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Variable Conductance Heat Pipe Radiator Trade Study for Lunar Fission Power Systems

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Abstract. Nuclear power systems for long-term Lunar and Martian missions present many challenges to thermal management systems, such as variable thermal loads, large temperature swings between day and night, and freezing of the working fluid. The radiator to reject the waste heat must be sized for the maximum power at the highest sink temperature. This radiator is then oversized for other conditions, such as the Lunar/Martian night, or periods when the power to be rejected is low. A Variable Conductance Heat Pipe (VCHP) radiator can passively accommodate changing thermal loads and environments. A trade study was conducted to select the best design for the heat exchanger/heat pipe interface. The interfaces that were examined were an annular evaporator, bent tubes inserted into the coolant channel and a POCO Foam saddle that engulfed the coolant channel and evaporator portion of the VCHP. The trade study showed the highest specific power was for the annular evaporator, roughly 10% higher specific power than the competing interface designs.

Keywords: Titanium/water heat pipe radiator, variable conductance heat pipe radiator, Lunar/Martian fission reactor thermal control

INTRODUCTION

Long-term Lunar and Martian systems present challenges to thermal systems, including changes in thermal load, and large changes in the thermal environment between Lunar (or Martian) day and night. A Variable Conductance Heat Pipe (VCHP) radiator can passively accommodate the changing thermal load and environment. In a VCHP, a non-condensable gas is added that blocks a portion of the condenser. The gas charge blocks more of the condenser as the heat pipe evaporator temperature is reduced. This variable thermal link allows the heat pipe evaporators (and any attached heat exchanger) to remain at an almost constant temperature during variations in power and heat sink temperature. In addition to passively controlling the thermal load, the gas allows the fluid in the heat pipe to freeze in a controlled fashion as the heat pipe is shut down, avoiding damage, and aids with start-up from a frozen condition.

NASA is currently considering Stirling power conversion systems for surface power applications with high-temperature heat-pipe radiators to reject the waste heat (Mason, Poston, and Qualls, 2008). A typical design showing the heat-pipe radiator panel assembly is shown in Figure 1. This type of modular design allows for the radiator panels to be compactly stowed during launch. During operation, waste heat from the Stirling convertors is removed by a pumped single-phase water loop. To reject the heat, the warm water passes through a series of heat exchangers with embedded heat pipes, which in turn are connected to the radiator panels. Since the heat pipes operate independently of one another, the system is inherently redundant and will continue to function effectively even if struck by micrometeoroid and orbital debris (MMOD). Studies have shown that a vertical radiator rejecting heat from both surfaces provides the highest efficiency. Orienting the radiator surfaces coplanar to the ecliptic provides the lowest maximum sink temperature at the equator (Dallas, Diaguila, and Saltsman, 1971). The vertical orientation (with respect to lunar gravity) also improves the performance of the embedded heat pipes. When the evaporator is placed at the lowest level, the heat pipe becomes completely gravity assisted and works as a thermosyphon. Unlike a heat pipe, a thermosyphon requires no wick in the condenser. Removal of the condenser wick provides the lowest possible liquid return pressure drop and allows the thermosyphon to transport more power than a fully-wicked heat pipe of similar geometry.

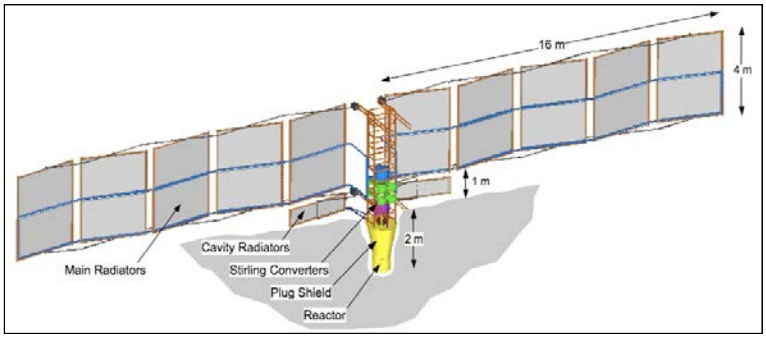


Figure 1. Fission Surface Power System Concept (Mason, Poston, and Qualls, 2008).

