENT RADIATION
BY REFLECTING SURFACES**

reflection process is shown for the meridional rays which are considered collimated. A "meridional" ray is defined as a ray that lies in a plane containing the surface normal of the reflecting surface.

If the reflecting surfaces are highly specular, or mirror-like, a large fraction of the incident energy is specularly reflected away from the shielded surface. The remaining fraction is either absorbed by the shield or is scattered in the off-specular directions. To minimize the net radiant energy transfer to the shielded surface, the non-specularly reflected fraction must be minimized.

To analyze shielding of the low-temperature surface it is necessary to consider also the non-meridional rays (skew rays). It is important to be able to predict the paths of all external radiation incident on the reflecting surfaces and to insure that only a well-controlled amount of the incident flux reaches the shielded surface. As will be shown in Section 2.4, the unshielded energy flux from an external heat source may represent a very sizeable heat input to the low-temperature surface.

One configuration which will screen and reflect non-meridional rays away from a low-temperature surface is a body of revolution. Since this is the design concept with which we are most familiar, we will discuss this configuration in some detail.

An illustration of the reflection process for two typical rays of radiant energy entering a specular cone is given in Figure 2. The process which will be discussed below is similar for other bodies of revolution.

In Figure 2, Ray I is considered to be coplanar with the cone axis--that is, a meridional ray. Ray II enters parallel to the plane containing Ray I and the axis, but is displaced from it by a distance slightly less than the radius of the conical opening. For the design shown, where the sum of the cone half-angle ϕ and the angle of incidence ψ is slightly greater than 90° , the coplanar Ray I will make only one intersection with the conical surface and then be reflected out of the

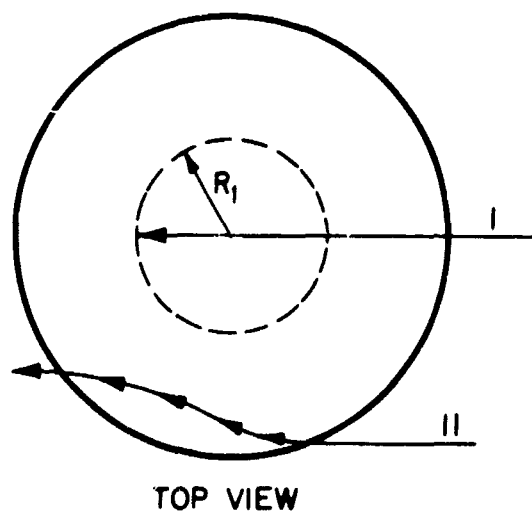
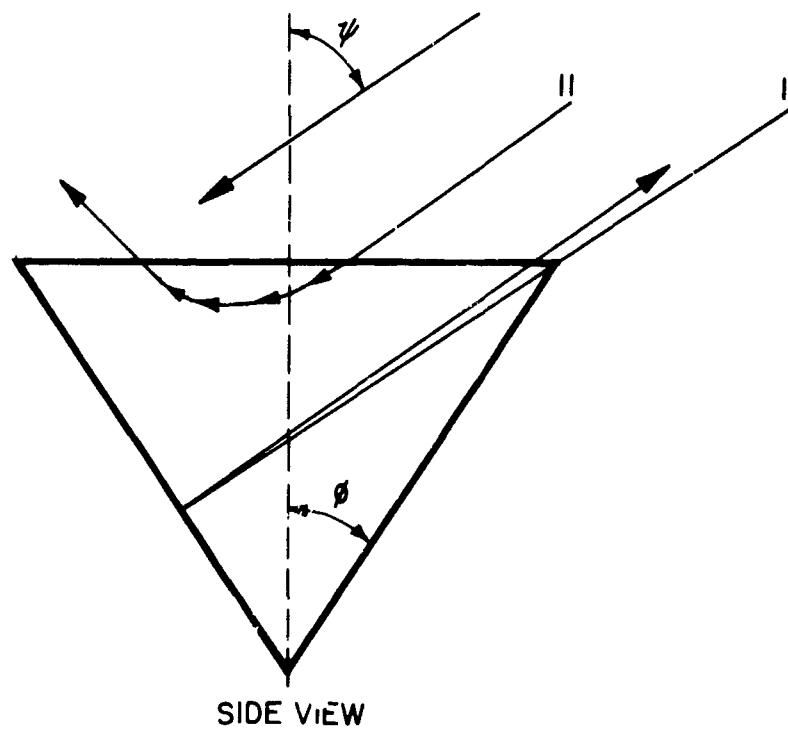


FIGURE 2 DIAGRAM OF REFLECTED RAYS

conical opening. On the other hand, Ray II will make several intersections with the surface before being directed out of the cavity. However, it can be demonstrated that all intersections will lie outside a minimum radius, R_1 , defined by the intersection of the conical surface and Ray I. This radius defines an imaginary plane below which the low-temperature surface may be located. All rays, both meridional and non-meridional, which enter the cone at angles greater than or equal to the angle ψ shown in Figure 2 will be reflected away from the imaginary plane.

The design discussed above can be modified somewhat to decrease the volume of the cone frustum. The meridional Ray I was reflected out of the cone after only one intersection with the specularly reflecting surfaces. However, it is only necessary that this ray not be reflected downward toward the low-temperature surface. This is illustrated in Figure 3 where the reflection pattern for a meridional ray is shown for two slightly different designs.

The larger cone frustum is similar to that shown in Figure 2 where the meridional ray is incident normal to the reflecting surface. Here, the cone half-angle is $\phi_1 = \pi/2 - \psi$. With the smaller cone, the meridional ray is reflected in a direction parallel to the shielded surface and directed out of the frustum at a subsequent reflection. The cone half-angle here is $\phi_2 = \frac{1}{2} (\frac{\pi}{2} - \psi)$.

It can be demonstrated that the non-meridional rays are also shielded from the lower surface for the so-called "minimum cone" shown in Figure 3. It is called the minimum cone since the bottom surface would not be shielded from the incident rays if the cone half-angle were decreased.

The difference in the size of the cone frustum required to shield the lower surface is significant for the two designs illustrated. There are other considerations, however, which may recommend the choice of one design over another. For example, the requirement on degree of specularity is more stringent where the cone half-angle is small. Nevertheless, from a shielding standpoint alone, it is necessary only that the cone half-angle be greater than or equal to the half-angle defining the "minimum cone".

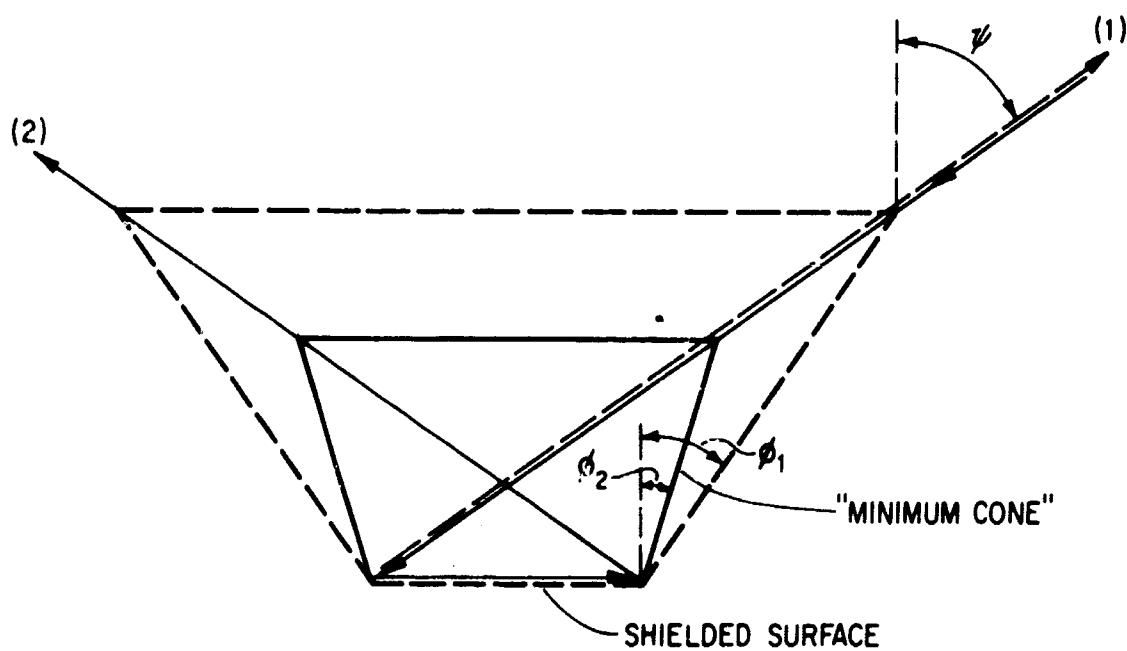


FIGURE 3 REFLECTION OF MERIDIONAL RAY FOR TWO CONE DESIGNS

2.2 Requirements for Limiting Radiant Interchange to Cold Stage

Within this section we will discuss the overall design concepts of a radiation cooler and the requirements for minimizing the temperature of the cold stage. We introduce this subject with reference to Figure 4. Here, the temperature of the cold stage (Surface 1) is shown as a function of power to the cold stage and temperature of the outer stage (Surface 2). The results are illustrated for a typical cooler design of cone half-angle 25° . The method of calculation of the radiant interchange between the two surfaces is given in Appendix I.

The cold-stage temperature is highly dependent on the power to the cold stage. This power can be comprised of bias and laser local oscillator power dissipation, conductive heat leaks, optical throughput and external heat fluxes not shielded by the outer stage. A design goal is to minimize each of these power inputs.

For a given power input to the cold stage, the temperature is also sensitive to the temperature and surface emittance of outer stage. Clearly, another design goal is to use a surface with a low emittance and to obtain a low outer-stage temperature.

2.2.1 Typical Cooler Design

In addition to shielding the low-temperature surface containing the detector, the surrounding surfaces must provide support for the detector without heating it above some specified operating temperature. A typical radiation cooler design is shown schematically in Figure 5. Many variations of this design are possible, but we consider this design to illustrate the heat transfer paths characteristic of radiation coolers.

The assembly consists basically of an outer stage mounted to the spacecraft through low-conductance supports and which itself supports the cold stage by low-conductance supports. Housed within the low-temperature stage is the detector. The internal surfaces of the outer stage are highly reflecting specular surfaces to shield the cold stage from external heat inputs.

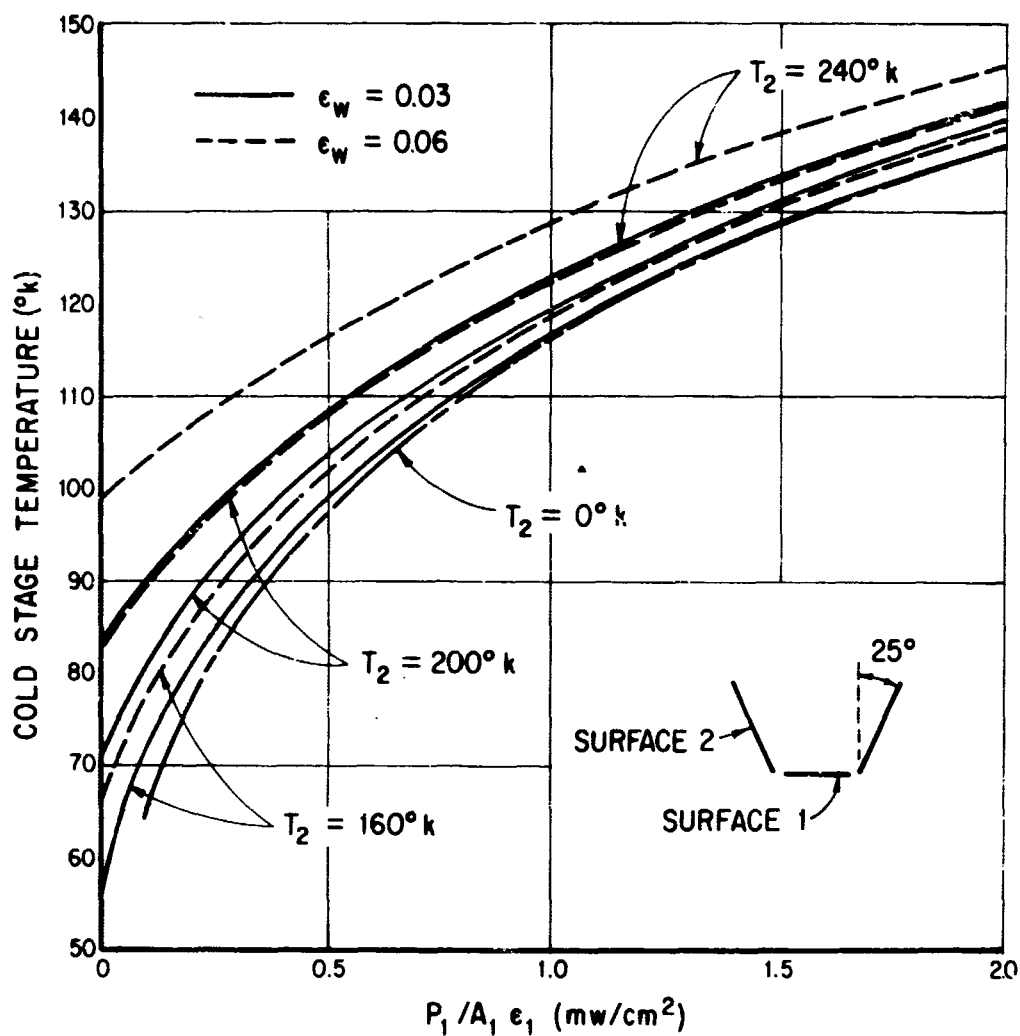


FIGURE 4 INFLUENCE OF CONE TEMPERATURE AND POWER TO COLD STAGE ON COLD-STAGE TEMPERATURE FOR TYPICAL DESIGN

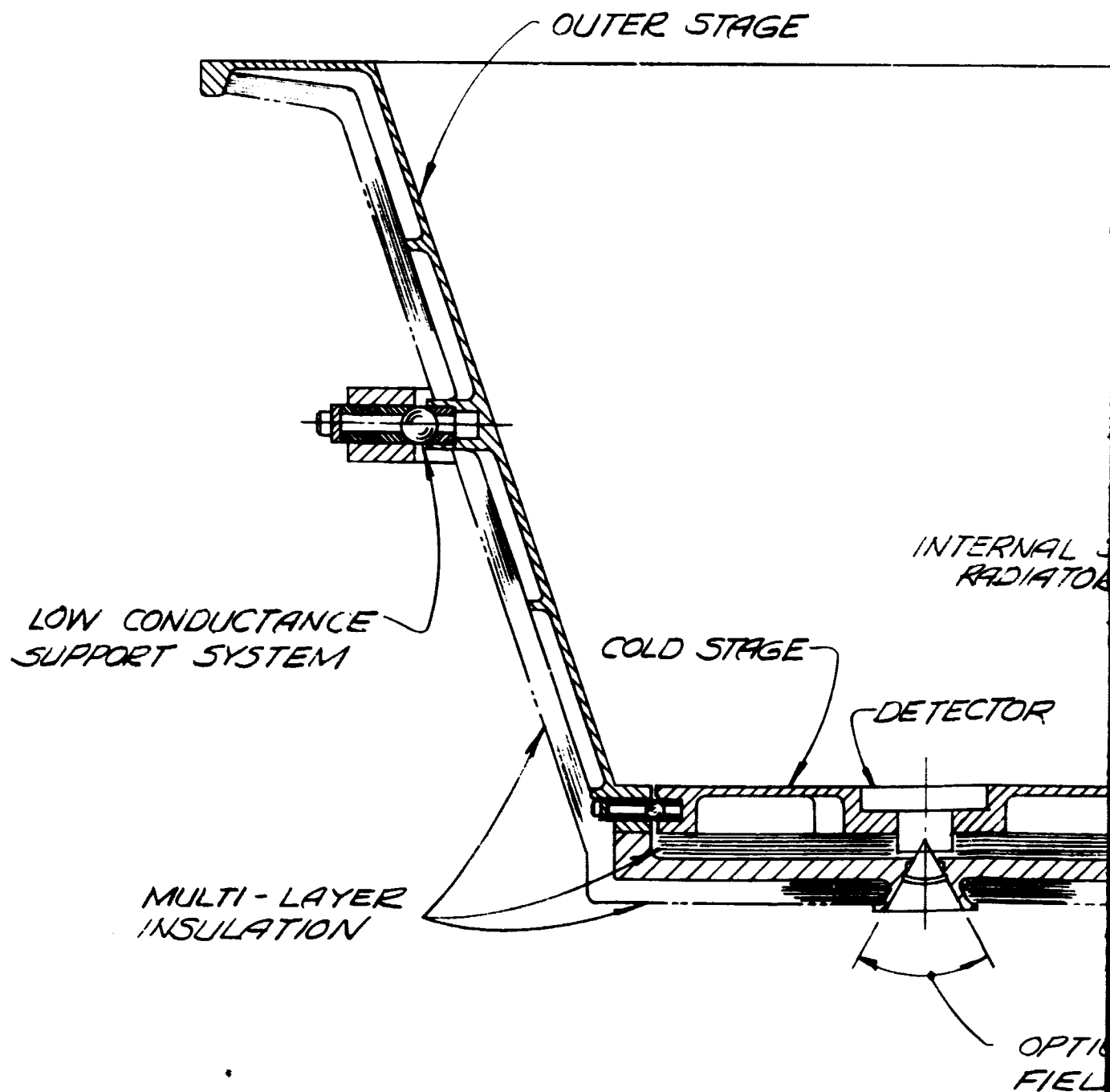
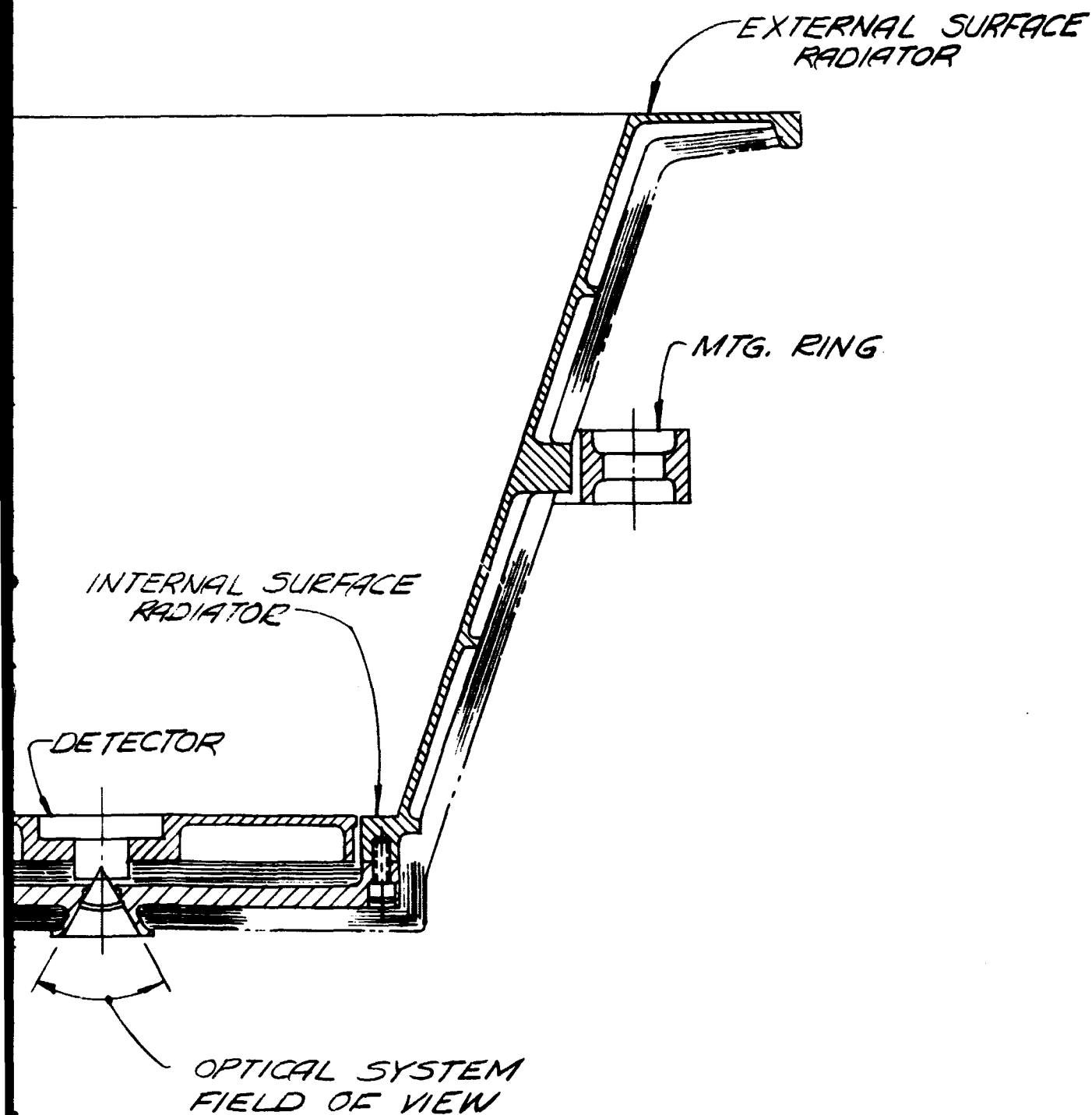


FIGURE-5 - TYPICAL RADIATION COOLER



TWIN COOLER DESIGN

The top surface of the cold stage is of high emittance to dissipate the net power on the stage by radiation to space. The bottom and side surfaces of the cold stage are low-emittance surfaces to minimize the radiant heat input from the warmer outer stage. Multilayer insulation is also used between the two stages to further decrease the energy transfer.

