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Analytical Investigation of a Reflux Boiler

FINAL REPORT

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Ultralight Fabric Reflux Tube
Thermal Model

About the Ultralight Fabric Reflux Tube Thermal Model Report

The purpose of supplying the final report in this electronic book form is to provide the user convenience and the ability to directly use the thermal model developed since all of the calculation and plotting features of Mathcad are live. A change in any desired parameter is accomplished by simply finding the value (or values if it is a range variable) and changing it (them) to some other value. All values will be recalculated (provided the calculation mode remains in the automatic mode) and plots depending on this (these) value will be redrawn.

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Click on the "TOC" button on the Electronic Book controls palette to go to the Table of Contents.

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Heat Transfer Modeling of a Single Ultralight Fabric Reflux Tube (UFRT)

About the Format of Electronic Books

About this Electronic Book

Table of Contents

Thermal Modeling of a Single Ultralight Fabric Reflux Tube

Executive Summary

A thermal model of a single Ultralight Fabric Reflux Tube (UFRT) was constructed and tested against data for an array of such tubes tested in the NASA-JSC facility. Modifications to the single fin model were necessary to accommodate the change in radiation shape factors due to adjacent tubes. There was good agreement between the test data and data generated for the same cases by the thermal model. The thermal model was also used to generate single and linear array data for the lunar environment (the primary difference between the test and lunar data was due to lunar gravity). The model was also used to optimize the linear spacing of the reflux tubes in an array. The optimal spacing of the tubes was recommended to be about 5 tube diameters based on maximizing the heat transfer per unit mass. The model also showed that the thermal conductivity of the Nextel fabric was the major limitation to the heat transfer. This led to a suggestion that the feasibility of jacketing the Nextel fiber bundles with copper strands be investigated. This jacketing arrangement was estimated to be able to double the thermal conductivity of the fabric at a volume concentration of about 12-14%. Doubling the thermal conductivity of the fabric would double the amount of heat transferred at the same steam saturation temperature.

The thermal model consisted of a one-dimensional, composite convection-conduction model. The one-dimensional assumption was very good on the shaded side of the tube that experienced condensation and was less so when the other side experienced solar insolation of sufficient magnitude to cause heat addition to the tube. The radiation was modeled as involving diffuse surfaces. On the shade side radiation exchange with the lunar surface and space was considered. On the sun side, a solar heat flux was input at the radiosity node of the fin since it really represented the net energy exchange with the sun.

Introduction

A project was initiated at Lamar University to design and test a thermal model of the new type heat transfer fin shown in the sketch below. The model results were compared with data obtained in NASA's facility at JSC (Building 33, Chamber E) and then used to predict performance on the lunar surface.

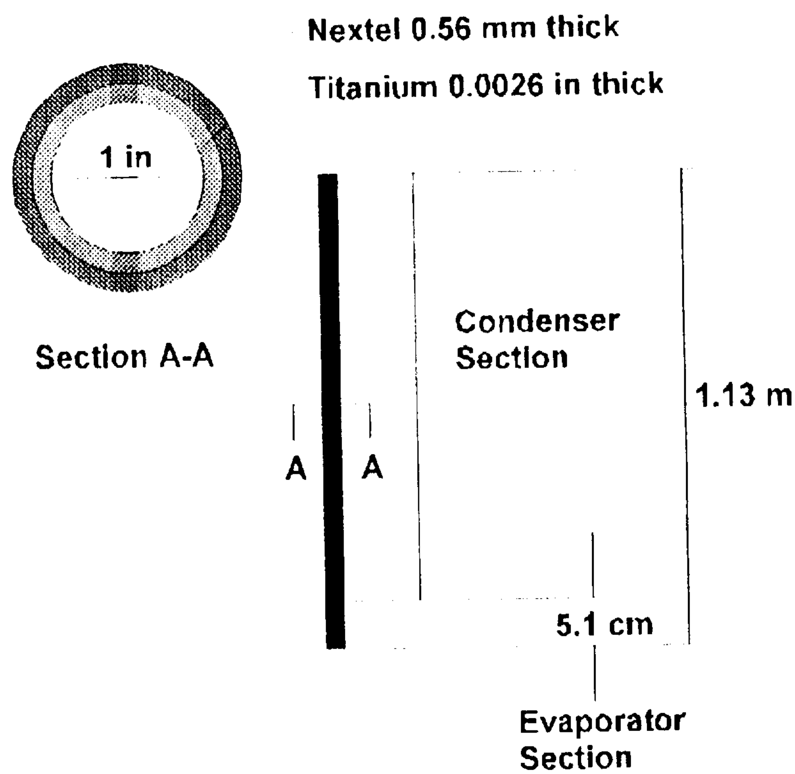


Figure 1. A sketch of the Ultralight Fabric Reflux Tube (UFRT) to be modeled.

The Model

The thermal model consisted of a sequence of quasi-one-dimensional processes: starting at the saturation temperature of the steam inside the tube the processes are; a convection process at the tube inside wall, a conduction process through the titanium liner, a conduction process through the containment fabric, and a radiation process from the fabric outer surface. The process is shown by the sketch of the electric circuit analogy shown in Figure 2. below.

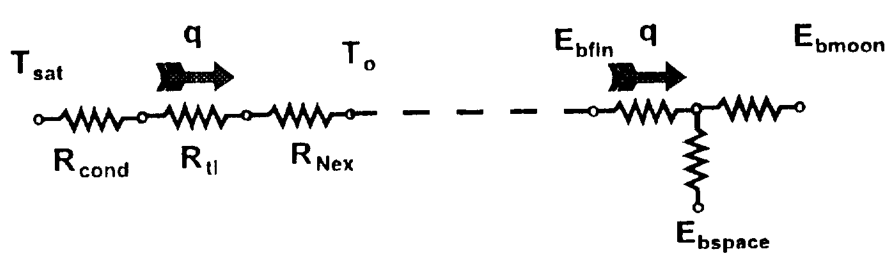


Figure 2. A sketch of the electric circuit analogy of the thermal model.

R_{cond} represents the condensation process inside the tube. In Appendix I it was shown that the condensation process could be modeled as laminar film condensation on a flat plate. Further it was shown that the magnitude of the temperature drop across the film for the range of interest of our study was less than 0.1 degree Celsius. The heat transfer coefficient for the condensation process was taken at a constant value for the JSC test and for the lunar environment at the values shown below.

$$h_{JSC} = 60 \cdot \frac{kW}{m^2 \cdot K}$$

Conservative Estimates from Appendix I

$$h_{lunar} = 40 \cdot \frac{kW}{m^2 \cdot K}$$

Unfortunately, it was postulated in Appendix II and demonstrated in analyzing test data in Appendix III that a simple one-dimensional model would not adequately represent the behavior of the system. Since solar insolation impacts only one side of the tube, the model had to be divided into two parts as shown in the sketch of Figure 3. below.

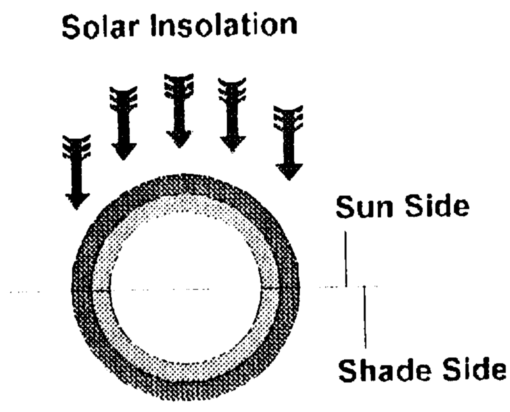


Figure 3. The two parts (sides) of the thermal model.

Some combinations of the radiant boundary conditions cause the direction of heat transfer to reverse. Such reversal changes the internal convective process from condensation to free convection. In Appendix VIII the free convection heat transfer coefficients are calculated to be:

$$h_{JSC} = 57 \cdot \frac{watt}{m^2 \cdot K}$$

Conservative Estimates from Appendix VIII

$$h_{lunar} = 31 \cdot \frac{watt}{m^2 \cdot K}$$

Comparison with Experiment

In Appendix III the thermal model was used to generate data for the cases of the NASA-JSC test.

Conclusions

1. The thermal model contained in this report closely reproduces the test data generated in the NASA-JSC test facility.

2. From a heat transfer perspective, the optimal tube length to diameter ratio does not appear to have been reached. It appears that the optimal value would be limited by the thickness of the laminar condensate layer at the bottom of the tube which could be increased above the values calculated.

3. A simple model of the tube, heat transfer fluid system, and support structure for the fins indicates that the optimal pitch of a linear tube array should be about 5 tube diameters. This result is based on a mass of heat transfer fluid, fluid lines and support structure which would be about three times that of the tube itself per meter of pitch spacing. The parameter optimized, heat transfer per Kg of mass, is very sensitive to smaller values of pitch to diameter ratio but much less sensitive to larger values (see Appendix VI).

4. As expected, the thermal conductivity of the fabric containment limits the heat

transfer from the tube. A method of introducing a small volume fraction (about 12-14%) of copper fibers into the Nextel fabric is suggested in <u>Appendix VII</u> which it is estimated would double the effective thermal conductivity of the fabric and thereby double the heat transfer from the tube at the same saturation temperature of the steam inside the tubes.

End of Section

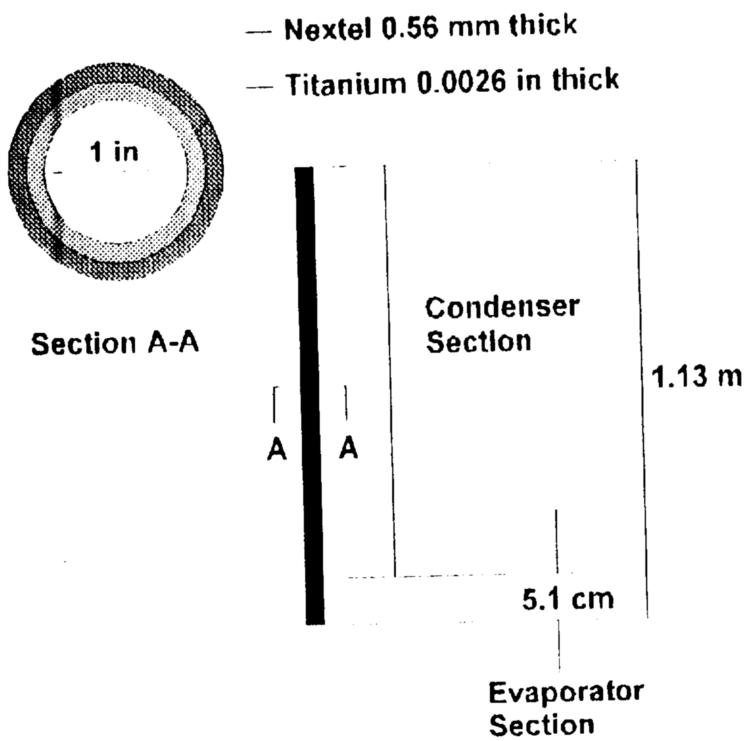
Table of Contents

Next Section

APPENDIX I

Thermal Modeling of the Condensation Process Inside a Single Ultralight Fabric Reflux Tube (UFRT)

Project Objective: To formulate and test a thermal model of the new type heat transfer fin shown in the sketch below. The model is to be tested against data obtained in NASA's facility at JSC and then used to predict performance on the Lunar surface.



$k_{nex} = 0.00012 \cdot \frac{kW}{m \cdot K}$

$k_{ti} = 8.4 \cdot \frac{kW}{m \cdot K}$ Ti-2Al-2Mn

$Vol_{water} = 20 \cdot cm^3$

The first step will be to examine the condensation process in the condenser to determine the complexity of the model needed. We will begin by making simple assumptions of laminar filmwise condensation on a vertical flat plate with no interface shear between the vapor and liquid film. We must test to see whether each of these assumptions is valid.

To begin, we will take saturated steam at 20 lb/in² abs condensing on the inside of a vertical 1 in diameter tube 1.13 m tall. The tube wall temperature is constant and varied from 227.8 to 227.95 °F. The heat transfer coefficient, the heat transfer and the condensate flow will be calculated for each wall temperature. Take T_{sat} = 227.96 °F and h_{fg} = 960.1 Btu/lb.

$j := 0..15$ $\qquad t_j := 227.8 + j \cdot .01$ $\qquad g := 9.8 \cdot \frac{m}{sec^2}$

$T_{w_j} := (460 + t_j) \cdot R \qquad T_{sat} := (460 + 227.96) \cdot R \qquad h_{fg} := 960.1 \cdot \frac{BTU}{lb}$

$T_{f_j} := \frac{T_{w_j} + T_{sat}}{2 \cdot K} \qquad \Delta T_j := T_{sat} - T_{w_j} \qquad L := 1.13 \cdot m - 5.1 \cdot cm$

$d := 1 \cdot in$

$N := \mathrm{READ}(sz\_h2o)$ Number of data points in water properties

$i := 0 .. (N - 1)$

$Th_i := \mathrm{READ}(T\_h2o) \qquad \mu h_i := \mathrm{READ}(mu\_h2o) \qquad \rho h_i := \mathrm{READ}(ro\_h2o)$

$kh_i := \mathrm{READ}(k\_h2o) \qquad GrPr_i := \mathrm{READ}(GrPr\_h2o)$

$Chp_i := \mathrm{READ}(CP\_h2o)$

$\mu_{f_j} := \mathrm{linterp}\left(Th, \mu h, T_{f_j}\right) \cdot \frac{kg}{m \cdot sec}$

$\rho_{f_j} := \mathrm{linterp}\left(Th, \rho h, T_{f_j}\right) \cdot \frac{kg}{m^3}$

