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Optimization of a Low-Gravity Two-Phase System for Lunar Heat Rejection

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A mathematical model of a low-gravity reflux boiler system for lunar-surface heat rejection, including the performance of a single tube and a linear array of tubes, is described. Model verification is demonstrated through correlation with NASA vacuum chamber test data. Performance comparisons are made between three manufactured configurations of the tubes for similar environmental conditions and with the tube dimensions normalized with respect to one another. Finally, the optimization of a linear array of composite tubes for operation in a worst-case lunar surface thermal environment is presented. A pitch-to-diameter ratio of between 4 and 5 is recommended because that choice provides over 95% of the maximum heat transfer with a more logistically feasible array.

Nomenclature

A	=	area of a radiative body, m^2
E_b	=	blackbody emittance, W/m^2
F	=	view factor
g	=	local gravitational constant, m/s^2
$\bar{h}$	=	average heat-transfer coefficient, W/m^2-K
h_{fg}	=	difference between saturated vapor and saturated liquid enthalpies, kJ/kg-K
J_{fin}	=	radiosity of the reflux boiler, assumed to be a fin, W/m^2
k	=	thermal conductivity of condensate, W/m-K
$\dot{m}$	=	mass flow rate of condensate, kg/s
P_{cr}	=	critical buckling load, lbf
Q	=	total heat transfer, W
q	=	heat flux, W/m^2
T_{sat}	=	saturation temperature of condensate, K
T_{wall}	=	temperature of reflux boiler wall, K
u	=	vertical velocity of the condensate, m/s
w	=	mass flow rate (per unit depth into plate) of condensate, kg/m-s
δ	=	condensate layer thickness, m
ε	=	emmisivity of the material
μ	=	liquid viscosity, kg/m-s
ρ	=	density of condensate, kg/m^3
ρ_v	=	density of saturated vapor, kg/m^3
σ	=	Stefan–Boltzmann constant equal to 5.67×10^{-8}, W/m^2-K^4

Introduction

THE need for a lunar base from which to begin man's exploration of the universe has been debated for decades, with considerably increased discussion since the last Apollo lunar landing in late 1972 (Ref. 1). Because the moon is Earth's closest neighbor, it is obviously the easiest and most practical goal for initial planetary colonization. Another factor affecting the prospect of lunar colonization is the new International Space Station.[2] This Earth-orbiting satellite will serve not only as a permanent observatory and facility for scientific research, but the moon could also be a staging base for the development of a planetary base on the moon or Mars. Recent discoveries of ice trapped beneath the lunar surface have renewed interest in a lunar base.

A lunar base must be designed to accommodate the harsh indigenous environment of the lunar surface. With no atmosphere to contain its heat or to repel the sun's energy, thermal environmental control is particularly difficult. Thermal loads from the waste heat of power generation systems such as fuel cells and photoelectric generators, from the heat generated in electronics packages within computers and communications systems, and from crew metabolic loads must be dissipated even under worst-case conditions of system operation and environmental thermal conditions. The moon's surface temperature varies from 88 K at the termination of lunar night to a maximum of approximately 388 K at the peak of a typical lunar day. With radiation as the sole means of heat rejection, special requirements and design considerations are involved for a lunar-surface thermal control system.

In addition to providing adequate thermal performance, any lunar-surface heat-rejection device must be readily transportable from earth, easy to assemble, and simple in its operation. The Ultra-Light Fabric Reflux Tube (UFRT),[3] with its self-contained working fluid, is a good example of such a device. This 2.54-cm-diam, 1-m-long tube facilitates easy and rapid assembly by a crew member in a space suit. It can be attached and physically clamped to a "thermal bus" for heat rejection from a habitability or service module. Because weight is at a premium because it directly affects launch cost and greatly influences logistics for any space mission, the UFRT was conceived for ease of transportation, ready assembly, and operational redundancy. The moon is constantly being bombarded by meteorites and meteoroids from space; therefore, designers favor an array of tubes, each with its own self-contained fluid. This configuration would provide minimum exposed area, as well as operational independence of each tube with respect to the others, minimizing the probability of damage and/or total failure of the thermal control system. A typical array of these tubes is shown in Fig. 1.