In a conventional heat pipe radiator, the heat pipes are totally passive and the radiator is sized to carry the highest power at the highest sink temperature. This radiator is then oversized for other conditions, such as the Lunar/Martian night, or periods when the power to be rejected is low. As the environment changes, the heat exchanger and heat pipe temperatures must be monitored. As sink temperature drops, the heat pipes dump more power and the reactor output must be controlled to keep the secondary loop and heat pipe temperatures constant. By substituting VCHPs for the heat pipes, the radiator will automatically adapt to changing conditions. Such a design eliminates the need for control of the reactor output to prevent freezing in the heat pipes. With VCHPs, freezing of the working fluid becomes permissible since the non-condensable gas adds freeze/thaw tolerance to the heat pipes.

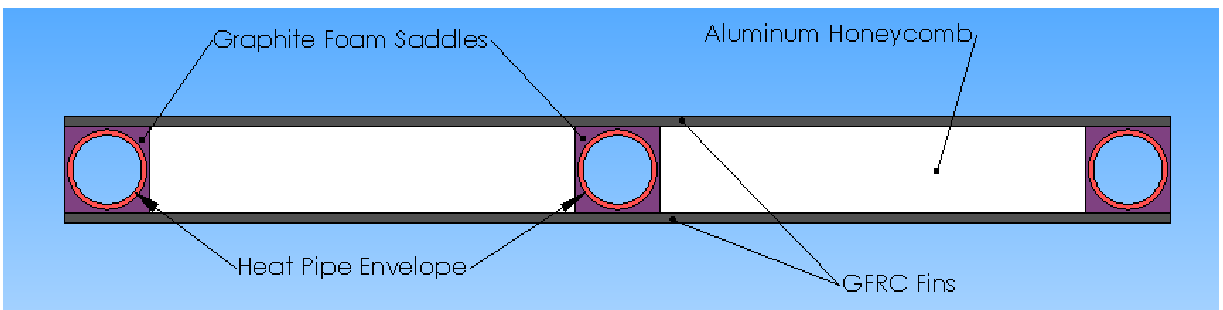


Figure 2. Radiator Panel Cross-Section, with Titanium Heat Pipes and POCO Foam Saddles.

VCHP RADIATOR DESIGN

The typical operating temperature for the radiator responsible for waste heat rejection is 380 K to 400 K. Under a program with NASA GRC, Anderson et al. (Stern and Anderson, 2005; Anderson et al., 2006) demonstrated that titanium/water heat pipes are suitable for radiators in this temperature range. For this reason, titanium heat pipes were assumed for the trade studies with a similar radiator design was chosen for this program; see the cross-section through the radiator panel in Figure 2. The heat pipe configuration assumes that a series of round heat pipes (with integral saddles) are embedded in the radiator panel to distribute heat. The radiator panel consists of the following: 1. A series of variable conductance titanium/water heat pipes to transfer heat from the secondary fluid to the radiator panel, 2. High conductivity POCO foam saddles to form an interface between the circular heat pipe and the flat fin, 3. High conductivity fins of Graphite Fiber Reinforced Composites (GFRC), 4. Aluminum honeycomb to provide stiffness to the structure, and 5. Bonding material applied at the heat pipe/foam interface and the foam/fin interface. Thermal Specifications for the VCHP radiator are given in Table 1.

HEAT EXCHANGER TRADE STUDY

Three different concepts were evaluated for the heat exchanger between the coolant lines and the VCHP radiator. Design considerations include ease of fabrication, modularity and redundancy potential, heat transfer performance, mass, and failure risk reduction. The primary selection criterion is specific power, the system's ability to radiate the waste heat with the lowest possible mass. The three designs considered are:

1. Graphite foam saddle between the coolant channel and the heat pipe.
2. Bent tube inserted into the coolant channel
3. Annular evaporator over coolant channel

Table 1. System Thermal Specifications

Radiator Thermal Power	70 kW_t
Radiator Outlet Temperature	370 K
Radiator Inlet Temperature	400 K
Individual Thermosyphon Power	≈500 W
Sink Temperature - Shackleton Crater	114 K to 212 K
Sink Temperature - Equator	101 K to 314 K
Minimum Wall Thickness	0.051 cm (0.020 in.)
Condenser Length	2 m
Panel Emissivity	0.9
Facesheet Thermal Conductivity (in-plane)	350 W/m K
Facesheet Thermal Conductivity (out of plane)	1 W/m K
Facesheet density	1.75 gm/cm^3
Facesheet Thickness	0.31 mm (0.012 in.)
Aluminum Honeycomb Density	0.05 g/cm^3

The heat exchanger/heat pipe radiator specific power was calculated using the method developed by Anderson and Stern (2005). The following temperature drops occur from the water to the radiator fin:

- Temperature drop from the water in the heat exchanger to the outside of the heat pipe wall. This calculation depends on the chosen interface, and is discussed for each interface below.
- Temperature drop through the evaporator wall.
- Temperature drop through the evaporator wick.
- Temperature drop in the vapor (neglect).
- Temperature drop through the condenser wick.
- Temperature drop through the condenser wall.
- Temperature drop through the graphite saddle.
- Temperature drop into the middle of the fin.
- Temperature drop along the fin.