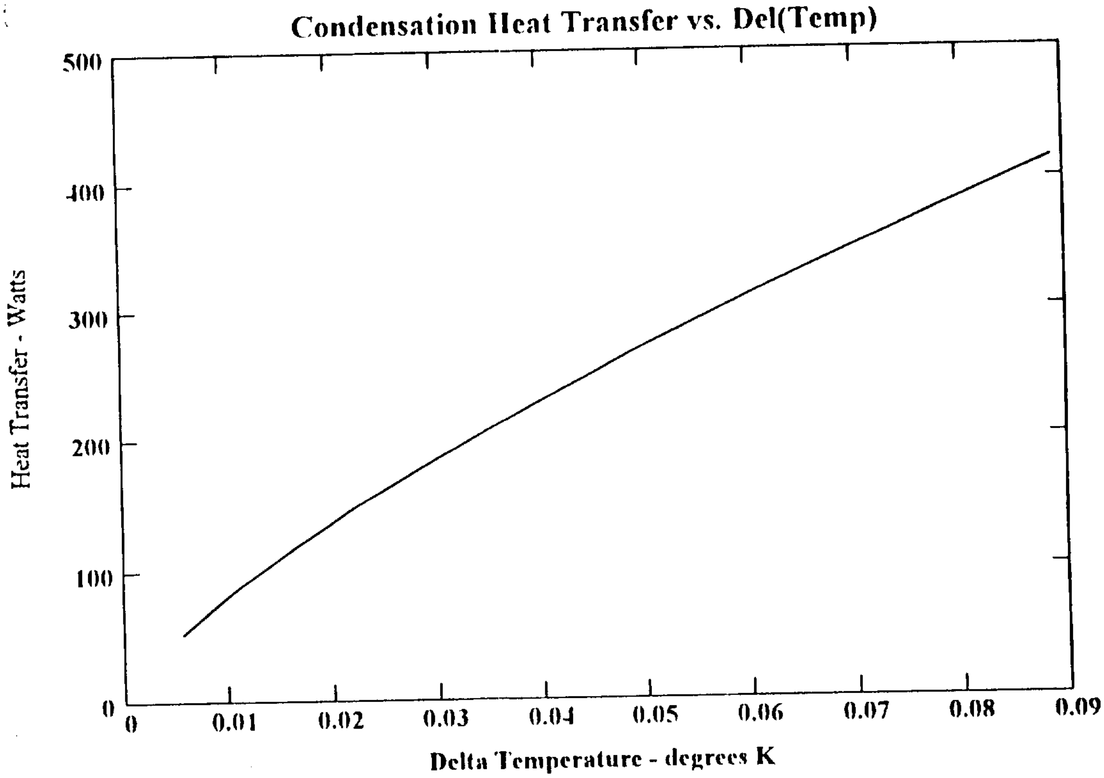
$k_{f_j} := \mathrm{linterp}\left(Th, kh, T_{f_j}\right) \cdot \frac{watt}{m \cdot K}$

$\rho_{v1_j} := 0.5542 + \frac{T_{f_j} - 400}{50} \cdot (0.4902 - 0.5542)$

$\rho_{v_j} := \frac{100 \cdot \rho_{v1_j}}{14.7} \cdot \frac{kg}{m^3}$

$h_j := 0.943 \cdot \left| \frac{\rho_{f_j} \cdot \left(\rho_{f_j} - \rho_{v_j}\right) \cdot g \cdot h_{fg} \cdot \left(k_{f_j}\right)^3}{\mu_{f_j} \cdot d \cdot \left(T_{sat} - T_{w_j}\right)} \right|^{\frac{1}{4}}$ Eqn. 9-10 in Holman

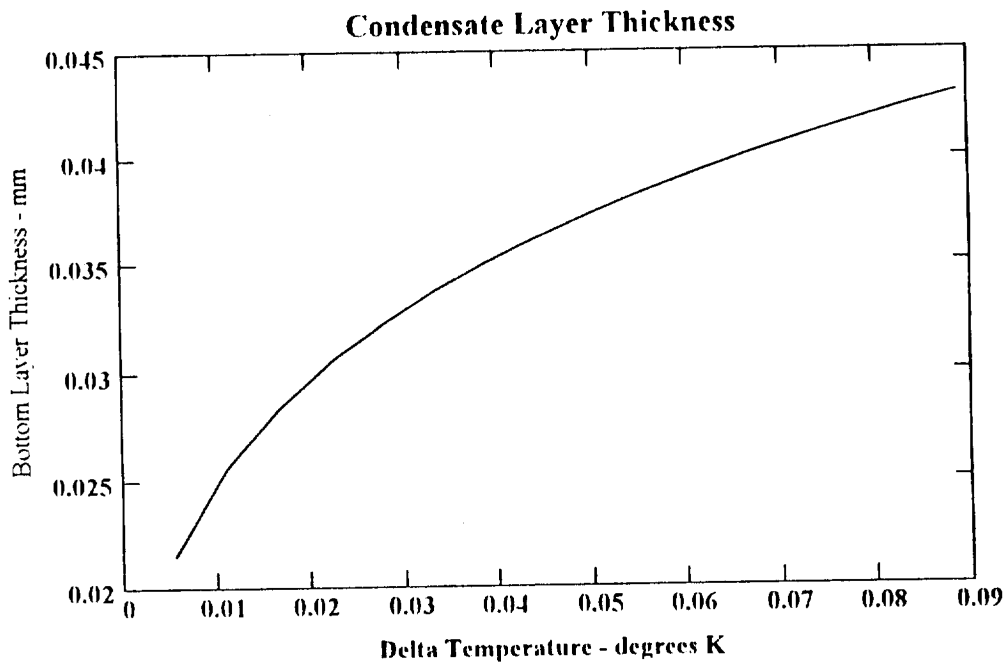
$q_j := h_j \cdot \pi \cdot d \cdot L \cdot \left(T_{sat} - T_{w_j}\right)$ Eqn. 9-15 in Holman

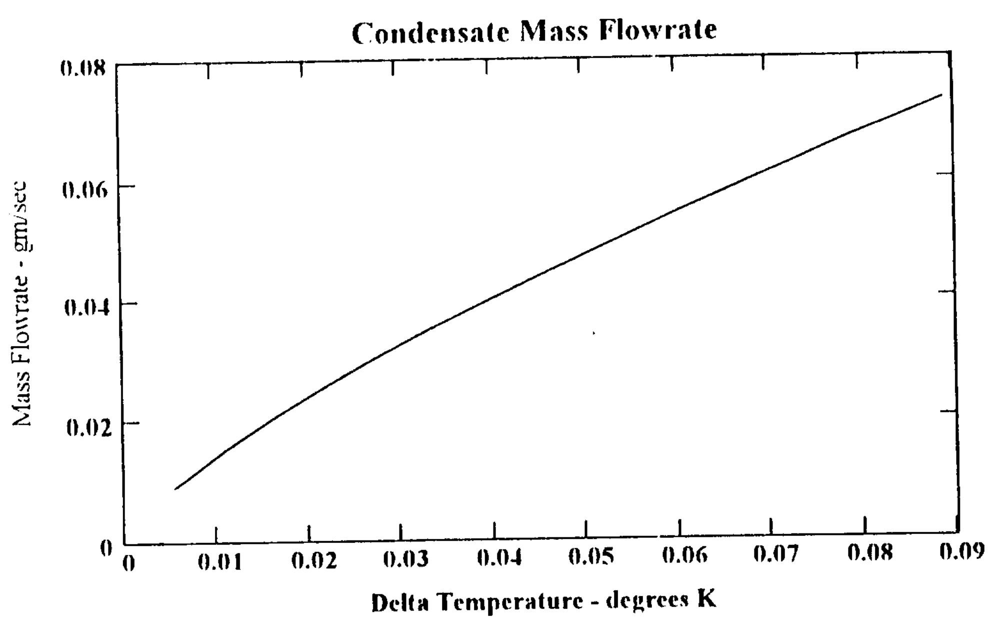


$$\delta_j = \left[\frac{4 \cdot \mu_{f_j} \cdot k_{f_j} \cdot L \cdot \left(T_{sat} - T_{w_j} \right)}{g \cdot h_{fg} \cdot \left[\rho_{f_j} \cdot \left(\rho_{f_j} - \rho_{v_j} \right) \right]} \right]^{\frac{1}{4}}$$

Eqn. 9-6 in Holman

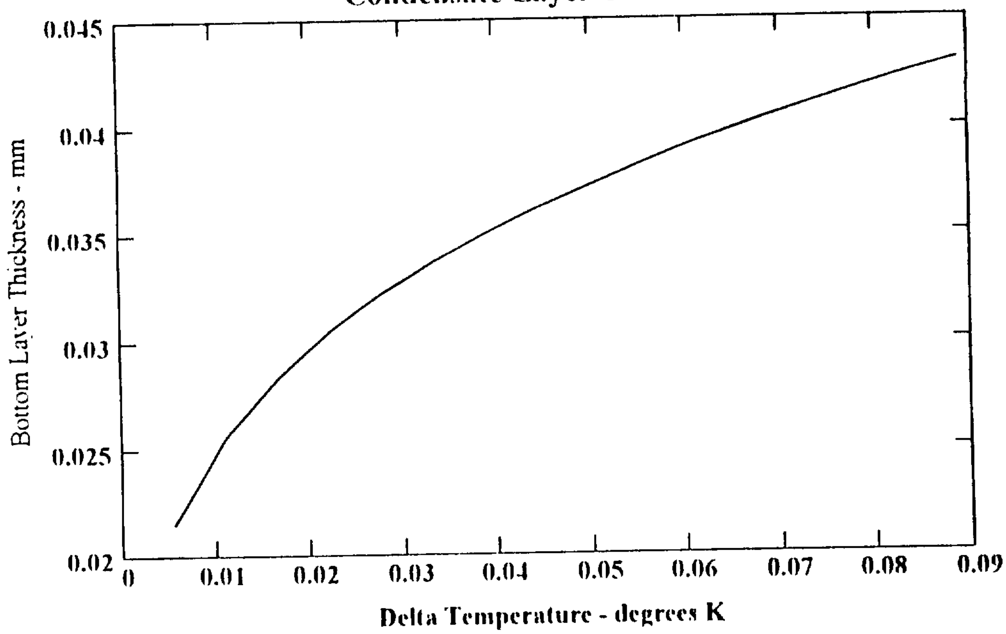
$$w_j = \frac{\rho_{f_j} \cdot \left(\rho_{f_j} - \rho_{v_j} \right) \cdot g \cdot \left(\delta_j \right)^3 \cdot \pi \cdot d}{3 \cdot \mu_{f_j}}$$



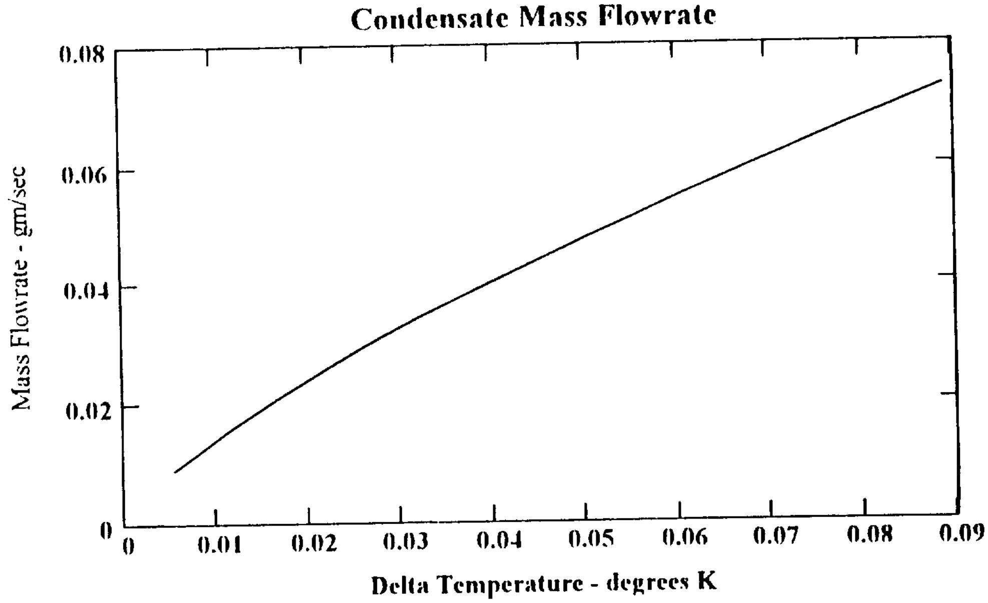


$$Re_{f_j} = \frac{4 \cdot w_j}{\pi \cdot d \cdot \mu_{f_j}}$$ Eqn. 9-13 in Holman

Condensate Layer Thickness

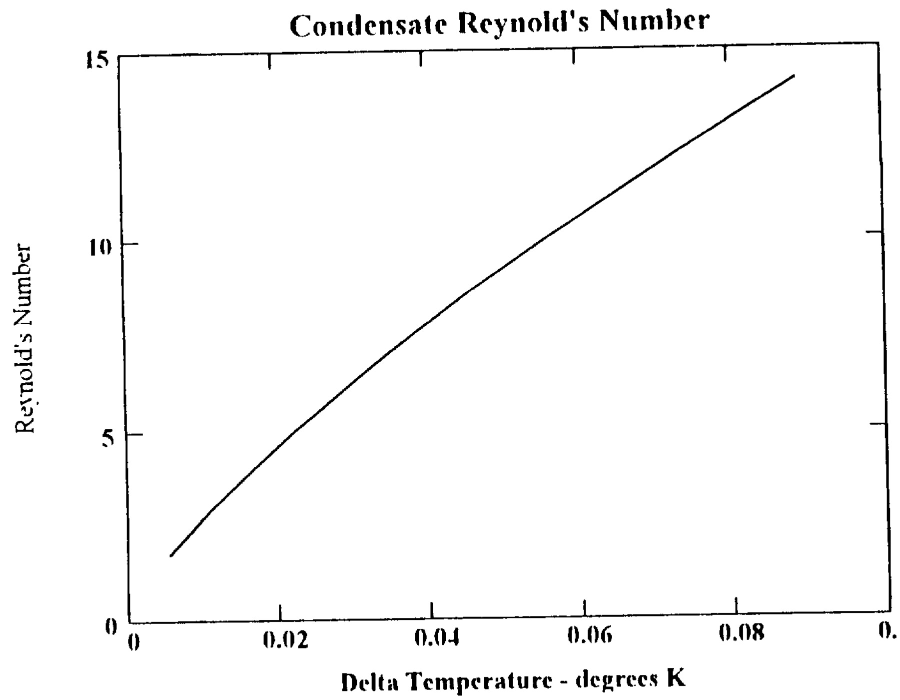


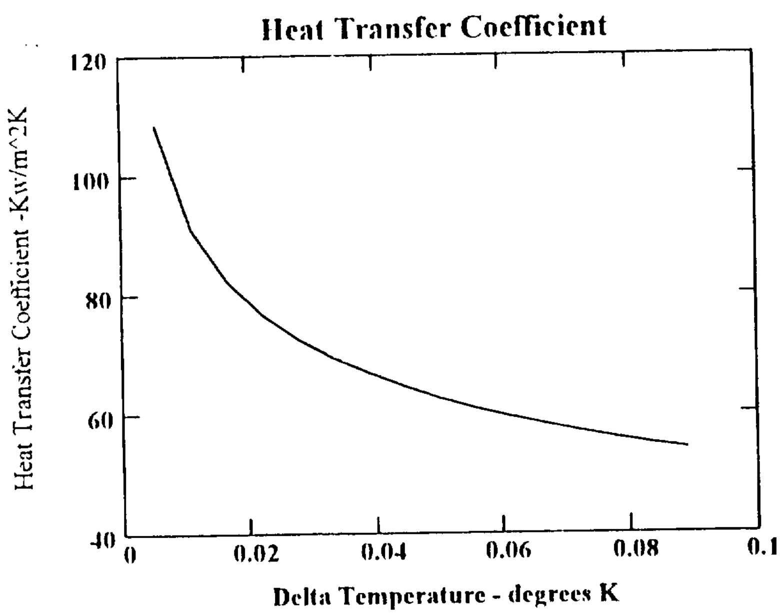
Condensate Mass Flowrate



$$Re_{f_j} \quad \frac{4\cdot w_j}{\pi\cdot d\cdot \mu_{f_j}}$$

Eqn. 9-13 in Holman





From which we can see that assuming an h = 60 Kw/m2oK would be conservative.

$$h_{cond} \quad 60 \cdot \frac{kW}{m^2 \cdot K}$$

Therefore, we can examine our assumptions. The filmwise condensation process is laminar as shown by Reynold's Numbers in the range of 2 to 14 in the figure above. The figure for condensate layer thickness at the bottom of the tube gives a maximum of:

$$\delta_{max} = 3.5 \cdot 10^{-5} \cdot m$$

at the maximum heat dissipation that we are interested in of 150 Watts.

