

Considerations for Radiator Design in Multi-Megawatt Nuclear Electric Propulsion Applications

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A key performance parameter determining the feasibility and performance of a multi-megawatt nuclear electric vehicle is the power systems specific mass (mass per unit of electric power output). The specific mass of the main radiators is the largest single element within the power system and can be greatly affected by numerous assumptions and design considerations. Among these are the number of parallel fluid loops, the overall geometry and view factor of the radiator, and for high temperature systems the method for accommodating the temperature limit of water heat pipes. Analyses are performed to develop an understanding of this trade space to enable a more accurate mass estimate and to help guide technology development efforts.

I. Nomenclature

A	= radiator area	σ_{SB}	= Stefan-Boltzmann constant
w	= panel width	ϵ	= emissivity
θ	= shadow shield half-angle	T_{avg}	= average temperature
α_{PS}	= specific mass of the NEP power system	s	= heat pipe spacing
ρ_{areal}	= areal density	F	= view factor
ΔP	= pressure drop	k	= thermal conductivity
f	= friction factor	t_{panels}	= panel thickness
ρ	= fluid density	η	= fin effectiveness
L	= pipe length	γ	= non-dimensional equivalent temperature parameter
d	= pipe diameter	T_{sink}	= sink temperature
u	= fluid velocity	$Q_{fin,real}$	= fin heat transfer
P_{max}	= max pressure	$Q_{fin,perfect}$	= heat transfer of perfect iso thermal fin
t_{pipe}	= thickness of pipe		
ζ	= non-dimensional fin effectiveness parameter		

II. Introduction

Nuclear electric propulsion (NEP) is under consideration by NASA as propulsion for a crewed Mars mission. The space nuclear propulsion (SNP) project is developing a technology maturation plan to guide development of the NEP subsystem technologies [1] [2]. An NEP system is composed of a nuclear reactor and power conversion cycle that

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generates electricity to run electric thrusters. NEP allows for high specific impulse at high power, enabling difficult opposition-class trajectories for a round-trip Mars mission to be considered in order to reduce crew time in space [3] [4]. One of the key performance parameters in determining the performance and feasibility of an NEP system is the specific mass of the power system, α_{ps} , defined as kilogram per kilowatt of electric power produced. The power system consists of the reactor and shield, power conversion, power management and distribution, and primary heat rejection subsystems. Of these, the primary heat rejection subsystems or PHRS has been found by Analytical Mechanics Associates (AMA) [5] [6] [7] [8] [9], and others [10] [11] [12], to be one of the largest contributors to the mass of the power system.

The PHRS mass model in this work has been developed to be adaptable and able to explore the trade space encompassed by a variety of different configurations; however, it does assume particular technology choices aligned with those built in Ref. [13]. The PHRS will be comprised of a combination of a pumped loop delivering the heat to individual carbon composite panels and heat pipes distributing the heat throughout the panels. Both heat pipes and a pumped loop are needed due to the size of the PHRS. Heat pipes are an ideal method to transfer heat since they can do so with almost no temperature drop between the hot and cold ends. This is accomplished using the liquid/gas phase change of the working fluid. At the hot end or evaporator, the fluid evaporates and is driven by a pressure gradient to the cold end or condenser. Here the fluid changes back to a liquid, and is wicked by capillary forces back to the evaporator. Another benefit of heat pipes is they are independent and the puncture of one has no effect on the others. Despite these advantages, heat pipes are fairly limited in the distance they can transport heat, requiring the pumped loop for the very large areas expected for the PHRS on a multi-megawatt class NEP vehicle. Eutectic sodium potassium (NaK) is chosen as the working fluid in the pumped loop due to its high density and favorable heat transfer properties. Another option is water, but this was discounted due to the high pressure needed to maintain its liquid phase at the expected operating temperatures. The NaK selection also allowed electromagnetic pumps to be selected to power the fluid loop. Water is, however, the best candidate for the heat pipe fluid, with the grooved wicking structure and shell assumed to be titanium. The panels will be made of a carbon-carbon composite with a high emissivity coating. The panels are assumed to consist only of the composite face sheet and include no structural core. The trade space explored in this paper will be within the bounds of these basic technology assumptions.