The tubes must have a high thermal emissivity with low solar absorptivity, and the thermodynamic properties of the working fluid must be consistent with the temperature level required for heat rejection. By vacuum filling the tubes, the tubes can be made to operate at lower pressure, lowering the boiling point of the fluid to provide the proper temperature difference between the heat source and the heat sink. By utilizing the latent heat associated with its phase change, the amount of working fluid needed for heat rejection with such a device is minimized.

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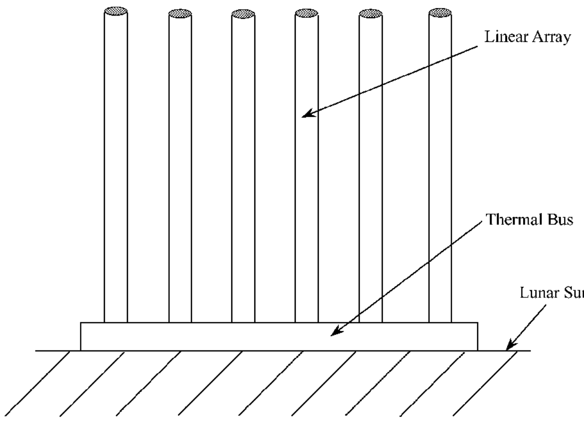


Fig. 1 Linear array of UFRTs on the lunar surface.

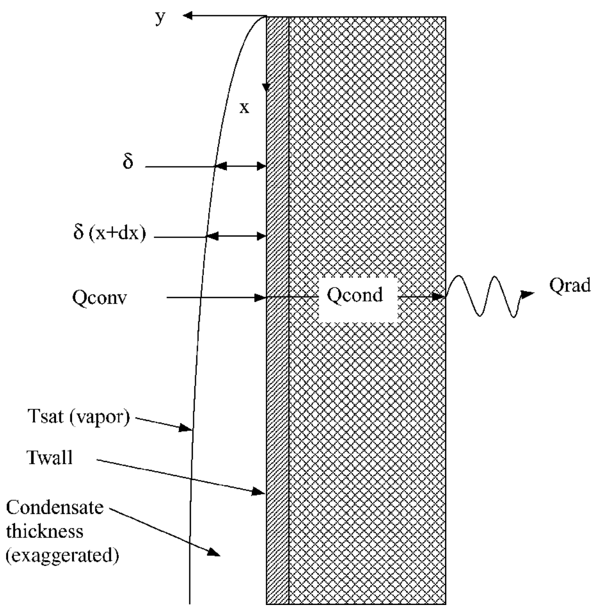


Fig. 2 Combined heat-transfer process for the vertical tube.

This paper describes an analysis developed for a single tube operating under various conditions on the lunar surface, with subsequent extension of the analysis to a linear array of tubes. Analysis results for one particular configuration, the UFRT, are then verified through performance comparisons with NASA thermal vacuum test data with the lunar thermal environment simulated by solar lamps. After verification of the model, the model is then used to compare the performance of three different types of tubes operating under similar conditions. Following this, an optimization study is presented, first for a single tube and then for a linear array of tubes on the lunar surface.

Mathematical Model

Classical thermodynamic, fluid dynamic, and heat-transfer methods were applied to a vertical cylinder containing water undergoing saturated nucleate pool boiling from a horizontal flat surface at the bottom of the tube.[4] Saturated steam rises from the evaporator section, condensing on the inner wall and forming a thin condensate layer. The condensate travels down the tube surface, returning to the evaporator. Simultaneously along the wall, heat energy is then conducted through the inner wall to the outside surface of the tube, where the energy is radiated from the outer surface to the moon and space, as shown in Fig. 2.

For the analysis the following assumptions were made: 1) negligible pressure differential along the vertical axis of the tube (from earlier tests); 2) laminar flow of the rising vapor in the tube; 3) the fluid is condensing on a vertical flat plate with laminar film-wise condensation; 4) there is no interface shear between the up-flowing vapor and the downflowing liquid film at the boundary-layer interface (assumptions 2, 3, and 4 are later verified in the analysis); and 5) a linear temperature distribution exists in the condensate layer between the wall and the vapor interface. Given these assumptions, the weight of an infinitesimal fluid element of thickness $\mathrm{d}x$ between y and δ is balanced by the viscous shear force at y and the buoyancy force caused by the displaced vapor, as shown in Eq. (1):

$$\rho g(\delta - y)\,\mathrm{d}x = \mu \frac{\mathrm{d}u}{\mathrm{d}y}\,\mathrm{d}x + \rho_v g(\delta - y)\,\mathrm{d}x \tag{1}$$

