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LOW-TEMPERATURE BRAYTON RADIATOR PERFORMANCE IN A VARIABLE-SINK-TEMPERATURE ENVIRONMENT

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16. Abstract

This is an analysis of the performance of a low-temperature Brayton space radiator in variable-sink-temperature surroundings. The cylindrical radiator with circumferential tubes is subjected to an equivalent sink temperature environment that varies along the fluid flow path. The radiator performance as measured by fluid outlet temperature is seen to be dependent on the orbital position and the location of the radiator inlet relative to the sink temperature environment. An approximate dynamic analysis of Sun-shade performance is included.

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SUMMARY

The performance of a low-temperature flight-type cylindrical radiator in a variable-sink-temperature environment is studied using basic heat-transfer relations. The sink temperature varies around the cylinder and also with the change in orbital location. The location of the fluid inlet is shown to affect the performance of the radiator because of the radiative interrelationship of the particular sections of the radiator with the given sink temperature environment. By the same token, flow direction is seen to have some effect on performance.

For a low-temperature radiator it is shown that in some orientations the radiator absorbs net heat over a portion of its length. The effect of this on performance of the particular radiator studied is shown.

Techniques suggested for decreasing the effects of variable sink temperature include flow splitting and control of the rate of flow. Both are shown to be effective in decreasing variations in both the circumferential and flow directions. However, they are ineffective in decreasing the Sun-shade variations in performance.

Finally, an approximate dynamic analysis is presented to determine the effects of thermal lag on performance. It is shown that there is substantial smoothing when comparing the steady-state and dynamic results. This means that it may be reasonable to design a radiator for an effective sink temperature that is below the peak steady-state value.

INTRODUCTION

The performance of a low-temperature flight-type cylindrical radiator in a variable-sink-temperature environment is studied using basic heat-transfer relations. A cylindrical radiator in Earth orbit is subjected to external thermal energy inputs which are nonuniform around its circumference; in addition, such a radiator is subjected to thermal energy inputs which vary with orbital location. A low-temperature radiator operating in such a constantly changing environment will reject a constantly changing net amount of heat unless some control is imposed on the radiator. The severity of such variation in radiator performance is analyzed, and several schemes for minimizing these effects are proposed.

Analysis of the severity of performance variation is important in determining whether controls are needed on the radiator. This study analyzes performance of the radiator but does not investigate the effects on the rest of a power system. Such a study is dependent on the characteristics of the particular system in which the radiator would be used. Radiators are typically designed for the highest sink temperature condition to be encountered and are therefore overdesigned at other than this worst condition. It is desirable to know whether dynamic thermal lag effects are such that the radiator could be smaller than predicted by steady-state analyses. This problem is treated herein by an approximate analysis. Finally, low-temperature radiators are sensitive to extreme sink temperature variations and, in fact, the sink temperature may at some point exceed the radiator temperature. Such a case is discussed herein and the performance in this unique situation is determined.

RADIATOR CONFIGURATION

The cylindrical radiator configuration of reference 1 is chosen for this study because it is a low-temperature flight-type radiator. It is basically a tandem radiator consisting of a series arrangement of high- and low-temperature sections (fig. 1). This radiator was designed for a 7.1 kilowatt-electric isotope Brayton power system. The coolant, a silicone oil having a viscosity of 2 centistokes at 298.3 K (537° R), enters the radiator at a temperature of 414.4 K (746° R) and a rate of 0.064 kilogram per second (0.14 lb/sec). It flows through circumferential channels for 8.90 meters (29.2 ft) where it passes into a manifold and is mixed with a lower temperature flow of 0.100 kilogram per second (0.22 lb/sec). This new combined flow of 0.163 kilogram per second (0.36 lb/sec) enters the low-temperature section and flows for a distance of 9.08 meters (29.8 ft). The outlet temperature of the high-temperature section is used as the inlet temperature of the low-temperature section.

The flow channels of the two sections are trapezoidal and the dimensions are shown in figures 2 and 3. The heat-transfer correlations of reference 2 were used and conduction in the flow direction was ignored as insignificant (ref. 3). The use of these correlations was verified by a tube panel test in reference 1 where good agreement was found.

SINK TEMPERATURE

The effective sink temperature of a radiator in Earth orbit is determined by energy inputs to the radiator from three sources: (1) directly from the Sun, (2) indirectly reflected from the Earth (albedo), and (3) thermal radiation from the Earth. Reference 4 was used to obtain sink temperature as a function of orbital altitude, radiator shape and orientation, and radiator coating properties.

The orbit chosen for this analysis is a 555.6-kilometer (300-n-mi) Earth orbit in the plane of the ecliptic. The axis of the cylindrical radiator is oriented perpendicular to the plane of the orbit. The emittance of the radiator is 0.88 for sink temperature calculations; a ratio of solar absorptance to thermal emittance of 0.34 was used. This corresponds to a solar absorptance of about 0.30.

In such an orbit, the cylindrical radiator is subjected to nonuniform sink temperature around its circumference as well as to sink temperatures which vary with orbital position. In order to analyze the effects of the variable sink temperature, the circumferential radiator is treated as 36 serially connected flat sections. Figure 4 shows the notation used to locate the orbital position of the radiator relative to the Sun and to locate the 36 flat sections on the cylinder. The flat sections are numbered counterclockwise with station 1 being directly opposite the Sun's rays at all orbital positions.

Four orbital positions are studied in detail for effects of varying sink temperature. These positions are determined by θ_s, the angle between the Earth-Sun and Earth-radiator vectors (fig. 4). The "high-noon" position is at $\theta_s = 0^\circ$. Since the radiator is in the Earth's shadow for θ_s between 113° and 247°, a second position studied is midway through the shadow at $\theta_s = 180^\circ$. The other two positions studied are $\theta_s = 60^\circ$ and the position just prior to entering the shadow ($\theta_s = 113^\circ$). It is noted that there is symmetry around the other half of the orbit.

Figure 5 shows the variation in equivalent sink temperature with station number for each of the four orbital positions. Tables of reference 4 along with a ratio of solar absorptance to thermal emittance of 0.34 were used to generate these curves. It is noted that the variability around the cylinder can very great, the most severe being just before entering the shadow at $\theta_s = 113^\circ$ where the Sun-side sink temperature is 319 K (575° R) while the shade-side sink temperature is 0 K (0° R).