The outer stage is also designed to operate at a low temperature to decrease the conductive and radiant energy transfer from the cone to the cold stage. Multilayer insulation is used to decouple the outer stage from the relatively warm spacecraft, and the internal surfaces of the outer stage are of low emittance and absorptance in both the solar and infrared spectral regions. It should be noted that both infrared emittance and solar absorptance must be as low as possible--not the usual α/ϵ ratio. Radiative cooling from the spacecraft is provided mainly by the external surface radiator and to a lesser extent by the internal surface radiator. On the external surface radiator, the ratio α/ϵ is important--and here it must be small.

In theory, the internal surface radiator is shielded from external heat inputs so that the absorptance is not important for this surface. However, the presence of the internal surface radiator serves another important function. It provides an extra safety margin for shielding the cold stage from external radiation reflected off the highly polished cone walls. If the conical surfaces are not completely specular, most of the reflected radiant energy which is not directed away from the cold stage will be intercepted by the internal surface radiator. Therefore, we also specify the internal radiator surface to be of low solar absorptance.

These are the general characteristics of a passive radiation cooler. Variations of this design--including staging--are possible. Some of these possibilities will be discussed in the sections which follow. However, the general design considered here will be useful in allowing us to focus our attention on specific aspects of a given design and to identify critical areas.

2.2.2 Geometrical Considerations

The geometry of the radiative cooler is determined by the external heat inputs associated with the orbital plane of the spacecraft and by the location of the cooler on the spacecraft. The anticipated location for the radiative cooler in the ATS-F configuration is shown in Figure 6. Environmental heat fluxes to a north- or south-viewing surface include direct solar inputs (which vary with the time of day and season) having a maximum sun angle of 23.5° , direct emission and solar reflection from the antenna and solar panel, and infrared emission and reflection of solar energy from the earth.

For the purpose of the following discussion, we will assume that preliminary calculations have shown that the heat input from the solar panel can be significant. Under this assumption, it is desired to shield the cold stage from heat inputs from the solar panel as well as from direct solar heating. Figure 7 shows the radiation cooler geometry in relation to the spacecraft axes. At a solstice, sunlight can enter the cooler aperture at 23.5° from the Z-Z spacecraft axis. Radiation from the solar panel enters the cone mouth at larger angles with respect to the spacecraft axis. According to the principles presented in Section 2.1, the cooler axis is tilted to bisect the include angle between the limiting rays, and a cone equal to or larger than the "minimum cone" is constructed. This construction is shown for meridional rays. However, as discussed previously, with a specularly reflecting body of revolution the non-meridional--or skew--rays will also be prevented from striking the low-temperature surface.

The application of this principle depends on the properties of the reflecting inner surfaces of the outer stage. We turn to a discussion of these requirements in the next section.

2.2.3 Surface Thermal/Optical Properties

There are several areas of the radiative cooler which must have different thermo-optical coatings. In the following discussion we will relate surface property values to the two wavelength regions of

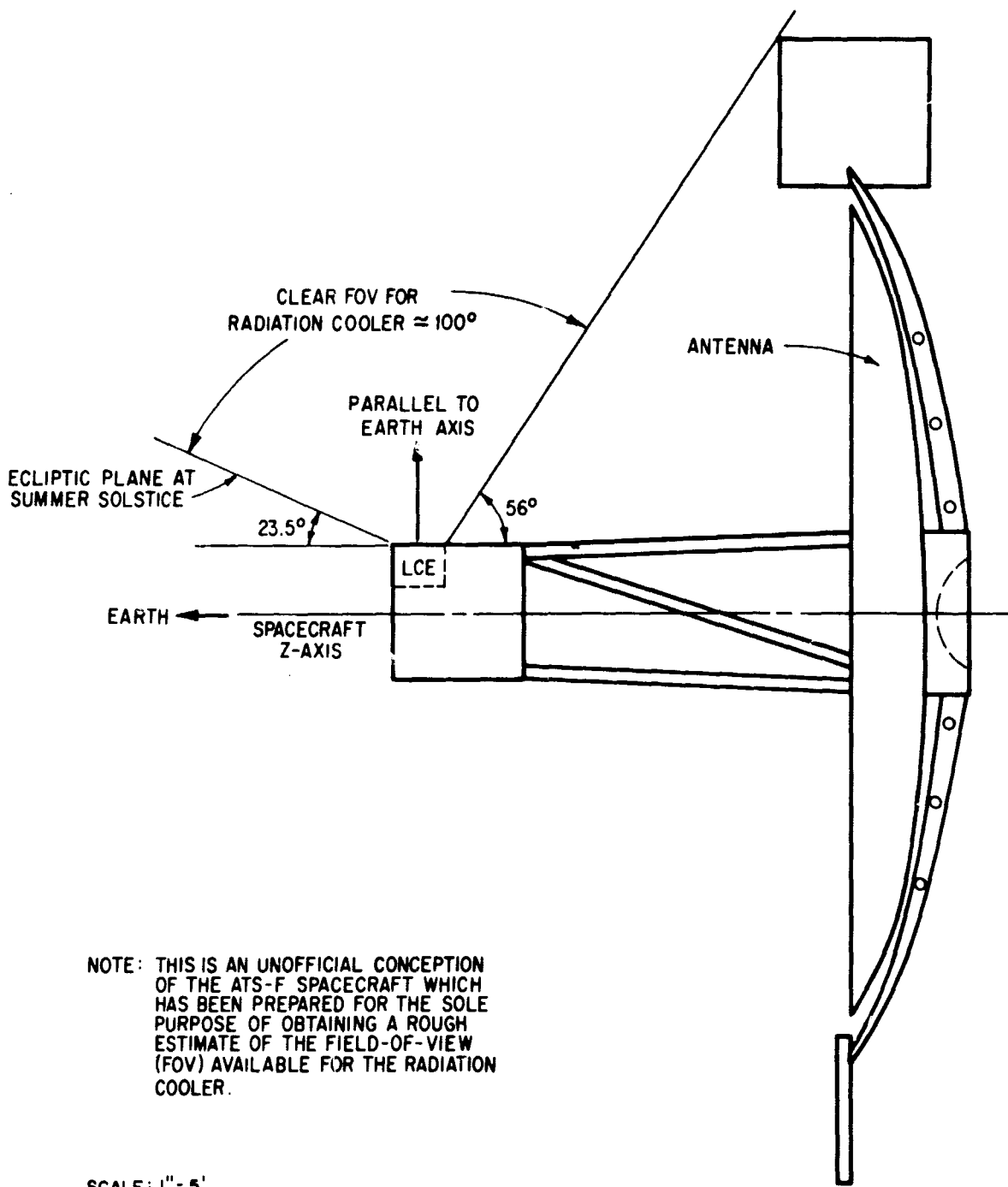
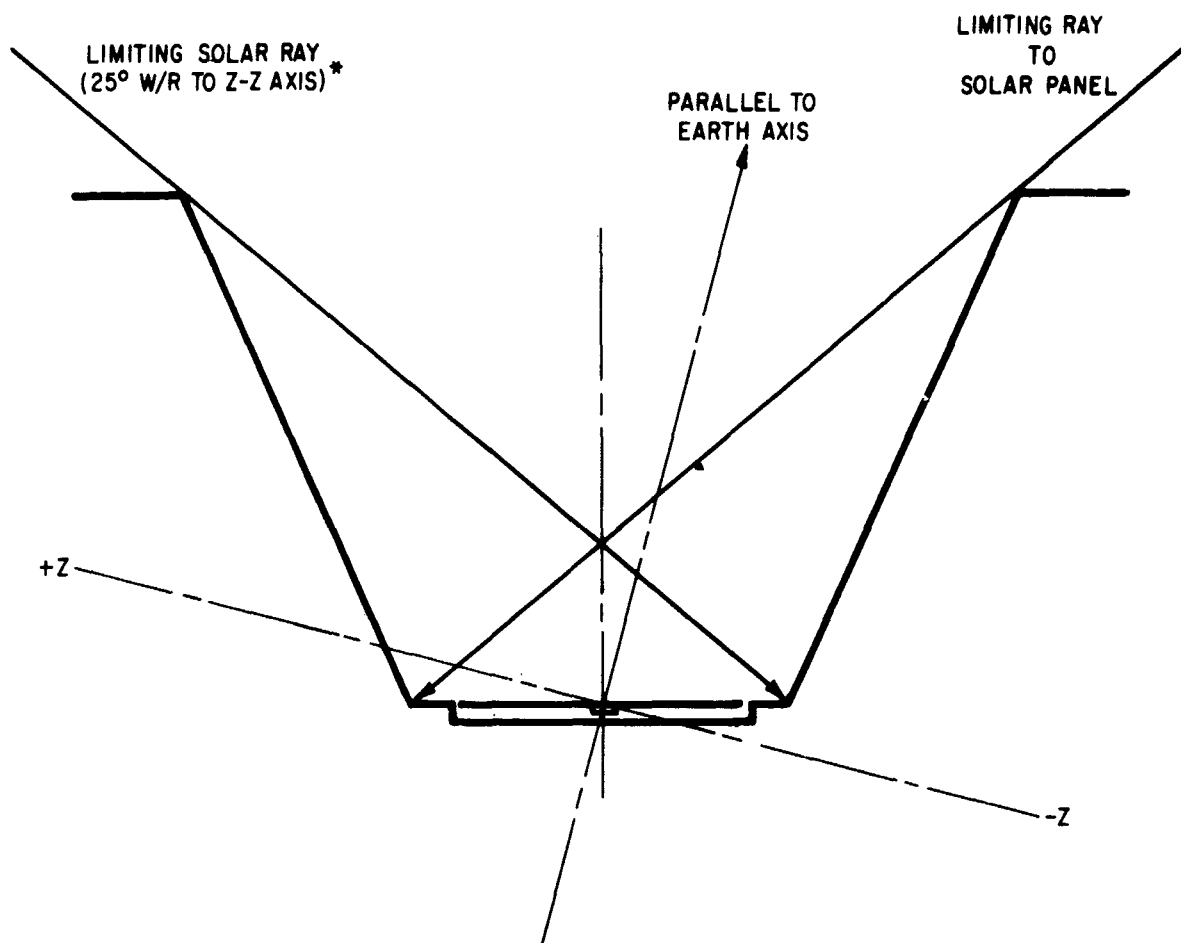


FIGURE 6 CONCEPTUAL VIEW OF ATS-F (From GSFC X-524-69-227)



* 25° SUN ANGLE INCLUDES 1.5° ALLOWANCE
FOR DEGRADED ATTITUDE CONTROL SYSTEM

**FIGURE 7 RADIATIVE COOLER GEOMETRY
IN RELATION TO SPACECRAFT AXES**

interest, viz., the solar spectrum from 0.3 to 3.0 microns and the infrared region from 5 to approximately 50 microns. The solar absorptance α_s is defined as the total absorptance to the solar spectrum. The emittance ϵ is the total hemispherical emittance at near room temperature.

The most critical area is the interior of the outer-stage cone frustum which must protect the low-temperature radiator from incident radiation due to sunlight, earth albedo, earthshine and thermal radiation from the spacecraft. In addition, the interior surface coating must minimize the radiative transfer between the walls and the low-temperature radiator. The interior walls are constructed geometrically to preclude external radiation from falling on the low-temperature radiator by the principles of geometric optics described earlier. To utilize this principle, the walls must be specular reflectors with a minimum scattering component.⁵

The interior walls of the cone frustum must therefore have the following properties:

1. Low solar absorptance,
2. Low infrared emittance,
3. Specular surface, and
4. Long-term stability in the space environment.

A surface which meets these requirements is vacuum-deposited aluminum applied to a highly polished electroless nickel coating on the aluminum cone. This technique is commonly used in fabricating metal mirrors for optical systems. Aluminum has the lowest solar absorptance of the pure metals with the exception of silver which must be overcoated to prevent tarnishing.

Typical property values of carefully controlled first-surface, vacuum-deposited aluminum coatings are given by Drummeter, et al.,⁶ and Hass⁷:

5. The "figure" of the surface is unimportant; however, the RMS surface roughness must be small to minimize the diffuse reflectance component.
6. Drummeter, L. F., and Hass, G., Physics of Thin Films, 2, pp 305-361, 1964.
7. Hass, G., Solar Absorptance and Emittance of Evaporated Metal Films, 1967.

$$\epsilon = 0.015 - 0.020$$

$$\alpha_s = 0.08 - 0.10$$

The emittance is virtually unaffected by aging due to the oxide film which is formed on the surface. The solar absorptance is also unaffected by exposure to the space environment.⁸

We believe that the basic problem with the aluminum coating will be cleanliness rather than changes that occur due to the space environment. For example, the deposition of foreign materials, on the internal surfaces of the cone could significantly affect the thermal optical properties.

The surface radiator at the aperture of the first-stage cone must have a relatively low solar absorptance and a high infrared emittance as discussed earlier. A number of choices are available for this coating. They include "Optical Solar Reflectors" (OSR coatings are second-surface mirrors of vacuum-deposited silver or aluminum on fused silica or FEP Teflon) or thermal control paints. Fused-silica OSR coatings (approximately 8 mils thick) are extremely stable, and have very low α_s/ϵ ratios (approximately 0.07-0.10); however, they must be adhesively bonded to the aluminum structure in a mosaic pattern with the attendant problems of differential expansion, etc. Silver-coated Teflon is also a stable coating with low α_s/ϵ ; however, the adhesion problem is not fully solved and the coating process requires further development. An alternate coating which avoids these development problems is the Z-93 paint coating which has been flown on many satellite systems. Z-93 is a relatively stable $\text{ZnO/K}_2\text{SiO}_3$ paint with initial α_s/ϵ values of 0.13 to 0.20. Data from Pegasus II indicate a change in α_s/ϵ from 0.21 at 240 days after launch to 0.24 at 460 days after launch.

The coating on the exterior surface of the low-temperature radiator (the cold stage) should have a high infrared emittance near 100°K.

8. Millard, J. P., Results from the Thermal Control Experiment on OSO III, AIAA 3rd Thermophysics Conference, June 1968.

For this surface 3M "Nextel" Optical Black Velvet or "Parsons" black paint will provide an emittance in the range of 0.90 to 0.93.

The periphery of the low-temperature radiator must have a low-emittance surface since it may not be possible to effectively insert multilayer insulation in the circumferential gap between the outer cone and the radiator. An emittance of 0.021-0.024 on clean, machined (20-35 μ inch RMS) surfaces of 6061 Al is typical.

2.3 Principles of External Shielding and Staging

Here, we will review the principles of the use of external shields and staging to reduce the temperature of the low-temperature stage. In the normal use of external shields, the net heat input to the cold stage is reduced indirectly by partially shielding the outer-stage, thereby allowing it to operate at a lower temperature. With the outer-stage temperature decreased, the radiant and conductive energy transfer to the cold stage is reduced. On the other hand, staging of the radiation cooler reduces the net heat input to the cold stage directly by isolating it from the outer stage conductively and--in some designs--radiatively.

We cannot attempt here to treat each of these possibilities in detail, but we can discuss the basic principles and their advantages and disadvantages. For this purpose, we will fall back again upon illustrations involving meridional ray traces.

2.3.1 External Shields

An external shield reduces the heat input to a surface by intercepting the energy and then reradiating in the infrared. The temperature at which the external shield radiates is determined by the complex interchange between the shield and the spacecraft, taking into account absorption of incident solar energy.

It is important that the energy radiated by the external shield is not incident on the cold stage. Allowing some of this energy to be incident on the cold stage defeats the purpose of the outer-stage design. This means that the outer stage can be only partially shielded from external heat inputs. For the design illustrated in Figure 7, the

shield cannot project above the lines showing the "limiting rays" from external sources without itself having a view to the cold stage.

It is important also to minimize the reflection of energy off the shields onto the outer stage. For example, if a shield were placed at the outer edge of the external surface radiator shown in Figure 7, it would have to be at an angle from the normal at least equal to that of the "minimum cone" half-angle. Otherwise, incident solar energy could be reflected off the shield onto the external surface radiator and into the cone mouth. This additional heat load would far outweigh the decrease in energy flux from the spacecraft or solar panels. An additional requirement is that the external shield be highly specular. If not highly specular, the shield could reflect incident energy to the surface radiator or into the cone mouth at all angles of incidence and for any shield construction.

It is difficult to make general statements about whether external shields should or should not be used. However, we feel that the preceding remarks are useful and that the specific design analysis should make clear the justification for an external shield when one is used.

2.3.2 Staging of Radiative Cooler

The basic purpose of staging is to reduce the heat load on the cold stage by reducing the temperature of its immediate environment. This is done by placing intermediate stages between the low-temperature surface and the outer stage--connecting them by low-conductance mounts. These intermediate stages equilibrate by radiation to space.

The penalty paid by staging is the increased overall size of the radiator and increased difficulties in maintaining optical alignment. This latter subject is outside the scope of the present work but must be considered in an overall design analysis.