Integrating and using the boundary condition that $u = 0$ at $y = 0$, the velocity of the liquid in the x direction (vertically down) is given by Eq. (2):

$$u = [(\rho - \rho_v)g/\mu]\left(\delta y - \tfrac{1}{2}y^2\right) \tag{2}$$

The mass flow rate $\dot{m}$ of condensate through any x position of the film is obtained when the preceding equation is integrated along $\mathrm{d}y$ from 0 to δ for unit depth into the plate, as shown in Eq. (3):

$$w = \rho(\rho - \rho_v)g\delta^3\big/3\mu \tag{3}$$

with the heat transfer at the wall area expressed by Eq. (4):

$$q_x = -k\,\mathrm{d}x \frac{\delta T}{\delta y}\bigg]_{y=0} = k\,\mathrm{d}x \frac{T_{\mathrm{sat}} - T_w}{\delta} \tag{4}$$

As the flow proceeds from x to $x + \mathrm{d}x$, the film thickness grows from δ to $\delta + \mathrm{d}\delta$ as a result of the influx of additional condensate. Writing an expression for the amount of condensate added between x and $x + \mathrm{d}x$ and integrating with the boundary condition $\delta = 0$ at $x = 0$, the thickness of the condensate layer can be shown to be given by Eq. (5):

$$\delta = \left[\frac{4\mu k x(T_{\mathrm{sat}} - T_w)}{g h_{\mathrm{fg}}\rho(\rho - \rho_v)}\right]^{\frac{1}{4}} \tag{5}$$

The thickness of the condensate layer is a function of the independent variable delta-temperature ΔT, which is the difference between the saturation temperature T_{sat} (228.96°F for water at an operating pressure of 20 psia) and T_w (unknown wall temperature).

The heat-transfer coefficient is next determined by equating the vapor-to-condensate-layer convection heat transfer to the heat conducted through the layer to the tube wall and integrating over the length of the tube. In this manner the average heat-transfer coefficient can be expressed, assuming a linear temperature distribution through the condensate layer, as shown in Eq. (6):

$$\bar{h} = 0.943\left[\frac{\rho(\rho - \rho_v)g h_{\mathrm{fg}} k_f^3}{L\mu_f(T_{\mathrm{sat}} - T_w)}\right]^{\frac{1}{4}} \tag{6}$$

A special condition was analyzed for the case when, as a result of an adverse planetary thermal environment, the net heat transferred from the tube can be negative. In this situation an estimate of the free convection heat-transfer coefficient between the inside of the tube wall and the saturated steam from the evaporator was needed for the case when the radiant boundary conditions produce an external surface temperature above the saturated steam temperature. A further refinement of the model might be to include the effect of radiation inside the tube, but that effect was neglected for this analysis. The negative heat-transfer condition, using free convection, was included in the thermal model because certain environmental conditions produce this effect. The calculation for the free-convection coefficient for this case resulted in a value of 31 W/m^2-K, which is more than three orders of magnitude lower than the condensing heat-transfer coefficient of 40,000 W/m^2-K inside the tube. Obviously the "dark" side of the tube will never experience this condition because T_{sat} will always be above the outer surface temperature. At certain times of the lunar day, however, condensation may cease on the "hot" side of the tube, and heat can actually be transferred into the tube by free convection of the vapor, whereas on the "cold" side condensation will continue to occur.

The internal analysis was linked to a radiative model in which thermal energy is dissipated diffusely to the vacuum environment by radiation to the lunar surface and to deep space, taking into account solar incident heat flux upon the tube. The model used to simulate the radiative heat transfer is radially one-dimensional because there

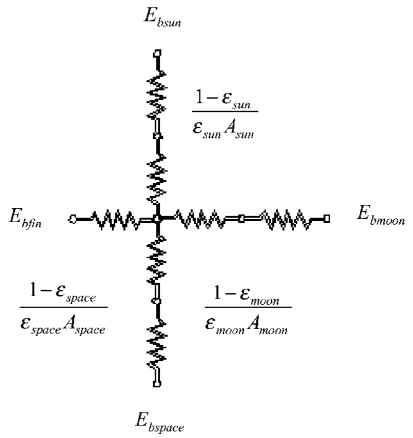


Fig. 3a Standard circuit for model.