The PHRS and associated structure can make up half or more of the total power system mass, and simple estimates of the mass of the heat rejection system contain a large degree of uncertainty stemming from the wide variety of assumptions and configurations that could be used in a flight system. The purpose of this paper is to explore the PHRS trade space and develop an understanding of how these assumptions impact the mass. Given the low technology readiness level (TRL) of NEP systems, including the PHRS [14], the goal of the current modeling effort is to explore different options and not to recommend a particular design choice. The PHRS mass model developed in this paper is still subject to a number of assumptions and uncertainty due to the low TRL, and results should be interpreted in this context. The exact values discussed could change, however, the general trends and conclusions are expected to remain the same.

III. Mass Model Formulation

AMA has developed a model for the mass of the NEP power system [7] that scales off of electric power level, power conversion cycle mass flow rate, efficiency, and other factors. The PHRS mass model is expanded from work done by Siamadis [15] for the Jupiter Icy Moons Orbiter (JIMO) [10]. Siamadis's radiator concept assumes a pumped fluid loop supplies waste heat to radiator panels, which have embedded heat pipes to distribute the heat across the panels. Siamadis assumes a combination of series and parallel piping supplying the pumped fluid. This assumption worked well for JIMO which was designed to be a 100 kW_e system. A crewed vehicle to Mars would require a power level on the order of 2 MW_e, and at this power level significantly more of the piping needs to be laid out in parallel in order to avoid a massive pressure drop or excessive piping and fluid mass. To maintain a given pressure drop, the pipe diameter must be increased to accommodate a higher mass flow rate and power level. This results in the overall PHRS area density increasing significantly at high power levels. In order to account for this in Ref. [7], a maximum mass flow rate of 30 kg/s was assumed, and any mass flow rate beyond this needed to transport the overall heat load through the PHRS was allocated to a separate parallel loop, thus limiting the piping and fluid mass increases to a reasonable level.

Improvements made to the model in this paper have eliminated the need for the maximum mass flow rate simplification. The number of parallel loops can now be set as an input parameter, and its effect on the PHRS areal mass quantified. In addition to the series/parallel configuration of the piping, other improvements made since the last iteration include taking into account heat pipe physics, different geometries and view factors, and varying certain

design parameters. This improves the accuracy of the model and allows it to be more flexible and better able to guide the technology development of NEP systems.

A. Thermodynamic Cycle

A Brayton cycle is used to convert the fission reactor's thermal energy to electricity that can be used to power electric thrusters such as the Hall-effect or magnetoplasmadynamic thruster. One of the primary drivers of cycle performance is the power conversion inlet temperature (PCIT). Fig. 1 shows a reference case for a 1200 K PCIT, 2 MW_e, Brayton cycle [9]. The specific state points and mass flow rate in this reference case will be used as inputs to the PHRS model for all the trade studies presented in this paper. The Brayton cycle comprises a recuperator, turbine, compressor, reactor, and heat exchanger interfacing the Brayton cycle and PHRS as shown in Fig. 1. The heat exchanger is needed to transfer energy from the Brayton cycle working fluid, He-Xe in this reference case, to the NaK in the PHRS. This heat exchanger is needed so that NaK can be used for the pumped working fluid in the PHRS rather than He-Xe which has a low density and poor heat transfer properties. In addition, a separate pump can more efficiently recover the PHRS pressure loss than the compressor could if the Brayton working fluid were used as the heat transport medium. Using NaK results in a lower overall mass of the PHRS plumbing [7].