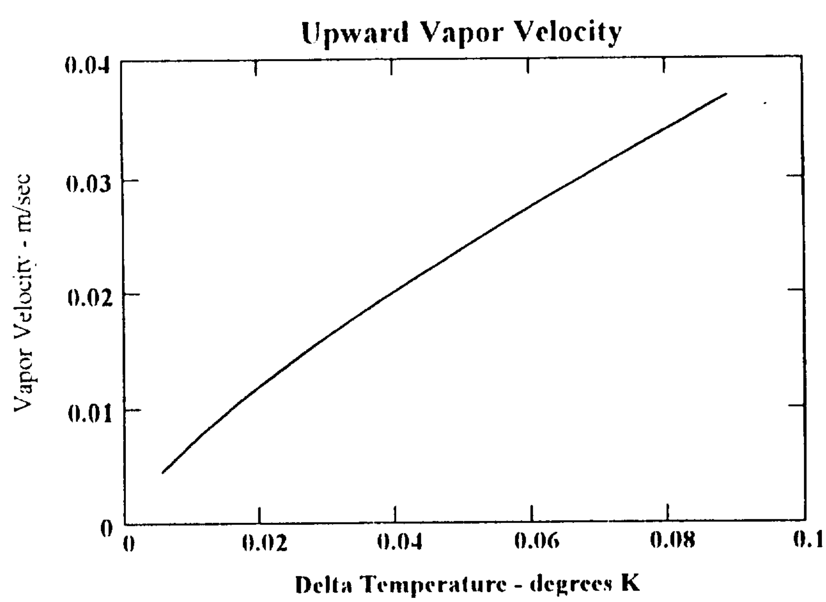
The ratio of diameter to condensate thickness is then:

$$ratio = \frac{\delta_{max}}{d} \qquad ratio = 1.378 \cdot 10^{-3}$$

which validates our vertical flat plate assumption.

We will calculate and plot the upward vapor velocity to determine whether our no interface shear assumption is approximately correct. We know that for steady state the upward mass flowrate of vapor must be equal to the downward flow of condensate.

$$A_{v_j} = \frac{\pi \cdot \left(d - 2 \cdot \delta_j\right)^2}{4} \qquad w_{vapor} = w_{condensate} \qquad V_{v_j} = \frac{w_j}{\rho_{v_j} \cdot A_{v_j}}$$



which validates our assumption of no interface shear.

We know that the initial water charge is 20 cm³. Let us check to see the maximum volume of water contained in the film to be sure that the evaporator section does not go dry. The volume of water contained in the film is given by:

$$\text{Volume} = \int_0^L \pi \cdot d \cdot \delta \, dx$$

$$\text{Volume} = \pi \cdot d \cdot \int_0^L \left[\frac{4 \cdot \mu \cdot k \cdot x \cdot (T_{sat} - T_w)}{g \cdot h_{fg} \cdot \left[\rho_f \cdot (\rho_f - \rho_v) \right]} \right]^{\frac{1}{4}} dx$$

$$\text{Volume} = \pi \cdot d \cdot \left[\frac{4 \cdot \mu \cdot k \cdot (T_{sat} - T_w)}{g \cdot h_{fg} \cdot \left[\rho_f \cdot (\rho_f - \rho_v) \right]} \right]^{\frac{1}{4}} \cdot \frac{(L)^{\frac{5}{4}}}{\left(\frac{5}{4}\right)}$$

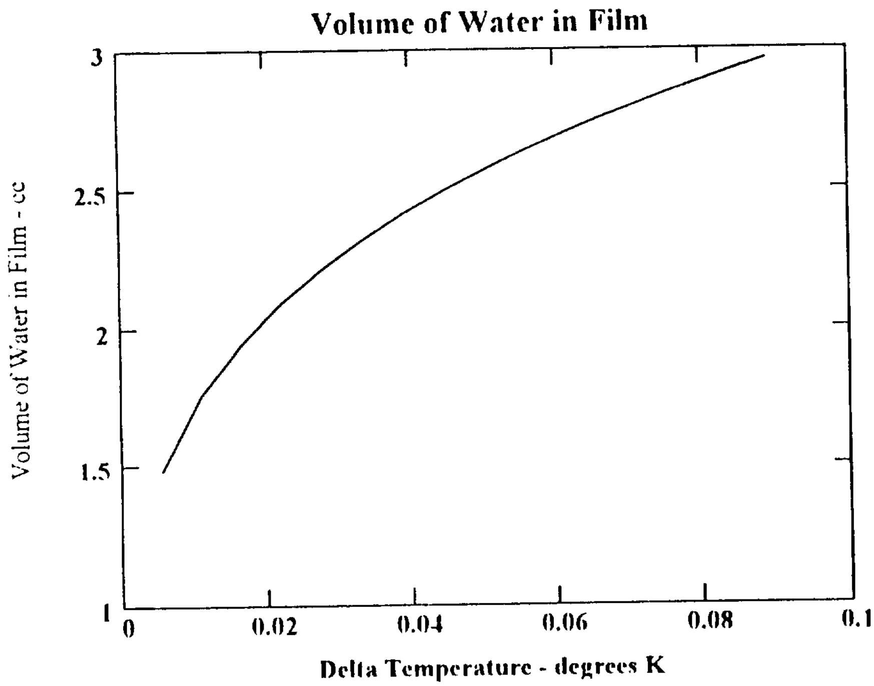
$$\text{Volume}_j := \pi \cdot d \cdot \left[\frac{4 \cdot \mu_{f_j} \cdot k_{f_j} \cdot (T_{sat} - T_{w_j})}{g \cdot h_{fg} \cdot \left[\rho_{f_j} \cdot (\rho_{f_j} - \rho_{v_j}) \right]} \right]^{\frac{1}{4}} \cdot \frac{(L)^{\frac{5}{4}}}{\left(\frac{5}{4}\right)} \cdot 100^3$$

$$v_g := 20.089 \cdot \frac{ft^3}{lb}$$

$$m_{vapor} := \frac{\pi \cdot d^2}{4} \cdot \frac{L + 5.1 \cdot cm}{v_g} \qquad m_{vapor} = 4.566 \cdot 10^{-4} \cdot kg$$

but at $T_{sat} := 500 \cdot K$ $\qquad v_g := 3.788 \cdot \frac{ft^3}{lb}$

$$m_{vapor} \quad \frac{\pi \cdot d^2}{4} \cdot \frac{L + 5.1 \cdot cm}{v_g} \qquad\qquad m_{vapor} = 2.421 \cdot 10^{3} \cdot kg$$



So about 10% of the 20 cm³ charge is contained in the film at the maximum anticipated heat load of 150 Watts, however, the mass of water vapor becomes the same order of magnitude for a saturation temperature of 500°K.

Lunar Environment

Now we need to repeat the calculations for the lunar environment.

$$g \quad \frac{9.8}{6} \cdot \frac{m}{sec^2}$$

Again, we will take saturated steam at 20 lb/in² abs condensing on the inside of a vertical 1 in diameter tube 1.13 m tall. The tube wall temperature is constant and varied from 227.8 to 227.95 °F. The heat transfer coefficient, the heat transfer and the condensate flow will be calculated for each wall temperature. Take T_{sat} = 227.96 °F and h_{fg} = 960.1 Btu/lb.

$j := 0..15 \qquad t_j := 227.8 + j \cdot .01$

$T_{w_j} := (460 + t_j) \cdot R \qquad T_{sat} := (460 + 227.96) \cdot R \qquad h_{fg} := 960.1 \cdot \dfrac{BTU}{lb}$

$T_{f_j} := \dfrac{T_{w_j} + T_{sat}}{2 \cdot K} \qquad \Delta T_j := T_{sat} - T_{w_j} \qquad L := 1.13 \cdot m - 5.1 \cdot cm$

$d := 1 \cdot in$

$N := READ(sz\_h2o)$ Number of data points in water properties

$i := 0..(N - 1)$

$Th_i := READ(T\_h2o) \qquad \mu h_i := READ(mu\_h2o) \qquad \rho h_i := READ(ro\_h2o)$

$kh_i := READ(k\_h2o) \qquad GrPr_i := READ(GrPr\_h2o)$

$Chp_i := READ(CP\_h2o)$

$\mu_{f_j} := linterp\left(Th, \mu h, T_{f_j}\right) \cdot \dfrac{kg}{m \cdot sec}$

$\rho_{f_j} := linterp\left(Th, \rho h, T_{f_j}\right) \cdot \dfrac{kg}{m^3}$

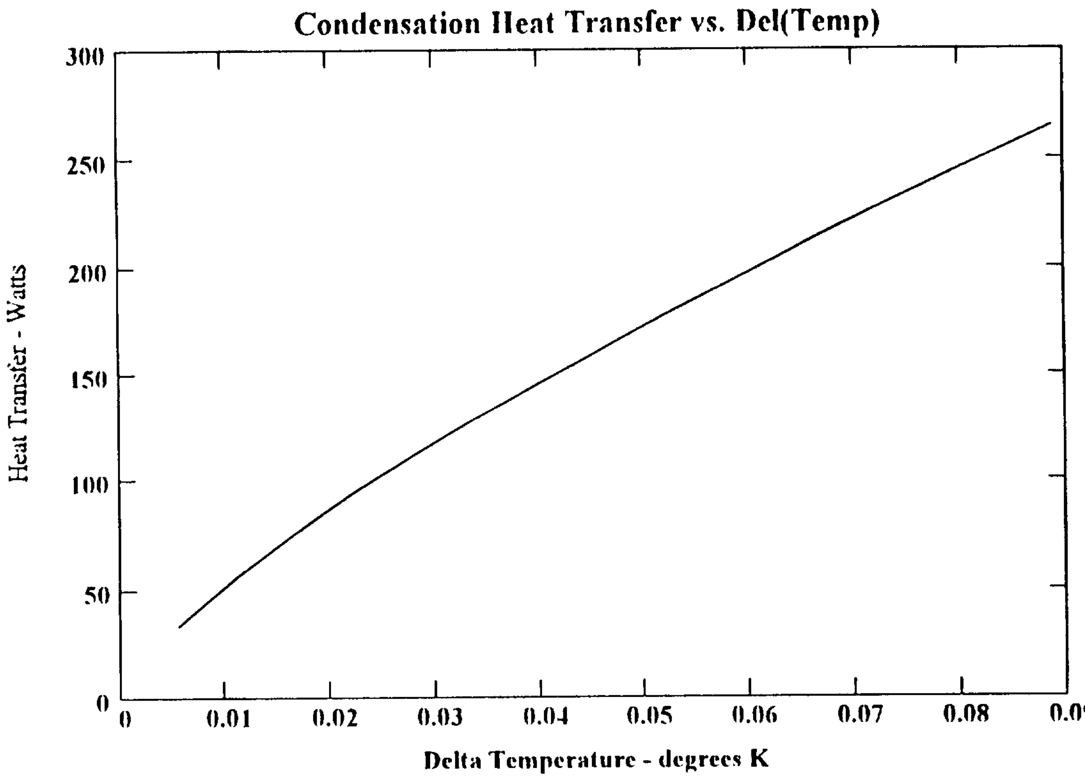
$k_{f_j} := linterp\left(Th, kh, T_{f_j}\right) \cdot \dfrac{watt}{m \cdot K}$

$\rho_{v1_j} := 0.5542 + \dfrac{T_{f_j} - 400}{50} \cdot (0.4902 - 0.5542)$

$\rho_{v_j} := \dfrac{100 \cdot \rho_{v1_j}}{14.7} \cdot \dfrac{kg}{m^3}$

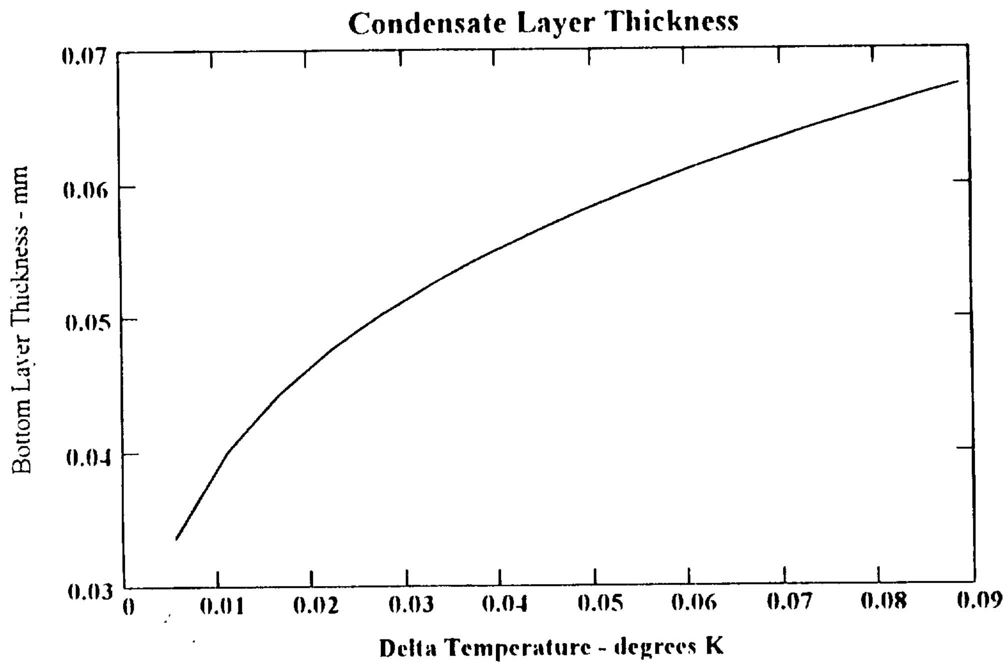
$$h_j := 0.943 \cdot \left[\frac{\rho_{f_j} \cdot \left(\rho_{f_j} - \rho_{v_j}\right) \cdot g \cdot h_{fg} \cdot \left(k_{f_j}\right)^3}{\mu_{f_j} \cdot d \cdot \left(T_{sat} - T_{w_j}\right)} \right]^{\frac{1}{4}}$$