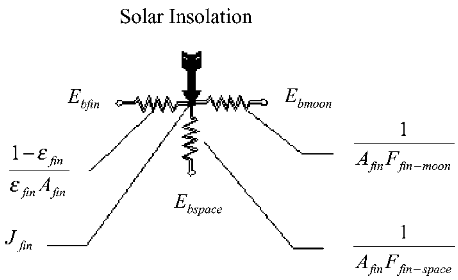


Fig. 3b Reduced circuit for the mathematical model.

is insufficient axial variation in tube temperature to warrant breaking the tube into sections and utilizing a finite difference analysis. All radiation is absorbed or emitted, with no transmittance, and the radiation is assumed to be diffuse. The tube is segmented into two sides, a shaded side and a nonshaded, or sun side, in lieu of angular numerical integration in a cylindrical coordinate system. This technique was found to work quite well in simulating actual test data.

Assuming the radiant energy to be diffuse, the standard electric circuit analogy for the radiant energy exchange between the fin (or tube), the lunar surface, the sun and space can be represented by Fig. 3a, where E_{bfin} is the blackbody emittance of the tube at its outer surface temperature.

Because the areas of the sun, deep space, and the lunar surface are much greater than the area of the fin, their associated surface resistance can effectively be neglected, resulting in the simplified electrical circuit of Fig. 3b.

The resulting energy balance for the "sun side" of the tube is given by Eq. (7):

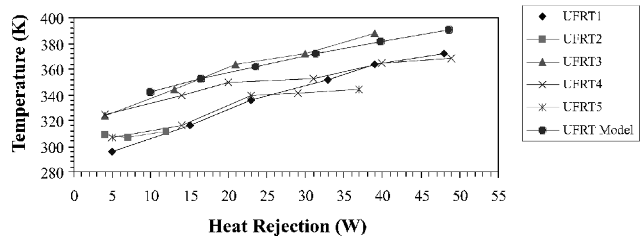
$$\frac{E_{\text{bfin}} - J_{\text{fin}}}{(1-\varepsilon_{\text{fin}})/\varepsilon_{\text{fin}}A_{\text{fin}}} + \frac{E_{\text{bspace}} - J_{\text{fin}}}{1/A_{\text{fin}}F_{\text{fin-space}}} + \frac{E_{\text{bmoon}} - J_{\text{fin}}}{1/A_{\text{fin}}F_{\text{fin-moon}}} + q_{\text{sun}} = 0 \tag{7}$$

In contrast, the "shade-side" heat flow equation is given by Eq. (8):

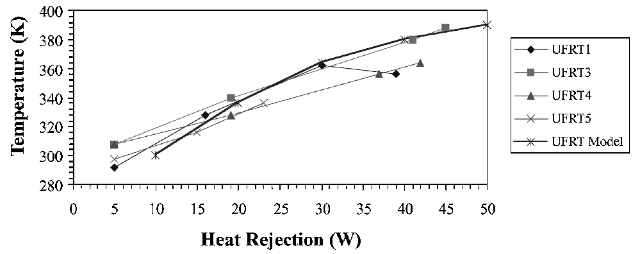
$$\frac{E_{\text{bfin}} - J_{\text{fin}}}{(1-\varepsilon_{\text{fin}})/\varepsilon_{\text{fin}}A_{\text{fin}}} + \frac{E_{\text{bspace}} - J_{\text{fin}}}{1/A_{\text{fin}}F_{\text{fin-space}}} + \frac{E_{\text{bmoon}} - J_{\text{fin}}}{1/A_{\text{fin}}F_{\text{fin-moon}}} = 0 \tag{8}$$

In both of these equations, E_{bspace} is taken as zero, E_{bfin} is equal to σT_{fin} (Ref. 4), $E_{\text{bmoon}} = \sigma T_{\text{moon}}$ (Ref. 4), and ε_{fin} is the emissivity of the tube at its outer surface. The shape factors are 0.5 for a single fin to space and 0.5 for a fin to the lunar surface, assuming the lunar surface is an infinite plane, with deep space being infinite.