Heat Exchanger/Heat Pipe Interface

Graphite Foam Saddle

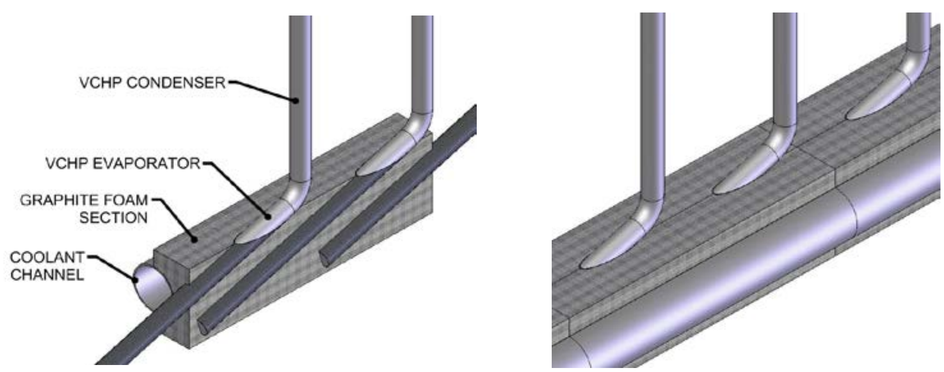
In the Graphite Foam Saddle design, heat transfer from the coolant line to the VCHP occurs through an intermediate saddle made from graphite foam, such as POCO HTC. Figure 3(a) shows a cross-section, while Figure 3b shows a typical assembly. In the graphite saddle design, heat transfer will encounter the following resistances from coolant to the heat pipe evaporator O.D.:

- Convection in the coolant channel,
- Conduction through the coolant channel wall,
- Conduction through the adhesive between the coolant channel and graphite foam saddle,
- Conduction through the graphite foam saddle,
- Conduction through the adhesive layers required to bind the graphite foam to the VCHP,

The individual resistances are shown in Table 2 and add up to a total resistance of 0.089 K/W for the graphite foam saddle concept. The majority of the resistance is found in the convective heat transfer from the coolant to the pipe wall, the VCHP wick, and the graphite foam saddle. Note that, in this design, only half of the heat transfer area available to each channel is used. This effectively doubles the convective thermal resistance. Conduction losses through the pipe wall and adhesive layers are minimal as these thicknesses are small, 8.9x10^{-4} m (0.035 inch) and 1.3x10^{-4} m (0.005 inch), respectively.

Table 2. Individual Resistances for Saddle Concept.

Component	Thermal Resistance
Convection	0.022 K/W
Coolant Piping	0.0035 K/W
Adhesive Layer at Coolant Pipe	0.0052 K/W
Saddle	0.02 K/W
Adhesive Layer at VCHP	0.0069 K/W
VCHP Wall	0.0054 K/W
VCHP Wick	0.026 K/W
Total	**0.089 K/W**



(a) **(b)**
Figure 3. (a) Graphite Saddle Concept Cross-Section, (b) Assembled Graphite Saddles.

The variables varied to maximize the specific power for the graphite saddle concept are:

- Fin Width
- POCO Graphite vs. POCO HTC graphite
- Heat Pipe Evaporator Length
- Heat Pipe Outside Diameter

Table 3. Optimal Variables for POCO Saddle Design (Shackleton Crater).

Variable	Value
Fin Width	18.8 cm
Evaporator Length	30.5 cm
Coolant Channel O.D.	2.54 cm
Angle	5°
Power per Heat pipe	459 W
Specific Power	733 W/kg

The heat pipe evaporator angle was fixed at 5°, since this minimizes the amount of POCO foam required. Table 3 shows the optimal design for the POCO saddle design. Using these variables the optimal design has a total mass of 95.81 kg using 153 heat pipes. As shown below, the foam saddle design outperforms the submerged evaporator design, but not the annular evaporator design. The foam saddle design also adds assembly and maintenance complexities, reducing its appeal.