STEADY-STATE RESULTS

In order to study the effects of this variable sink temperature on radiator performance, the radiator is treated as 36 sequential equal-size segments. The flow rate into the radiator and the fluid inlet temperature are kept fixed at the design conditions previously specified. The segments are capable of rejecting an amount of heat which varies both with the radiator temperature and sink temperature. Therefore, it is of interest to study the effects of inlet location and flow direction on performance. In the steady state, this means that the radiator is fixed in each of the four orbital positions for a sufficiently long time that there is no thermal lag. Also there is no spinning of the radiator.

The fluid outlet temperature is taken as a measure of radiator performance. From figure 6 it is seen that the outlet temperature at $\theta_s = 0^\circ$ can vary by as much as 5.6 K (10° R) depending on inlet location and flow direction. This suggests that judicious orientation of the radiator would result in a significant improvement in performance. For $\theta_s = 60^\circ$, the outlet temperature can vary by 7.8 K (14° R) depending again on flow direction and orientation. Just before entering the Earth's shadow at $\theta_s = 113^\circ$, the variability is 8.9 K (16° R). In the shadow, the variability is about 2.8 K (5° R).

The variability in performance throughout an orbit can be substantial if fluid flow is kept constant as it was for those cases shown in figure 6. Looking at the extremes of figure 6, it is noted that the outlet temperature can vary from about 273.3 K (492° R) in the shade ($\theta_s = 180^\circ$) to about 304.4 K (548° R) when the radiator is directly between the Earth and the Sun ($\theta_s = 0^\circ$). Since this 31.1 K (56° R) variability may be unacceptable to the system, it may be necessary to introduce some control on the radiator flow rate to decrease this variability.

Figure 7 shows the sink temperature and the temperature profile of a radiator just about to enter the Earth's shadow at $\theta_s = 113^\circ$. The severe variations in sink temperature around the cylinder cause the radiator to absorb net heat over about one quarter of its circumference near the outlet. This particular case, with the inlet at station 25 and counterclockwise flow, was chosen to show just how much of the radiator could possibly serve as a heat sink in such a low-temperature application. For this arrangement the fluid outlet temperature is 293.6 K (528.5° R).

In order to decrease the effects of inlet location and flow direction on performance, radiators were analyzed with the flow split in half and run halfway around the circumference. In this situation, there are two high-temperature and two low-temperature sections. The two exit fluids from the low-temperature radiators are combined and mixed, and the new mixed fluid temperature is taken as the measure of performance.

When the inlet manifold of the high-temperature radiator is 180° from the outlet manifold of the low-temperature radiator, there are three possible flow patterns:

(1) two clockwise flow sections, (2) two counterclockwise flow sections, and (3) one counterclockwise, one clockwise. The variation resulting from (3) is much greater than (1) or (2), so it is dropped from consideration.

It is noted from figure 8 that flow splitting results in decreased variability in fluid outlet temperature. For $\theta_s = 0^{\circ}$, it is seen that the outlet temperature can vary by as much as 3.3 K (6° R). For $\theta_s = 60^{\circ}$, the variability in outlet temperature is of the order of 2.8 K (5° R). Just before entering the shadow at $\theta_s = 113^{\circ}$, the variability is only about 2.2 K (4° R). Finally, in the shadow the variability is very small (about 0.3 K (0.5° R)).

The way to eliminate the effects of inlet location and flow direction is to have vertical tubes in the radiator where each tube is subjected to a constant sink temperature along its entire length. The inlet temperature of all such vertical tubes is constant, but because of changing sink temperature from tube to tube, and hence nonuniform heat rejection capability, the outlet temperatures will be unequal. The fluid exiting from all of these tubes is mixed in a manifold and the mixed temperature is dependent on θ_s only. The steady-state performance of such a radiator is shown in figure 9. It is noted that the radiator steady-state outlet temperature changes by as much as 26.7 K (48° R) during an orbit.

The mixed outlet temperature shown in figure 9 is the mean of the outlet temperatures of the individual flat plate radiators. The outlet temperatures of those individual radiators are shown in figure 10 for four values of θ_s. It is seen that the outlet temperatures vary from 265 to 334.4 K (477° to 602° R).

DYNAMIC RESULTS

It is apparent that the orbital position variations in steady-state performance are greater than those due to inlet location and flow direction. In order to get an approximation of the dynamic effects, the radiator was treated as a mass of aluminum weighing 388 kilograms (855 lb) and having a radiating area of 54.8 square meters (590 ft^2). The orbital sink temperature variation of a cylinder was obtained by using tables in reference 4 and is shown in figure 11. Reference 4 contains average view factors directly usable in obtaining the effective "average" sink temperature of a cylinder. It is seen that the sink temperature is at a low of 186.7 K (336° R) in the Earth's shadow and reaches 266.1 K (479° R) when the radiator is directly between the Earth and the Sun (high noon). The steady-state fluid outlet temperature as a function of sink temperature was obtained by running the radiator program with the conditions of reference 1 for several fixed sink temperatures; that is, the sink temperature was fixed for the entire radiator. The resultant fluid outlet temperatures are shown in figure 12. The design outlet temperature of 295 K (531° R) of reference 1 is noted in the figure.

Figures 11 and 12 are combined in figure 13 to show the steady-state outlet temperature as a function of orbital position under fixed flow rate conditions of the reference 1 radiator. It is seen that throughout an orbit the outlet temperature varies by about 27.2 K (49° R), similar to figure 9, where the exit fluids of 36 vertical radiators were combined and mixed.

In order to perform the approximate dynamic analysis, the data of figure 12 were converted to an equivalent mean radiating temperature. The mean radiating temperature is defined as that temperature at which an isothermal radiator would reject an amount of heat equivalent to the nonisothermal radiator. This relationship is shown in figure 14 for the flow conditions of the radiator of reference 1.

Figure 15 shows the steady-state and dynamic mean radiating temperatures. The steady-state curve was obtained by combining figures 11 and 14. The dynamic curve was obtained by an iterative procedure in which initial conditions were assumed and the mass of aluminum was cycled through several orbits until its temperature response stabilized to that shown in the figure. Since the flow rate is kept constant at the design level of reference 1, the amount of heat rejected varies throughout an orbit. It is seen that the mean dynamic temperature is about 1.1 K (2° R) below the peak steady-state mean temperature and about 0.6 (1° R) above the lowest steady-state mean temperature.