For an array of tubes (assumed here to be a linear array with the plane of the tubes perpendicular to the plane of the solar vector), the shape factor for two tubes that "see" each other was calculated using Hottel's "crossed-strings" method. Because all tubes or fins in the array are assumed to be operating at the same temperature, the heat flow circuit for each tube will be the same, except for the effect that adjacent tubes have in blocking radiation from a central tube. This effect of blocked radiation is accounted for in the model.



a) Vacuum test series 1 with UFRT model; UFRT surface temperatures, solar angle 90 deg



b) Vacuum test series 4 with UFRT model; UFRT surface temperatures, solar angle 45 deg

Fig. 4 Comparison of analytical and test performance data.

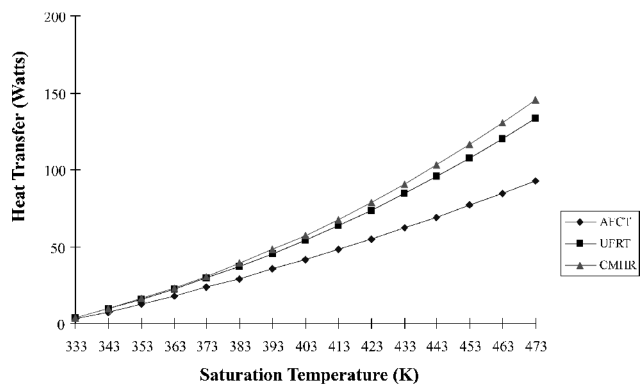


Fig. 5 Performance of three manufactured tube configurations, with normalized dimensions.

Based on the preceding analytical considerations, an integrated system thermal model was developed and verified through correlation with NASA vacuum chamber test data.[4] Two examples of the comparison of analytical and test performance are shown in Figs. 4a and 4b. Figure 4a shows UFRT surface temperatures as a function of measured heat flux at the evaporator for a test run simulating lunar noon (a solar angle of 90 deg and a lunar surface temperature of 388 K) for five tubes in a linear array (UFRT 3 is the middle tube in the array), with the predicted (model) performance for a single interior superimposed on the test data. Figure 4b shows the results of another test run with a four-tube array, corresponding to a solar angle of 45 deg and a lunar-surface temperature of 238 K. In both cases the model predicts temperatures that corresponds well to vacuum chamber test data.

The model was then used to predict the performance of three different manufactured reflux boiler tubes for similar environmental conditions and with the tube dimensions normalized with respect to one another (Fig. 5). The three tubes tested were the UFRT[2], the Composite Material Heat Pipe Radiator (CMHR),[5] and the Aluminum Foil-Lined Composite Tube (AFCT).[6]

The CMHR was selected as the device of interest for the optimization analysis becuase it exhibited the highest thermal performance of the three manufactured devices. This tube was the result of a follow-on development effort by the manufacturer to improve the original UFRT configuration. The composite tube has similar dimensions to the original UFRT (Fig. 6).

The CMHR consists of a titanium liner 0.076 mm thick and a composite material mesh of thickness 0.38 mm, resulting in a total wall thickness of approximately 0.456 mm. It has an overall thermal resistance of 0.05 K/W, the lowest of the three tubes tested. The composite material mesh is made of carbon fiber and resin, providing added strength for the pressure vessel. The total length of the CMHR is 1.14 m including its evaporator and fill tube sections, giving it a mass per unit length of 0.145 kg/m. The advantages of the metal inner liner are that it 1) provides a vacuum seal (this is needed because the resin does not fill the cracks in the carbon fiber); 2) keeps noncondensable gases of the outer wrapping from entering the working fluid environment (these gases, if permitted to outgas into the tube could potentially halt the condensation process); 3) prevents erosion of the outer composite layer (throughout the life of the tube, the composite material will erode if in direct contact with the working fluid); and 4) provides sufficient wall thickness at both ends for conventional welding of the end caps to the titanium liner.

The emissivity of the CMHR was measured in laboratory tests[5] to be 0.86, with a solar absorptivity of approximately 0.10.