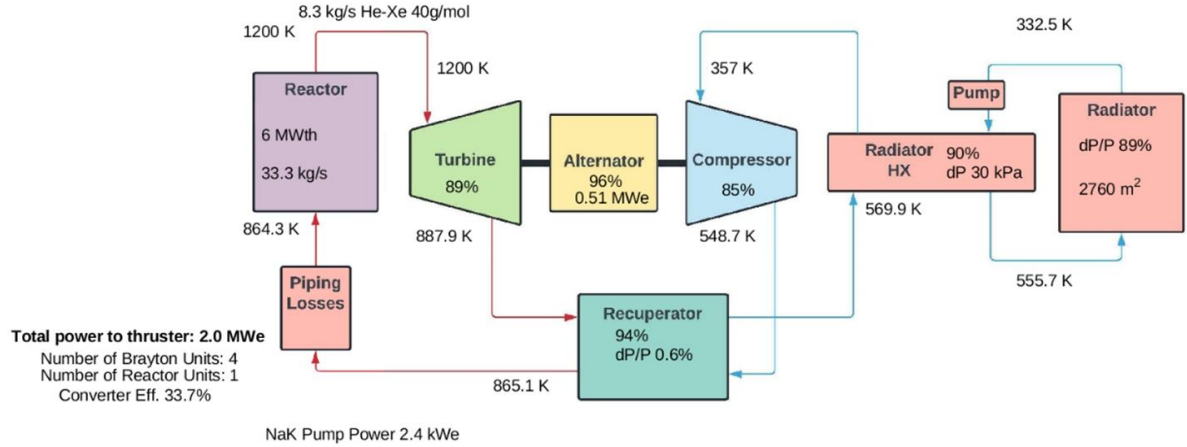


Fig. 1 Reference power cycle diagram for 1200 K PCIT [9].

B. PHRS Geometry

The geometric layout of the PHRS is simplified to be a function of as few variables as possible. The total area is broken up into different ‘wings’, these are symmetrical sets of radiator panels rotated about the central structure of the NEP vehicle. Fig. 2 shows one individual wing. The triangular shape is defined by the reactor radiation shield half-angle, θ . Due to concerns of radiation scattering by the PHRS and significantly increased surface coating degradation [16] [10], the PHRS is located in the protected region behind the shield. The triangular wing is composed of uniform rectangular panels; the width of these panels, w (direction away from the central axis), is an input to the model and the length (direction along the central axis) is a function of the shield half-angle and the width such that the panels can be stacked in a stair-step pattern in the triangle defined by the shield half-angle. Fig. 3 shows how multiple wings can be arranged around the central axis of the NEP vehicle.

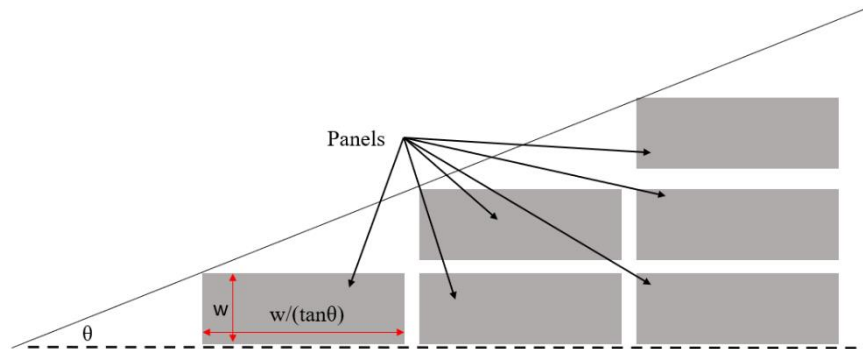


Fig. 2 Triangular panel layout of one wing.

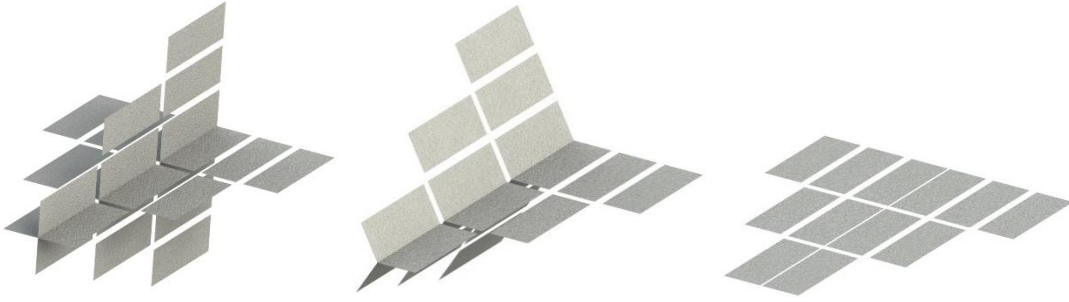


Fig. 3 4-wing configuration (left), 3-wing configuration (middle), and 2-wing configuration (right).