The difference between peak dynamic and steady-state mean temperatures is of interest to the designer. By accounting for thermal lag, it may not be necessary to design for the "worst" sink temperature condition. It is seen by using the peak temperatures from figure 15 along with figure 14 that there is effectively a decrease of about 3.3 K (6° R) in the "worst" equivalent sink temperature.

Because of changing sink temperature throughout an orbit, a constant flow rate radiator will reject a varying amount of heat and the outlet temperature will vary (see figs. 12 and 13). Since this variation in temperature could adversely affect the rest of the system, some control on the radiator may be necessary. Reference 1 has a design heat rejection of 17.62 kilowatts of thermal power. The flow rate through the radiator can be continuously controlled throughout the orbit by bypassing a portion of the flow in such a way that the heat rejected is held constant at 17.62 kilowatts. When the radiator fluid is mixed with the bypass fluid, the mixed temperature entering the Brayton waste heat exchanger will remain fixed and the Brayton cycle operating conditions will be unaffected by the orbital variations. This case is shown in figure 16. It is noted that the mean radiating temperature varies from 296.1 to 327.2 K (533° to 589° R) for the steady-state analysis and from 300 to 323.9 K (540° to 538° R) for the dynamic analysis. This difference of 3.3 K (6° R) in peak mean temperatures translates into a difference of about 6.1 K (11° R) in "worst" equivalent sink temperature.

SUMMARY OF RESULTS

From an analysis of the performance of a low temperature space radiator in variable sink temperature surroundings, the following were determined:

(1) The performance of such a circumferential tube space radiator is affected by location of the fluid inlet relative to the sink temperature environment.

(2) The direction of flow in a nonsymmetrical sink temperature environment also affects the performance but to a lesser degree.

(3) Severe variability in the sink temperature can result in the radiator acting as a heat sink over part of its length.

(4) Splitting the flow, and thereby having tubes "see" only a portion of the circumferential sink temperature variability, appears to decrease the effect of inlet station location for fixed orbital position. The Sun-shade perturbation is more severe, however, and is only slightly affected by flow splitting.

(5) The dynamics of the radiator tend to substantially smooth out the temperature profile predicted by the steady-state analysis. This smoothing results in a decrease in the peak effective equivalent sink temperature.

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

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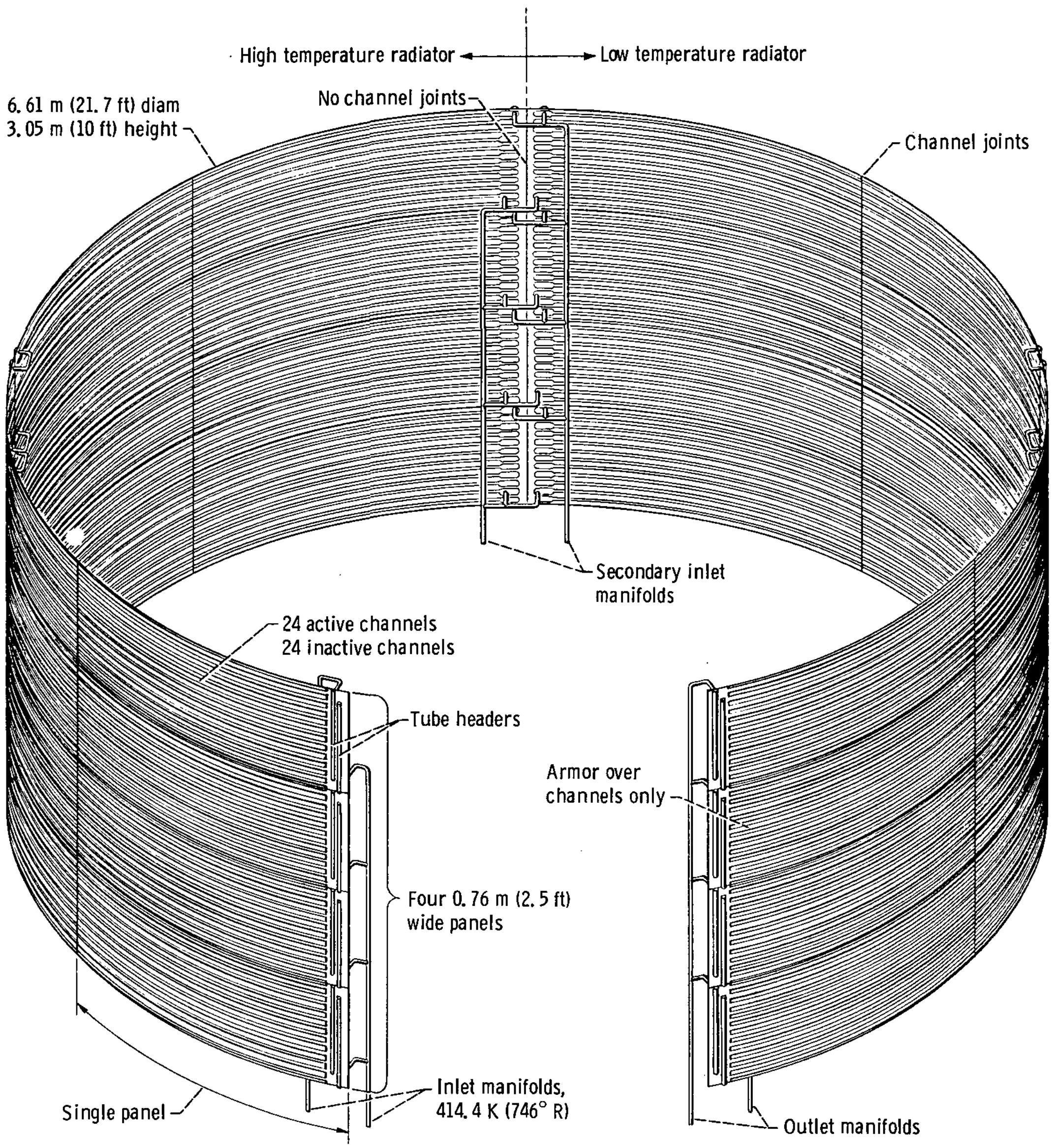

Figure 1. - Brayton cycle radiator details.