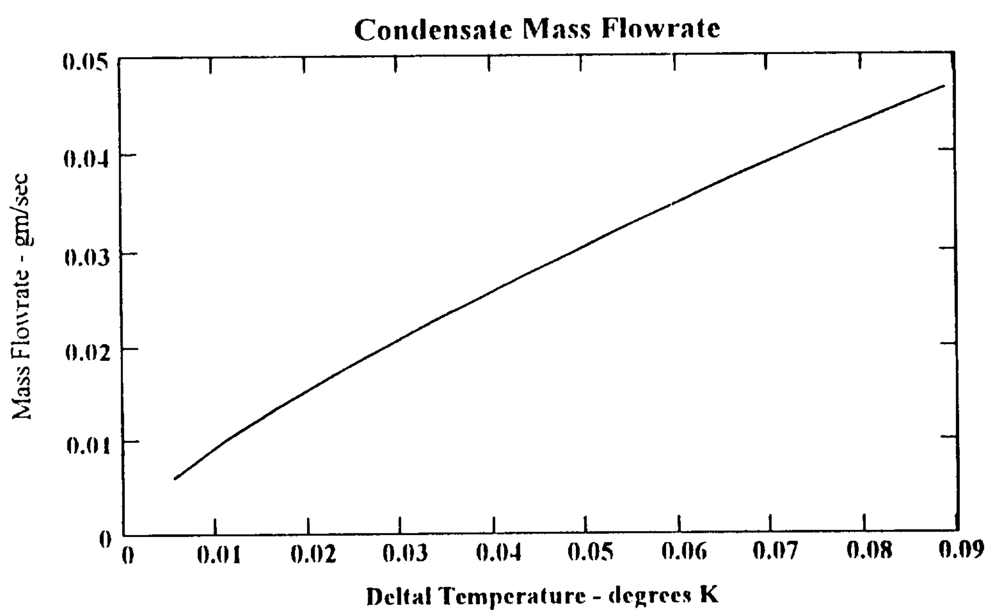
$q_j := h_j \cdot \pi \cdot d \cdot L \cdot \left(T_{sat} - T_{w_j}\right)$



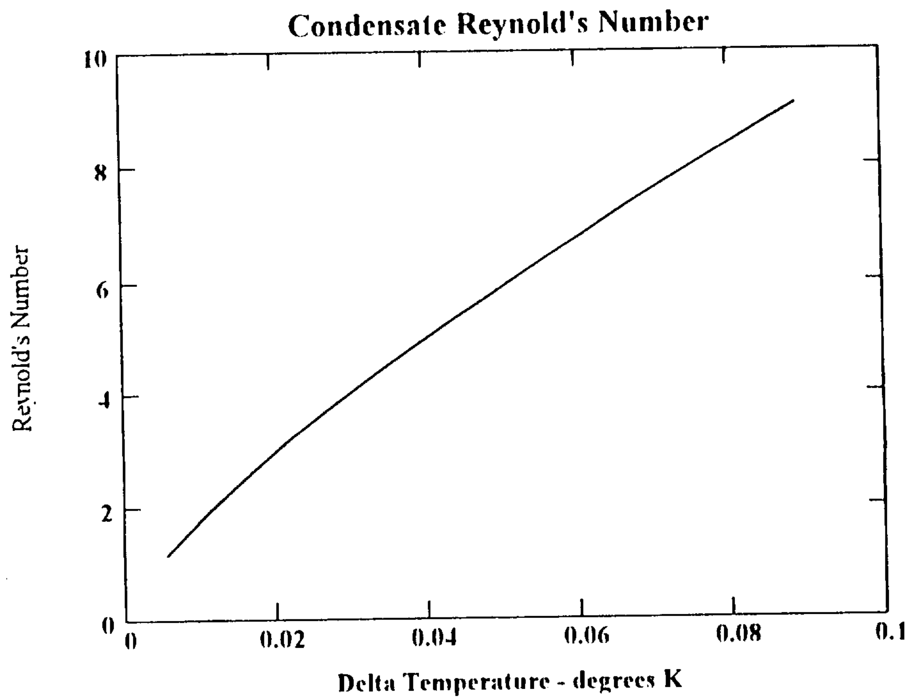
$$\delta_j = \left[\frac{4 \cdot \mu_{f_j} \cdot k_{f_j} \cdot L \cdot \left(T_{sat} - T_{w_j}\right)}{g \cdot h_{fg} \cdot \left[\rho_{f_j} \cdot \left(\rho_{f_j} - \rho_{v_j}\right)\right]} \right]^{\frac{1}{4}}$$

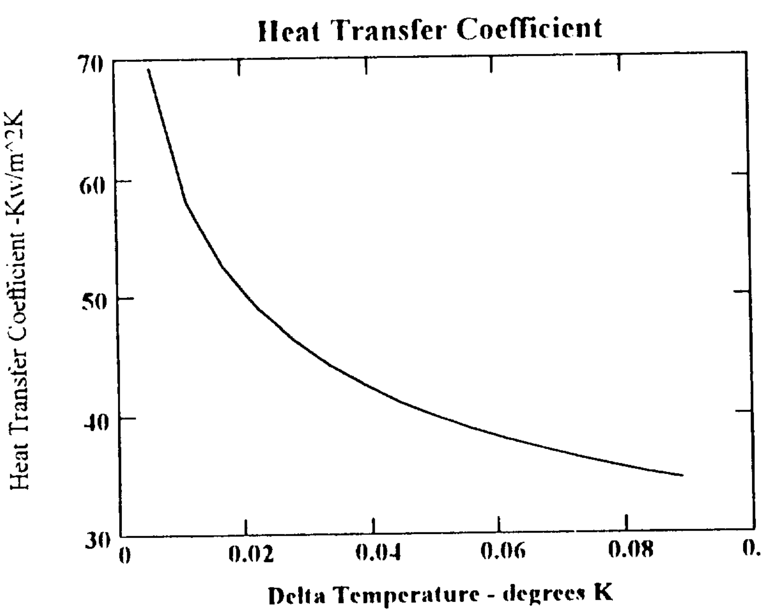
$$w_j = \frac{\rho_{f_j} \cdot \left(\rho_{f_j} - \rho_{v_j}\right) \cdot g \cdot \left(\delta_j\right)^3 \cdot \pi \cdot d}{3 \cdot \mu_{f_j}}$$


Condensate Layer Thickness
Bottom Layer Thickness - mm
0.07
0.06
0.05
0.04
0.03
0
0.01
0.02
0.03
0.04
0.05
0.06
0.07
0.08
0.09
Delta Temperature - degrees K


Condensate Mass Flowrate
Mass Flowrate - gm/sec
0.05
0.04
0.03
0.02
0.01
0
0
0.01
0.02
0.03
0.04
0.05
0.06
0.07
0.08
0.09
Deltal Temperature - degrees K

$$Re_{f_j} = \frac{4 \cdot w_j}{\pi \cdot d \cdot \mu_{f_j}}$$





From which we can see that assuming an h = 40 Kw/m2oK would be conservative.

conservative.

$$h_{cond} := 40 \cdot \frac{kW}{m^2 \cdot K}$$

Therefore, we can examine our assumptions. The filmwise condensation process is laminar as shown by Reynold's Numbers in the range of 2 to 10 in the figure above. The figure for condensate layer thickness at the bottom of the tube gives a maximum of:

$$\delta_{max} := 4.2 \cdot 10^{-5} \cdot m$$

at the maximum heat dissipation that we are interested in of 150 Watts.

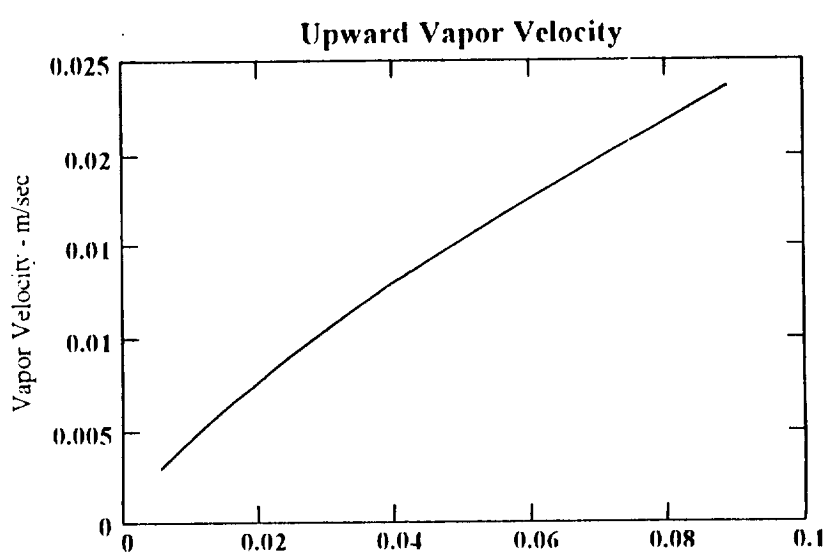
The ratio of diameter to condensate thickness is then:

$$ratio := \frac{\delta_{max}}{d} \qquad ratio = 1.654 \cdot 10^{3}$$

which validates our vertical flat plate assumption.

We will calculate and plot the upward vapor velocity to determine whether our no interface shear assumption is approximately correct. We know that for steady state the upward mass flowrate of vapor must be equal to the downward flow of condensate.

$$A_{v_j} := \frac{\pi \cdot (d - 2 \cdot \delta_j)^2}{4} \qquad w_{vapor} = w_{condensate} \qquad V_{v_j} := \frac{w_j}{\rho_{v_j} \cdot A_{v_j}}$$



Delta Temperature - degrees K

which validates our assumption of no interface shear.

We know that the initial water charge is 20 cm³. Let us check to see the maximum volume of water contained in the film to be sure that the evaporator section does not go dry. The volume of water contained in the film is given by:

$$\text{Volume} = \int_0^L \pi \cdot d \cdot \delta \, dx$$

$$\text{Volume} = \pi \cdot d \cdot \int_0^L \left[\frac{4 \cdot \mu \cdot k \cdot x \cdot (T_{sat} - T_w)}{g \cdot h_{fg} \cdot \left[\rho_f \cdot (\rho_f - \rho_v) \right]} \right]^{\frac{1}{4}} dx$$

$$\text{Volume} = \pi \cdot d \cdot \left[\frac{4 \cdot \mu \cdot k \cdot (T_{sat} - T_w)}{g \cdot h_{fg} \cdot \left[\rho_f \cdot (\rho_f - \rho_v) \right]} \right]^{\frac{1}{4}} \cdot \frac{(L)^{\frac{5}{4}}}{\left(\frac{5}{4}\right)}$$

$$\text{Volume}_j := \pi \cdot d \cdot \left[\frac{4 \cdot \mu_{f_j} \cdot k_{f_j} \cdot \left(T_{sat} - T_{w_j}\right)}{g \cdot h_{fg} \cdot \left[\rho_{f_j} \cdot \left(\rho_{f_j} - \rho_{v_j}\right) \right]} \right]^{\frac{1}{4}} \cdot \frac{(L)^{\frac{5}{4}}}{\left(\frac{5}{4}\right)} \cdot 100^3$$

$$v_g := 20.089 \cdot \frac{ft^3}{lb}$$

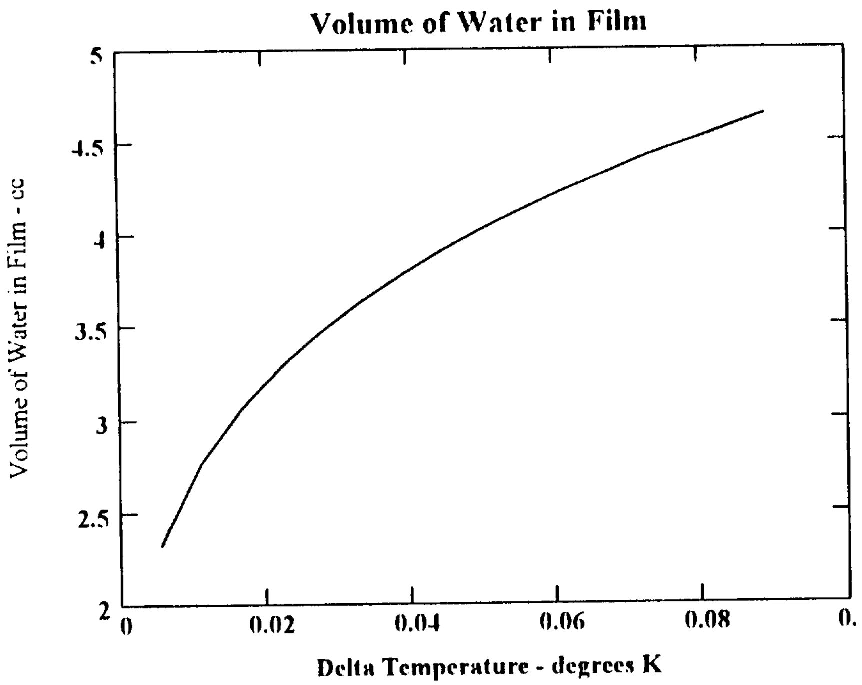
m_{vapor} $\quad \pi \cdot d^2 \cdot L + 5.1 \cdot cm$

$m_{vapor} = 4.566 \cdot 10^{-4} \cdot kg$

$$m_{vapor} := \quad \frac{\quad}{4} \quad \frac{\quad}{v_g} \qquad m_{vapor} = 4.566\ 10 \quad kg$$

but at $T_{sat} \quad 500 \cdot K \qquad v_g \quad 3.788 \cdot \frac{ft^3}{lb}$

$$m_{vapor} := \frac{\pi \cdot d^2}{4} \cdot \frac{L + 5.1 \cdot cm}{v_g} \qquad m_{vapor} = 2.421 \cdot 10^{\ 3} \cdot kg$$



So about 25% of the 20 cm³ charge is contained in the film at the maximum anticipated heat load of 150 Watts, however, the mass of water vapor increases the total to about 35% at a saturation temperature of 500°K.

End of Section

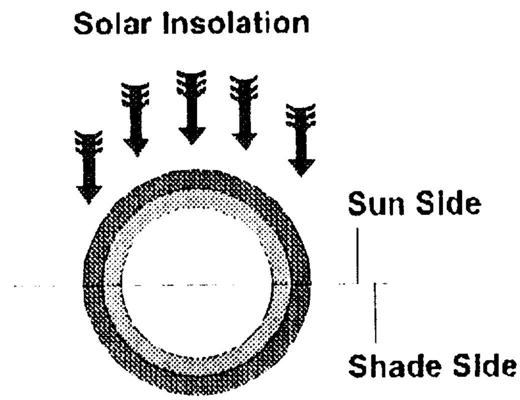
Table of Contents

APPENDIX II

Radiation Modeling

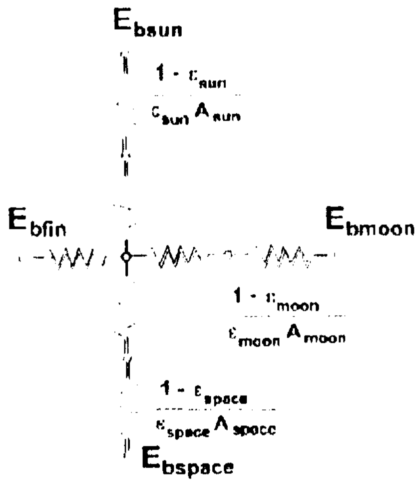
Radiation Heat Transfer

If solar insolation is present, we might note that the insolation is incident on only half the fin area as shown in the sketch below.