Finally, it is important that the basic principle of shielding the low-temperature surface from external heat inputs not be compromised.

2.4 Effect of External Heat Inputs on Design Configuration

Within the preceding sections we discussed the importance of designing the basic cooler configuration to shield the cold stage containing the detector from the major external heat inputs. We wish to emphasize here that nearly any external heat input can be a major heat input if not shielded.

Consider, for example, a small spacecraft appendage--say a solar panel--that has a direct line of sight to the cooler aperture outside the limiting rays defined in Figure 7. Radiation entering the cooler at angles of incidence less than that corresponding to the limiting rays will be transmitted through the cooler outer stage to the cold stage. Since the cooler aperture area is much larger than the area of the cold stage, the outer stage actually acts as a collector of radiation which is funneled to the lower surface.

For the sake of illustration, let us assume that the view factor from a unit area of the cooler aperture to the external heat source is only 0.01. Let us further assume that the source emits energy at 300°K and has a surface emittance of 0.8. The energy incident on a unit area of cooler aperture is then 0.37 mw/cm^2 . For the cooler design shown as an insert in Figure 4, the net energy transmitted to the cold stage (Surface 1) could be approximately 0.50 mw per cm^2 of the cold stage. From the results illustrated in Figure 4, it is clear that this represents a sizeable heat load on the cold stage. In fact, the minimum temperature that can be obtained by a perfectly black surface with no other heat inputs is only slightly less than 100°K.

This external heat input should, therefore, be shielded from the cold stage. According to the principles developed earlier, this would be done by considering the external source as defining a new limiting ray and by constructing the cooler as shown in Figure 7.

In summarizing this section, the points to be made are that the transfer of energy from an external heat source should be carefully considered in a design analysis and that what may appear to be a small heat

input may actually be very large because of the funneling effect of the radiative cooler outer stage. In this case, the cooler design should allow the cold stage to be shielded from the external source.

3.0 TABULATION OF HEAT INPUTS

Heat inputs to the radiation cooler include external radiative inputs, conduction from the spacecraft and power inputs through the detector subsystem. The external radiative inputs include direct solar heating; reflected solar energy from the earth, spacecraft and spacecraft appendages; and emitted energy from the earth, spacecraft and spacecraft appendages.

The net energy incident depends not only on the design configuration, but on the time of day and year. In this section, information will be presented which may be useful in calculating the net heat inputs for various radiation cooler design configurations expected for ATS-F application.

3.1 Sun and Earth Heat Inputs

3.1.1 Direct Solar Heating

The solar energy transport to a unit area of a control surface representing the cooler aperture is presented in the following table for two times of the year--the solstice and the equinox--and for several values of inclination of the control surface normal with respect to the earth axis.

A drawing illustrating the orientation is presented in Figure 8. The angle β represents the orbital position of the control surface measured from the subsolar point. Angle γ is the inclination angle of the cooler with respect to the earth axis. At the solstice the sun angle $\theta = 23 \frac{1}{2}^\circ$; at the equinox, $\theta = 0^\circ$. A table of the heat fluxes is given below.

3.1.2 Earth Inputs

Heat inputs from the earth to the control surface arise from reflection of solar energy and direct emission. We will tabulate values based on an orbital height of 19,326 nautical miles, an assumed earth emission of 0.021 watt/cm^2 and a planetary albedo of 0.40.⁹ Table 2 lists the heat inputs from direct emission. Maximum values of heat input by planetary reflection of solar energy are given in Table 3.

9. J. A. Stevenson, J. C. Grafton, "Radiation Heat Transfer Analysis for Space Vehicles," ASD Technical Report 31-119, Part I.

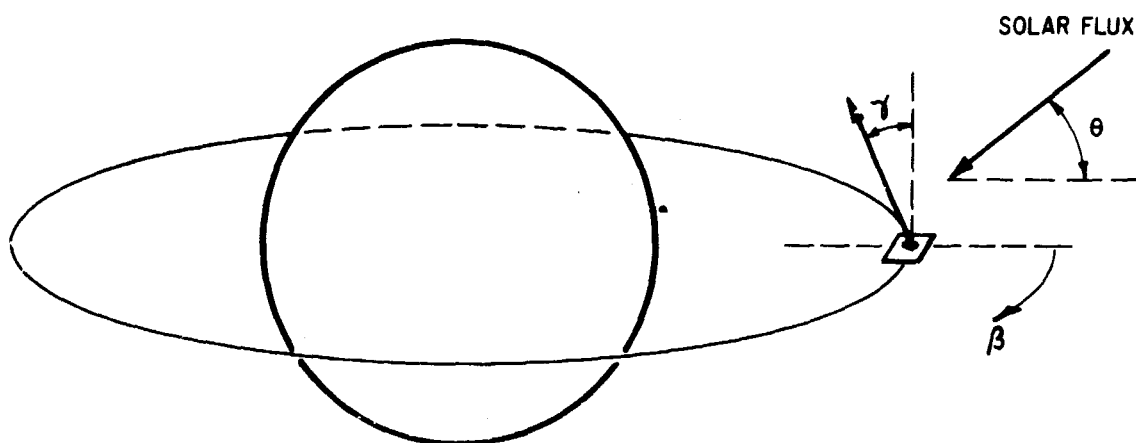


**FIGURE 8 SOLAR HEATING OF UNIT AREA OF CONTROL SURFACE;
DEFINITION OF ANGLES**

TABLE 1. SOLAR FLUXES TO CONTROL SURFACE

Sun Angle, θ	Inclination Angle, γ	Orbital Position, β	Incident Solar Flux, watt/cm ²
23.5° (solstice)	0°	0°-360°	0.0558
	10°	0°	0.0327
		20°	0.0340
		40°	0.0379
		60°	0.0438
		80°	0.0511
		100°	0.0588
		120°	0.0661
		140°	0.0720
		160°	0.0759
		180°	0.0773
	20°	0°	0.0085
		20°	0.0112
		40°	0.0188
		60°	0.0305
		80°	0.0448
		100°	0.0661
		120°	0.0744
		140°	0.0861
		160°	0.0937
		180°	0.0964
0° (equinox)	0°	0°-360°	0.0
	10°	0°-90°	0.0
		100°	0.0042
		120°	0.0122
		140°	0.0186
		160°	0.0228
		180°	0.0243
	20°	0°-90°	0.0
		100°	0.0083
		120°	0.0239
		140°	0.0367
		160°	0.0450
		180°	0.0479

TABLE 2. HEAT INPUT BY PLANETARY EMISSION

Inclination Angle, γ	Incident Flux, watt/cm ²
0°	0.00002
10°	0.00008
20°	0.00016

TABLE 3. HEAT INPUT BY PLANETARY REFLECTION OF SOLAR ENERGY

Inclination Angle, γ	Incident Flux, watt/cm ²
0°	0.00003
10°	0.00019
20°	0.00034

As expected for the altitude of interest these incident fluxes are extremely small.

3.2 ATS-F Solar Panel and Antenna Heat Inputs

Heat inputs from the antenna and from the solar panel arise from direct emission of infrared radiation and reflection of solar energy. The maximum expected values of heat input from reflected sunlight and emitted energy as a function of the temperature of the emitting surface are given for the following surface properties:

Antenna (75% open area):

$$\epsilon = 0.04$$

$$\alpha = 0.13$$

Solar Panels:

$$\epsilon = 0.84$$

$$\alpha = 0.80$$

The antenna was assumed to be constructed of an open-mesh metal-coated screen. Since the structure has approximately 75% open area, the effective emittance and reflectance values were taken to be 0.01 and 0.22, respectively. The solar panel properties assumed are characteristic of fused silica.

Heat inputs were calculated for two spacecraft configurations labeled "A" and "B". The configuration factors for a unit area on the control surface (representing the cone aperture) to the antenna and to the solar panels were determined using computer routines. A table of these configuration factors is presented in Table 4.

Based on an effective antenna emittance of 0.01, calculated values of direct emission to a unit area control surface are illustrated in Figure 9. The "tilt" angles are the angles of inclination of the control surface with respect to the earth axis. The maximum likely temperature of the antenna is 450°K. Corresponding values of heat input from the solar panel are shown in Figure 10.

TABLE 4. CONFIGURATION FACTORS FROM CONTROL SURFACE TO
ANTENNA AND SOLAR PANEL

Inclination Angle, γ	"A"		"B"	
	Antenna	Solar Panel	Antenna	Solar Panel
0°	0.100	0.019	0.084	0.008
10°	0.058	0.016	0.058	0.006
20°	0.028	0.013	0.027	0.004

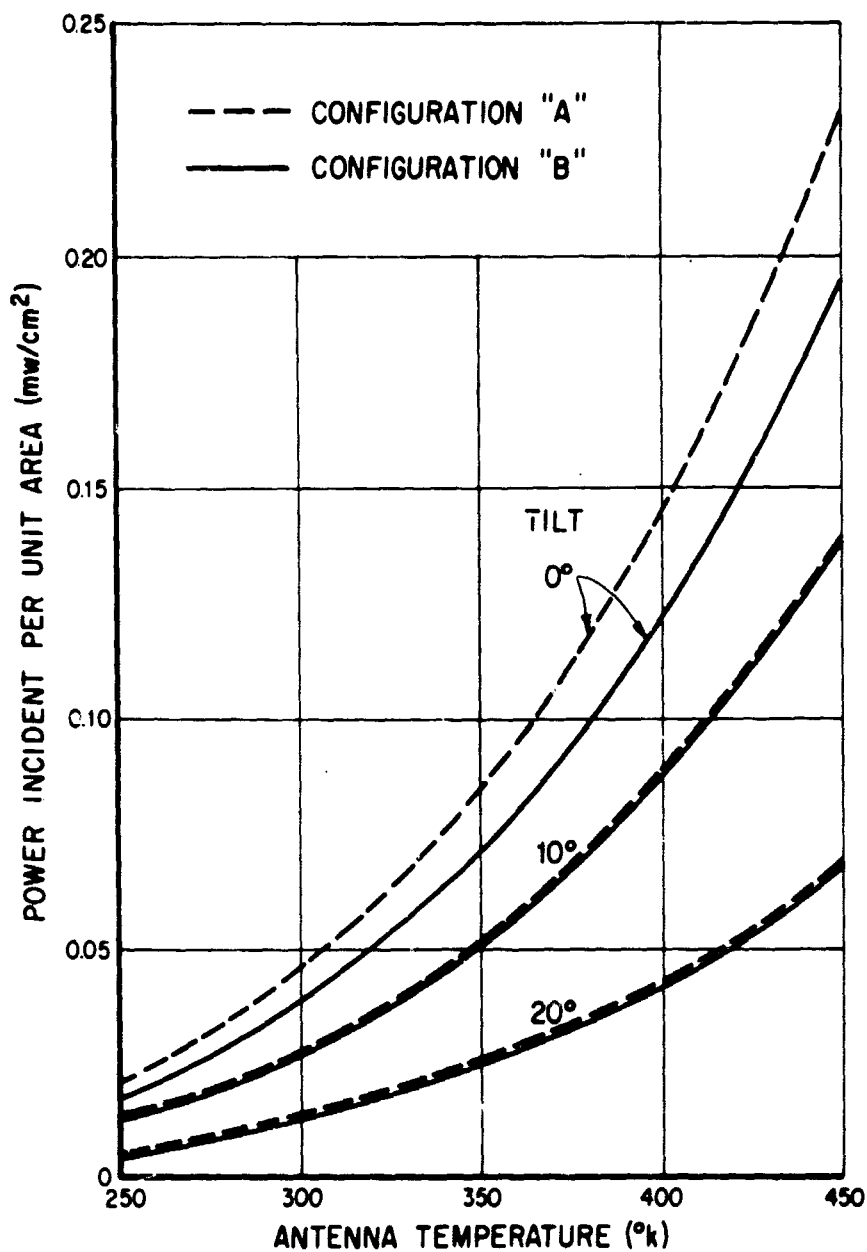


FIGURE 9 ENERGY EMITTED BY ANTENNA TO CONTROL SURFACE

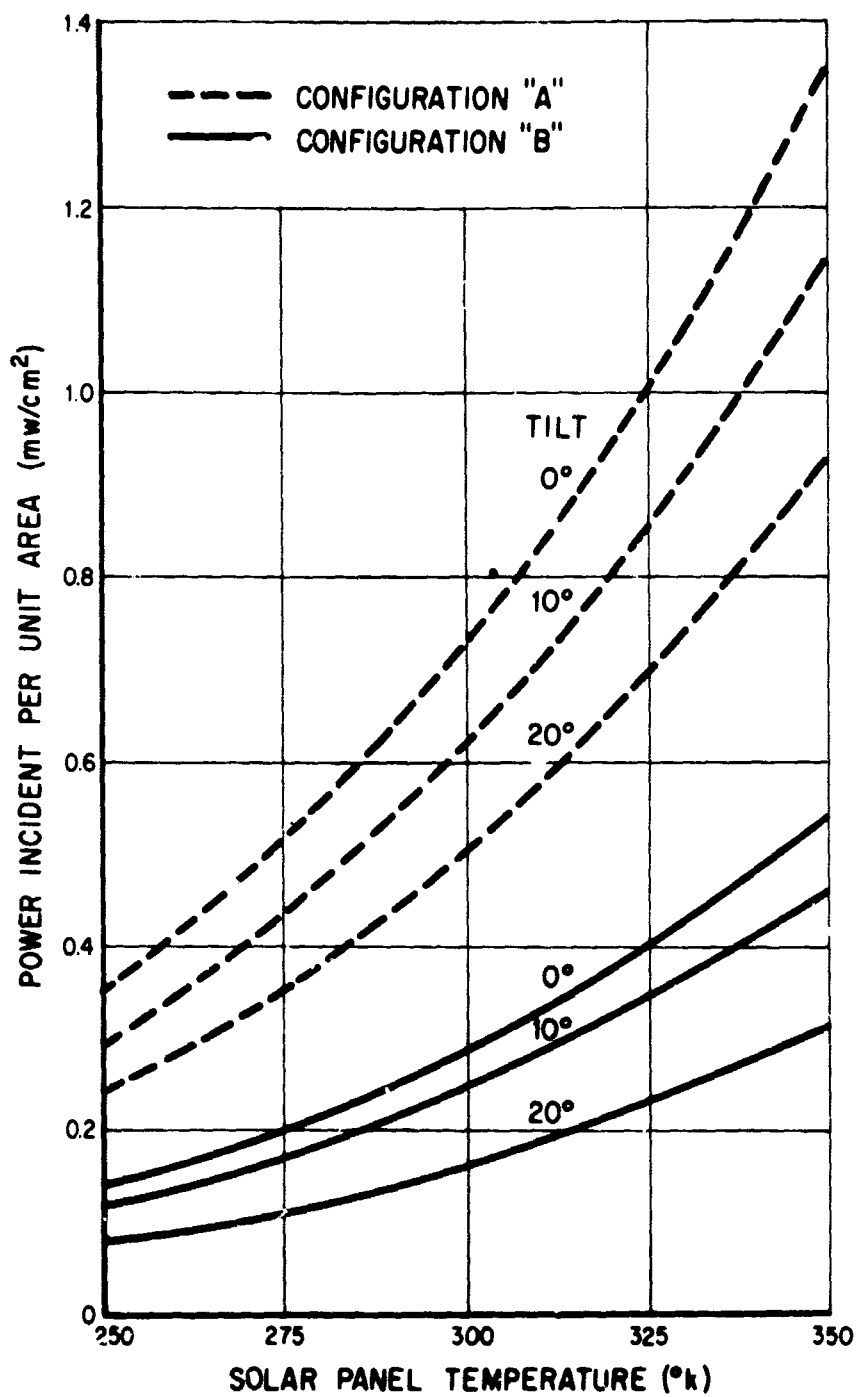


FIGURE 10 ENERGY EMITTED BY SOLAR PANEL TO CONTROL SURFACE

Values of heat input by reflection of solar energy are much less certain than the values for direct emission. It is suggested for the purpose of design that the maximum likely values be used. These are given in Table 5.

3.3 Detector Subsystem

The power to the cold stage associated with the detector subsystem is from bias power and laser local oscillator power: dissipation, conduction in the lead wires, I^2R losses in the lead wires, transmitted energy through the optics from the spacecraft interior, and direct radiation from the lens mount.

It is beyond the scope of the present work to comprehensively review various detector systems to determine values for bias power and local oscillator power dissipation. However, for the LCE experiment, a typical configuration might consist of 4 detectors with a power dissipation of 9 milliwatts each. Thus, with no other power inputs present, the detectors account for a 36-milliwatt heat load to the cold stage. This provides a convenient point of reference for discussion of other heat loads associated with the detector subsystem.

3.3.1 Lead Wires

The design of a lead wire system requires a tradeoff between mechanical considerations and thermal considerations. To minimize heat leaks, it is advantageous to use the smallest wires possible without compromising the mechanical integrity.

In this section we will review the ranges of heat loads associated with the use of various lead wire configurations. Both conductive and I^2R losses will be considered. For the purpose of discussion, it is assumed that:

1. Each wire is 1" long.
2. Each wire experiences a 100°K temperature drop along its length.

TABLE 5. MAXIMUM VALUES OF HEAT INPUT TO CONTROL SURFACE
BY SOLAR REFLECTION FROM ANTENNA AND SOLAR PANEL
FOR TWO SPACECRAFT CONFIGURATIONS

Inclination Angle, γ	"A"		"B"	
	Antenna (mw/cm ²)	Solar Panel (mw/cm ²)	Antenna (mw/cm ²)	Solar Panel (mw/cm ²)
0°	3.04	0.532	2.55	0.213
10°	1.76	0.448	1.76	0.182
20°	0.82	0.364	0.82	0.123

Table 6 lists the conductive losses down a single lead wire for various size wires of stainless steel, nickel and "copper weld" (30% copper, 70% iron). The range of values shows very clearly the desirability of using very small wires of low conductance. In terms of the conductive heat leak, stainless steel wires are an order of magnitude better than either nickel or copper-weld wires.

Table 7 presents a comparison of the heat load by I^2R losses in a single lead wire of the sizes and materials considered above. These results are based on an assumed current of 10 milliamperes. The I^2R losses for other current ratings can easily be determined by noting that the power dissipation is proportional to the square of the current.

In comparison with the conductive heat loads, the I^2R loads are very small. Even for a #40 gauge stainless-steel lead wire, the heat load is less than 0.2 milliwatt. However, it should be noted that in a typical system there may be five lead wires with one carrying four times the current of the others. The I^2R losses in this wire would be sixteen times greater than those listed in Table 7.

To put things into proper perspective, the total power dissipation is presented in Tables 8-10 for three lead wire configurations. It is assumed that each configuration consists of four wires of the same material each carrying 1 milliamperes and that a fifth wire carries 41 milliamperes.

3.3.2 Optical Aperture

The heat load on the cold stage from the optical aperture consists of transmitted energy from the spacecraft interior and direct radiation from the lens mount. The intent in this section is to demonstrate a method of determining this heat load and to arrive at a "typical" value for comparison with other heat inputs.