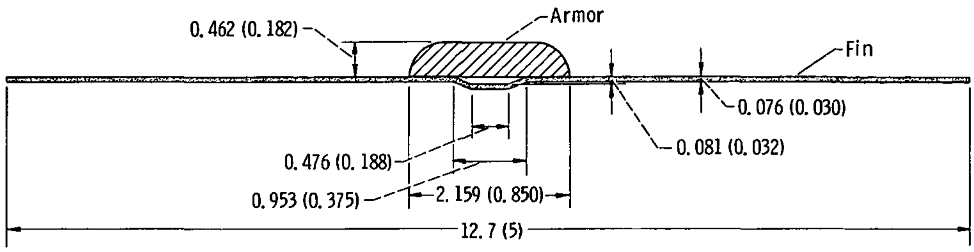

Figure 2. - Channel-fin dimensions, primary radiator. Dimensions are in centimeters (in.).

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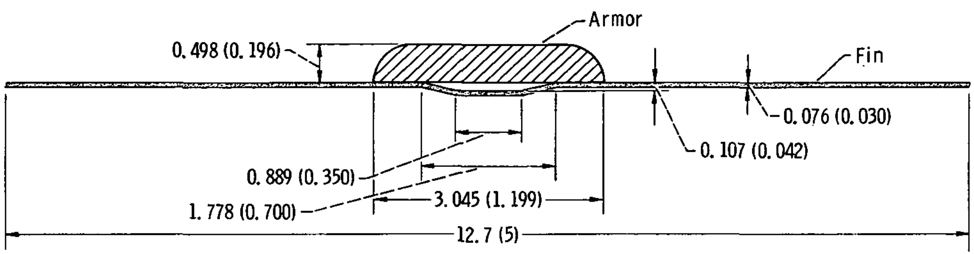

Figure 3. - Channel-fin dimensions, secondary radiator. Dimensions are in centimeters (in.).

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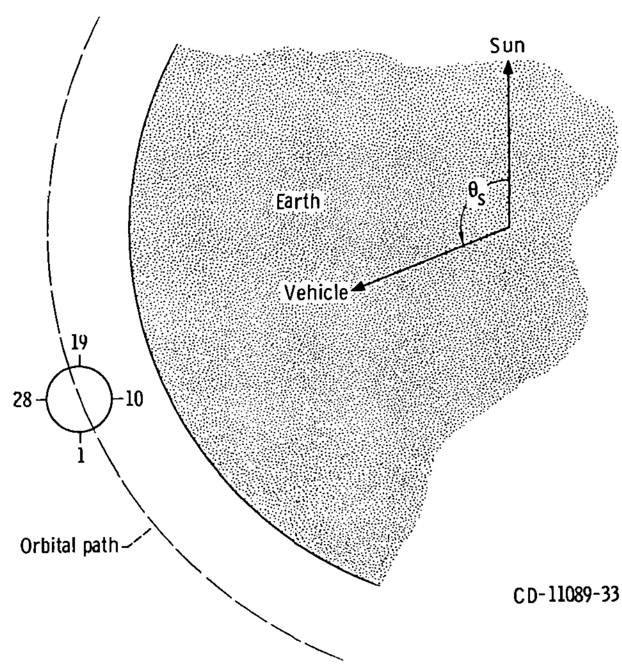

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Figure 4. - Notation for orbital position and orientation.

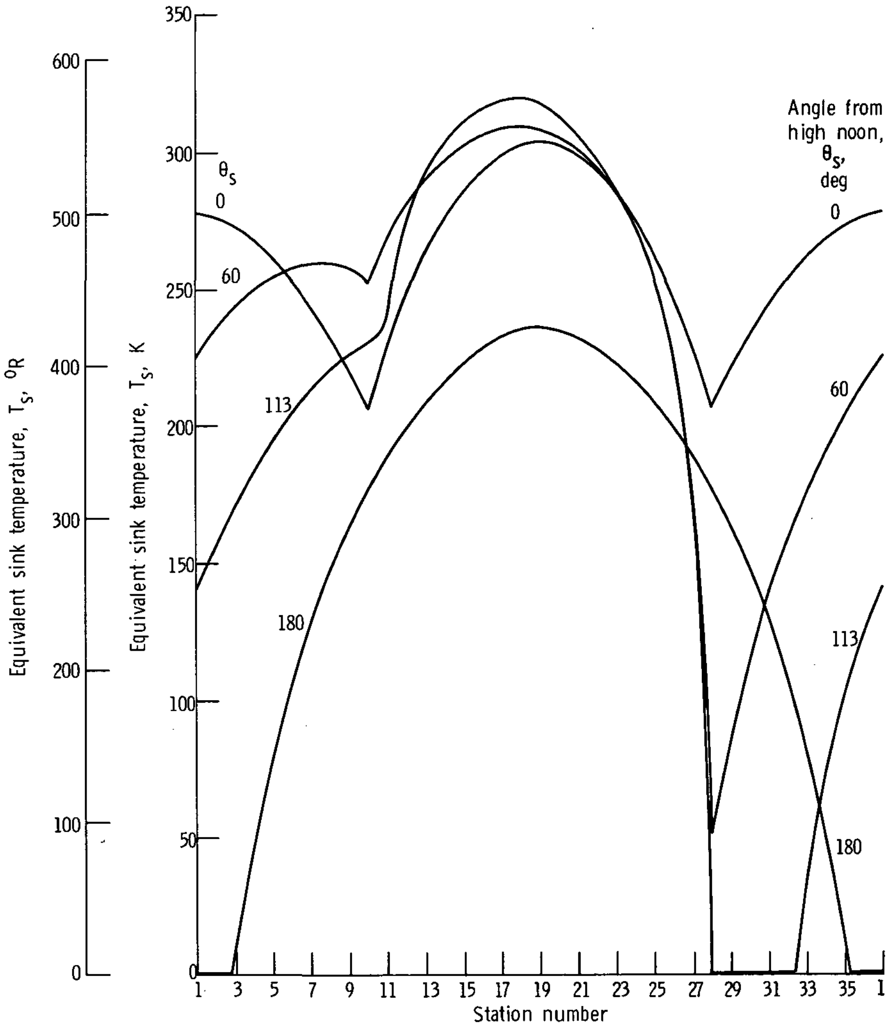

Figure 5. - Variation in equivalent sink temperature around a cylinder for several orbital positions.