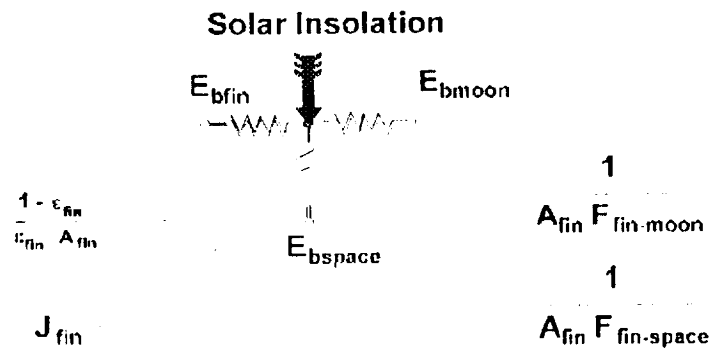


The area of the fin will be composed of equal areas on the sun side and on the shade side.

Radiant energy exchange will be assumed to be diffuse. The electric circuit analogy for the radiant energy exchange between the fin, the lunar surface, the sun and black space can be represented by the partial sketch below.



The reason we have presented only a partial sketch is that we will show that it is all that is necessary to determine the net radiant energy exchange with the fin. In each of the surface resistances there is an area term in the denominator. All other terms will be expressed with the area of term in the denominator. All other terms will be expressed with the area of the fin in the denominator. Since A_{space}, A_{moon}, and A_{sun} are much much larger than A_{fin}, their associated resistances can be taken as zero. In addition, we know the net heat transfer with the sun to be the solar insolation value. Hence, the modified electric circuit analogy would look like that below.



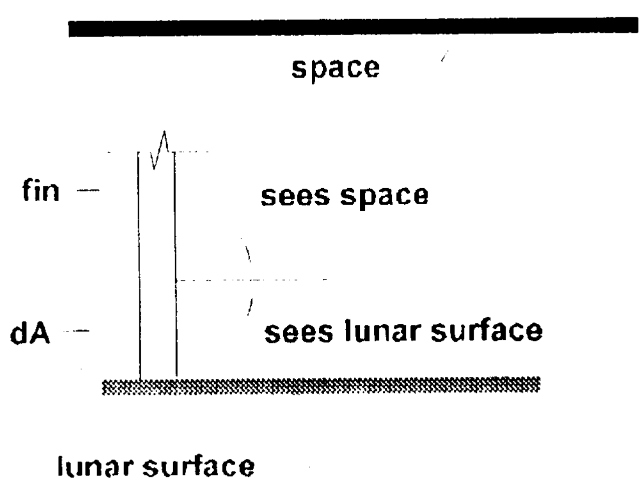
For which we can sum the heat flows into the node J_{fin} and set them to zero.

$$\frac{E_{bfin} - J_{fin}}{\left(\frac{1-\varepsilon_{fin}}{\varepsilon_{fin} \cdot A_{fin}}\right)} + \frac{E_{bspace} - J_{fin}}{\left(\frac{1}{A_{fin} \cdot F_{fin\_space}}\right)} + \frac{E_{bmoon} - J_{fin}}{\left(\frac{1}{A_{fin} \cdot F_{fin\_moon}}\right)} + q_{sun} = 0$$

which applies on the sun side and a similar equation would apply on the shade side.

$$\frac{E_{bfin2} - J_{fin2}}{\left(\frac{1-\varepsilon_{fin}}{\varepsilon_{fin} \cdot A_{fin}}\right)} + \frac{E_{bspace} - J_{fin2}}{\left(\frac{1}{A_{fin} \cdot F_{fin\_space}}\right)} + \frac{E_{bmoon} - J_{fin2}}{\left(\frac{1}{A_{fin} \cdot F_{fin\_moon}}\right)} = 0$$

In these equations we know that $E_{bspace}=0$, $E_{bfin}=\sigma T_{fin}^4$, $E_{bmoon}=\sigma T_{moon}^4$ and $\varepsilon_{fin}=0.89$. The configuration factors may be determined in the following manner: consider the sketch below,



From the figure above it is obvious that $F_{dA\text{-}moon}$ and $F_{dA\text{-}space}$ are both 0.5 for any position on the surface of the fin (assumes lunar surface is an infinite flat plate). Therefore $F_{fin\text{-}moon}$ and $F_{fin\text{-}space}$ are 0.5 for either the whole fin area or half the fin area (divided into sun side and shade side). The equations above then reduce to on the sun side taking q_{sun} to be the energy incident on the fin per unit area times the perpendicular area:

$$J_{fin} = \varepsilon_{fin} \cdot \sigma \cdot T_{fin}^4 + (1 - \varepsilon_{fin}) \cdot \left(\frac{\sigma \cdot T_{moon}^4}{2} + \frac{q_{sun}}{A_{fin}}\right)$$

and on the shade side:

$$J_{fin2} = \varepsilon_{fin} \cdot \sigma \cdot T_{fin2}^{4} + (1 - \varepsilon_{fin}) \cdot \left(\frac{\sigma \cdot T_{moon}^{4}}{2} \right)$$

where A_{fin} is half the total fin area, T_{fin} is the outside temperature of the Nextel exposed to the sun, and T_{fin2} is the Nextel temperature on the shade side.

We can now examine the simple but important case of what minimum heat transfer is necessary to keep the fin from freezing-up during the lunar night. For this time we will take T_{moon} = 88 °K, T_{sat} = 273°K and no solar insolation.

$d := 1 \cdot in \qquad L := 1.13 \cdot m$

$\varepsilon_{fin} := 0.89 \qquad T_{sat} := 273 \cdot K \qquad T_{moon} := 88 \cdot K$

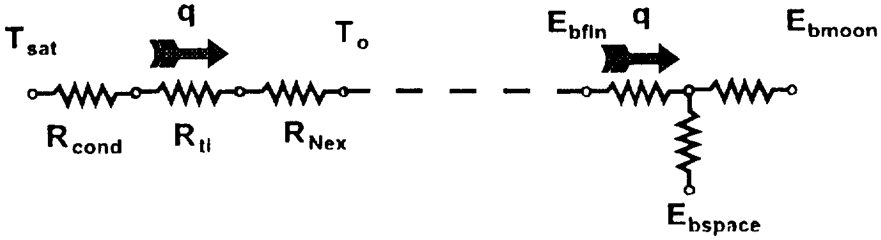
$d_o := d + 2 \cdot .00056 \cdot m + 2 \cdot .0026 \cdot in \qquad d_o = 0.027 \cdot m$

We will iterate for the outside Nextel surface temperature such that the heat loss by radiation is equal to the heat convected and conducted through the cylindrical fin wall. We will take only half the fin area so that we will be consistent later on when we must consider sun side and shade side.

$A_{fin} := \frac{\pi \cdot d_o \cdot L}{2} \qquad A_{fin} = 0.047 \cdot m^{2}$

$k_{ti} := 8.4 \cdot \frac{kW}{m \cdot K} \qquad k_{nex} := 0.00012 \cdot \frac{kW}{m \cdot K} \qquad h_{cond} := 60 \cdot \frac{kW}{m^{2} \cdot K}$

The iteration process is shown in the sketch below.



$$R_{cond} := \frac{1}{h_{cond} \cdot \pi \cdot d \cdot L} \qquad R_{ti} := \frac{\ln\left(\frac{d + .0026 \cdot 2 \cdot in}{d}\right)}{\pi \cdot k_{ti} \cdot L}$$

$$R_{Nex} := \frac{\ln\left(\frac{d_o}{d + .0026 \cdot 2 \cdot in}\right)}{\pi \cdot k_{nex} \cdot L}$$

There is a more compact way to do the calculations below, however, this method has been used to try to illustrate the calculation methodology.

Assume $T_o := 271.694 \cdot K$

$$q_1 := \frac{T_{sat} - T_o}{R_{cond} + R_{ti} + R_{Nex}} \qquad q_1 = 12.935 \cdot watt$$

$$\sigma := 5.669 \cdot 10^{-8} \cdot \frac{watt}{m^2 \cdot K^4}$$

$$J_{fin} := \varepsilon_{fin} \cdot \sigma \cdot T_o^{\,4} + (1 - \varepsilon_{fin}) \cdot \left(\frac{\sigma \cdot T_{moon}^{\,4}}{2}\right) \qquad J_{fin} = 275.114 \cdot \frac{watt}{m^2}$$

$$q_2 := \frac{\sigma \cdot T_o^{\,4} - J_{fin}}{\left(\frac{1 - \varepsilon_{fin}}{\varepsilon_{fin} \cdot A_{fin}}\right)} \qquad q_2 = 12.935 \cdot watt$$

Therefore it takes $q_{tot} := 2 \cdot q_1$ $\quad q_{tot} = 25.871 \cdot watt$

to keep the fin from freezing during the lunar night.

Increasing the saturation temperature to 300°K. $\quad T_{sat} := 300 \cdot K$

$kj := 0 .. 4 \qquad T_{s_{kj}} := 0 \qquad q_{t_{kj}} := 0 \qquad T_{s_0} := 273 \qquad q_{t_0} := \frac{q_{tot}}{watt}$

Assume $T_o := 298.104 \cdot K$

$$q_1 := \frac{T_{sat} - T_o}{R_{cond} + R_{ti} + R_{Nex}} \qquad q_1 = 18.779 \cdot watt$$

$$\sigma := 5.669 \cdot 10^{-8} \cdot \frac{watt}{m^2 \cdot K^4}$$

$$J_{fin} := \varepsilon_{fin} \cdot \sigma \cdot T_o^{\,4} + \left(1 - \varepsilon_{fin}\right) \cdot \left(\frac{\sigma \cdot T_{moon}^{\,4}}{2}\right) \qquad J_{fin} = 398.631 \cdot \frac{watt}{m^2}$$

$$q_2 := \frac{\sigma \cdot T_o^{\,4} - J_{fin}}{\left(\frac{1 - \varepsilon_{fin}}{\varepsilon_{fin} \cdot A_{fin}}\right)} \qquad q_2 = 18.778 \cdot watt$$

$$q_{tot} := 2 \cdot q_1$$

at $T_{sat} = 300 \cdot K$ $\quad q_{tot} = 37.558 \cdot watt$ $\quad T_{s_1} := 300$ $\quad q_{t_1} := \frac{q_{tot}}{watt}$

Increase $T_{sat} := 350 \cdot K$

Assume $T_o := 346.532 \cdot K$

$$q_1 := \frac{T_{sat} - T_o}{R_{cond} + R_{ti} + R_{Nex}} \qquad q_1 = 34.349 \cdot watt$$

$$J_{fin} := \varepsilon_{fin} \cdot \sigma \cdot T_o^{\,4} + \left(1 - \varepsilon_{fin}\right) \cdot \left(\frac{\sigma \cdot T_{moon}^{\,4}}{2}\right) \qquad J_{fin} = 727.749 \cdot \frac{watt}{m^2}$$

$$q_2 := \frac{\sigma \cdot T_o^{\,4} - J_{fin}}{\left(\frac{1 - \varepsilon_{fin}}{\varepsilon_{fin} \cdot A_{fin}}\right)} \qquad q_2 = 34.348 \cdot watt$$

$$q_{tot} := 2 \cdot q_1$$

at $T_{sat} = 350 \cdot K$ $\quad q_{tot} = 68.698 \cdot watt$ $\quad T_{s_2} := 350$ $\quad q_{t_2} := \dfrac{q_{tot}}{watt}$

Increase $\quad T_{sat} := 400 \cdot K$

Assume $\quad T_o := 394.189 \cdot K$

$$q_1 := \frac{T_{sat} - T_o}{R_{cond} + R_{ti} + R_{Nex}} \qquad q_1 = 57.556 \cdot watt$$

$$J_{fin} := \varepsilon_{fin} \cdot \sigma \cdot T_o^4 + (1 - \varepsilon_{fin}) \cdot \left(\frac{\sigma \cdot T_{moon}^4}{2} \right) \qquad J_{fin} = 727.749 \cdot \frac{watt}{m^2}$$

$$q_2 := \frac{\sigma \cdot T_o^4 - J_{fin}}{\left(\dfrac{1 - \varepsilon_{fin}}{\varepsilon_{fin} \cdot A_{fin}} \right)} \qquad q_2 = 57.558 \cdot watt$$

$$q_{tot} := 2 \cdot q_1$$

at $T_{sat} = 400 \cdot K$ $\quad q_{tot} = 115.111 \cdot watt$ $\quad T_{s_3} := 400$ $\quad q_{t_3} := \dfrac{q_{tot}}{watt}$

Increase $\quad T_{sat} := 500 \cdot K$

Assume $\quad T_o := 486.507 \cdot K$

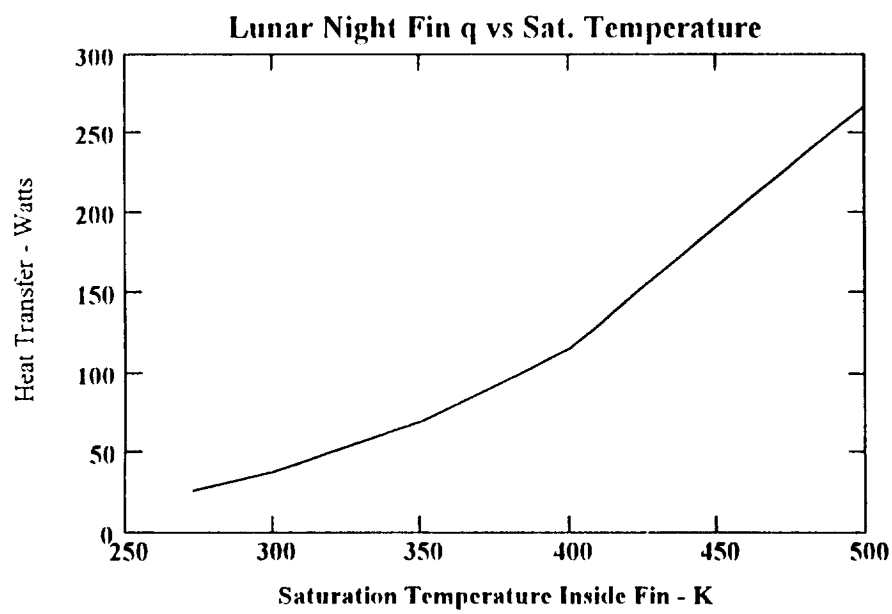
$$q_1 := \frac{T_{sat} - T_o}{R_{cond} + R_{ti} + R_{Nex}} \qquad q_1 = 133.643 \cdot watt$$

$$J_{fin} := \varepsilon_{fin} \cdot \sigma \cdot T_o^{4} + (1 - \varepsilon_{fin}) \cdot \left(\frac{\sigma \cdot T_{moon}^{4}}{2} \right) \qquad J_{fin} = 2.827 \cdot 10^{3} \cdot \frac{watt}{m^{2}}$$

$$q_2 := \frac{\sigma \cdot T_o^{4} - J_{fin}}{\left(\frac{1 - \varepsilon_{fin}}{\varepsilon_{fin} \cdot A_{fin}} \right)} \qquad q_2 = 133.644 \cdot watt$$

$$q_{tot} := 2 \cdot q_1$$

at $T_{sat} = 500 \cdot K$ $\quad q_{tot} = 267.286 \cdot watt$ $\quad T_{s_4} := 500$ $\quad q_{t_4} := \frac{q_{tot}}{watt}$