A schematic diagram of an optical aperture is shown in Figure 11. The lens mount is considered to be part of the outer stage of the cooler or at the same temperature. Furthermore, the interior

TABLE 6. CONDUCTIVE HEAT LEAK DOWN A SINGLE LEAD WIRE

Wire Gauge	Power, mw		
	Stainless Steel	Nickel	Copperweld
24	9.67	145.86	160.78
26	6.09	76.06	101.16
28	3.83	47.82	63.60
30	2.41	30.08	40.00
32	1.51	18.92	25.16
36	0.60	7.48	9.95
40	0.24	2.95	3.94

TABLE 7. I^2R HEAT LOAD FROM A SINGLE LEAD WIRE WITH
A CURRENT OF 10 MILLIAMPERES

Wire Gauge	Power, mw		
	Stainless Steel	Nickel	Copperweld
24	0.004	-	-
26	0.006	-	-
28	0.010	-	-
30	0.015	-	-
32	0.025	0.001	0.001
36	0.062	0.002	0.002
40	0.157	0.005	0.005

TABLE 8. TOTAL HEAT LOAD WITH FIVE STAINLESS STEEL WIRES^{*}

I (milliamps)	Power, mw		
	32 Gauge	36 Gauge	40 Gauge
5	7.77	3.30	1.98
10	8.14	4.24	4.34
15	8.76	5.80	8.29
20	9.62	7.98	13.81
25	10.73	10.78	20.90
30	12.09	14.21	29.57

^{*} See text for discussion of distribution of current.

TABLE 9. TOTAL HEAT LOAD WITH FOUR STAINLESS STEEL WIRES
AND ONE COPPERWELD WIRE*

I (milliamps)	Power, mw		
	32 Gauge	36 Gauge	40 Gauge
5	31.24	12.41	5.06
10	31.32	12.62	5.59
15	31.46	12.97	6.48
20	31.65	13.46	7.72
25	31.91	14.10	9.32
30	32.21	14.87	11.28

* See text for discussion of distribution of current.

TABLE 10. TOTAL HEAT LOAD WITH FIVE NICKEL WIRES *

I (milliamps)	Power, mw		
	32 Gauge	36 Gauge	40 Gauge
5	94.60	37.46	14.92
10	94.65	37.60	15.28
15	94.75	37.84	15.88
20	94.98	38.18	16.72
25	95.05	38.60	17.81
30	95.26	39.13	19.13

* See text for discussion of distribution of current.

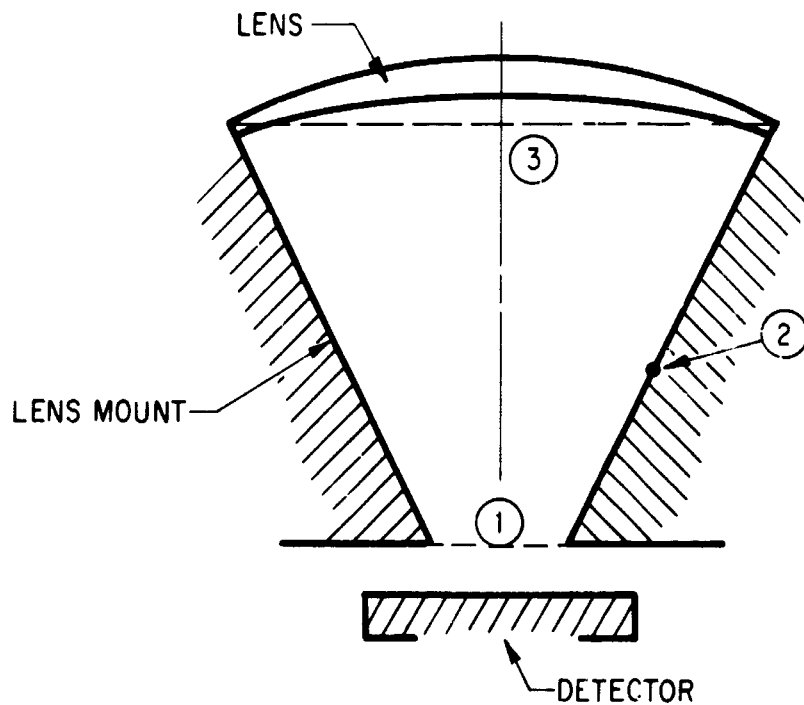


FIGURE 11 SCHEMATIC DIAGRAM OF OPTICAL APERTURE AND LENS MOUNT

walls of the lens mount, surface 2, are considered to be nearly black to prevent stray light from reaching the detector.

The energy transmitted through the optics is considered to be emitted from a black 300°K source (representing the telescope interior which acts as a cavity). Of this energy incident on the lens only a small fraction is transmitted because of a filter on the lens. Of the energy transmitted, only the fraction represented by the view factor from the lens to the detector is considered to pass through the aperture (surface 1) to the detector. Finally, the assumption is made that all energy transmitted through the aperture is absorbed by the cold stage. This latter assumption is conservative. The heat load from transmitted energy can be written as

$$q = A_1 F_{1-3} \tau_f E_b (300^\circ\text{K})$$

where

A_1 = area of aperture

F_{1-3} = configuration factor from aperture to lens

τ_f = transmittance of filter

E_b = black-body emissive power

The value of the transmittance depends on the bandwidth of the filter. For the calculations we will make below, the filter will be assumed to have a 90% transmittance from 9 to 11 microns, and a 10% transmittance elsewhere. For 300°K radiation, this corresponds to an effective transmittance of 20.78%.

To obtain an estimate of the magnitude of the heat load due to transmitted energy we make the following assumptions:

Diameter of aperture = 0.5 cm

Configuration factor from aperture to lens = 0.25

With an effective filter transmittance of 20.78%, the heat load to the cold stage from a 300°K spacecraft interior would be 0.47 mw. Hence, even with conservative estimates, this heat load is typically small in comparison with other heat loads.

Because the inner surfaces of the lens mount are typically near black, we can estimate the heat transfer between the lens mount and cold stage by considering the mount to be a cavity of unity emittance. Since the cavity is at the temperature of the outer stage, the heat load is given by the equation

$$q = A_{\perp} \Delta (\sigma T^4)$$

where $\Delta (\sigma T^4)$ is difference in the emissive powers of the outer stage and the cold stage. For the example considered above of an aperture diameter of 0.5 cm and assuming an outer-stage temperature of 240°K, the heat load would be approximately 3.5 mw. This represents a fairly significant heat load and indicates that the interchange between the cold stage and the lens mount must be considered in a thermal design analysis.

4.0 WEIGHT AND VOLUME CONSIDERATIONS FOR LCE COOLER

In this section we will develop a method for determining the size of radiative cooler required to achieve a desired range of operating temperatures. The emphasis will be on achieving a given cold-stage temperature with the smallest and lightest cooler. We will focus our attention on the "A" and "B" spacecraft configurations. The radiation cooler designs will be slightly different for these two cases, and they will be treated separately.

For reference, Figure 12 shows a schematic representation of the cooler. As discussed in Section 2, we will assume the shielding surface to be a body of revolution - in this case a cone frustum. The various radii defined in the figure will be used during parametric studies.

4.1 ATS-F Configuration "A"

For this spacecraft configuration we postulate the construction of an outer stage which rejects all radiation incident at angles greater than 45° from the normal. The outer stage is assumed to be a specularly reflecting cone frustum.

In the thermal design of a radiative cooler, it is important to be able to predict the fraction of external heat inputs which is transmitted through the outer stage to the cold stage and the fraction which is absorbed by the outer stage. As an example of the type of analysis necessary, illustrations of the transmittance and absorptance of a specular cone frustum designed to reject radiation incident at angles greater than 45° are shown in Figures 13 and 14. The apparent absorptance and transmittance are defined by the following equations:

$$P_A = (\pi R_2^2) (S \cos \psi) \alpha_a$$

$$P_T = (\pi R_2^2) (S \cos \psi) \tau$$

where

P_A = power absorbed on cone walls

P_T = power transmitted through lower opening of cone
frustum (to cold stage)

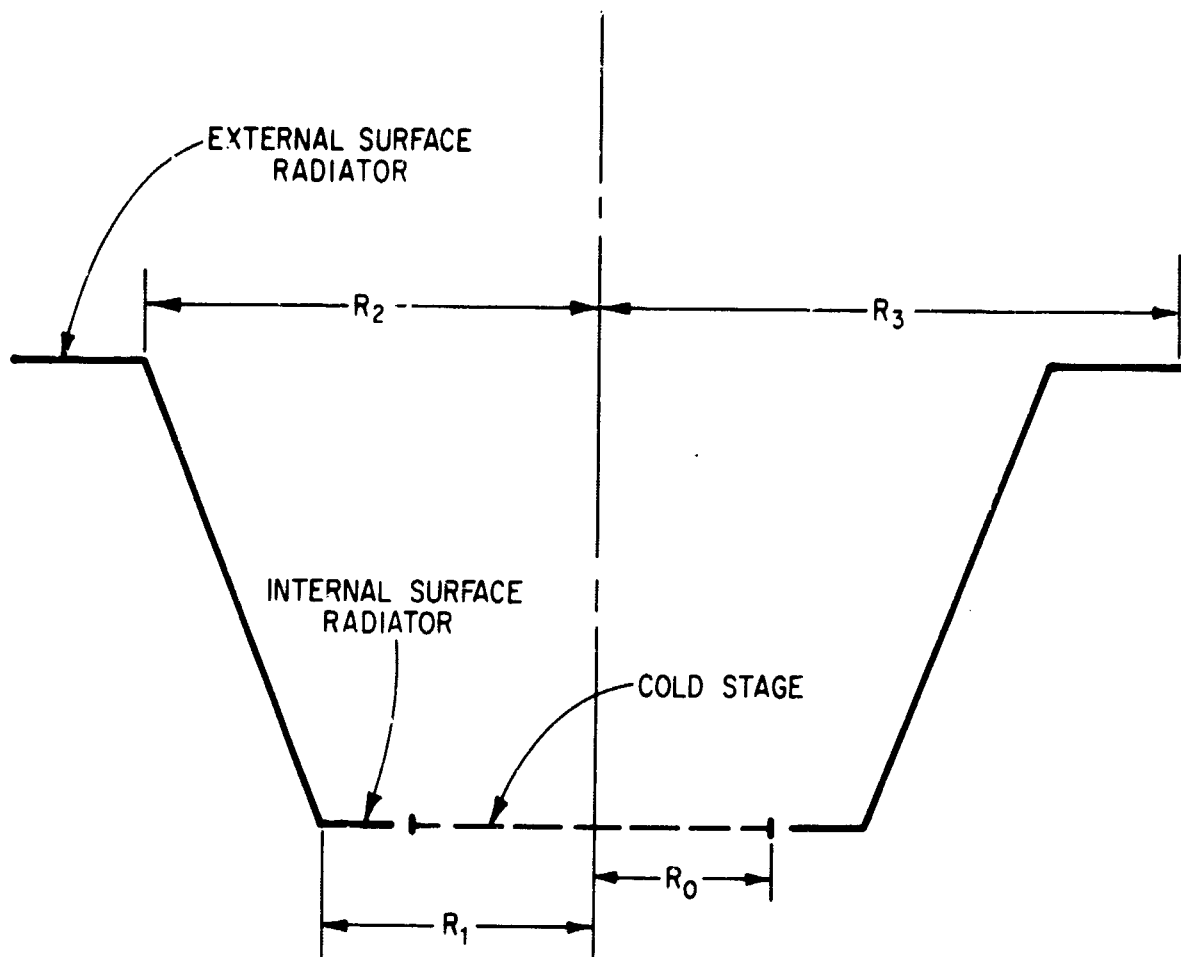


FIGURE 12 SCHEMATIC REPRESENTATION OF RADIATION COOLER

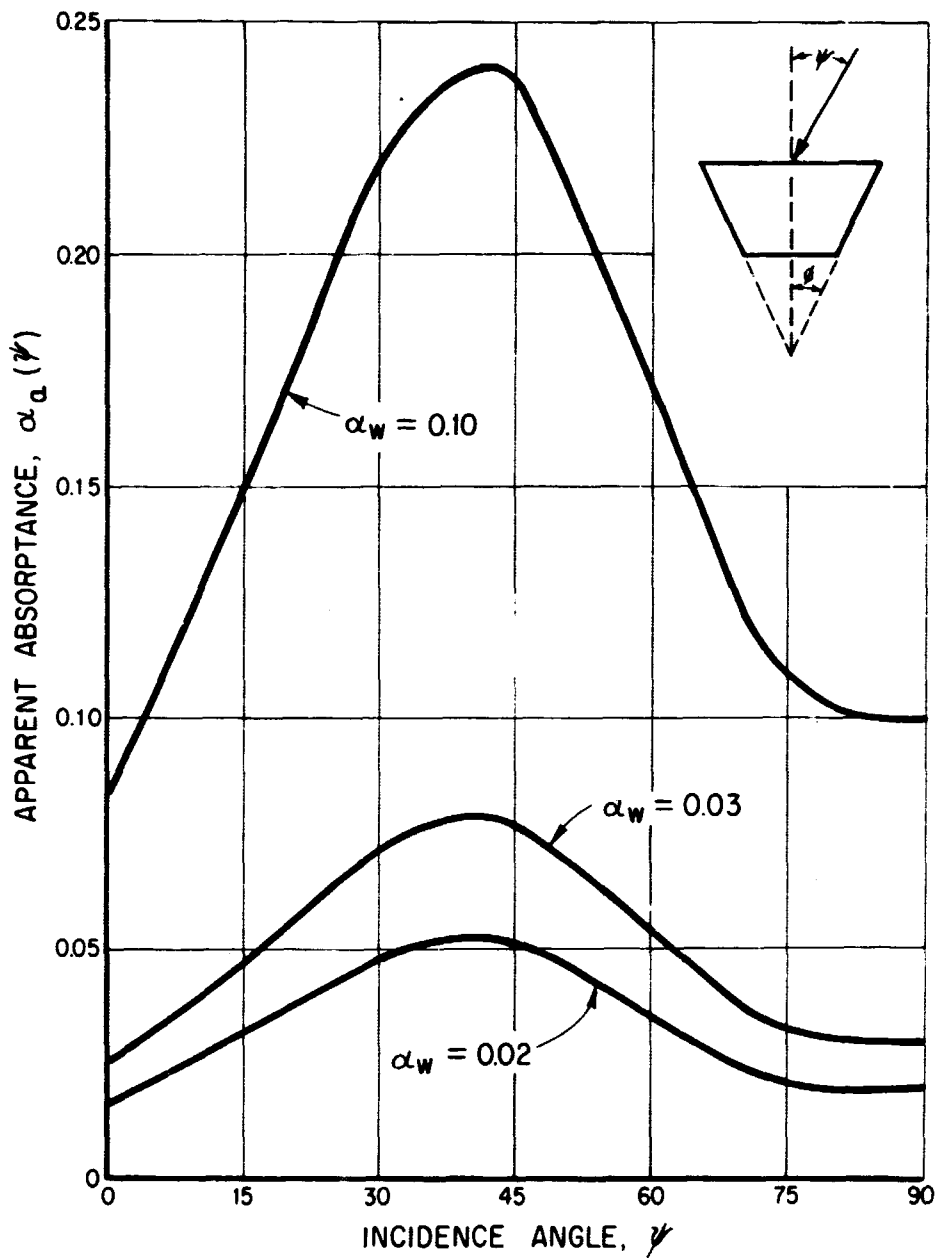


FIGURE 13 APPARENT ABSORPTANCE OF SPECULAR CONE FOR PARALLEL INCOMING RADIATION

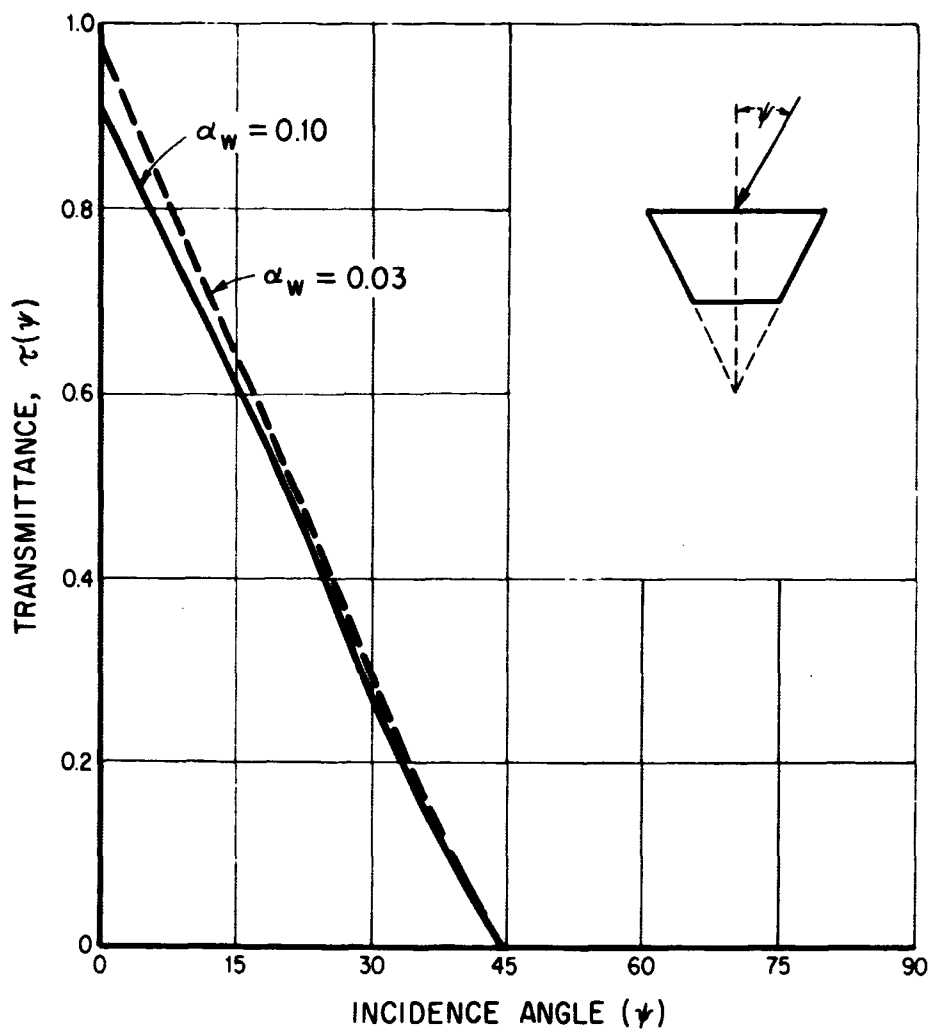


FIGURE 14 TRANSMITTANCE OF SPECULAR CONE FOR PARALLEL INCOMING RADIATION

R_2 = radius of conical opening

S = intensity of incident radiation

The method for computing these quantities is described in Appendix I.

Determination of the absorptance as a function of incidence angle is important for determining the power input to the outer stage from direct solar heating as a function of orbital position. As will be shown later, direct solar heating represents the major power input to the radiative cooler. Figure 13 shows the absorptance to vary significantly with incidence angle. Values for the absorptance and transmittance are also needed to determine the radiative interchange view areas between the cone and the cold stage as developed in Appendix I.

As discussed in Section 2, heat input to the cold stage is related to the cone temperature. A goal of the design is to minimize this input by maintaining a low outer-stage temperature. The usual method is by the use of surface radiators of low α/ϵ material. We will investigate here the range of operating temperatures likely to be achieved.