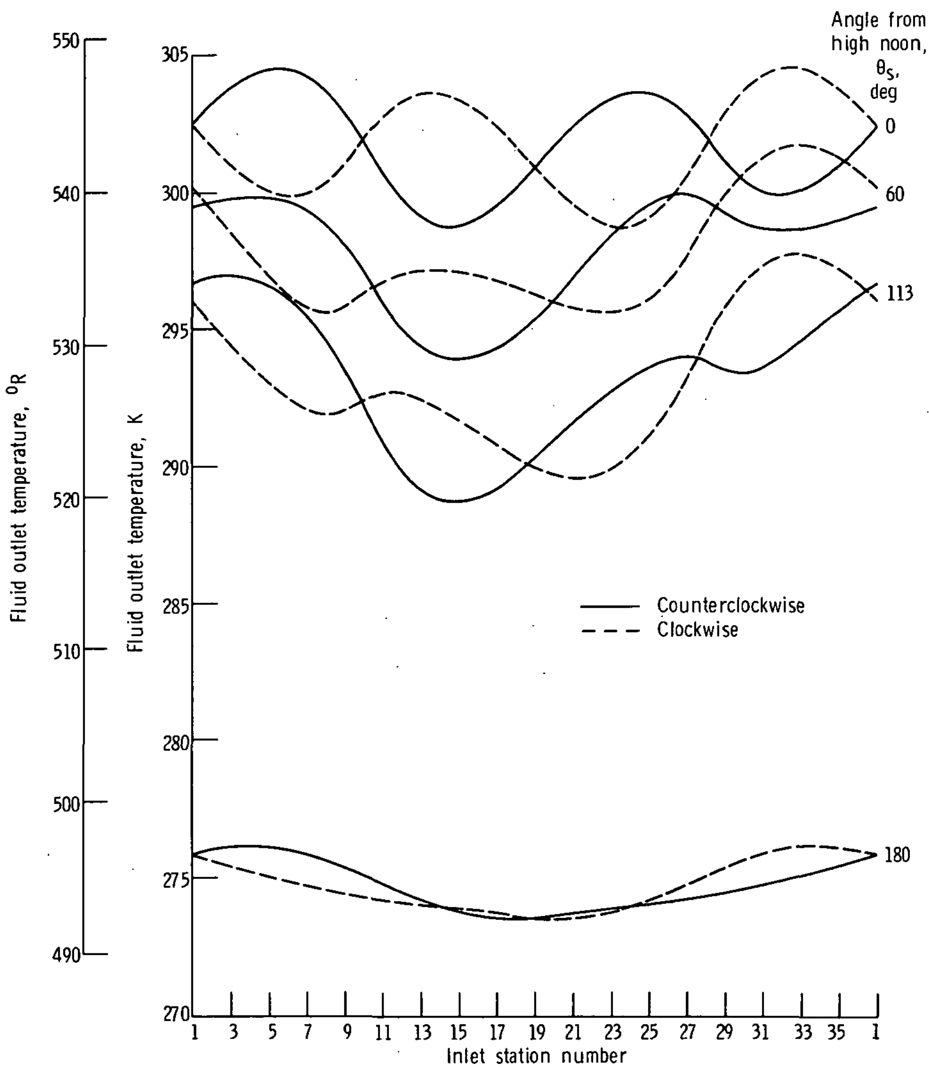

Figure 6. - Variation of fluid outlet temperature with inlet station location and flow direction (outlet 360^0 from inlet).

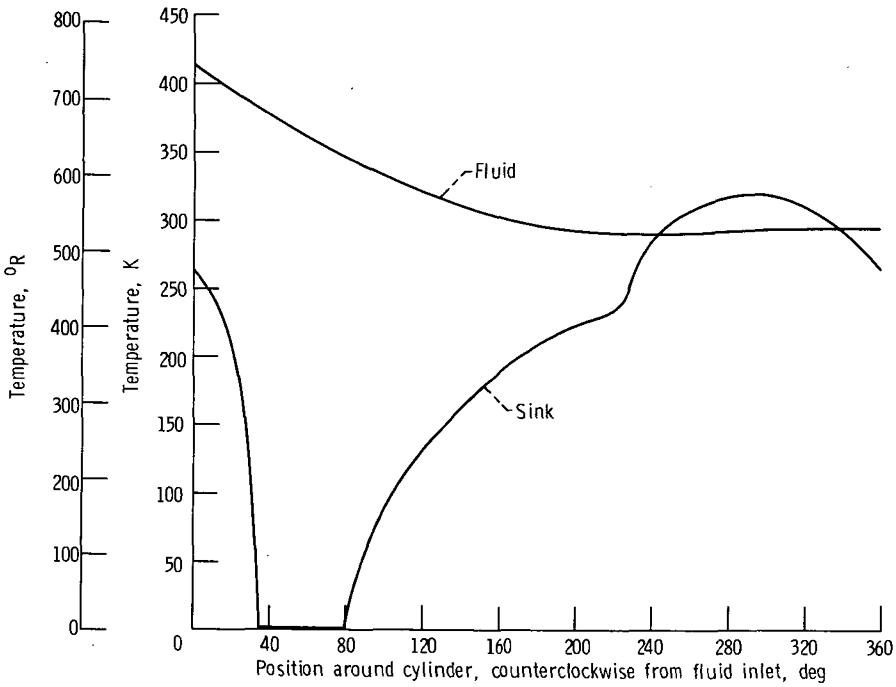

Figure 7. - Temperature profile of radiator fluid for $\theta_s = 113^0$ with inlet at station 25.