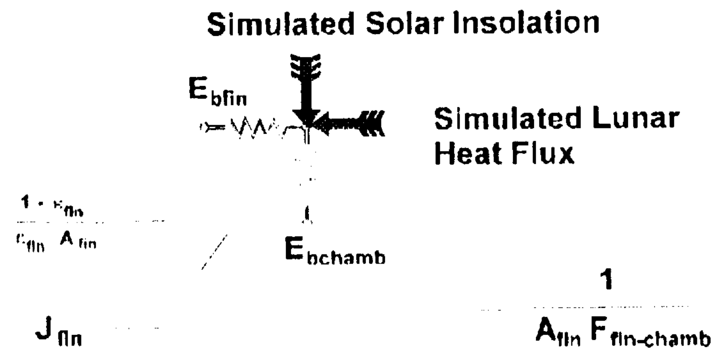
End of Section

Table of Contents

APPENDIX III

Thermal Model of the JSC Test

JSC Test Model



$k_{nex} := 0.00012\cdot\frac{kW}{m\cdot K}$ $\quad L := 1.13\cdot m - 5.1\cdot cm$ $\quad h_{cond} := 60\cdot\frac{kW}{m^2\cdot K}$

$k_{ti} := 8.4\cdot\frac{watt}{m\cdot K}$ $\quad d := 1\cdot in$ $\quad \sigma := 5.669\cdot 10^{-8}\cdot\frac{watt}{m^2\cdot K^4}$

$\varepsilon_{fin} := 0.89$

$d_o := d + 2\cdot .00056\cdot m + 2\cdot .0026\cdot in$ $\quad d_o = 0.027\cdot m$ $\quad h_{fc} := 57\cdot\frac{watt}{m^2\cdot K}$

$A_{fin} := \frac{\pi\cdot d_o\cdot L}{2}$ $\quad A_{fin} = 0.045\cdot m^2$

$$R_{cond} := \frac{1}{h_{cond}\cdot\pi\cdot d\cdot\frac{L}{2}} \qquad R_{ti} := \frac{\ln\left(\frac{d + .0026\cdot 2\cdot in}{d}\right)}{\pi\cdot k_{ti}\cdot L} \qquad R_{Nex} := \frac{\ln\left(\frac{d_o}{d + .0026\cdot 2\cdot in}\right)}{\pi\cdot k_{nex}\cdot L}$$

$$R_{eqv} := R_{cond} + R_{ti} + R_{Nex} \qquad R_{eqv2} := \frac{h_{cond}}{h_{fc}}\cdot R_{cond} + R_{ti} + R_{Nex}$$

Therefore the energy balance at the radiosity node is:

$$\frac{E_{bfin} - J_{fin}}{\left(\frac{1 - \varepsilon_{fin}}{\varepsilon_{fin} \cdot A_{fin}}\right)} + q_{ss} + q_{sl} + \frac{E_{bchamb} - J_{fin}}{\left(\frac{1}{A_{fin} \cdot F_{fin\_chamb}}\right)} = 0$$

$$J_{fin} = \frac{\left[\frac{1}{(1 - \varepsilon_{fin})} \cdot \varepsilon_{fin} \cdot \sigma T_{osun}^{4} + \frac{q_{ss}}{A_{fin}} + \frac{q_{sl}}{A_{fin}} + F_{fin\_chamb} \cdot \sigma \cdot T_{chamb}^{4}\right]}{\left[\frac{1}{(1 - \varepsilon_{fin})} \cdot \varepsilon_{fin} - F_{fin\_chamb}\right]}$$

or without solar:

$$J_{fin} = \frac{\left[\frac{1}{(1 - \varepsilon_{fin})} \cdot \varepsilon_{fin} \cdot \sigma T_{osh}^{4} + \frac{q_{sl}}{A_{fin}} + F_{fin\_chamb} \cdot \sigma \cdot T_{chamb}^{4}\right]}{\left[\frac{1}{(1 - \varepsilon_{fin})} \cdot \varepsilon_{fin} \quad F_{fin\_chamb}\right]}$$

Then an energy balance setting the energy transfer through the composite wall equal to the energy radiated from the outer surface gives:

$T_{chamb} := 120 \cdot K$ $\qquad$ $F_{fin\_chamb} := 1$

$q_{sl} := 16120 \cdot \frac{watt}{m^2} \cdot A_{fin}$ $\qquad$ $q_{ss} := 1370 \cdot \frac{watt}{m^2} \cdot d \cdot L$

$$F(T_o) := \frac{\left[\frac{1}{(1 - \varepsilon_{fin})} \cdot \varepsilon_{fin} \cdot \sigma \cdot T_o^{4} + \frac{q_{sl}}{A_{fin}} + F_{fin\_chamb} \cdot \sigma \cdot T_{chamb}^{4}\right]}{\left[\frac{-1}{(1 - \varepsilon_{fin})} \cdot \varepsilon_{fin} - F_{fin\_chamb}\right]}$$

$$Z(T_s, T_o) := \frac{T_s - T_o}{R} - \frac{\sigma \cdot T_o^{4} - F(T_o)}{1 - \varepsilon}$$

$$s \quad o \qquad R_{eqv} \qquad \left(\frac{1-\varepsilon_{fin}}{\varepsilon_{fin}\cdot A_{fin}}\right)$$

$$F_s(T_o) := \frac{\left[\frac{1}{(1-\varepsilon_{fin})}\cdot\varepsilon_{fin}\cdot\sigma\cdot T_o^{\,4} + \frac{q_{sl}}{A_{fin}} + \frac{q_{ss}}{A_{fin}} + F_{fin\_chamb}\cdot\sigma\cdot T_{chamb}^{\,4}\right]}{\left[\frac{1}{(1-\varepsilon_{fin})}\cdot\varepsilon_{fin} + F_{fin\_chamb}\right]}$$

$$Z_s(T_s, T_o) := \frac{T_s - T_o}{R_{eqv}} - \frac{\sigma\cdot T_o^{\,4} - F_s(T_o)}{\left(\frac{1-\varepsilon_{fin}}{\varepsilon_{fin}\cdot A_{fin}}\right)}$$

$$Z_{s2}(T_s, T_o) := \frac{T_s - T_o}{R_{eqv2}} - \frac{\sigma\cdot T_o^{\,4} - F_s(T_o)}{\left(\frac{1-\varepsilon_{fin}}{\varepsilon_{fin}\cdot A_{fin}}\right)}$$

$T_{sat} := 373\cdot K \qquad T_{on} := 300\cdot K$

$T_{osh} := \mathrm{root}(Z(T_{sat}, T_{on}), T_{on}) \qquad T_{osh} = 433.294\cdot K$

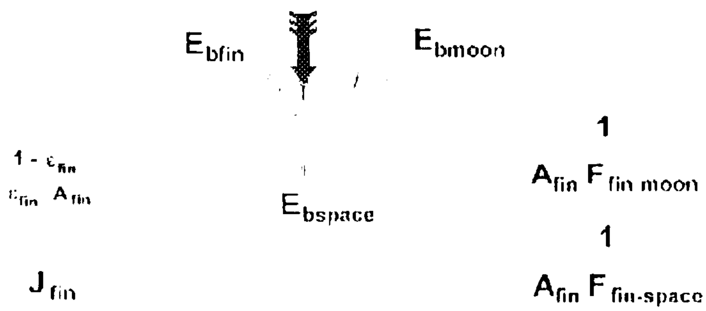
End of Section

Table of Contents

APPENDIX IV

Modeling of a Single Fin on the Lunar Surface

Single Fin Model for the Lunar Environment



Test Series 6: Single Fin - Lunar night just prior to sunrise

$k_{nex} \quad 0.00015 \cdot \frac{kW}{m \cdot K}$ $\qquad L \quad 1.13 \cdot m \quad 5.1 \cdot cm$ $\qquad h_{cond} \quad 40 \cdot \frac{kW}{m^2 \cdot K}$

$k_{ti} \quad 8.4 \cdot \frac{watt}{m \cdot K}$ $\qquad d \quad 1 \cdot in$ $\qquad \sigma \quad 5.669 \cdot 10^{-8} \cdot \frac{watt}{m^2 \cdot K^4}$

$\varepsilon_{fin} \quad 0.89$

$d_o \quad d + 2 \cdot .00056 \cdot m + 2 \cdot .0026 \cdot in$ $\qquad d_o = 0.027 \cdot m$ $\qquad h_{fc} \quad 31 \cdot \frac{watt}{m^2 \cdot K}$

$A_{fin} \quad \frac{\pi \cdot d_o \cdot L}{2}$ $\qquad A_{fin} = 0.045 \cdot m^2$

$$R_{cond} \quad \frac{1}{h_{cond} \cdot \pi \cdot d \cdot \frac{L}{2}} \qquad R_{ti} \quad \frac{\ln\left(\frac{d + .0026 \cdot 2 \cdot in}{d}\right)}{\pi \cdot k_{ti} \cdot L} \qquad R_{Nex} \quad \frac{\ln\left(\frac{d_o}{d + .0026 \cdot 2 \cdot in}\right)}{\pi \cdot k_{nex} \cdot L}$$

$$R_{eqv} \quad R_{cond} + R_{ti} + R_{Nex} \qquad R_{eqv2} \quad \frac{h_{cond}}{h_{fc}} \cdot R_{cond} + R_{ti} + R_{Nex}$$

$T_{moon} \quad 88 \cdot K$

$E_{bm} \quad \sigma \cdot T_{moon}^{\ 4}$

$$Z\left(T_{oe}, T_{sate}\right) := \frac{T_{sate} - T_{oe}}{R_{eqv}} - \frac{\sigma \cdot T_{oe}^{4} - \left|\varepsilon_{fin} \cdot \sigma \cdot T_{oe}^{4} + \left(1 - \varepsilon_{fin}\right) \cdot \left(\frac{E_{bm}}{2}\right)\right|}{\left(\frac{1 - \varepsilon_{fin}}{\varepsilon_{fin} \cdot A_{fin}}\right)}$$

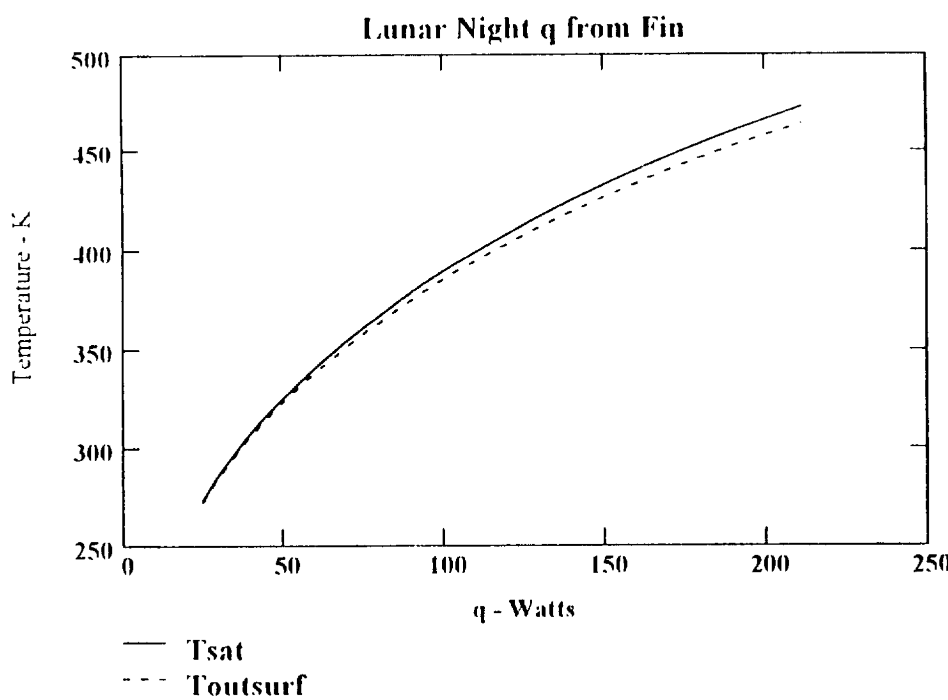
$$j := 0 .. 20 \qquad T_{s_j} := (273 + 10 \cdot j) \cdot K$$

$$T_{on} := 300 \cdot K$$

$$T_{o2_j} := \mathrm{root}\left(Z\left(T_{on}, T_{s_j}\right), T_{on}\right)$$

$$q_{1_j} := \frac{T_{s_j} - T_{o2_j}}{R_{eqv}} \cdot 2$$

Single Fin, Just Prior to Sunrise



Test Series 1 - 5

To model the different test series, set the model input parameters as shown - results plotted below.