Fig. 4 shows how the liquid NaK and heat pipes interface in each radiator panel; the NaK runs down the length of one side of the panel and all of the heat pipes are oriented perpendicular to this flow, with their condenser sections running down the width of the panel. The NaK makes two passes of the panel and enters and exits the panel at the same point. On the first pass, every other heat pipe is in contact with the NaK, with the remaining heat pipes being serviced on the second pass. This is to spread out the hottest parts of the panel and improve the average radiating temperature and overall radiative effectiveness. The pumped NaK is assumed to be in direct contact with the heat pipes, transferring energy via convection. For the purpose of this model the interface is assumed to be perfect, with no temperature loss between that part of the NaK and the heat pipe. The effect of a small change in temperature at this interface is relatively small, requiring only a little additional radiator area, which should be accounted for in a mass growth allowance when these results are used in a mission study.

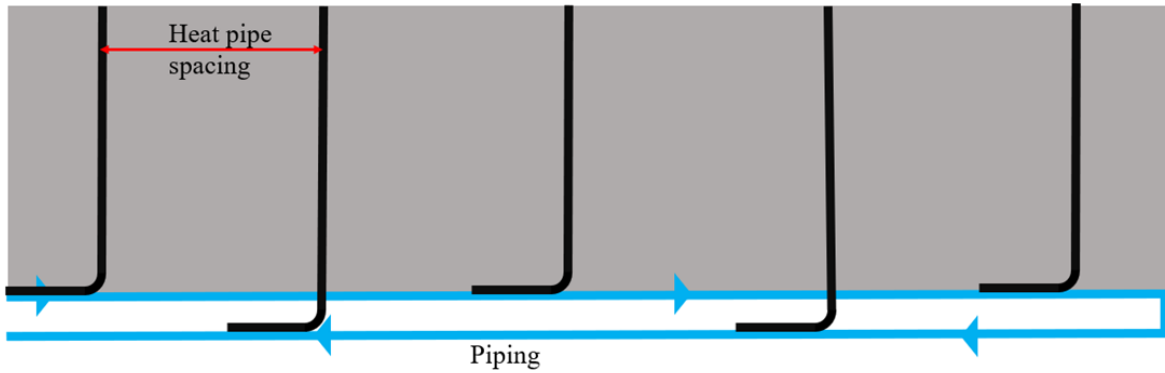


Fig. 4 Heat pipes assumed to have direct interface with pumped fluid.

Fig. 5 shows the extremes cases for how the parallel loop trade is set up. In the fully series configuration, each panel is plumbed in series with the others. In this case one single pipe snakes its way across an entire PHRS wing, servicing each panel in turn. In the fully parallel configuration, each panel is serviced by its own pipe, this means that for each panel there is a supply and return pipe running along the central axis. Fig. 5 is intended only as a visual aid to demonstrate the differences between the series and parallel piping configurations, not as a representation of the true piping path, as it neglects certain details such as the second pass through each panel. Any combination of series and parallel pipes between one fluid loop per wing and one fluid loop per panel can be used. This allows for the number of loops to be directly traded for the lowest system mass.

The view factor will not be constant for different values of the shield half-angle or different numbers of wings. With two wings the view factor is exactly one since none of the PHRS can see any other part of it, and as shown in Fig. 6 when more wings are added the view factor is significantly reduced. In addition, the view factor increases for larger shield half-angles as a smaller fraction of the total PHRS area is concentrated close to the central axis.