The calculations were performed assuming a power input to the radiator equal to the average over an orbit of the summer solstice. Heat inputs to the cooler included direct solar heating, reflected solar energy from the spacecraft and the earth and direct emission of energy from the spacecraft and earth. The interior surfaces of the cone were assumed to have an emittance of 0.03 and a solar absorptance of 0.10.

As illustrated in Figure 12, the cooler was assumed to have both an inner surface radiator and an external surface radiator. The ratios of the inner to outer radii of the surface radiators are the parameters R_1/R_0 and R_3/R_2 , respectively. For the majority of calculations, the properties of the surface radiator were assumed to be:

$$\epsilon_{\text{RAD}} = 0.85$$

$$\alpha_{\text{RAD}} = 0.20$$

These values are characteristic of a white-painted thermal control surface. Some calculations were also performed assuming the external surface radiator has the properties of a second-surface mirror (OSR) with an emittance of 0.85 and a solar absorptance of 0.06.

Figure 15 illustrates the range of operating temperatures at the summer solstice for various surface radiator configurations. Incorporation of an external surface radiator results in a significant decrease in the cone temperature. At a radius ratio of $R_3/R_2 = 1.4$, the area of the external surface radiator is nearly equal to the area of the cone mouth.

The influence of the radius ratio of the inner-surface radiator on the temperature is much less than that of the external surface radiator. For the range of radius ratios considered here the area of the inner-surface radiator is small in comparison with the area of the external surface radiator.

Even using a very large external surface radiator, the cone temperature is approximately 280°K. The reason this high temperature is achieved is because the solar absorptance of the external surface radiator and the apparent absorptance of the specular cone are high (see Figure 13). By using an OSR coating for the external surface radiator, the power input to the radiator from solar heating is significantly decreased. At a radius ratio of $R_3/R_2 = 1.4$, the use of this surface results in a 30°K decrease in the temperature of the outer stage. Subsequent calculations will be based on the use of an OSR-coated external surface radiator.

The ratio of the radius of the external surface radiator to the radius of the cold patch is also shown in Figure 15. Numerical values for the radius of the base and the height of the cone will be presented below.

Once a design for the outer stage (cone) of the radiation cooler has been chosen, the temperature of the cold stage as a function of the power to the cold stage can be determined. These results are

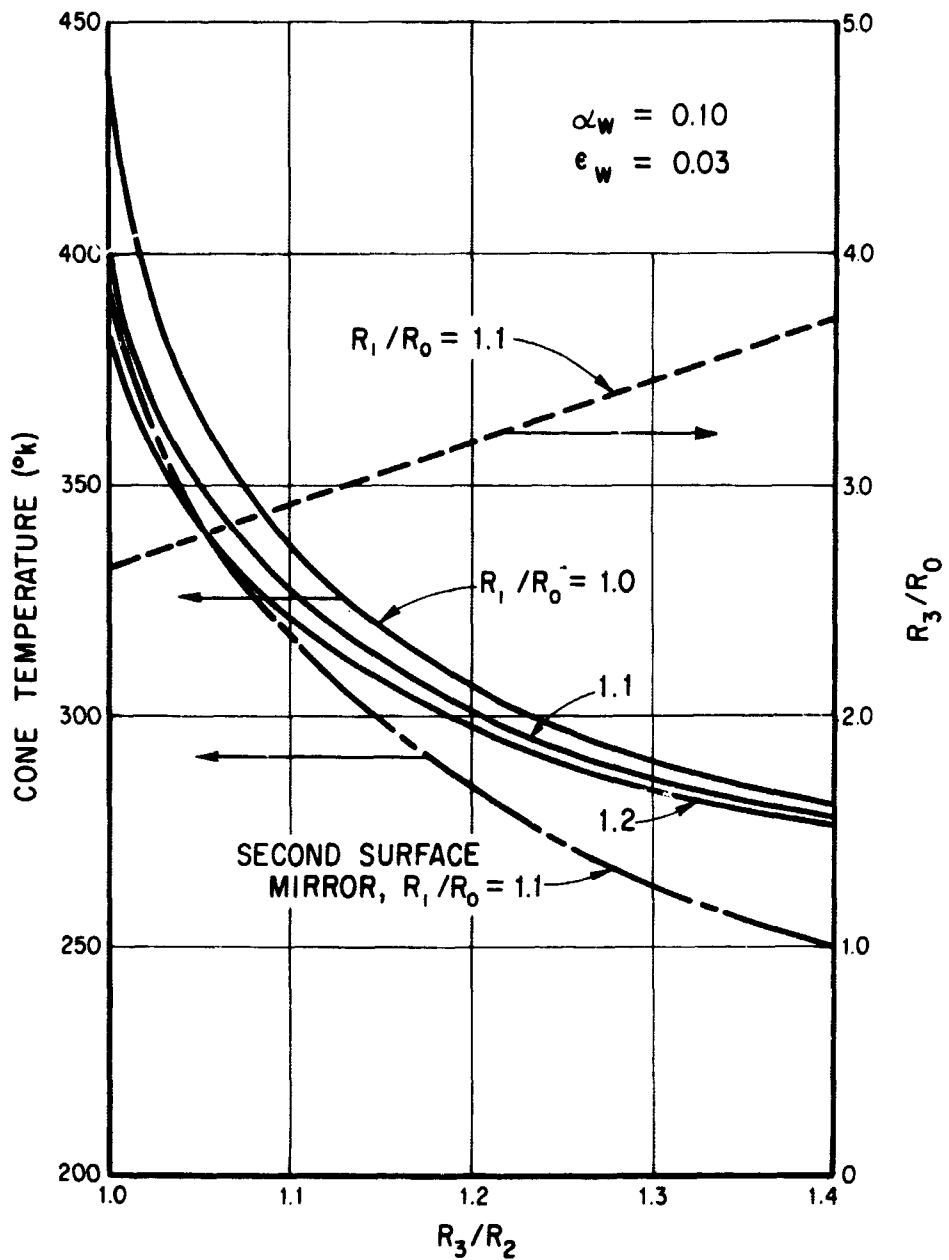


FIGURE 15 INFLUENCE OF SURFACE RADIATOR DESIGN ON CONE TEMPERATURE; CONFIGURATION "A" DESIGN

shown in Figure 16 for surface radiator radii ratios of $R_1/R_0 = 1.1$ and $R_3/R_2 = 1.4$ (see Figure 12). The power to the cold stage includes all power inputs other than radiative interchange between the cone and the cold stage. For these calculations, the emittance of the cold stage was taken to be 0.93; the emittance of the specular conical surface was taken to be 0.02. As before, these calculations were performed assuming a power input to the cooler equal to the average over an orbit at the summer solstice.

For a given size radiator, the temperature of the cold stage is highly dependent on the net power to the cold stage; and at a given power level, using an OSR coating on the external surface radiator results in a lowering of the cold-stage temperature of approximately 7-8°K. If the solar absorptance of the inner surfaces of the outer stage were decreased from 10% to 8%, the temperature drop of the cold stage would be approximately 2-3°K.

The results also show that even with no power input to the cold stage, the temperature at the summer solstice would be from 110°K to 120°K depending upon the type of external surface radiator used. These calculations are for the time of year when the heating rates are maximum. Detailed performance calculations for a typical design will be presented in Section 5 for both the summer and winter solstice and the equinox.

Figure 16 can be used to size the radiator needed to obtain a given operating temperature for a specified power to the cold stage. These results are shown in Figures 17 and 18. The weight calculations were performed assuming a characteristic cone wall thickness of 1/16 inches. Because of mechanical considerations, the thickness might be different from this and may even be a function of the size of the radiator. Nevertheless, the values listed provide a frame of reference for evaluating the various designs. It was assumed that the cone was constructed of 6061 aluminum.

As discussed earlier, the power input to the cold stage by the detector may be in the neighborhood of 40 milliwatts. For the sake

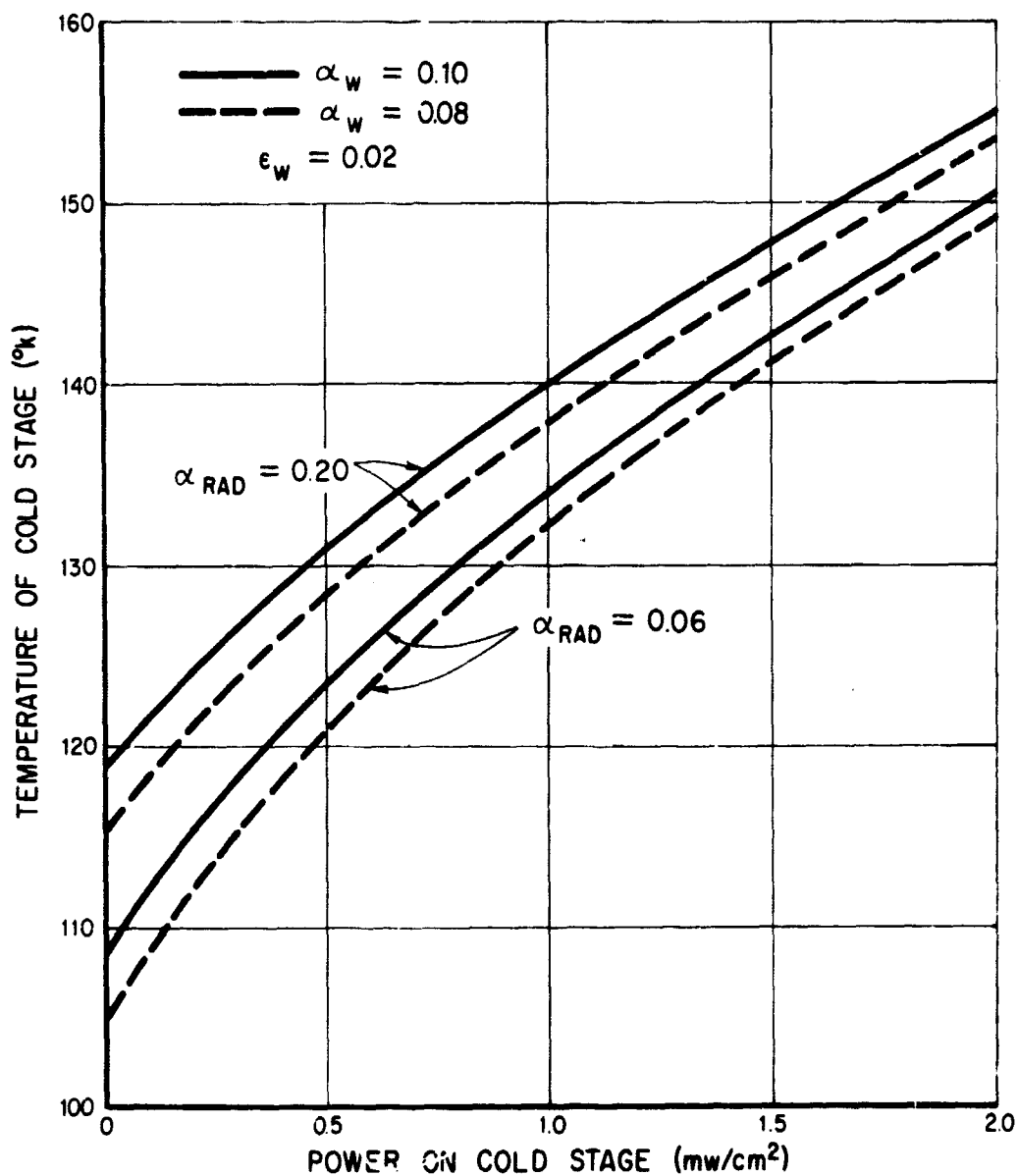


FIGURE 13 INFLUENCE OF POWER TO COLD STAGE ON COLD-STAGE TEMPERATURE FOR CONFIGURATION "A" DESIGN

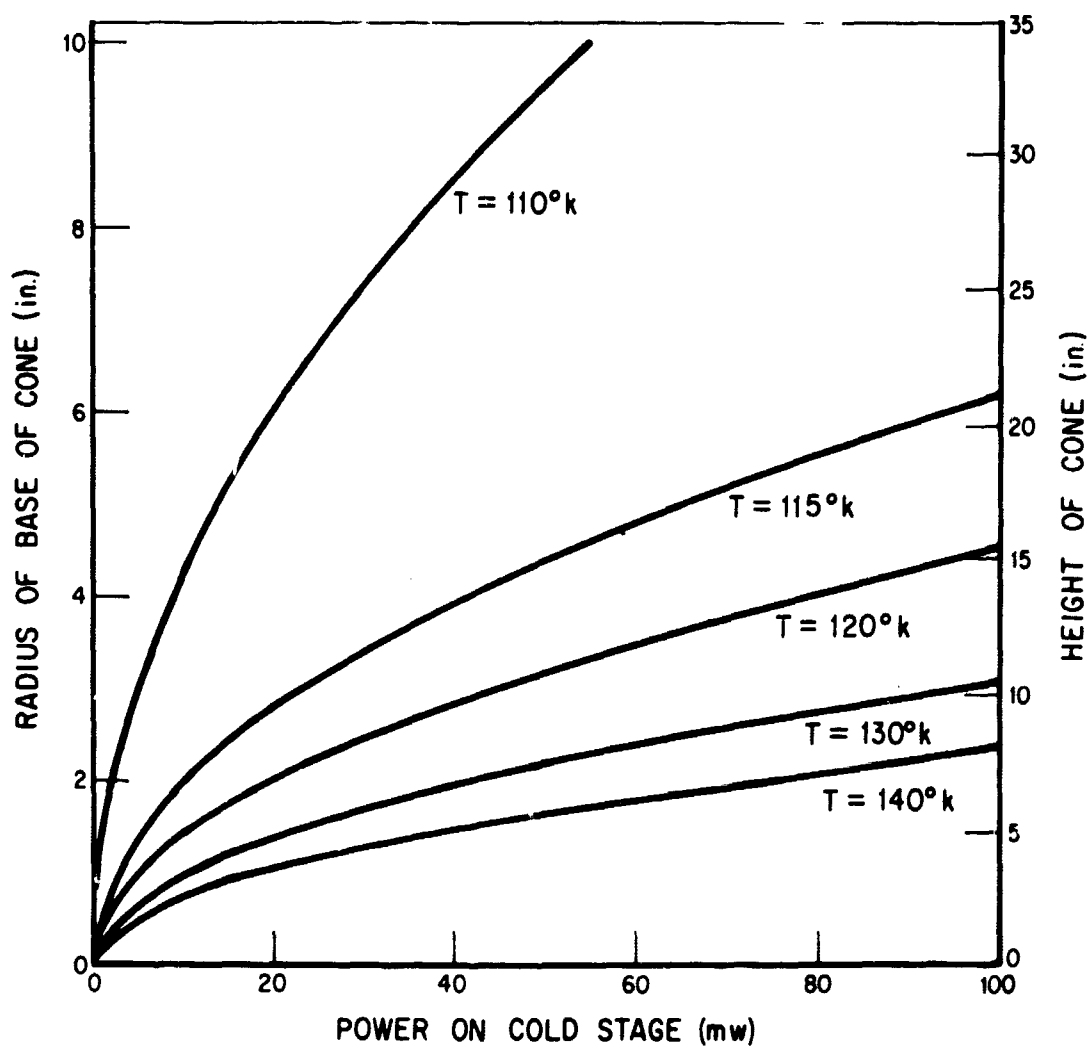


FIGURE 17 RADIATIVE COOLER SIZE NEEDED TO MAINTAIN SPECIFIED COLD-STAGE TEMPERATURE FOR CONFIGURATION "A" DESIGN

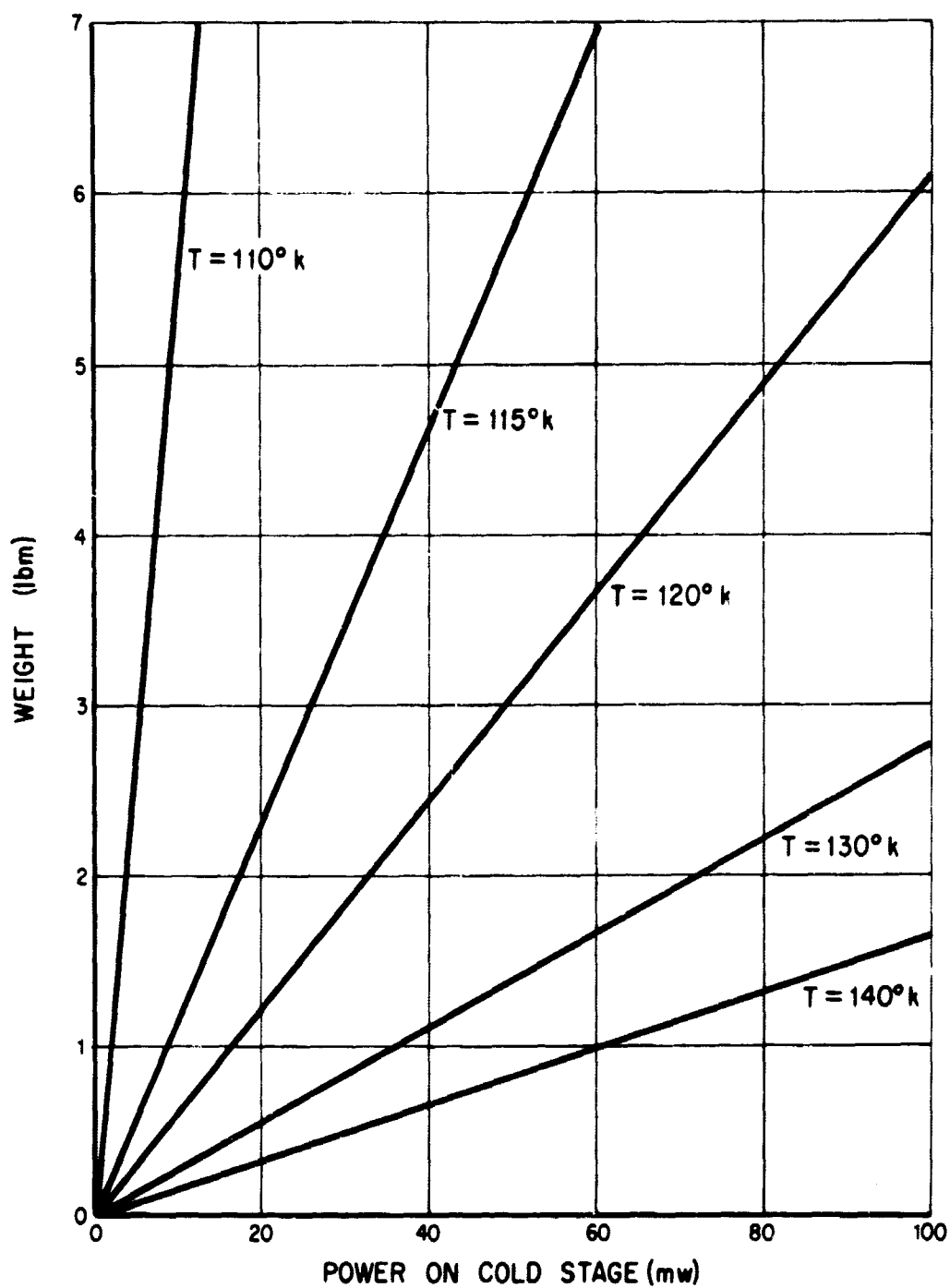


**FIGURE 18 APPROXIMATE WEIGHTS OF CONFIGURATION "A"
RADIATIVE COOLER; THICKNESS = 1/16 INCH**

of discussion, let us assume that other power inputs are in the neighborhood of 20 mw, giving a total of 60 mw of power to be dissipated by the cold stage.