Optimization

Single Tube

An attempt was made to determine an optimal length and diameter for a single composite tube (CMHR) under "lunar night" conditions. It was not possible to achieve a true maximum because the heat transfer and tube mass increase linearly with increasing length and diameter, yielding a constant value for the heat transfer per unit mass, with tube thickness and all other parameters remaining constant. Diameters greater than approximately 1 in. (2.54 cm) are not considered practical from the standpoint of handling during assembly.

Buckling Investigation

Because it was anticipated that if the length of the tube is increased enough, buckling will at some point occur; a buckling analysis was conducted for a vertical tube using an algorithm combining the moduli of elasticity of the composite material and resin, based on the volume fraction of the two materials and neglecting the thin titanium liner.[7] It was discovered that buckling will not occur until the tube reaches a length of approximately 26 m, which is well beyond any practical length for lunar colony construction.

Linear Array Optimization

An optimization analysis was performed for a linear array of composite tubes operating at a saturation temperature of 373 K, with a lunar surface temperature of 80 K and a solar angle of 45 deg.[8] The optimization parameter of interest is heat transfer per unit mass (q/m). The p/d ratio, where p is the distance between tube centers and d is the diameter of the tube, is incremented for a linear array. To account for mass variations of the entire array system, a parameter termed the mass fraction m is here defined as the ratio of the mass of interconnecting tubing and hardware to the mass of a single tube. As the p/d ratio increases, the heat transfer increases, as does the mass of the tubing and interconnecting material. After reaching a maximum, if p/d is further increased the heat transfer begins to decrease. Figure 7 shows the results of this analysis with the heat transfer per unit mass plotted as a function of pitch-to-diameter ratio for three values of mass fraction.

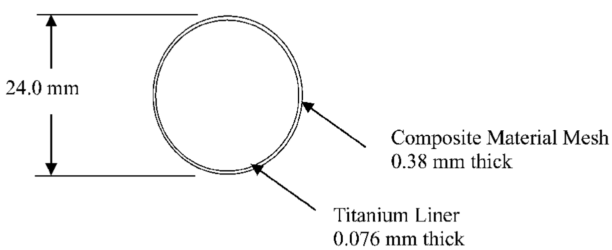


Fig. 6 Cross section of CMHR.

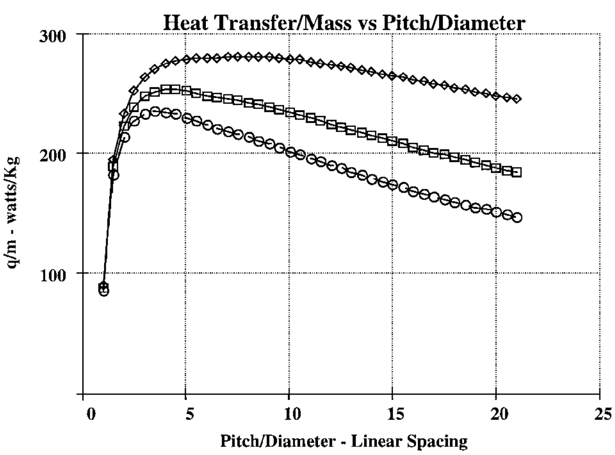


Fig. 7 Optimization of pitch/diameter ratio: ◇, mass fraction 1; □, mass fraction 2; and ○, mass fraction 3.

Conclusions

Because the best of the reflux boiler tubes (CMHR) was selected for the optimization analysis and because there is no true optimum length and diameter for a single tube, the optimization analysis, concentrating on a linear array of tubes operating on the lunar surface, produced the following conclusions:

1) For any given mass fraction the heat transfer per unit mass is not as sensitive to the right of the maxima as it is to the left. This is important because the object of the array design is to space the tubes as closely as possible, and as the tubes get very close together they tend to produce a large blockage effect on one another for radiation, thereby reducing the effective performance of each tube.

2) As the mass of the interconnecting tubing becomes large, i.e., for larger mass fraction ratios, the maximum heat-transfer point shifts downward and to the left. This is produced by a combination of the effects of increased mass and decreased heat transfer caused by interference with the radiative heat-transfer process from adjacent tubes.

3) A reasonable design pitch-to-diameter ratio between 4 and 5 for a mass fraction of 1.0 would provide over 95% of the heat transfer of the mathematical maximum ($p/d = 8$), allowing for a smaller and more logistically feasible array from an engineering design standpoint.

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