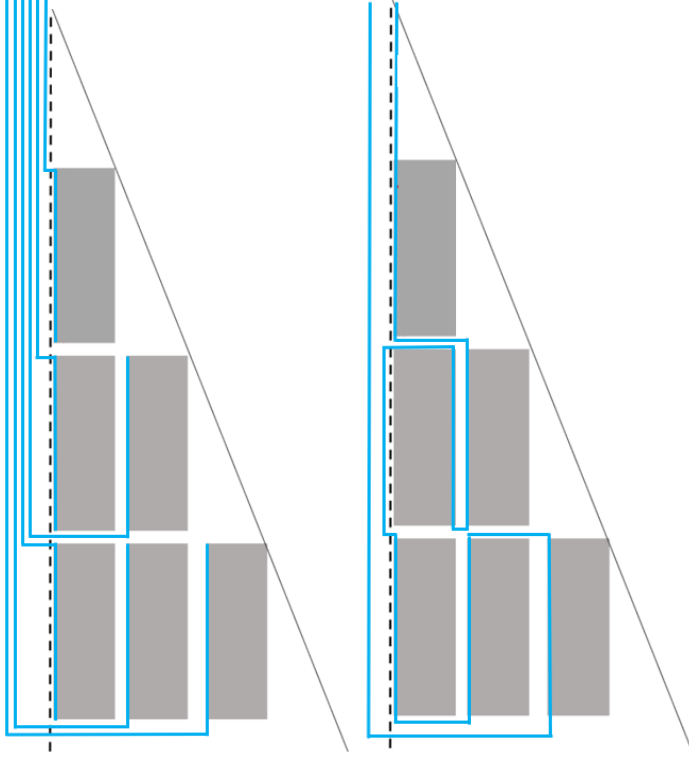


Fig. 5 Parallel piping layout (left) vs series piping layout (right).

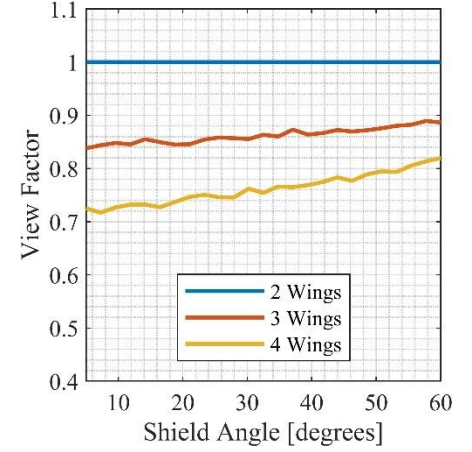


Fig. 6 Changes in view factor for different radiator configurations.

C. Piping

The NaK piping is assumed to be circular titanium tubes, with the diameter determined by defining an allowable pressure loss Eq. (1) [17]. As the allowable pressure drop increases the pipe can be made smaller. The PHRS pressure drop is an important variable determining the performance and mass of the NEP system as a whole. The optimum pressure loss has previously been estimated to be around 360 kPa [7], and recent changes to the model have not changed this value. With the pressure loss fixed at 360 kPa, the primary driver of piping diameter is the mass flow rate through the pipe. This is determined by the Brayton cycle temperature and power requirements, and the number of parallel loops used in the PHRS. The thickness of the piping is determined, using hoop stress Eq. (2), by the maximum pressure present along the piping. This pressure is tied to the pressure drop since the max pressure only need be a little above the max pressure drop.

$$\Delta P = f \rho \left(\frac{L}{d} \right) \left(\frac{u^2}{2} \right) \quad (1) \quad \sigma_y = \frac{Pr}{t} \quad (2)$$

D. Heat Pipes

Each heat pipe in the PHRS needs to be able operate at a specific temperature and power level. Using an operational limits-based heat pipe model similar to that in Ref. [18], a heat pipe geometry can be found for a given power level and temperature. Since it is undesirable to have each heat pipe in the PHRS be unique, five temperatures (350 K, 400 K, 450 K, 500 K, 550 K) and eight power levels (250 W – 2kW) were chosen to select heat pipe geometries for. Most of the heat pipes in the PHRS will be operating in between these values so the selected geometry will correspond to the next higher level.