Test Series	Tmoon	Theta
1	388	90
2	352	45
3	288	60
4	238	45
5	88	0

$$T_{moon} \quad 388 \cdot K \qquad\qquad 0 \quad \frac{90 \cdot \pi}{180}$$

$$q' \quad 442 \cdot \frac{BTU}{hr \cdot ft^2}$$

$$E_{bm} \quad \sigma \cdot T_{moon}^4 \qquad\qquad q_{sun} \quad q' \cdot d \cdot L \cdot \cos(0)$$

$$Eq \quad \left(\frac{E_{bm}}{2} + \frac{q_{sun}}{A_{fin}}\right)$$

$$Z_1(T_{oe}, T_{sate}) \quad \frac{T_{sate} \quad T_{oe}}{R_{eqv}} \quad \sigma \cdot T_{oe}^4 \quad \frac{\left|\varepsilon_{fin} \cdot \sigma \cdot T_{oe}^4 + (1 \quad \varepsilon_{fin}) \cdot Eq\right|}{\left(\frac{1 \quad \varepsilon_{fin}}{\varepsilon_{fin} \cdot A_{fin}}\right)}$$

$$Z_2(T_{oe}, T_{sate}) \quad \frac{T_{sate} \quad T_{oe}}{R_{eqv2}} \quad \sigma \cdot T_{oe}^4 \quad \frac{\left|\varepsilon_{fin} \cdot \sigma \cdot T_{oe}^4 + (1 \quad \varepsilon_{fin}) \cdot Eq\right|}{\left(\frac{1 \quad \varepsilon_{fin}}{\varepsilon_{fin} \cdot A_{fin}}\right)}$$

$$T_{o2n} \quad 300 \cdot K$$

$$j := 0 .. 10 \qquad T_{sat_j} := 273 \cdot K + j \cdot 10 \cdot K$$

$$T_{osun_j} := root\left(Z_1\left(T_{o2n}, T_{sat_j}\right), T_{o2n}\right)$$

$$T_{oshade_j} := root\left(Z\left(T_{o2n}, T_{sat_j}\right), T_{o2n}\right)$$

$$T_{osunfc_j} := root\left(Z_2\left(T_{o2n}, T_{sat_j}\right), T_{o2n}\right)$$

$$q_{sun_j} := \frac{T_{sat_j} - T_{osun_j}}{R_{eqv}}$$

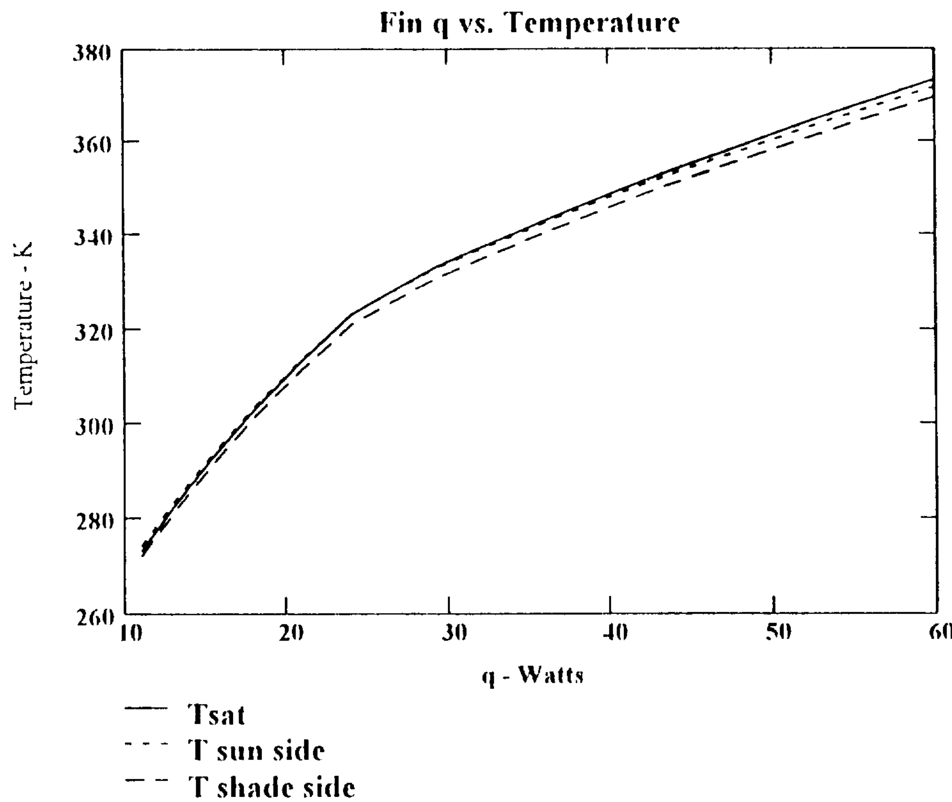
$$q_{shade_j} := \frac{T_{sat_j} - T_{oshade_j}}{R_{eqv}}$$

$$q_{sunfc_j} := \frac{T_{sat_j} - T_{osun_j}}{R_{eqv2}}$$

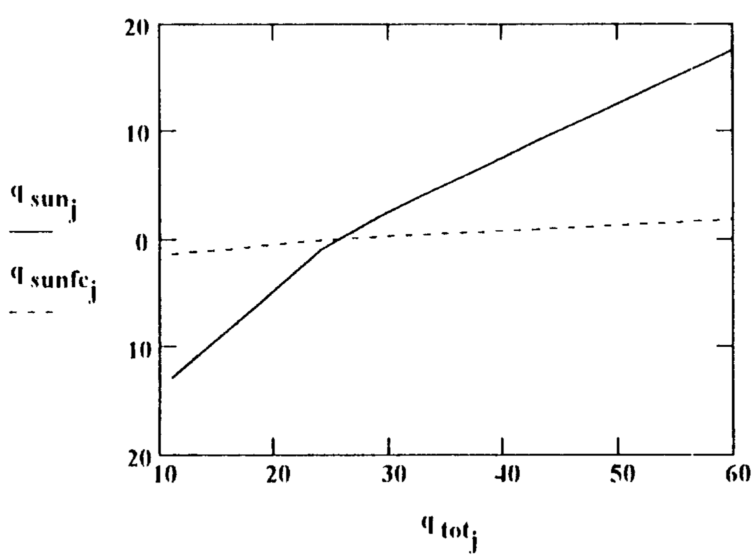
$$q_{tt}(kk) := \left| \begin{array}{ll} \left(q_{sun_{kk}} + q_{shade_{kk}}\right) & \text{if } T_{sat_{kk}} > T_{osun_{kk}} \\ \left(q_{sunfc_{kk}} + q_{shade_{kk}}\right) & \text{otherwise} \end{array} \right.$$

$$q_{tot_j} := q_{tt}(j)$$

Single Fin on the Lunar Surface



Free Convection and Condensation Heat Transfer



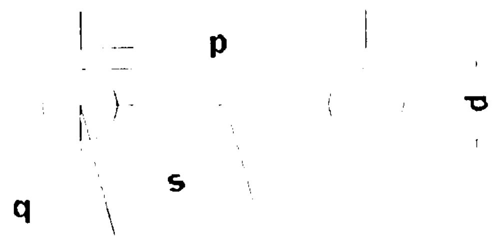
End of Section

Table of Contents

APPENDIX V

The Shape Factor Between Adjacent Tubes in a Linear Array

To determine the shape factor between adjacent tubes in a linear array we will treat them as infinitely long. Then the shape factor is simply the difference in the sum of the crossed strings minus the uncrossed strings divided by the base length. The sketch below shows two tubes of diameter, d, in such an array separated by a pitch distance p.



Then $F_{1\text{-}2}$ is:

$$F_{1\_2} = \frac{4\cdot(s+q) - 4\cdot\frac{p}{2}}{\pi\cdot\frac{d}{2}} = \frac{8}{\pi}\cdot\left(\frac{s}{d} + \frac{q}{d} - \frac{p}{2\cdot d}\right)$$

but $$s = \sqrt{\frac{p^2}{4} - \frac{d^2}{4}}$$ or $$\frac{s}{d} = \frac{\sqrt{\left(\frac{p}{d}\right)^2 - 1}}{2}$$

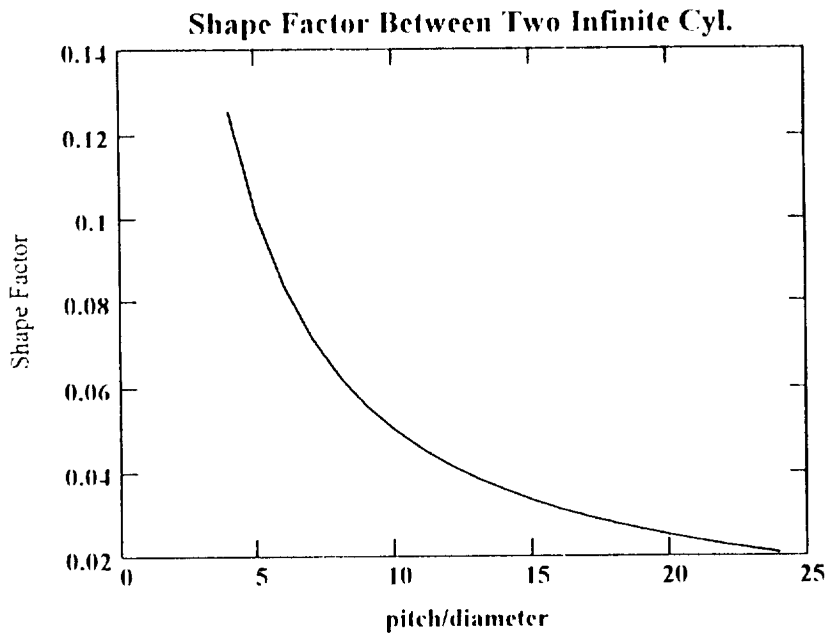
$$\alpha = \mathrm{asin}\left(\frac{d}{p}\right)$$ and $$\frac{q}{d} = \mathrm{asin}\left(\frac{d}{p}\right)$$

therefore

$$F_{1\_2} = \frac{4}{\pi}\cdot\left(\sqrt{\left(\frac{p}{d}\right)^2 - 1} + \mathrm{asin}\left(\frac{d}{p}\right) - \frac{p}{d}\right)$$

$$F_{cc}(pd) \quad \left| \sqrt{(pd)^2 \quad 1} + \operatorname{asin}\left(\frac{1}{pd}\right) \quad pd \right|$$

$$j \quad 0..20 \qquad pp_j \quad 4 + j \qquad F_{12_j} \quad F_{cc}(pp_j)$$

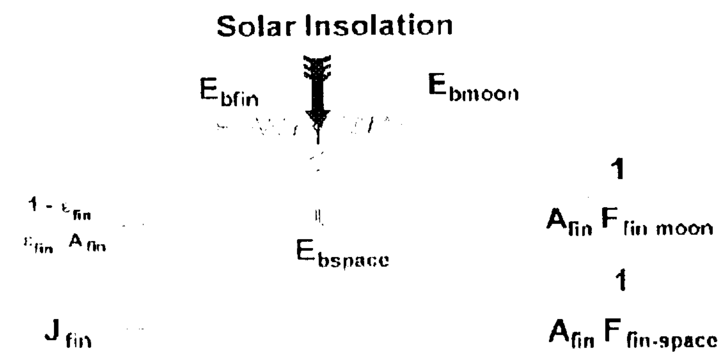


End of Section

Table of Contents

APPENDIX VI

Pitch Optimization of Linear Tube Arrays



$k_{nex} \quad 0.00012 \cdot \frac{kW}{m \cdot K}$ $\qquad L \quad 1 \cdot m \quad 5.1 \cdot cm$ $\qquad h_{cond} \quad 60 \cdot \frac{kW}{m^2 \cdot K}$

$k_{ti} \quad 8.4 \cdot \frac{watt}{m \cdot K}$ $\qquad d \quad 1 \cdot in$ $\qquad \sigma \quad 5.669 \cdot 10^{8} \cdot \frac{watt}{m^2 \cdot K^4}$

$\varepsilon_{fin} \quad 0.89$

$d_o \quad d + 2 \cdot .00056 \cdot m + 2 \cdot .0026 \cdot in$ $\qquad d_o = 0.027 \cdot m$ $\qquad h_{fc} \quad 31 \cdot \frac{watt}{m^2 \cdot K}$

$A_{fin} \quad \frac{\pi \cdot d_o \cdot L}{2}$ $\qquad A_{fin} = 0.04 \cdot m^2$

$R_{cond} \quad \frac{1}{h_{cond} \cdot \pi \cdot d \cdot \frac{L}{2}}$ $\qquad R_{ti} \quad \frac{\ln\left(\frac{d + .0026 \cdot 2 \cdot in}{d}\right)}{\pi \cdot k_{ti} \cdot L}$ $\qquad R_{Nex} \quad \frac{\ln\left(\frac{d_o}{d + .0026 \cdot 2 \cdot in}\right)}{\pi \cdot k_{nex} \cdot L}$

$R_{eqv} \quad R_{cond} + R_{ti} + R_{Nex}$ $\qquad R_{eqv2} \quad \frac{h_{cond}}{h_{fc}} \cdot R_{cond} + R_{ti} + R_{Nex}$

$$\frac{E_{bfin} - J_{fin}}{\left(\frac{1-\varepsilon_{fin}}{\varepsilon_{fin} \cdot A_{fin}}\right)} + \frac{0 - J_{fin}}{\left(\frac{1}{A_{fin} \cdot F_{f\_s}}\right)} + \frac{\sigma T_m^{\,4} - J_{fin}}{\left(\frac{1}{A_{fin} \cdot F_{f\_m}}\right)} + q_{sun} \cdot A_{fin} = 0$$

$$J_{fin} = \frac{\frac{\varepsilon_{fin} \cdot \sigma \cdot T_{oe}^{\,4}}{1-\varepsilon_{fin}} + F_{f\_m} \cdot (\sigma \cdot T_m)^4 + q_{sun}}{\left(\frac{\varepsilon_{fin}}{1-\varepsilon_{fin}} + 2 \cdot F_{f\_m}\right)}$$

but $F_{f\_s} = F_{f\_m}$ $\qquad q' := 442 \cdot \frac{BTU}{hr \cdot ft^2}$ $\qquad \theta := \frac{45}{180} \cdot \pi$

$T_m := 80 \cdot K \qquad q_{sun} := q' \cdot \cos(\theta)$

$$e_f := \frac{\varepsilon_{fin}}{1-\varepsilon_{fin}}$$

$$G(T_{oe}, F_{f\_m}) := e_f \cdot \sigma \cdot T_{oe}^{\,4} + \sigma \cdot F_{f\_m} \cdot T_m^{\,4} + q_{sun}$$

$$Z_1(T_{oe}, T_{sate}, F_{f\_m}) := \frac{T_{sate} - T_{oe}}{R_{eqv}} - \frac{\sigma \cdot T_{oe}^{\,4} - \left|\frac{G(T_{oe}, F_{f\_m})}{(e_f + 2 \cdot F_{f\_m})}\right|}{\left(\frac{1}{e_f \cdot A_{fin}}\right)}$$

$$Z(T_{oe}, T_{sate}, F_{f\_m}) := \frac{T_{sate} - T_{oe}}{R_{eqv}} - \frac{\sigma \cdot T_{oe}^{\,4} - \left|\frac{e_f \cdot \sigma \cdot T_{oe}^{\,4} + \sigma \cdot F_{f\_m} \cdot T_m^{\,4}}{(e_f + 2 \cdot F_{f\_m})}\right|}{\left(\frac{1}{e_f \cdot A_{fin}}\right)}$$

$$Z_2(T_{oe}, T_{sate}, F_{f\_m}) \quad \frac{T_{sate} \quad T_{oe}}{R_{eqv2}} \quad \frac{\sigma \cdot T_{oe}^4 \cdot \left| \frac{G(T_{oe}, F_{f\_m})}{(e_f + 2 \cdot F_{f\_m})} \right|}{\left(\frac{1}{e_f A_{fin}} \right)}$$