Bent Tube in Channel Design

The Bent Tube in Channel design, developed at NASA, inserts the VCHPs directly into the coolant channel, as shown in Figure 4. The major drawback of this design lies in long-term reliability. Each VCHP would need to be welded into the coolant channel. One weld failure would cause a leak in the coolant channel and render the entire setup incapable of operation. This is a significant concern for this concept. The advantage of the submerged evaporator design over the graphite saddle concept there is no increase in mass addition due to the heat exchanger. The mass per heat pipe of this design with a given spacing will be lower which will allow power per heat pipe to be lower and still keep the same specific power.

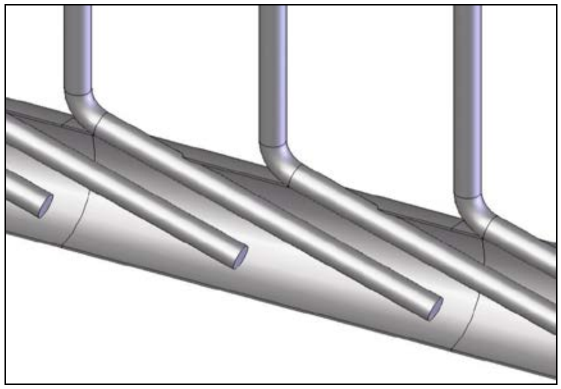


Figure 4. Intrusive VCHP Evaporators Welded to Coolant Channel at Entry Point.

In the bent tube, the temperature drop from the water to the heat pipe evaporator O.D. is due to convection in the water. The, the parameters varied include the evaporator angle and the coolant channel diameter. The temperature drops for the most part are the same as in the saddle concept, except for the initial heat transfer to the evaporator wall by the coolant channel. The variables varied to maximize the specific power for the graphite saddle concept are:

- Fin Width
- Heat Pipe Evaporator Angle
- Heat Pipe Evaporator Length
- Heat Pipe Outside Diameter

Since the evaporators will eventually run into one another at slight angles, the angles examined are between 5° and 15°. The length and angle will determine the size of the coolant channel as shown in Figure 5. The gap is set at 0.51 cm. A larger coolant channel will extend the allowable length of the evaporator (seen in Figure 5), which will reduce the temperature drop into the heat pipe. The coolant channels to be examined will be between 1 inch and 2.5 inches. Table 4 shows the variables taken into consideration.

The maximum specific power for the submerged evaporator design is shown in Table 5. Using this design 181 total heat pipes would be needed to dissipate the input power. That would put a total mass of the system at 104.43 kg. As shown below, the submerged evaporator design is over 10 kg heavier than the annular evaporator design. This design is not limited by the maximum allowable power for the heat pipe, since the output power per pipe was low due to the large temperature drop through the small evaporator area. Although the unit design is lighter, it took 35 more heat pipes to dissipate the total power.

Table 4, Variables for Submerged Evaporator Design (Shackleton Crater).

Variable	Value Range
Fin Width (Heat Pipe Spacing)	5.1 cm to 30.5 cm (2 inch to 12 inch)
Evaporator Length	0.20 m – Limit (8 inch – Limit)
Coolant Channel OD	2.5 cm to 6.4 cm (1 inch to 2.5 inch)
Angle	5° to 15°