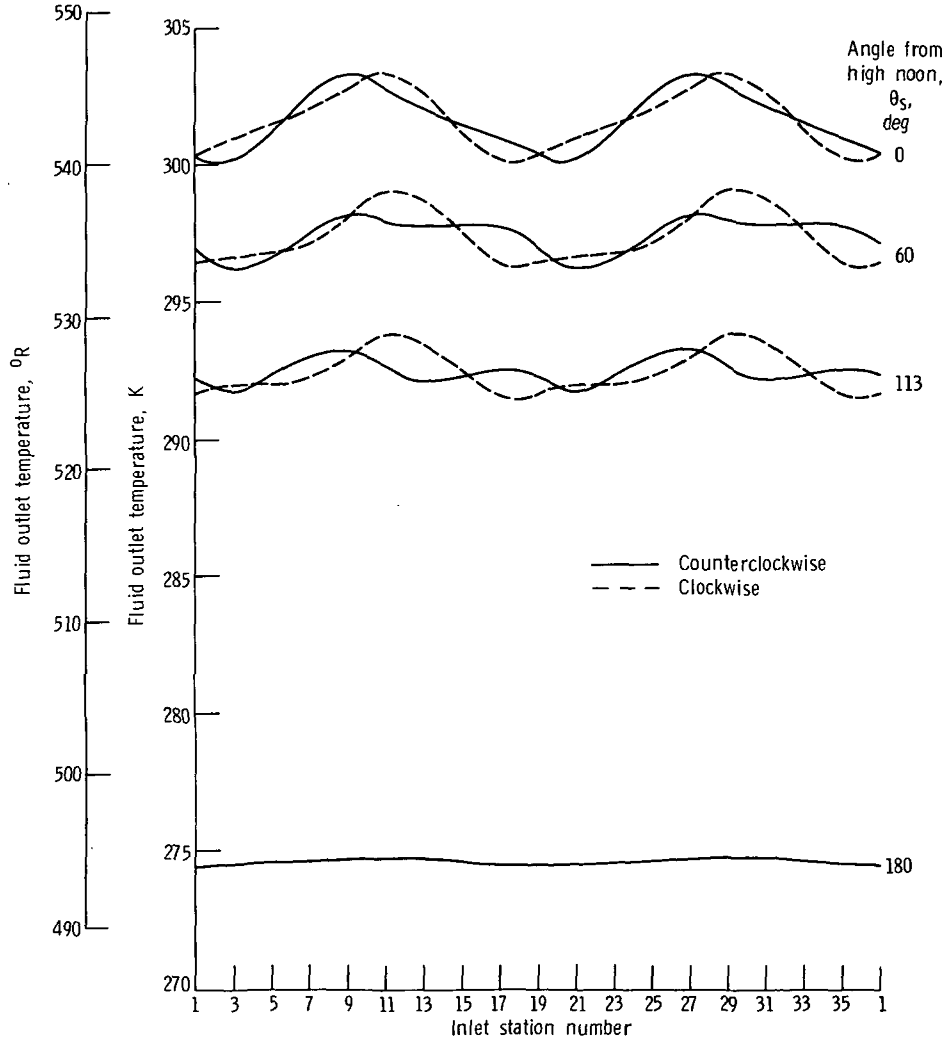

Figure 8. - Variation of fluid outlet temperature with inlet station location and flow direction (outlet 180° from inlet, flow splitting).

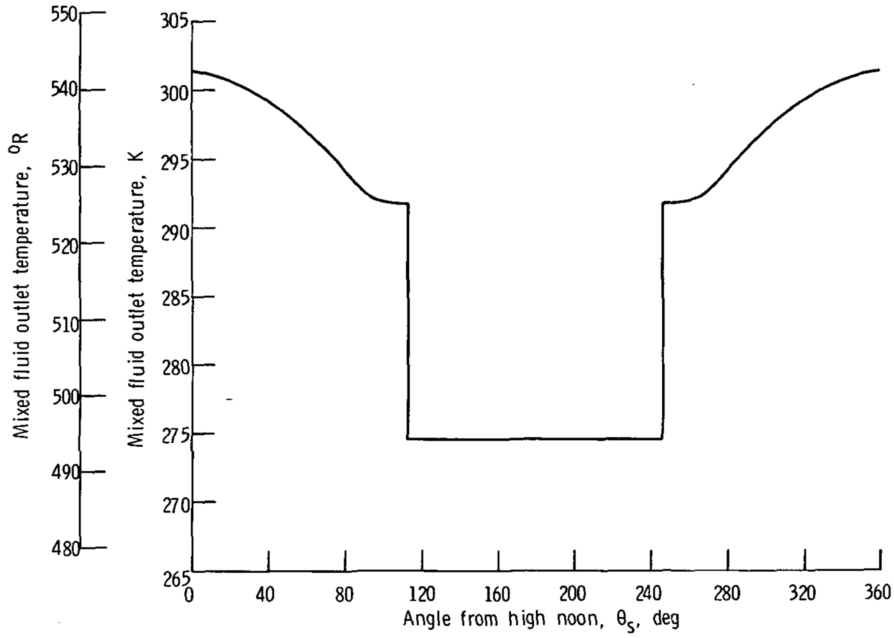

Figure 9. - Variation in fluid outlet temperature with orbital position for vertical tube radiator.

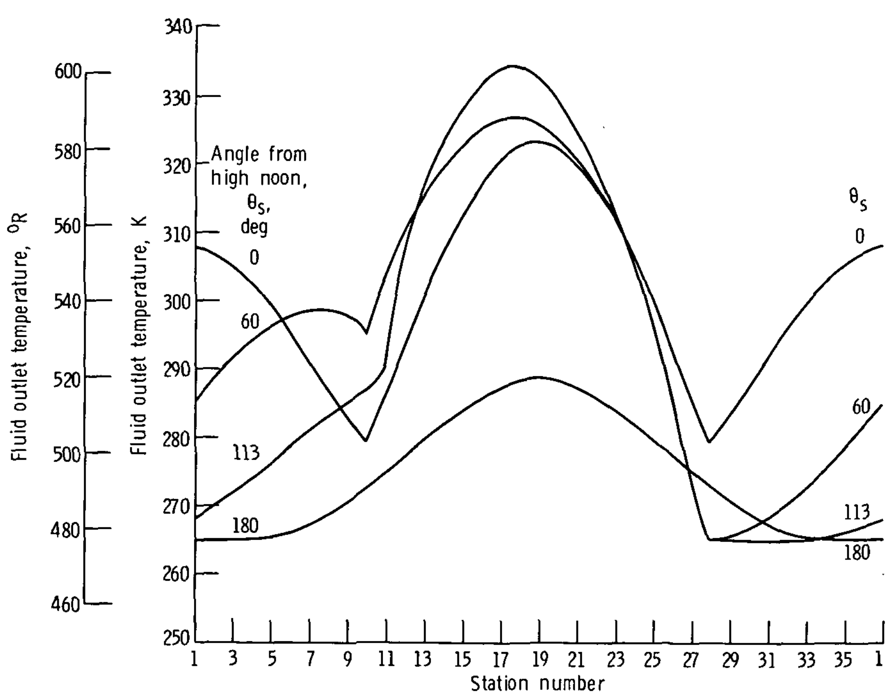

Figure 10. - Fluid outlet temperature distribution about a vertical tube radiator for several values of θ_s.