The condenser section of the heat pipe is the same as the width of the panels; two different values of this variable are considered: 1 meter and 0.75 meters. The adiabatic section of the heat pipe is the length between the evaporator and condenser and is assumed to be 5 cm, with the evaporator length dependent on the power and temperature requirements and set to be as small as possible. Other geometry parameters solved for include: the outer diameter, shell thickness, wick thickness, groove width, groove spacing, and number of grooves. Due to the unconstrained nature of the problem, there are many valid geometries for a given power level and temperature. A geometry was selected such that the mass of the heat pipe is minimized. However, the mass optimization was unstable due to the nonlinear and non-continuous nature of the heat pipe limits equations, and the heat pipe masses calculated may not have the

lowest possible. Fig. 7 shows the mass of heat pipes for the defined operational characteristics; the general trend is that the heat pipe mass will increase as the temperature or power level are increased. In particular, the mass of the 550 K heat pipes is significantly larger than that of the other temperatures, this is because 550 K is the upper limit at which water heat pipes can operate and still transfer around a kilowatt of power. Section IV further discusses the difficulty with operating heat pipes at or above 550 K.

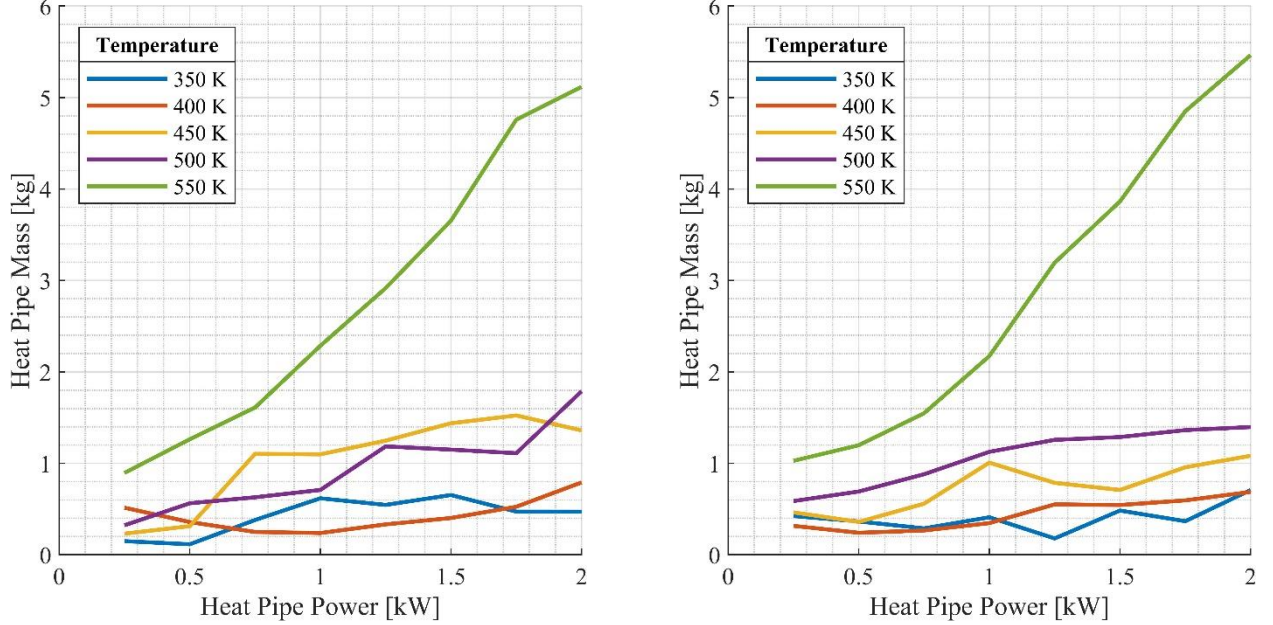


Fig. 7 Heat pipe mass as a function of temperature and power level, 1 meter condenser (left) and 0.75 meter condenser (right).