The size and weight of the radiator necessary to dissipate a given power from the cold stage increases rapidly as the specified cold-stage temperature is decreased. For a specified cold-stage temperature of 120°K and a net power of 60 mw on the cold stage, the cooler would be nearly 12" high and weigh approximately 3 1/2 lbs. By accepting operation at a cold-stage temperature of 130°K, the size and weight can be decreased considerably.

4.2 ATS-F Configuration "B"

For this spacecraft configuration we postulate the construction of a conical outer stage which rejects all radiation incident at angles greater than 52° from the normal. In all other respects, the radiator is considered similar to that discussed in Section 4.1. The method of analysis performed here is also similar.

Various size surface radiations were investigated to evaluate expected outer-stage temperatures. For the same radii ratios R_1/R_0 and R_3/R_2 , the outer-stage temperatures were approximately 10°K lower for the configuration "B" design. Surface radius ratios of $R_1/R_0 = 1.1$ and $R_3/R_2 = 1.4$ were chosen for calculations of cold-stage temperatures versus power to the cold stage. These ratios are the same as those used in the configuration "A" analysis.

Figure 19 illustrates the dependence of cold-stage temperature on power to the cold stage. As before, this power includes all power inputs other than the radiative interchange between the cold stage and the conical outer stage.

Comparison between Figures 19 and 16 reveals that the dependence of cold-stage temperature on power and on the optical properties of the external surface radiator is similar for the two cases. However, for the same conditions, the cold-stage temperature achieved in the configuration "B" design is approximately 10-15°K lower than the corresponding temperature for the configuration "A" design.

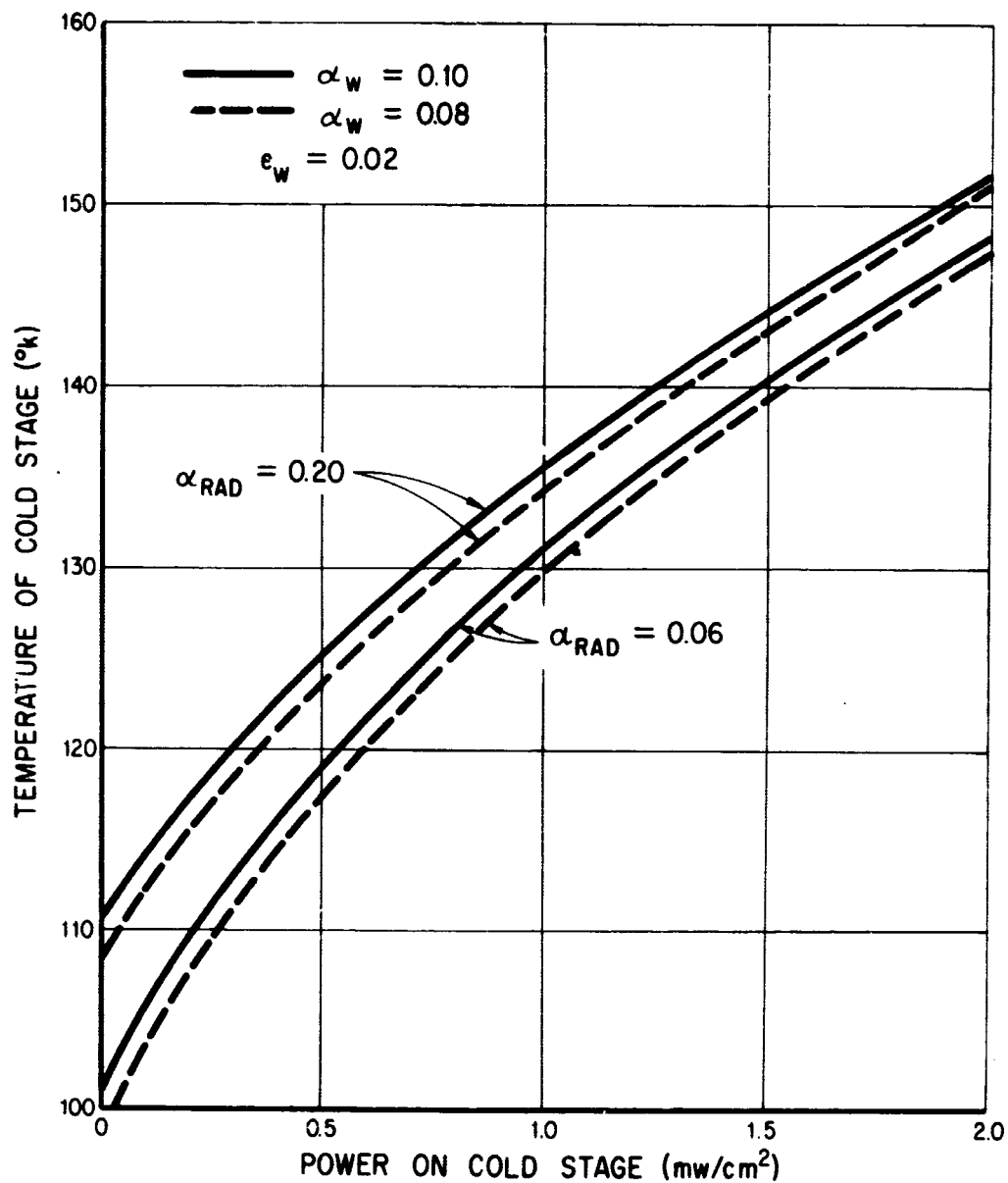


FIGURE 19 INFLUENCE OF POWER TO COLD STAGE ON COLD-STAGE TEMPERATURE FOR CONFIGURATION "B" DESIGN

This difference is reflected in the radiator size required to achieve a given cold-stage temperature. This can be seen by comparing Figures 20 and 17. As a basis of comparison, with a total of 60 milliwatts power dissipation to the cold stage and an operating temperature of 120°K, the radiator height would be about 6 inches for a configuration "B" design; in the configuration "A" design the required height was approximately 12 inches. Comparison of Figures 21 and 18 reveals that the corresponding weight of a configuration "B" design is about half that for a configuration "A" design.

The primary difference between the two designs is the cone angle required to reject the external heat inputs to the radiative cooler. The field of view of a control surface to external heat sources is larger in the configuration "A" case; hence, a cone with a smaller cone half-angle is needed to shield the low-temperature stage. This results in an increased view area from the cone to the cold stage - hence, the higher temperatures.

The results presented in this section were for the worst case of operation at the summer solstice. Results for other times of the year will be presented in the following chapter for a typical configuration "A" design.

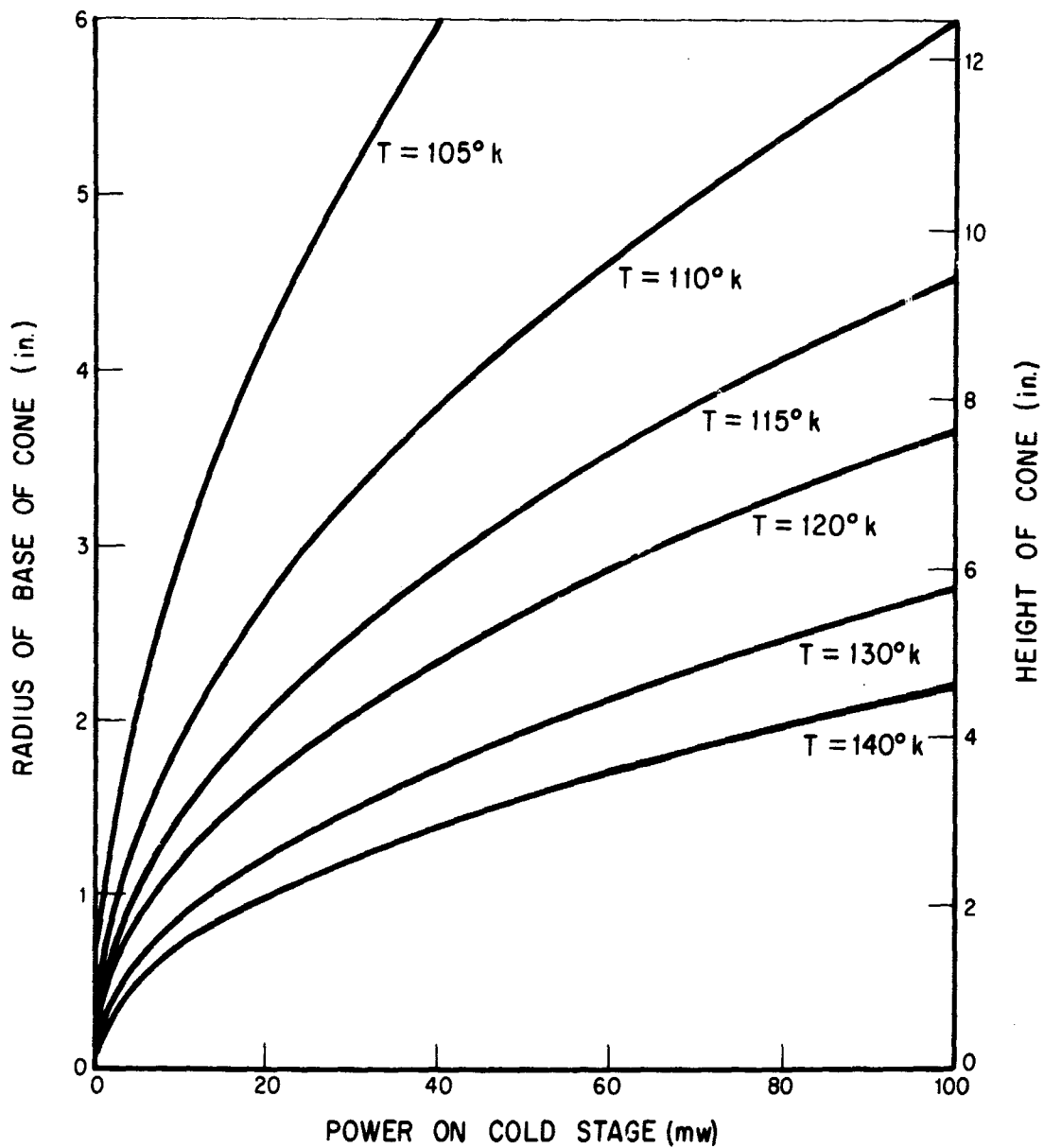


FIGURE 20 RADIATIVE COOLER SIZE NEEDED TO MAINTAIN SPECIFIED COLD-STAGE TEMPERATURE FOR CONFIGURATION "B" DESIGN

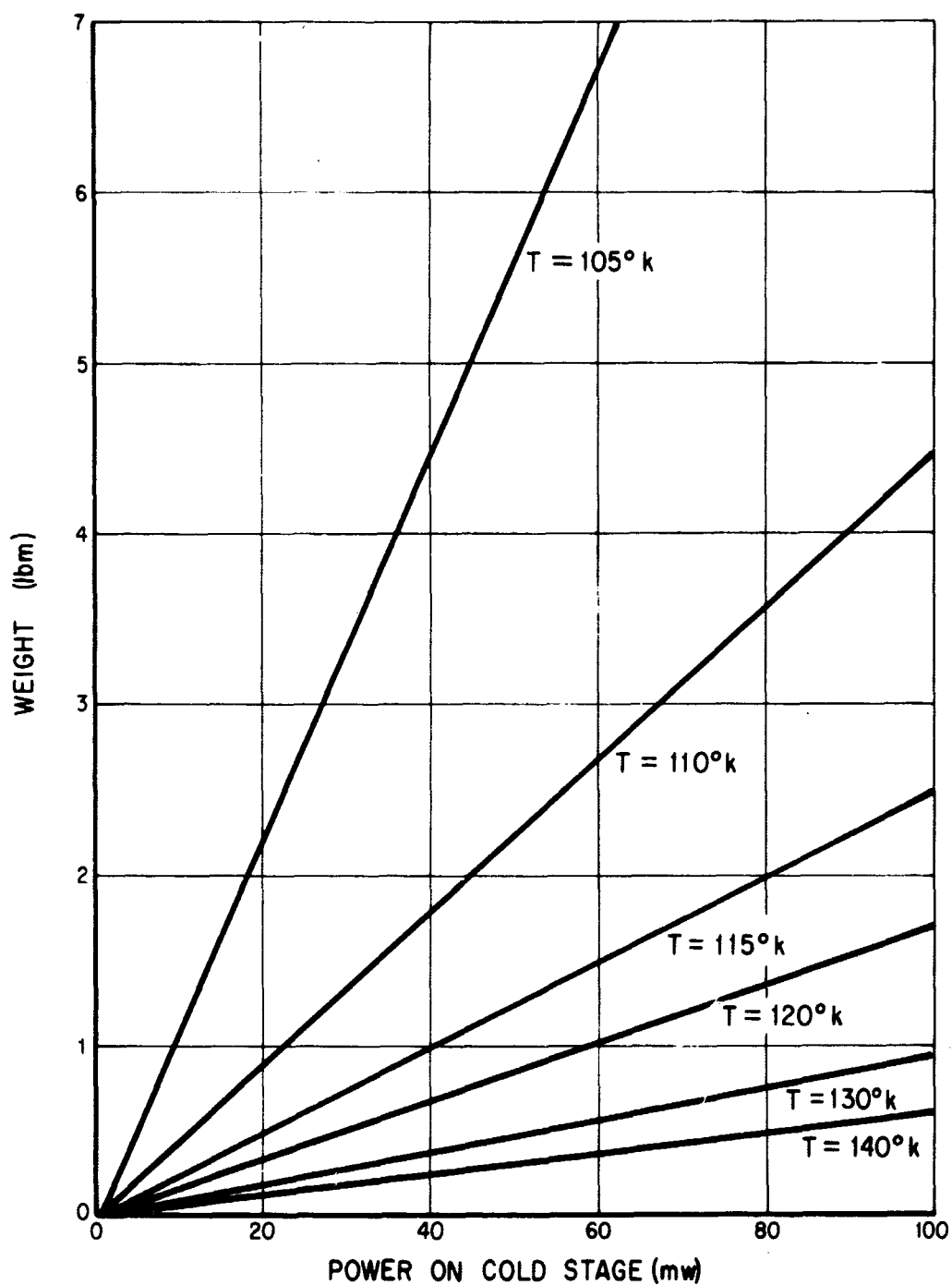


FIGURE 21 APPROXIMATE WEIGHTS FOR CONFIGURATION "B"
RADIATIVE COOLER; THICKNESS = 1/16 INCH

5.0 PERFORMANCE CALCULATIONS FOR SPACECRAFT CONFIGURATION "A"

In this section the transient thermal behavior for the typical configuration "A" radiative cooler discussed in Section 4 will be determined over an orbit for operation at three times of the year - the summer solstice, the winter solstice, and an equinox. A base design will be specified and detailed temperature distributions performed. Also, the influence on predicted temperatures of uncertainties in properties and/or changes in design will be discussed. These results are useful in identifying areas of the design which deserve very close attention.

5.1 Specification of Base Design

A conceptual view of the base design is shown in Figure 5. The outer stage (cone) is considered designed to reject all radiation entering at angles greater than 45° with respect to the cone axis; the cone axis is tilted at 20° from the earth axis. Based on the results presented in Figures 16 and 17, the area of the cold stage containing the detectors was chosen to be 125 cm^2 . Also from the results presented in Section 4 the radii ratios of the inner and outer surface radiators were chosen to be

$$R_1/R_0 = 1.1$$

$$R_3/R_2 = 1.4$$

(See Figure 12 for definition of radii.) The overall dimensions of the radiator are:

Diameter of base: 5.46 inches

Diameter of external
surface radiator: 18.48 inches

Height of radiator: 9.34 inches

The radiative properties of the surfaces are given in the following table.

	$\frac{\alpha}{s}$	ϵ
Upper surface - cold stage	0.93	0.93
Bottom and side of cold stage	-	0.025
Cone interior	0.10	0.02
External radiator	0.06	0.85
Internal surface radiator	0.20	0.85

The conductances between the spacecraft and cone and the cone and cold stage were taken to be 0.3 mW/K and 0.1 mW/K, respectively. The shielding factors¹⁰ of the MLI insulation from the spacecraft to the outer stage and from the outer stage to the cold stage were taken to be 100 and 200, respectively.

The effect of changes in these assumed values will be discussed later.

It was also assumed that both the radiator outer stage and cold stage are constructed of aluminum. The thermal masses associated with these stages are 0.61 watt-hr/°K and 0.011 watt-hr/°K for an equivalent outer stage thickness of 1/8" and a cold stage equivalent thickness of 0.10".

For the base case, the power to the cold stage is considered to be 40 milliwatts comprising bias and local oscillator power dissipation, I^2R heating through lead wires and the heat inputs from the optical aperture.

Heat input to the radiator outer stage and external surface radiator consist of a small constant power input, and a larger variable power input from direct solar heating. Values for the constant heat inputs are:

10. The shielding factor is defined as the reciprocal of the effective emittance of the MLI blanket.

Direct emission from antenna	0.02 watt
Reflection of solar energy off antenna	0.13 watt
Direct emission from solar panel	0.70 watt
Reflection of solar energy off solar panel	0.09 watt
Direct emission from earth	0.12 watt
Reflection of solar energy from earth	<u>0.05 watt</u>
TOTAL:	1.11 watt

Values used for direct solar heating are:

- (1) Summer solstice 1.18 to 24.68 watt
- (2) Equinox 0 to 7.58 watt
- (3) Winter solstice 0

The solar heating rates vary with position of the spacecraft around the orbit (see Figure 8).

5.2 Orbital Temperature Calculations

Figure 22 illustrates the cold-stage temperature history as a function of orbital position during the summer solstice. The calculations are for the conical cooler design specified in the preceding section.

For the base case of a cold-stage thermal mass of 0.011 watt-hr/°K, the temperature range is from 107°K to 125°K. An effect of the thermal mass is to decrease the temperature fluctuation that would result were the thermal mass very small. Doubling the thermal mass, in fact, decreases the temperature fluctuation by nearly 6°K; for this latter case the temperature range is from 110°K to 122°K. Hence, an important finding is that the temperature fluctuation experienced by the cold stage can be significantly influenced by its thermal mass.