$T_{o2n} \quad 300 \cdot K$

$j \quad 0 .. 40$ $\qquad pd_j \quad 1 + j \cdot .5$ $\qquad T_{sat} \quad 373 \cdot K$

$$F_{cc}(pd) \quad \left| \sqrt{(pd)^2 \quad 1} + \mathrm{asin}\left(\frac{1}{pd}\right) \quad pd \right|$$

$F_{fm_j} \quad 0.5 \quad F_{cc}(pd_j)$

$T_{osun_j} \quad \mathrm{root}\left(Z_1(T_{o2n}, T_{sat}, F_{fm_j}), T_{o2n}\right)$

$T_{oshade_j} \quad \mathrm{root}\left(Z(T_{o2n}, T_{sat}, F_{fm_j}), T_{o2n}\right)$

$T_{osunfc_j} \quad \mathrm{root}\left(Z_2(T_{o2n}, T_{sat}, F_{fm_j}), T_{o2n}\right)$

$$q_{sun_j} \quad \frac{T_{sat} \quad T_{osun_j}}{R_{eqv}}$$

$$q_{shade_j} \quad \frac{T_{sat} \quad T_{oshade_j}}{R_{eqv}}$$

$$q_{sunfc_j} := \frac{T_{sat} - T_{osun_j}}{R_{eqv2}}$$

$$q_{tt}(kk) := \begin{cases} \left(q_{sun_{kk}} + q_{shade_{kk}}\right) & \text{if } T_{sat} > T_{osun_{kk}} \\ \left(q_{sunfc_{kk}} + q_{shade_{kk}}\right) & \text{otherwise} \end{cases}$$

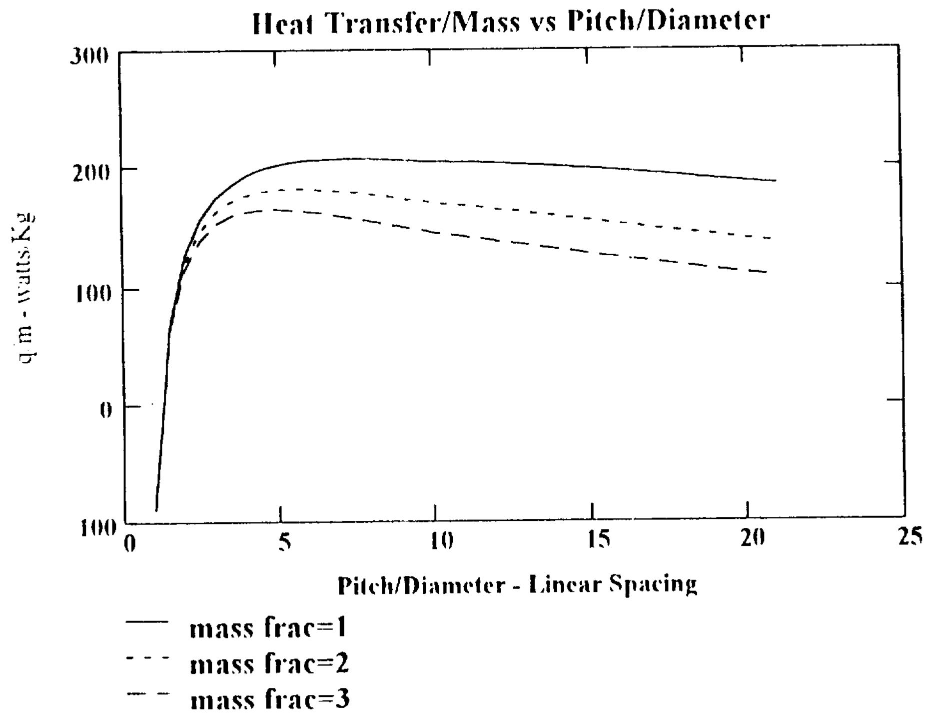
$$q_{tot1_j} := \frac{q_{tt}(j)}{131.7 \cdot gm + pd_j \cdot d \cdot 131.7 \cdot \frac{gm}{m}}$$

A mass fraction of 1, i.e. each meter of spacing adds a mass equal to the reflux tube mass.

$$q_{tot2_j} := \frac{q_{tt}(j)}{131.7 \cdot gm + pd_j \cdot d \cdot 131.7 \cdot \frac{gm}{m} \cdot 2}$$

$$q_{tot3_j} := \frac{q_{tt}(j)}{131.7 \cdot gm + pd_j \cdot d \cdot 131.7 \cdot \frac{gm}{m} \cdot 3}$$

For a Saturation Temperature of 373oK, Lunar Temperature of 80oK, and a Solar Angle of 45o



This figure indicates that a resonable design pitch/diameter ratio would be about 5. A higher saturation temperature would perhaps push this ratio down to 4 to 4.5 as would more massive structure and piping arrangements. Even in these cases, the use of a ratio of 5 would not be costly since the function is not as sensitive to the right of the peak as it is to the left.

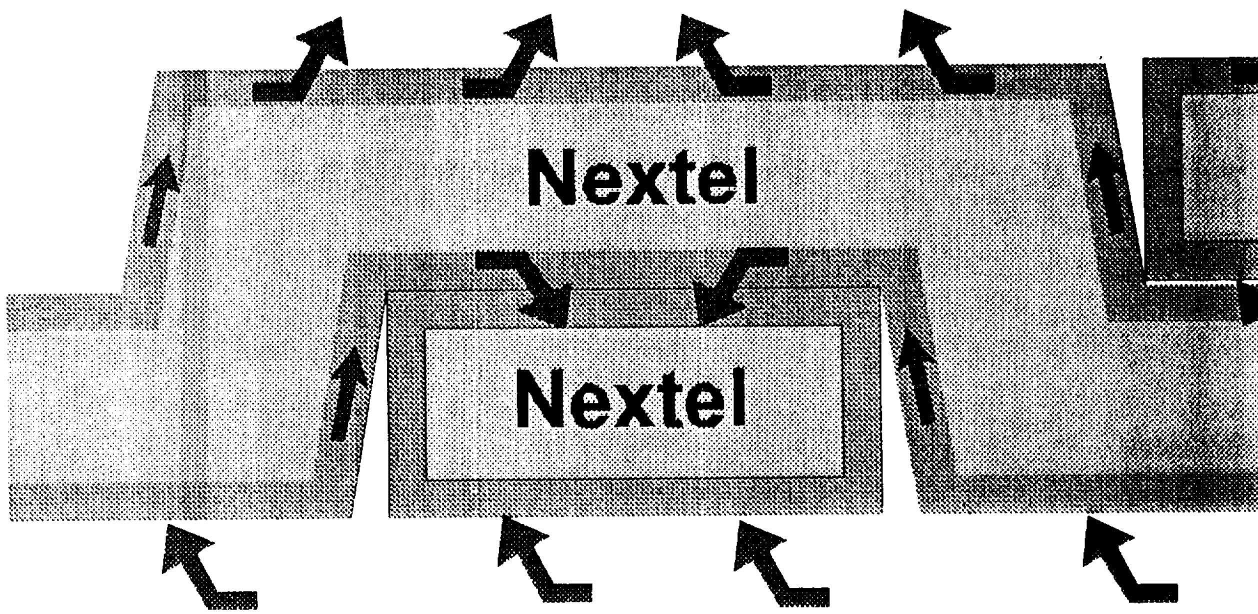
End of Section

Table of Contents

APPENDIX VII

Equivalent Thermal Conductivity of Concentric Layers of Copper and Nextel

The model below shows copper wires covering Nextel threads in a composite fabric weave. The purpose of the fine copper wires is to increase the thermal conductivity of the fabric while minimizing the loss of strength provided by the Nextel fibers. The copper layer is shown around the Nextel fibers so that the weave of the fabric allows the heat to be conducted along the copper wires or strands and thereby carries the heat from the inside surface to the cross layers of fibers and from these cross layers to the outside surface as shown in the figure below. Copper is used instead of gold to avoid an undesireable decrease in outside surface emmisivity.



$$k_{Nex} \cdot w^2 \cdot \frac{\Delta T}{t} + k_{cu} \cdot w \cdot \frac{t}{2} \cdot 2 \cdot x \cdot \frac{\Delta T}{w} = k_e \cdot w^2 \cdot \frac{\Delta T}{t}$$

$$k_e = \frac{\left(k_{Nex} \cdot \frac{w^2}{t} + k_{cu} \cdot t \cdot x\right)}{\left(w^2\right)} \cdot t = k_{Nex} + k_{cu} \cdot \left(\frac{t}{w}\right)^2 \cdot x$$

$$\frac{k_e}{k_{Nex}} = 1 + \frac{k_{cu}}{k_{Nex}} \cdot \left(\frac{t}{w}\right)^2 \cdot x$$

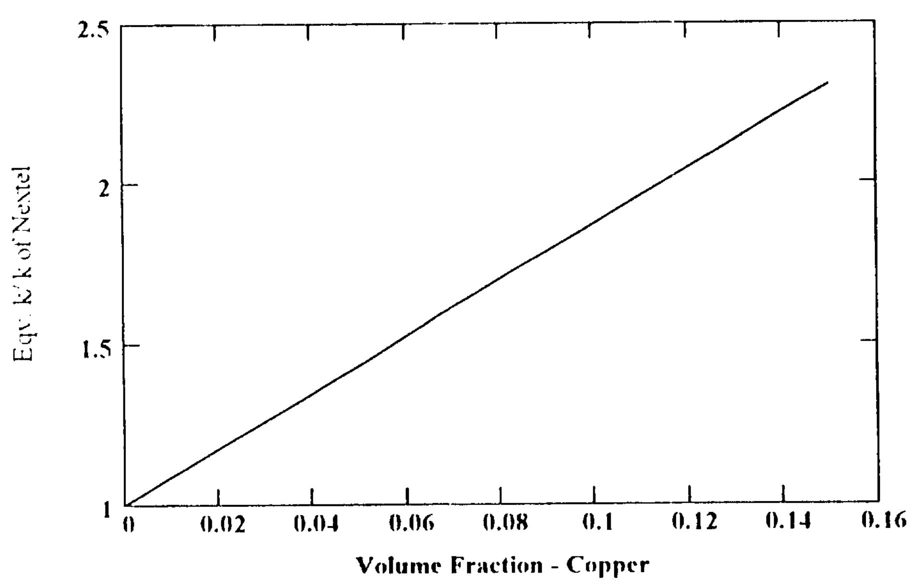
$k_{Nex} \quad 0.00012 \cdot \frac{kW}{m \cdot K} \qquad t \quad .0026 \cdot in \qquad w \quad .05 \cdot in$

$k_{cu} \quad 0.386 \cdot \frac{kW}{m \cdot K}$

$ke(xx) \quad 1 + \frac{k_{cu}}{k_{Nex}} \cdot \left(\frac{t}{w}\right)^2 \cdot xx \qquad \frac{k_{cu}}{k_{Nex}} = 3.217 \cdot 10^3$

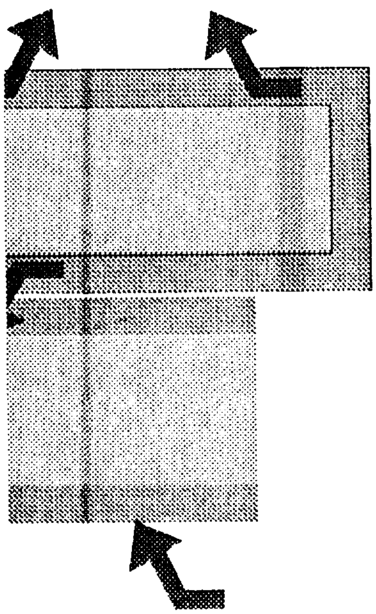
$i \quad 0..20 \qquad x_i \quad 0 + i \cdot .0075$

$krat_i = ke(x_i) \qquad \left(\frac{t}{w}\right)^2 = 2.704 \cdot 10^{-3}$



End of Section

Table of Contents



APPENDIX VIII

Estimation of the Free Convection Heat Transfer Coefficient Inside the Vertical Tube

Estimation of the free convection heat transfer coefficient between the inside tube wall and the saturated steam from the evaporator. This condition exists when the radiant boundary conditions produce an external surface temperature above the saturated steam temperature.

The heat transfer coefficient will be taken as that for a vertical flat plate even though the tube will not behave in this fashion. This will give a larger value which in our case will produce a higher heat transfer to the tube and thereby give conservative results.

Properties of steam at 573°K.

$\mu \quad 19.7 \cdot 10^{6} \cdot Pa \cdot sec \qquad Pr \quad 1.443$

$\rho \quad 37.4 \cdot \frac{kg}{m^3}$

$\beta \quad \frac{1}{573 \cdot K}$

$k \quad .0641 \cdot \frac{watt}{m \cdot K}$

$\Delta T \quad 5 \cdot K \qquad L \quad 1.13 \cdot m \qquad g \quad 9.8 \cdot \frac{m}{sec^2}$

$$Gr \quad \frac{g \cdot \beta \cdot \Delta T \cdot L^3}{\left(\frac{\mu}{\rho}\right)^2} \qquad Gr = 4.447 \cdot 10^{11}$$

therefore, turbulent

$$Nu \quad \frac{0.15 \cdot (Gr \cdot Pr)^{\frac{1}{3}}}{\left[1 + \left(\frac{0.492}{Pr}\right)^{\frac{9}{16}}\right]^{\frac{16}{27}}} \qquad Nu = 999.452$$

$$h \quad Nu \cdot \frac{k}{L} \qquad h = 56.695 \cdot \frac{watt}{m^2 \cdot K} \qquad h = 9.984 \cdot \frac{BTU}{hr \cdot ft^2 \cdot R}$$

Use h = 57 watt/m²°K for the JSC test and:

$$g \quad \frac{g}{6}$$

$$Gr \quad \frac{g \cdot \beta \cdot \Delta T \cdot L^3}{\left(\frac{\mu}{\rho}\right)^2} \qquad Gr = 7.412 \cdot 10^{10}$$

still turbulent

$$Nu \quad \frac{0.15 \cdot (Gr \cdot Pr)^{\frac{1}{3}}}{\left[1 + \left(\frac{0.492}{Pr}\right)^{\frac{9}{16}}\right]^{\frac{16}{27}}} \qquad Nu = 550.02$$

$$h \quad Nu \cdot \frac{k}{L} \qquad h = 31.2 \cdot \frac{watt}{m^2 \cdot K} \qquad h = 5.495 \cdot \frac{BTU}{hr \cdot ft^2 \cdot R}$$

h = 31 watt/m²°K for the lunar surface.

End of Section

[Table of Contents]

[Next Section]