E. Panels

The heat pipes are assumed to be isothermal, but the carbon-carbon composite panels will have a temperature drop between the heat pipes determined by their conductance and the power radiated away from the surface. To account for this, the PHRS area is increased by 10% compared to a constant temperature panel, allowing for the panel thickness to be calculated using equations (3), (4), (5), and (6) [19]. The panel thickness is governed by two primary assumptions: the heat pipe spacing and the temperature uniformity of the panels. Varying each of these will have the same result on the panel thickness, and since the temperature uniformity would likely be a design requirement, the heat pipe spacing is chosen to be the independent variable. The panels need to be thicker when the heat pipe spacing is larger in order to conduct the heat a further distance with the same temperature drop. A major source of uncertainty in the panel mass comes from the assumed thermal conductivity of the carbon-carbon composite, as the conductivity can vary widely depending on the exact composite used. Juhasz [20] provides the properties for several composites, and a conductivity of 400 W/m/K was chosen. This corresponds to the P95 WG composite, which is on the higher end of conductivities for composites that have actually been fabricated.

$$\zeta = 2 \left[\sigma_{SB} \epsilon T_{avg}^3 \left(\frac{S}{2} \right)^2 F \right] / [kt_{panels}] \quad (3)$$

$$\gamma = \frac{T_{sink}}{T_{avg}} \quad (4)$$

$$\eta = (1 - 1.2491\zeta + 1.0093\zeta^2)(1 - \gamma^4),$$

$$0.01 \leq \zeta \leq 0.2$$

$$\eta = (-0.4049 \log(\zeta) + 0.5321)(1 - \gamma^4), \quad (5)$$

$$0.2 \leq \zeta \leq 2.0$$

$$\eta = \frac{Q_{fin,real}}{Q_{fin,perfect}} \quad (6)$$

F. Other Components

1. Pump

Polzin [21] reviews the state of liquid metal pumps for space based nuclear power applications. A specific mass of 260 kg per kW of required pump power is the average of the electromagnetic pumps identified by Polzin. This value

is likely to be quite conservative since many of the electromagnetic pumps it is based on experienced poor performance and technical issues that could be remedied. In addition, mechanical pumps are another technology option and may result in a lower mass.

2. Accumulator

An accumulator is needed for the pumped loop section in order to account for the difference in fluid volume due to the temperature change between idle (assumed to be 300 K) and nominal operation modes. This volume differential is used to scale the accumulator mass, based off of the NaK accumulator design from Tournier and El-Genk [22].

3. Structure and Deployment

The support structure and deployment mechanisms together are assumed to have a constant areal mass of 1.6 kg/m². This value comes from the FEA work done on the central truss by the Aerojet-Rocketdyne [23]. They estimated a truss mass of 4,000 kg for a 2,500 m² PHRS.

IV. High PHRS Inlet Temperature Considerations

A. Limitations of Water Heat Pipes

As discussed previously, water heat pipes are not practical at operating temperatures exceeding 550 K. Above this temperature the mass of the heat pipes becomes exceptionally large and the power they can transfer decreases significantly. This is particularly problematic given that so much more heat can be rejected through radiation at these elevated temperatures, and if the heat pipes cannot transfer this additional energy they must be put closer together. Fig. 8 shows the limits of a heat pipe designed to transport 2 kW at 550 K. The viscous, sonic, and entrainment limits occur at a much higher power levels and are not relevant for this particular heat pipe. In addition to reduced heat transport capability at 550 K, the limiting factor transitions from the capillary limit to the boiling limit, which is more important to stay away from. Unlike with the capillary limit where the heat pipe is simply unable to transfer more power, exceeding the boiling limit can heat pipe evaporator wick can dry-out, preventing any heat transfer at all and leading to damage.