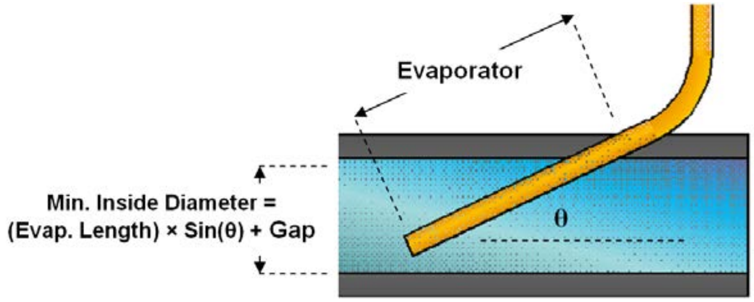


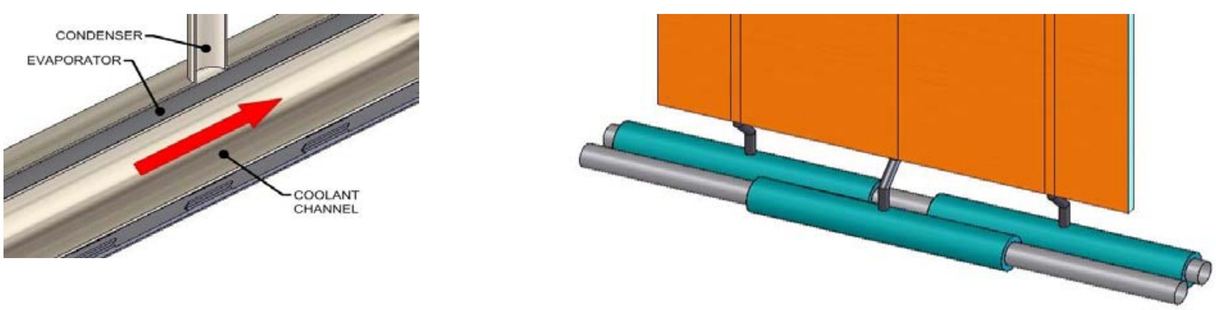
Figure 5. Coolant Channel Diameter Relationship with Evaporator Length and Angle.

Table 5. Optimum Variables for Submerged Evaporator Design (Shackleton Crater).

Variable	**Value Range**
Fin Width	14 cm (5.48 in)
Evaporator Length	48 cm (19 in)
Coolant Channel O.D.	5 cm (1.86 in)
Angle	5°
Power per Heat pipe	387.6 W
Specific Power	675.3 W/kg

Annular Evaporator

In this concept, shown in Figure 6, the VCHP evaporator surrounds the coolant flow. If the evaporators cover the entire available length, then this concept offers some MMOD protection, since the outer wall and screen wicks will "bumper" the impact. In addition, this design offers the largest evaporator heat transfer surface area of the three concepts. Assuming an equivalent evaporator of 10 inches (25.4 cm) and a channel diameter of 2.54 cm (1 inch), the annular evaporator heat transfer area is much higher than the heat transfer area of the bent tubes. The annular evaporator has 167% more heat transfer area than a 0.95 cm (0.375 inch) bent tube, 100 % more area than a 1.27 cm (0.50 inch) bent tube, and 33% more area than a 1.91 cm (0.75 inch) bent tube. Since there will be two coolant loops; every second annular evaporator will be attached to each coolant loop; see Figure 6(b). This allows the evaporator to be twice as long as the spacing between heat pipes.



(a) **(b)**

Figure 6. Annular Evaporator Concept. (a) Cross-Section. (b) Heat Pipe Evaporators Alternate between Coolant Loops.

In this concept, the outer surface of the coolant channel would be wicked. The evaporator shell with condenser arm would then be welded around the coolant channel. This approach would allow several VCHPs to be welded to a seamless section of pipe. Weld failure would only compromise a single VCHP, and not the entire coolant channel.

The entire system (evaporator, condenser, foam, fins, etc.) was examined to optimize the specific power in this design. As mentioned above, an advantage of the annular evaporator is the reduced temperature drop into the evaporator due to the large area exposed to the coolant channel. This allows the wick to take on less thickness to hold the same amount of liquid and thus less temperature drop through the liquid. In order to maximize specific power the variables shown in Table 6 were considered.

Table 6. Trade Study Variables for Annular Evaporator.

Variable	Range
Fin Width (Heat Pipe Spacing)	5.1 cm to 30.5 cm (2 in to 12 in)
Evaporator Length	2.5 cm to 30.5 cm (1 in to 12 in)
Condenser Diameter	1.0 cm, 1.3 cm, 1.9 cm (3/8 in, 1/2 in, 3/4 in)
Condenser Wall Thickness	Based on Stress
Evaporator Wall Thickness	Based on Stress
Fin Thickness	3.1 mm, 4.1 mm (0.012 in, 0.016 in)
Foam Type	POCO, POCO HTC
Radiator Location	Shackleton Crater