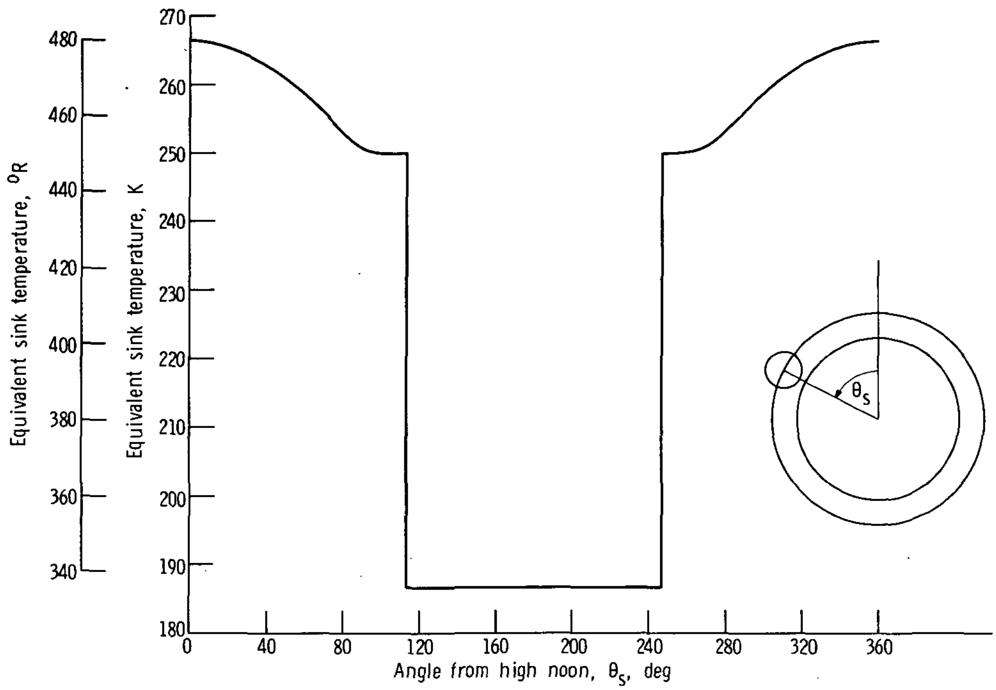

Figure 11. - Equivalent sink temperature of cylinder in 555.6-kilometer (300-n-mi) Earth orbit (α/ε = 0.34).

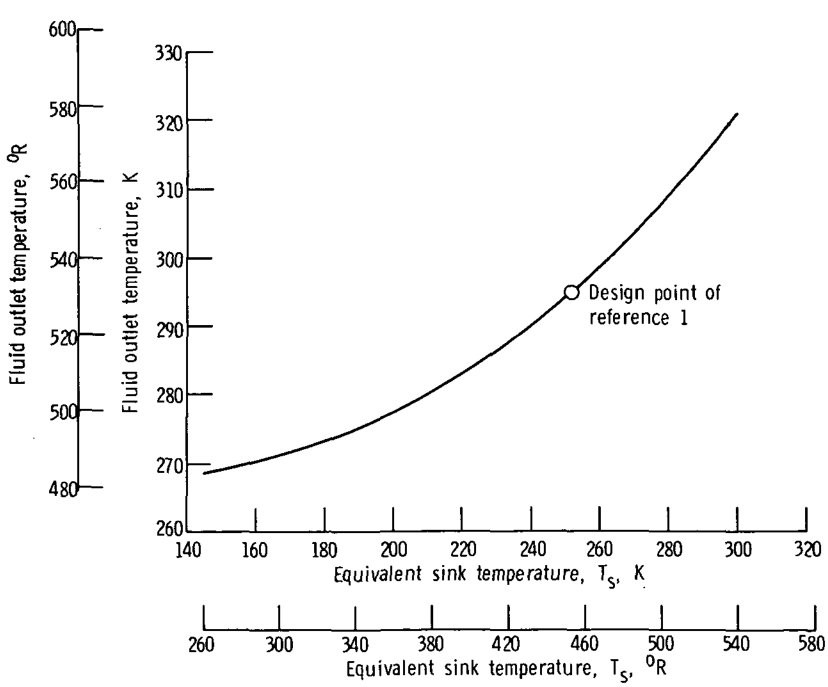

Figure 12. - Fluid outlet temperature as function of equivalent sink temperature.

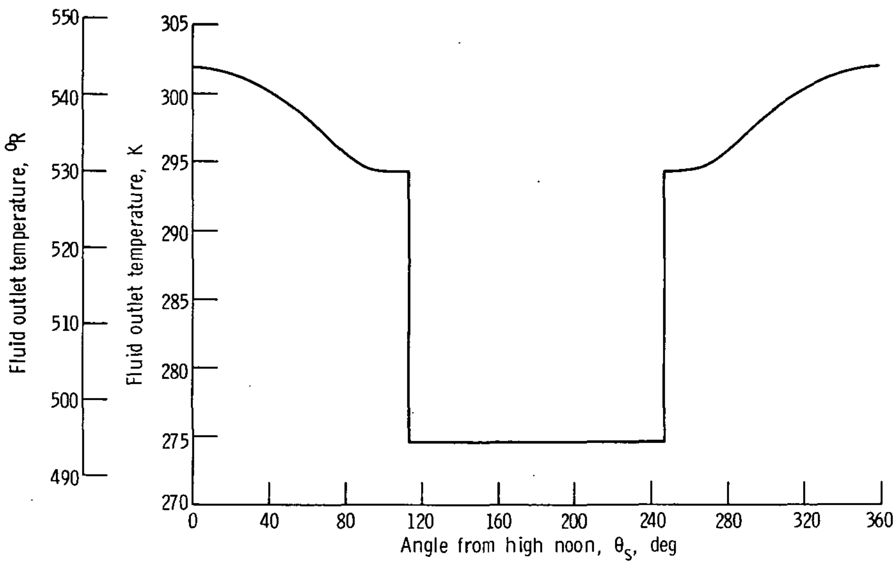

Figure 13. - Variation in fluid outlet temperature with orbital position for uniform equivalent sink temperature around cylinder.