Figure 23 illustrates the temperature history of the outer-stage cone at the summer solstice. Here, the influence of thermal mass on temperature response is seen to be much less than that found for the

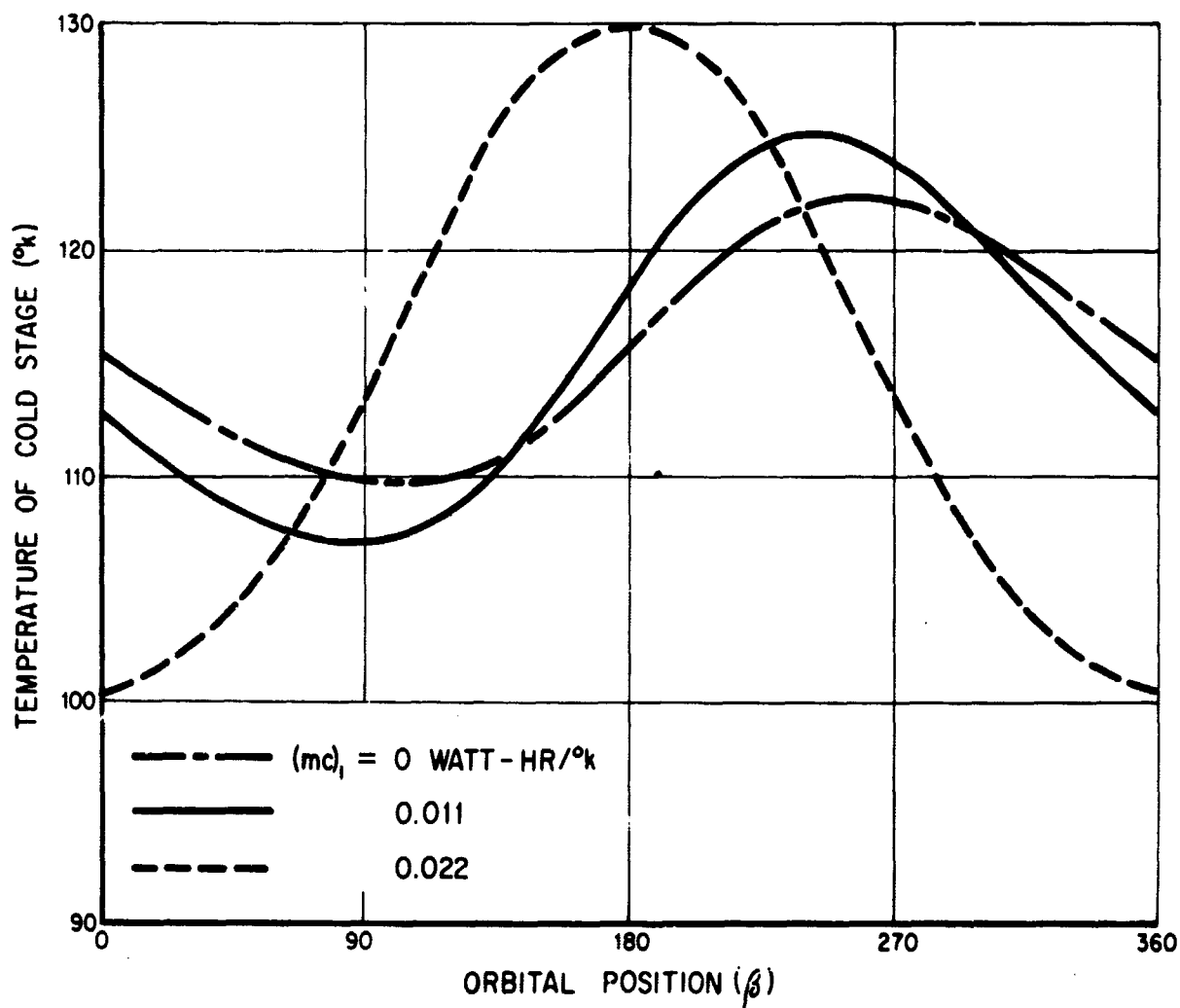


FIGURE 22 VARIATION OF CONFIGURATION "A" COLD-STAGE TEMPERATURE WITH ORBITAL POSITION AT SUMMER SOLSTICE; POWER TO COLD STAGE IS 40 MW

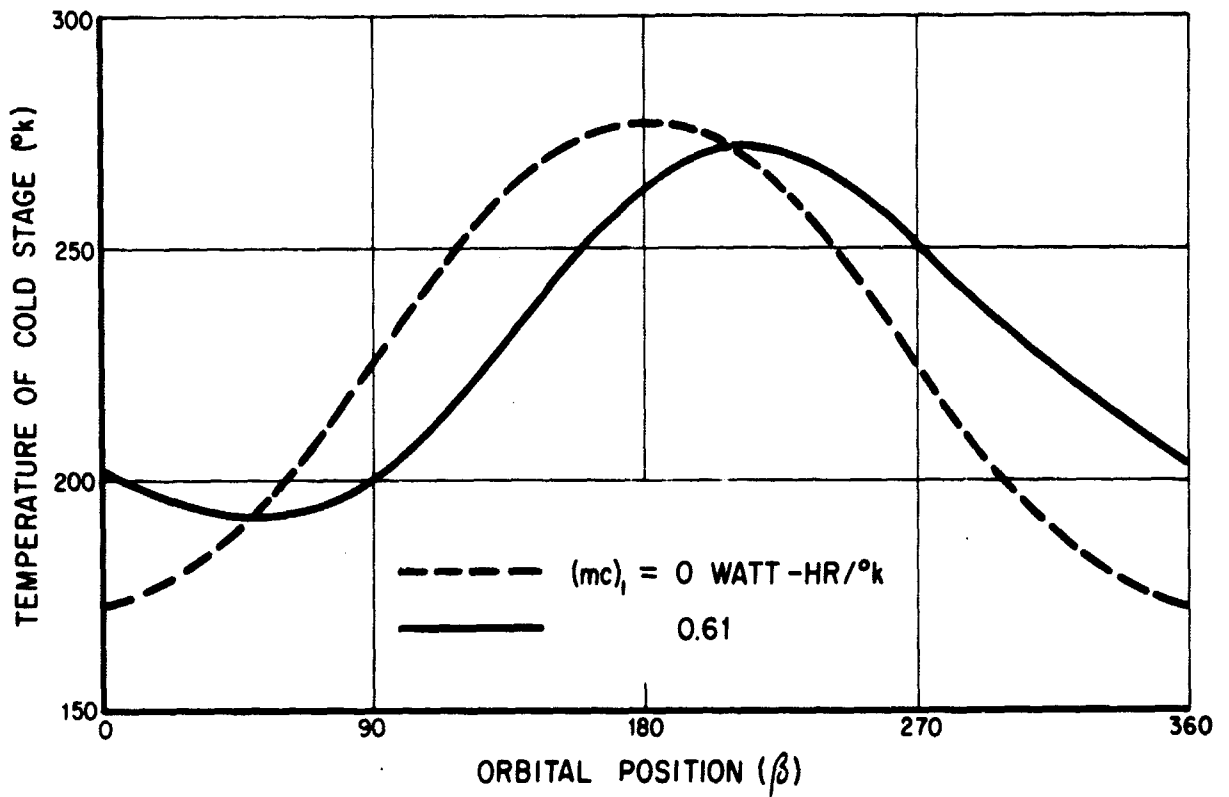


FIGURE 23 VARIATION OF CONFIGURATION "A" CONE TEMPERATURE WITH ORBITAL POSITION AT SUMMER SOLSTICE

cold stage. The mass calculated for the base case design decreases the cone temperature fluctuation by 27°K out of 105°K. While not shown in the figures, the corresponding decrease in the cold-stage temperature fluctuation is less than 6°K. Since most of the radiator weight is in the outer stage, the radiator outer stage can be designed to optimize weight and mechanical performance without significantly affecting the cold-stage temperature fluctuation.¹¹

The maximum and minimum temperatures experienced by the cold stage are illustrated in Figure 24 for operation at three different times of the year and as a function of the power input to the cold stage. Since the radiator axis is considered tilted 20° from the earth axis, some temperature fluctuation is present also during the equinox. During the winter solstice, the temperature will be essentially constant around an orbit.

The average operating temperature is also highly dependent on the time of year. For the base case of 40 mw power to the cold stage, the mean temperatures are approximately 115°K, 102°K and 96°K at the summer solstice, equinox, and winter solstice, respectively. The variation of temperature with power input is nearly the same for all times of the year.

5.3 Influence of Design Uncertainties on Predicted Temperatures

A determination of the influence of design uncertainties on predicted cold-stage temperatures is important not only as an aid in identifying aspects of the design which must be most carefully controlled but also as an aid for improving the design from a thermal viewpoint. We will demonstrate this approach here and point out especially important design uncertainties.

The relative importance of the various heat transfer paths on cold-stage temperature can be estimated from a heat budget to the cold stage. Table 11 presents a heat budget for a base case design radiator

11. One word of caution need be stated here. The outer stage should be a good thermal conductor to minimize local hot spots due to uneven solar heating.

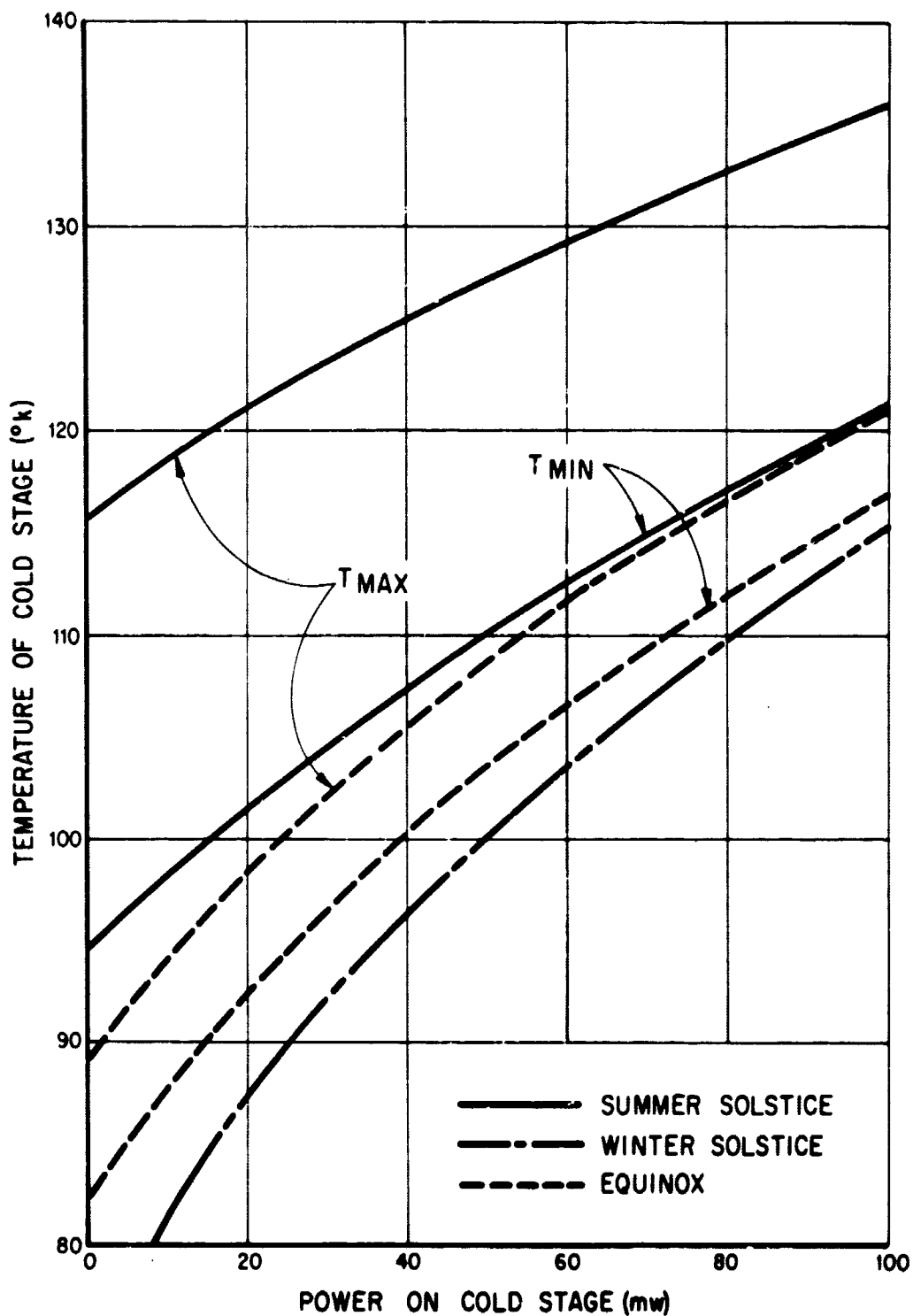


FIGURE 24 MAXIMUM AND MINIMUM COLD-STAGE TEMPERATURES AS A FUNCTION OF POWER TO COLD STAGE; CONFIGURATION "A" DESIGN

TABLE 11. COLD-STAGE POWER BUDGET FOR SUMMER SOLSTICE
AND AT 180° ORBITAL POSITION

	Power (mw)
1) Radiative interchange between cone walls and top surface of cold stage	73.82
2) Radiative interchange between cone and side of cold stage	16.63
3) Energy transfer through MLI to bottom of cold stage	16.46
4) Conduction from cone outer stage	14.54
5) Power input from bias power dissipation, local oscillator power dissipation, I^2R losses in lead wires, optical throughput	40.00
	161.45

with the spacecraft at an orbital position of 180° and during the summer solstice. The values listed in Table 11 show that the most important heat inputs are the power specified from bias power dissipator, etc., and from radiant interchange between the specular cone and the upper surface of the cold stage.

A series of calculations was performed to determine the affect of uncertainties associated with these powers and other heat inputs. The results have been reduced to the form of influence coefficients which are given in column 3 of Table 12. The table reveals that the cold-stage temperature is highly dependent on the emittance of the cone surfaces and the emittance of the side surface of the cold stage. To a somewhat lesser extent, it is influenced by the low-temperature support conductance, the solar absorptance of the cone walls, and the solar absorptance of the surface radiator.

Table 12 also provides an uncertainty analysis in thermal design predictions of cold-stage temperature. Listed in the table are the nominal values of variables used in the base design as well as examples of uncertainty intervals in these variables. The product of the uncertainty interval and the corresponding influence coefficient yields an uncertainty in the predicted temperature. The last column in the table is used to compute the RMS uncertainty in temperature. The calculation procedure is shown at the end of the table.

For the example given in Table 12 the uncertainty in cold-stage temperature was $\pm 4.5^\circ\text{K}$. The largest contributions to the uncertainty are related to the emittance of the conical surfaces and the external surface radiator emittance. These results would help identify these variables as areas where close attention should be paid.

The same type of analysis could be carried out for other times of the day or year. It should be stated that Table 12 is meant to be illustrative of a technique and not a detailed examination of uncertainties in a given design analysis.

TABLE 12. UNCERTAINTY ANALYSIS OF COLD STAGE;
SUMMER SOLSTICE; 180° ORBITAL POSITION

y - value of variable used in base design
T - maximum temperature of radiator
 Δy - uncertainty interval in variable
 δ - uncertainty interval in radiator temperature

	y	dT/dy	Δy	$(\frac{dT}{dy} \Delta y)^2$
1. Cone emittance	0.02	466K	0.005	5.43
2. Conductance of low-temperature supports (mw/°K)	0.1	23.7K ² /mw	0.05mw/°K	1.40
3. Insulation factor between cone and bottom of cold stage	200	-0.0312K	50	2.43
4. Emittance of side of cold stage	0.025	144.5K	0.002	0.08
5. Spacecraft temperature (K)	300	0.03	20	0.36
6. Solar absorptance of cone walls	0.10	68.5K	0.02	1.88
7. Emittance of cold stage	0.93	-16.50K	0.04	0.44
8. Emittance of external surface radiator	0.85	-28.0K	0.08	5.02
9. Solar absorptance of surface radiator	0.06	85.0K	0.02	2.89
10. Conductance between cone and spacecraft (mw/K)	0.3	0.1	0.1	-
11. Insulation factor between radiation cooler and spacecraft	100	-0.0156K	20	0.10

The RMS uncertainty in the temperature prediction is

$$\delta = \sqrt{\sum (\frac{dT}{dy} \Delta y)^2} = \pm 4.5^\circ\text{K}$$

6.0 SIMULATION REQUIREMENTS

The general requirements for environmental testing of components and experiments applicable to the LCE radiative cooler are outlined in an ATS-F and G specification.¹² However, because of the low-temperature requirements for the detector subsystem, consideration must be given to special test requirements not completely covered in the "Thermal Balance" and "Thermal-Vacuum" sections (paragraphs 4.1.8 and 4.1.9) of the referenced document. In the following discussion, attention will be directed toward the design qualification and flight-acceptance testing of the radiative cooler as a subsystem of the LCE experiment. These testing requirements will not differ appreciably from the requirements for the environmental testing on a spacecraft system level.

There are several important considerations in the thermal-vacuum testing of a passive radiator system designed to operate in the 100°K region. One consideration is the maximum allowable temperature range of the "black" cold-wall shroud used to simulate the deep-space environment. Another is the requirement for simulating direct solar heat inputs and the heat inputs from the ATS antenna and solar panels.

In any type of thermal-vacuum test the interior of the chamber must have cryogenically cooled walls. Most chambers are equipped with low-reflectance LN₂-cooled shrouds which operate in the 77 to 90°K range; however, only several facilities have the capabilities for cooling the shrouds below LN₂ temperature using a helium gas refrigerator.

A very simple model can be used to obtain the approximate difference between the measured temperature of a passive radiator in a thermal-vacuum chamber and the temperature that would occur in the space environment where the sink temperature is of the order of several degrees.

12. ATS-F and G Environmental Test Specification for Components and Experiments, S-230-ATS-2, NASA GSFC, January 1969.

Consider a plate (simulating the low-temperature stage) which is dissipating power to a "black" low-temperature sink. The difference between the radiator temperature measured in a simulated test and the design temperature of the radiator (calculated for the space environment) is given by

$$T_r^4 - T_d^4 = T_s^4$$

where

T_s - shroud temperature in the test chamber

T_r - radiator temperature measured in test

T_d - design temperature of the radiator for the space environment

For a typical design temperature, $T_d = 120K$, the measured test temperatures and the corresponding errors as a function of the temperature of the cold-wall shroud are given below:

T_s (K)	T_r (K)	Error (K)
20	120.2	+ 0.2
77	124.8	+ 4.8
90	128.4	+ 8.4

From this table it can be seen that the errors associated with testing in a "black" LN_2 -cooled shroud maintained near 77K are of the order of + 5K which would not appear to be significant since the range of allowable detector temperatures will most likely be significantly larger than the error. However, some care must be taken to maintain the shroud temperatures near the LN_2 boiling point. A poorly designed LN_2 shroud which operates in the 90-100°K region would introduce an error which may become significant. Thus, we conclude that a helium-cooled cryoshroud operating below 77K will not be required for testing the LCE radiator for detector-temperature operating levels of the order of 120°K.

A more precise model of the radiative transfer has been used to estimate the errors in measured radiator temperature as a function of the design operating temperature of the radiator in the space environment

for a fixed cryoshroud temperature of 77K. The complete analytical model described in Section 5 was used to predict the temperature errors. The results are presented in Figure 25. It can be seen that the errors are less than 6K for design operating temperatures above 105K.

Thus far, the effects of a finite-temperature cryoshroud on the temperature error have been discussed. An additional error is introduced by the finite reflectance of the "near-black" cryoshroud walls. If, for example, the cryoshroud surfaces are located close to the cooler aperture, a fraction of the energy emitted by the cone walls will be reflected back to the cold stage. The errors associated with energy leaving the cooler and being re-reflected onto the cold stage can be made vanishingly small by using a cryoshroud whose dimensions are large with respect to the cooler.

In Section 3 it was shown that direct sunlight and the emission from the solar panels were the dominant sources of power on the radiator. The power inputs from the antenna and from the earth are small and need not be considered in the simulation. Therefore, the thermal-vacuum testing should include a simulation of the direct solar inputs and the infrared emitted by the solar panels.