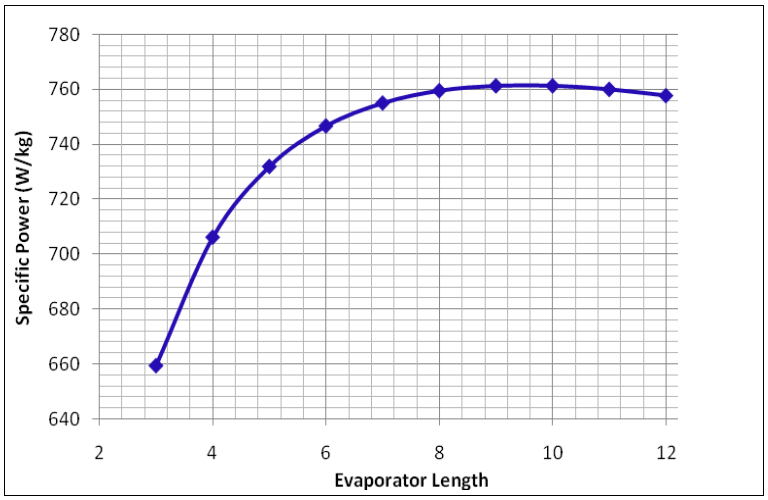


Figure 7. Highest Possible Specific Power for a Given Evaporator Length, Annular Evaporator.

The sink temperature was assumed to be 210 K, the coolant channel was chosen to be 1 inch, and the coolant temperature was chosen to be 385 K, which is the average of 370 K and 400 K. The sink temperature chosen is the maximum temperature at Shackleton. Figure 7 shows that evaporator lengths between 9 inches and 11 inches all show high specific powers. The evaporator length of 10 inches shows the highest specific power, but only slightly (0.04 W/kg) over the 9 inch evaporator. Table 7 shows both these designs. The 10 inch evaporator design would yield a total mass of 91.96 kg and would have a total of 146 heat pipes. The 9 inch design, though very close in specific power, would need one extra heat pipe (147) and that would give this design a total mass of 92.24 kg.

Table 7. Annular Heat Pipe Radiator Parameters (10 Inch Evap. versus 9 Inch Evap.)

Variable	Option 1	Option 2
Fin Width (Heat Pipe Spacing)	15.6 cm (6.145 in.)	15.7 cm (6.194 in.)
Evaporator Length	25.4 cm (10 in.)	22.9 cm (9 in.)
Condenser Outside Diameter	0.95 cm (3/8 in.)	0.95 cm (3/8 in.)
Fin Thickness	0.31 mm (0.012 in.)	0.31 mm (0.012 in.)
Foam Type	POCOFoam	POCOFoam
Power per Heat Pipe	479.5 W	477.7 W
Specific Power	761.3 W/kg	761.26 W/kg

Interface Comparison

The optimized specific power as a function of fin width for the 3 interfaces is compared in Figure 8. Due to the small temperature drop from the coolant channel to the vapor, the annular evaporator offers the most benefits. It is a fairly simple design for assembly; it requires the fewest heat pipes to dissipate the total input power, and has the lowest mass for the overall design. For optimized designs the annular evaporator has 3.8 % greater specific power than the foam saddle design, and 13 % greater specific power than the submerged evaporator design.

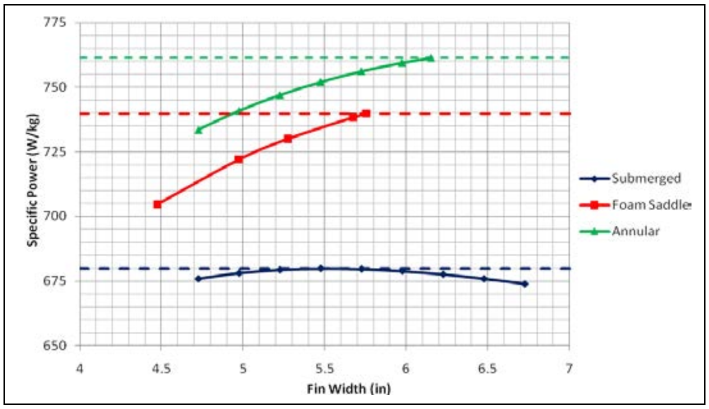


Figure 8. Comparison of Optimized Designs for Each Interface (Shackleton Crater).

CONCLUSIONS

NASA is currently considering Stirling power conversion systems for surface power applications Mason, Poston, and Qualls, 2008), with high-temperature heat-pipe radiators to reject the waste heat A trade study was conducted to select the best design for the heat exchanger/heat pipe interface. The interfaces that were examined were an annular evaporator, bent tubes inserted into the coolant channel and a POCO Foam saddle that engulfed the coolant channel and evaporator portion of the VCHP. The trade study showed the highest specific power was for the annular evaporator, roughly 10% higher specific power than the competing interface designs. . The benefits of this design include simplicity, easy to assemble, requires the fewest heat pipes to dissipate the total input power, and has the lowest mass for the overall design.

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