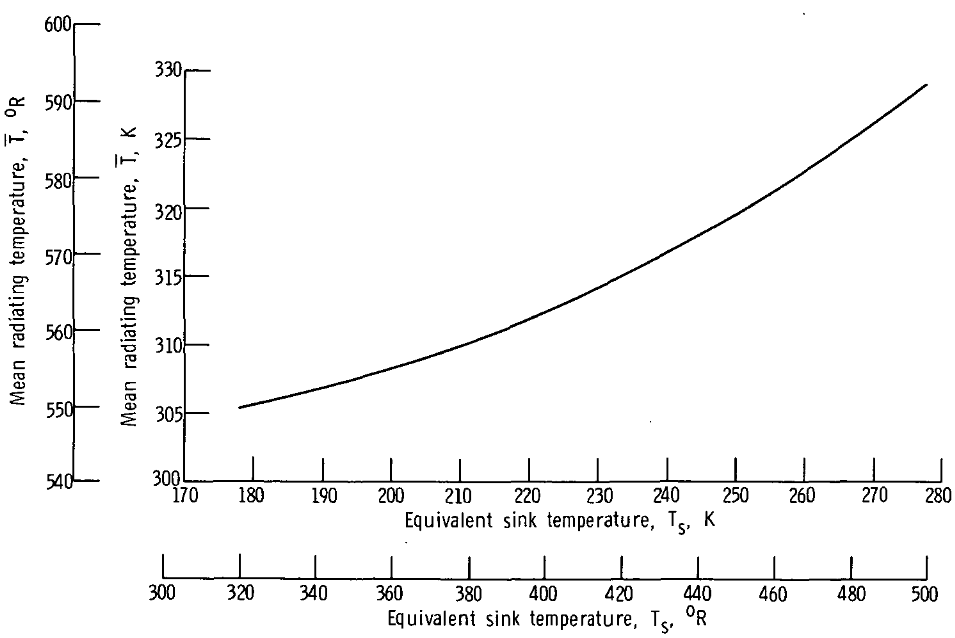

Figure 14. - Mean radiating temperature as function of equivalent sink temperature. Based on $Q = \sigma \varepsilon A (\bar{T}^4 - T_s^4)$ for variable Q; i.e., $Q = f(T_s)$.

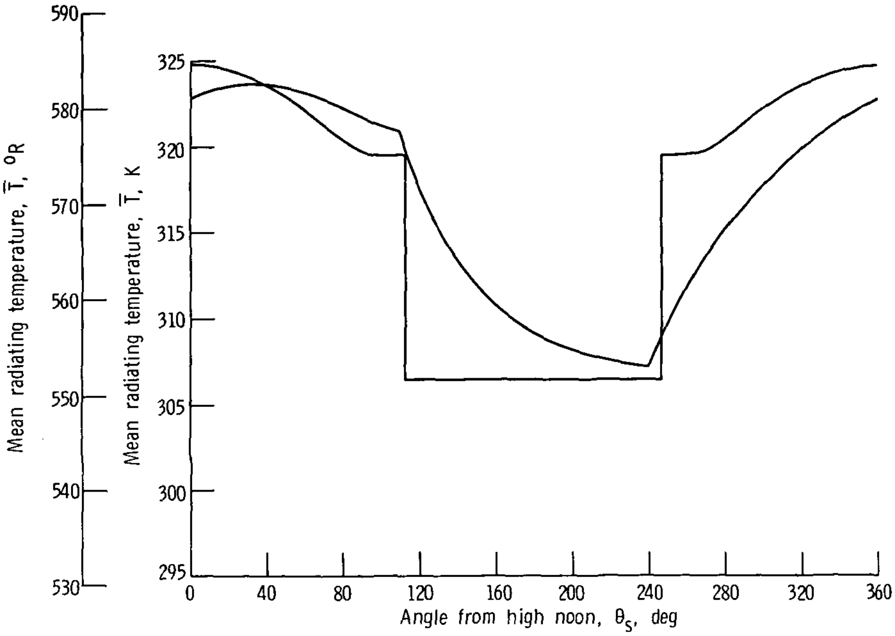

Figure 15. - Steady-state and dynamic isothermal radiating temperatures throughout 555.6-kilometer (300-n-mi) Earth orbit with constant fluid flow rate in radiator.

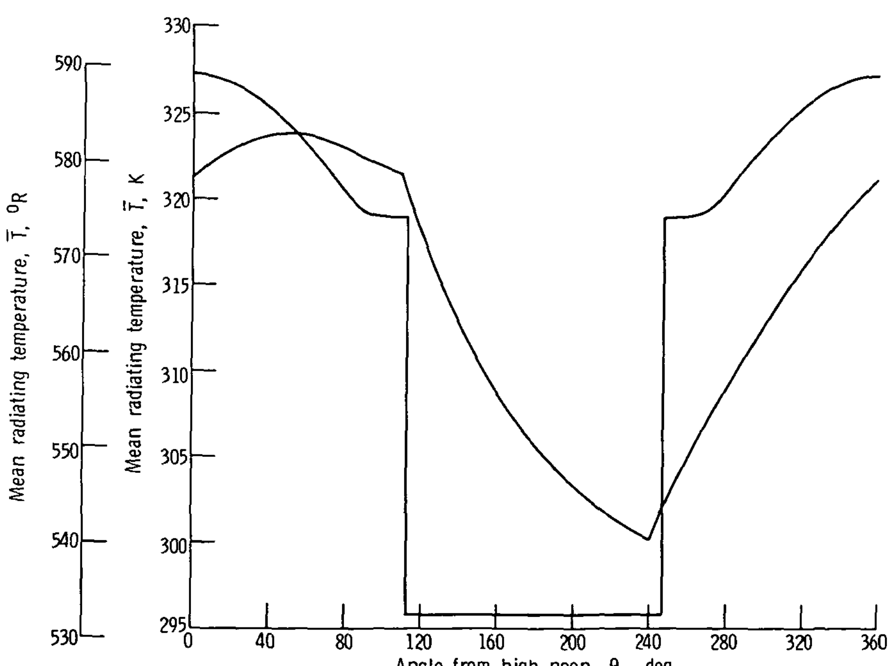

Figure 16. - Steady-state and dynamic isothermal radiating temperatures throughout 555.6-kilometer (300-n-mi) Earth orbits for constant radiator heat rejection.

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