First, consider the direct solar inputs. For the radiative cooler systems which have been discussed in this report, some type of specularly reflecting geometry is required to preclude direct radiation from falling on the low-temperature stage. Since the degree of specularity and the reflectance of the shielding surfaces are both critical to the design, we recommend that the design qualification and flight-acceptance tests be conducted with a simulated solar source to establish the maximum operating temperature of the detector. Furthermore, the solar simulation should be accomplished by a well-collimated beam having a reasonable approximation to the solar spectrum.¹³ We believe that an indirect solar

13. The errors associated with the type of source used, e.g., carbon-arc, mercury-xenon, etc., will depend on the surface coatings used. Further work would be required to determine the effects of the source on the experimental uncertainties.

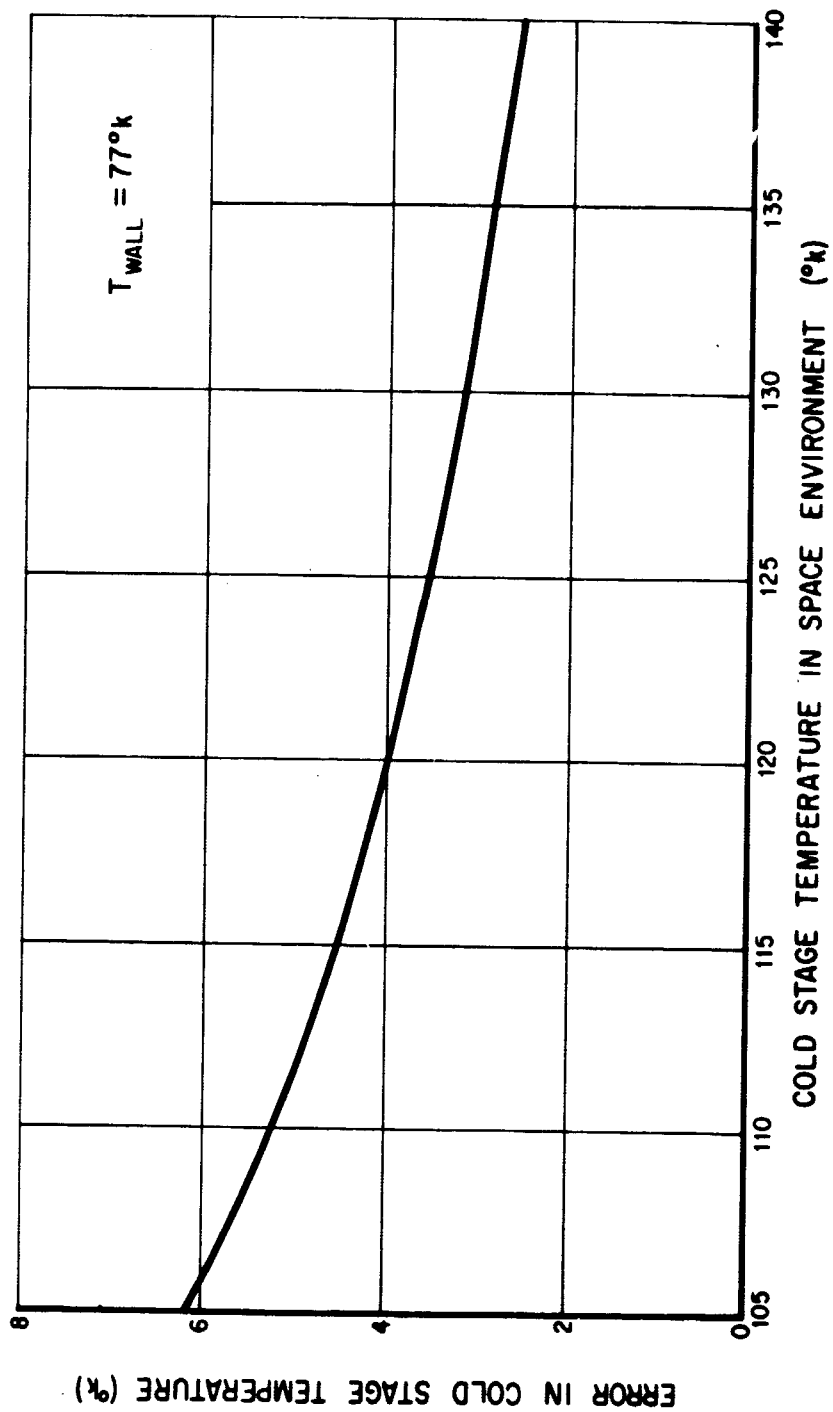


FIGURE 25 ERROR IN DETECTOR TEMPERATURE DUE TO ELEVATED SINK TEMPERATURE

simulation test which is made by measuring the solar absorptances of the surfaces involved, calculating the absorbed flux and using heaters to apply the power will not be appropriate since multiple reflections and the specularity of the surfaces will be critical to obtaining the detector temperature levels of interest.

There is, however, a problem associated with the use of a solar simulation beam which illuminates the outer stage of the radiator in a thermal-vacuum test. The problem arises due to spill-over of the beam and any subsequent stray-light reflections within the cryoshroud. In usual thermal-vacuum testing this effect is negligible because the temperatures of the component under test are near room temperature. In this case the design temperature of the radiator is low and any stray radiation introduces errors in the measured detector temperature.

Although the exact amount of diffusely scattered radiation will depend on the details of the test setup and the reflectance of the cryoshroud walls, it is possible to calculate the error introduced in detector temperature as a function of the amount of stray solar radiation diffusely incident on the aperture of the conically shielded radiator. The error in detector temperature as a function of the diffuse incident power for various detector design temperature levels is presented in Figure 26. From this figure it can be seen that an error of + 4K will be introduced in the measured detector temperature if the incident stray radiation is of the order of 0.1% of a solar constant. Since the cryoshroud walls may have a reflectance in the visible region as high as 10% even when coated with an "optical black" paint, it is possible that the stray radiation could introduce significant simulation errors. A similar problem exists in the background infrared radiation emitted by vacuum-pump ports or windows used to transmit the solar beam which are not properly baffled with cryogenically cooled shrouds and directly view the low-temperature radiator.

The example given here is not intended to provide a quantitative estimate of the experimental uncertainties associated with any test setup; it merely illustrates the importance of the design of the

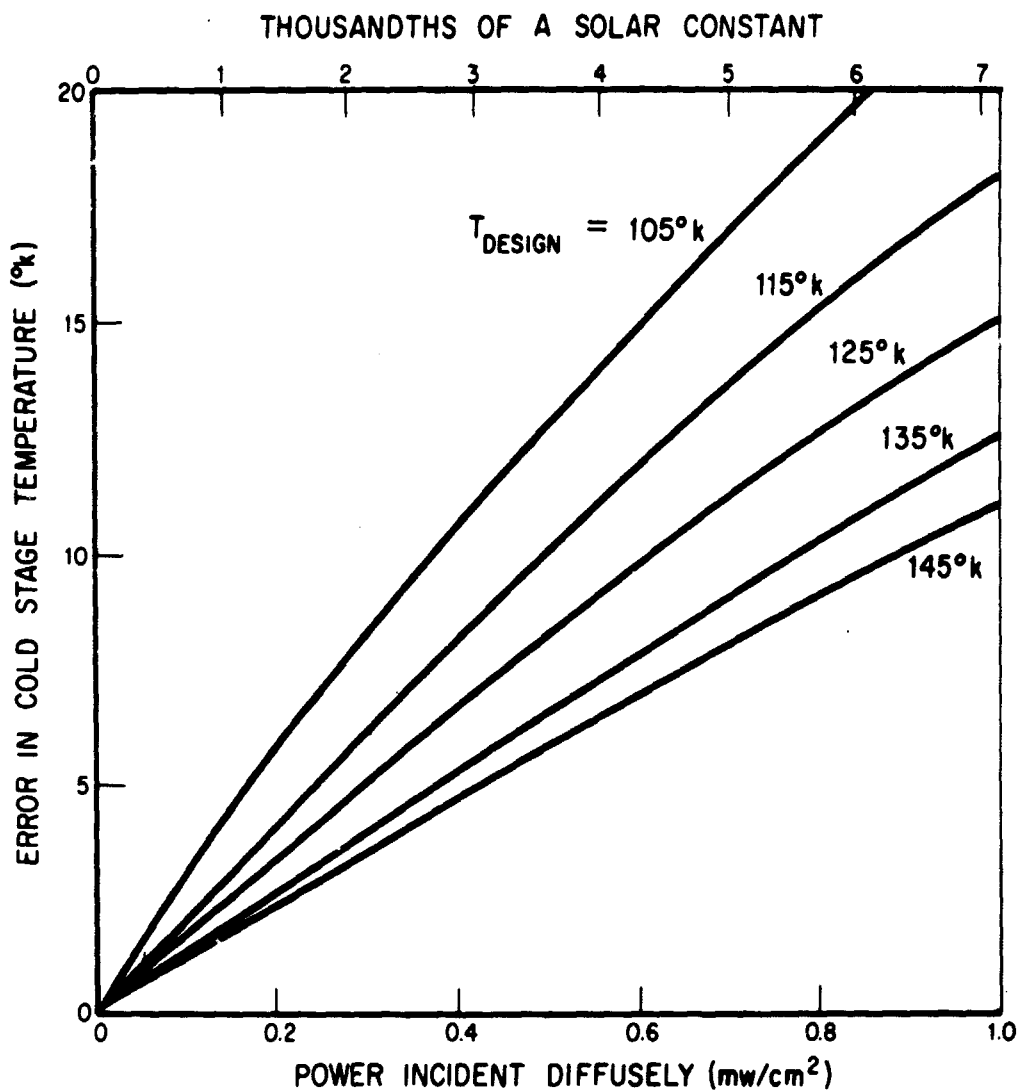


FIGURE 26 ERROR IN DETECTOR TEMPERATURE DUE TO SIMULATED SOLAR ENERGY INCIDENT DIFFUSELY ON CONE MOUTH

thermal-vacuum facility associated with the testing of low-temperature passive radiators.

In order to simulate the heat inputs associated with the emission from the ATS solar panels, we recommend the use of a temperature-controlled, heated plate located external to the radiator. A full-scale mock-up of the spacecraft solar panel is inordinately large; therefore, it is recommended that the heated plate be scaled down to simulate the radiant inputs from the solar panels. This type of simulation can be accomplished by adjusting the size of the heater plate, its location relative to the radiative cooler, and its temperature to provide the incident flux which would result from the full-scale solar panel configuration. Again, we recommend this type of simulation since indirect heating methods do not provide a meaningful test of the effects associated with multiple reflections between the specular surfaces used to shield the low-temperature radiator stage.

APPENDIX I

CALCULATION OF ABSORPTION AND EMISSION OF RADIANT ENERGY IN A SPECULAR CONE FRUSTUM

The general problem is illustrated in Figure 27. Parallel radiation is incident at angle ψ on a cone frustum of half-angle ϕ and height h with specularly reflecting surfaces. The special case of $h/L = 1$ represents a conical cavity. Shown also in Figure 27 is the path of a typical ray. If the ray represents a bundle of energy, a fraction α_w , equal to the absorptance of the conical surface, is absorbed at each point where the beam strikes the cone. The remaining fraction, $\rho_w = 1 - \alpha_w$, undergoes specular reflection. If the beam undergoes N reflections before leaving the frustum and the surface absorptance is directional independent, the fraction of incident energy absorbed by the cone walls is $1 - \rho_w^N$. The fraction of energy leaving the frustum is ρ_w^N .

A different view of the reflection process is shown in Figure 28. Here, an incident beam enters the conical opening through a differential area centered at (x_o, y_o) . The first two intersections with the conical surface are represented by locations 1 and 2. The quantity $\Delta\theta$ represents the change in polar angle between these two locations. By integrating the fraction of energy absorbed for each beam over the conical opening, the apparent directional absorptance is found:

$$\alpha_a(\psi) = \frac{1}{A} \int_A (1 - \rho_w^N) dA \quad (1)$$

Note that in general the quantity N depends on the location (x_o, y_o) of the beam as it enters the opening.

The net energy absorbed with parallel radiation incident at angle ψ is:

$$\text{Energy absorbed} = \alpha_a(\psi) S (A \cos \psi) \quad (2)$$

where S is the energy associated with the incoming radiation per unit area normal to the beam, A is the area of the conical opening, and α_a is

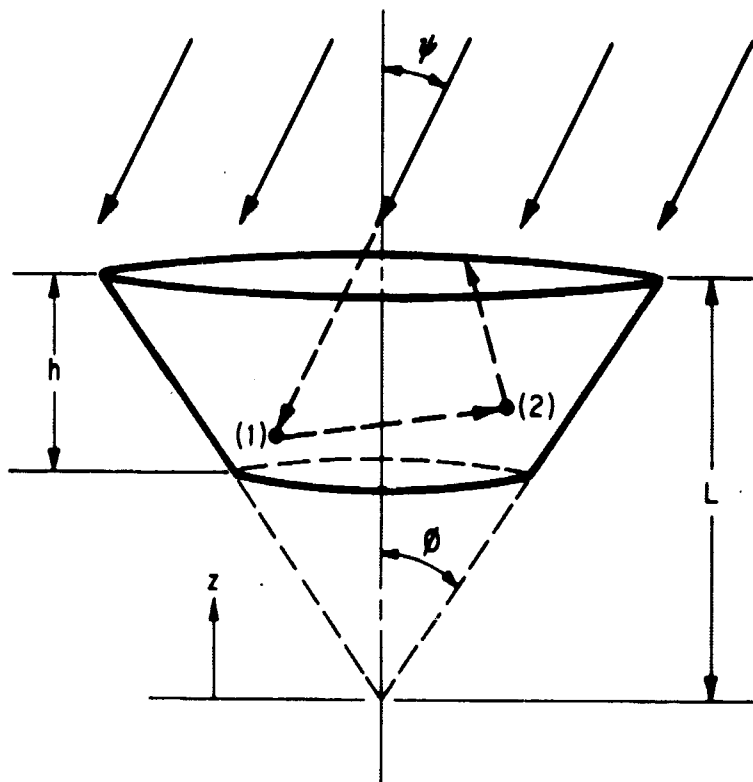


**FIGURE 27 REFLECTION OF PARALLEL INCIDENT RADIATION IN
A SPECULAR CONE FRUSTUM**

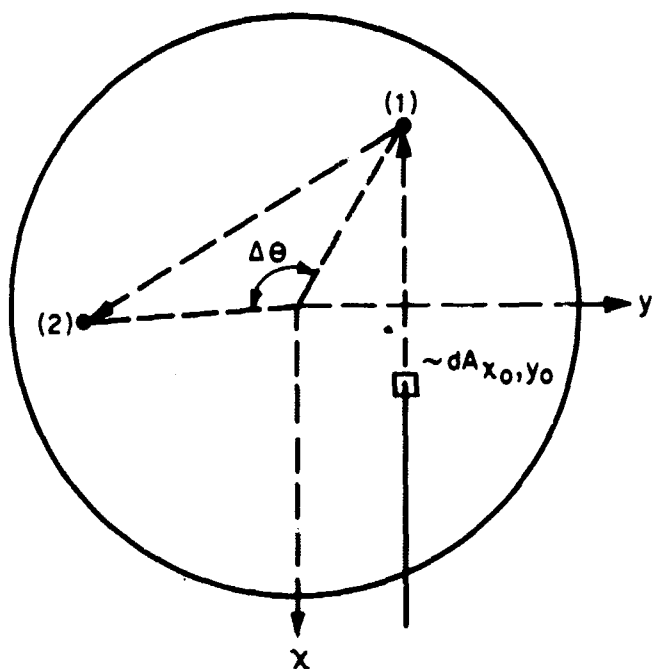


FIGURE 28 TOP VIEW OF CONE SHOWING REFLECTION OF INCIDENT RAY

the apparent directional absorptance. The apparent hemispherical absorptance for diffusely incident radiation can be calculated from

$$\bar{\alpha} = 2 \int_0^{\pi/2} \alpha_a(\psi) \cos \psi \sin \psi d\psi. \quad (3)$$

The transmittance (through the lower conical opening) for parallel incident radiation is determined in the same manner as the absorptance. The fraction of energy transmitted through the frustum is integrated for each beam entering the upper conical opening as in Equation (1). The net energy transmitted with radiation incident at angle ψ is then

$$\text{Energy transmitted} = \tau(\psi) S (A \cos \psi), \quad (4)$$

where $\tau(\psi)$ is the directional transmittance. By integrating the directional transmittance over all angles of incidence according to Equation (3) with $\tau(\psi)$ substituted for $\alpha_a(\psi)$, the transmittance for diffuse incoming radiation is computed.

Calculation of the number of reflections N undergone by a beam is performed by a ray trace technique. After determination of the location of the first intersection - point 1 in Figure 28 - the direction of the reflected beam is found by applying the following two rules:

1. The angle of reflection equals the angle of incidence.
2. The incident ray, reflected ray and surface normal at the intersection point are co-planar.

Once the second intersection point is found, the vertical coordinates z of succeeding reflections are found from the recursion relation developed by Lin¹:

$$z_n = \frac{z_{n-1} z_{n-2}}{K z_{n-2} - z_{n-1}} \quad (5a)$$

1. S. H. Lin and E. M. Sparrow, "Radiant Interchange Among Curved Specularly Reflecting Surfaces - Application to Cylindrical and Conical Cavities", Trans. ASME, J. Heat Transfer, 87, 299-307 (1965).

$$K \equiv 2 - 2 \sin^2 \phi (1 - \cos \Delta\theta) \quad (5b)$$

Subscript n designates the n-th intercept. The ray is followed until the value z_n is greater than the height of the cone L, is a positive quantity less than L-h, or is negative (representing an imaginary intersection). The case of z_n less than L-h but greater than zero corresponds to transmission through the lower opening.

By application of the reciprocity rule, values for the absorptance and transmittance for diffuse incoming radiation can be used to predict the net radiant energy emerging from either end of an isothermal diffusely emitting and specularly reflecting tapered tube. This can be seen by imagining the radiation entering the upper surface as coming from a black disk covering the opening and by assuming that the lower surface is a black surface at zero degrees.

From the reciprocity rule, the view area from the lower to upper conical opening is simply the product of the upper surface area and the apparent hemispherical transmittance. Since no energy emitted from the bottom surface can be reflected back, the view area from the bottom surface to the specular cone is immediately determined. Applying reciprocity, the view area from the cone to the bottom conical opening is found. Similarly, the view area from the cone to upper conical opening is the product of the upper surface area and the apparent hemispherical absorptance.

NOMENCLATURE

A = area of conical opening

h = height of cone frustum

K = constant defined by Equation (5b)

L = height of cone

N = number of reflections undergone by an incident ray

S = energy carried by incoming radiation per unit area normal to direction of travel

x,y,z = coordinates shown in Figures 27 and 28

α_w = surface absorptance
 $\alpha_a(\psi)$ = apparent directional absorptance for radiation incident at angle ψ
 $\bar{\alpha}$ = apparent hemispherical absorptance
 ρ_w = surface reflectance
 $\tau(\psi)$ = directional absorptance for radiation incident at angle ψ
 $\bar{\tau}$ = apparent hemispherical transmittance
 $\Delta\theta$ = change in polar angle between reflections, see Figure 28
 ϕ = cone half-angle
 ψ = incidence angle for parallel incoming radiation