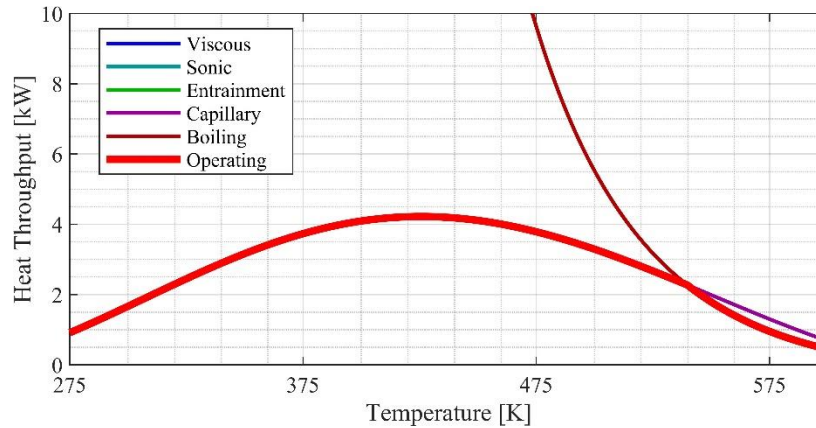


Fig. 8 Water heat pipe limit diagram.

B. Higher Temperature Heat Pipe Fluid Options

Due to the limitations of water heat pipes, they will not be used if the temperature exceeds 550 K. This however creates a problem as the power conversion system as a whole typically needs to sink energy to the PHRS at higher temperatures. This is particularly true when higher PCITs are considered as an option to improve the cycle efficiency. Higher temperature heat pipes are an option, but Cesium, the next common heat pipe fluid candidate, will not have a power limit in the kilowatt range until temperatures greater than 650 K [24]. There are a number of other potential fluid candidates for heat pipes in the 550 K to 650 K range including mercury, sulfur, sulfur/iodine, iodine, halides (SnCl₄, TiCl₄, GaCl₃, TiBr₄, AlBr₃, SbBr₃, SbCl₃, etc.), and organics (aniline, diphenyl, naphthalene, toluene, hydrazine, N-octane, phenol, etc.). Each of these fluids is not currently viable due to issues such as low TRL, toxicity, corrosiveness, or dissociation of fluid molecules when exposed to radiation [25] [26] [27].

C. Solutions to Water Heat Pipe Temperature Limit

Due to the lack of viable fluids in the required temperature range, in this work, heat pipes will not be considered as an option for those parts of the PHRS that exceed 550 K. Instead, other solutions are developed to work around the limitations of water heat pipes.

1. Limit Power Conversion Cycle

One option is to limit the power conversion cycle so that the PHRS inlet temperature will be below 550 K. This is the simplest solution as it requires no modification to the PHRS, however it will entail operating the power conversion cycle away from its optimal temperature. The power conversion cycle is typically optimized so as to reduce the overall α_{ps} . The specific mass can optimize at higher PHRS temperatures due to the reduced radiator area needed to reject heat at higher temperatures, despite resulting in a less efficient cycle. For the 1200 K PCIT reference case the PHRS inlet temperature is only 556 K so operating at a lower 550 K would have a small effect, but at a PCIT of 1400 K the PHRS inlet temperature optimizes much higher. Some of the benefit of going to a higher PCIT is lost if higher radiating temperatures cannot be taken advantage of.

2. Tap-Off

Another option is to use a tap-off, shown in Fig. 9, where some of the PHRS exit flow is mixed with the inlet flow to result in a temperature of 550 K going into the PHRS. Essentially this results in trading a lower temperature inlet for a higher mass flow rate through the PHRS. This method has similar drawbacks to limiting the power conversion cycle, in that the benefit of radiating at a higher temperature is not realized.

3. Pre-Cooling

Another option that allows for radiation at higher temperatures is pre-cooling, shown in Fig. 9. This option assumes that NaK is pumped through each of the panels that are too hot for water heat pipes to function, resulting in a hybrid PHRS using both pumped NaK and heat pipes in the paneling. The panels are assumed to be the same size and shape regardless of the method of cooling, although this need not be the case. The drawbacks of this method include much more exposure of the NaK pipes to micrometeorite and orbital debris (MMOD) impact, where a single impact can result in NaK leaking out of large parts of the system and causing a significant decrease in performance to the NEP system. Damage to a heat pipe on the other hand results in graceful degradation where only a small part of the PHRS is lost.

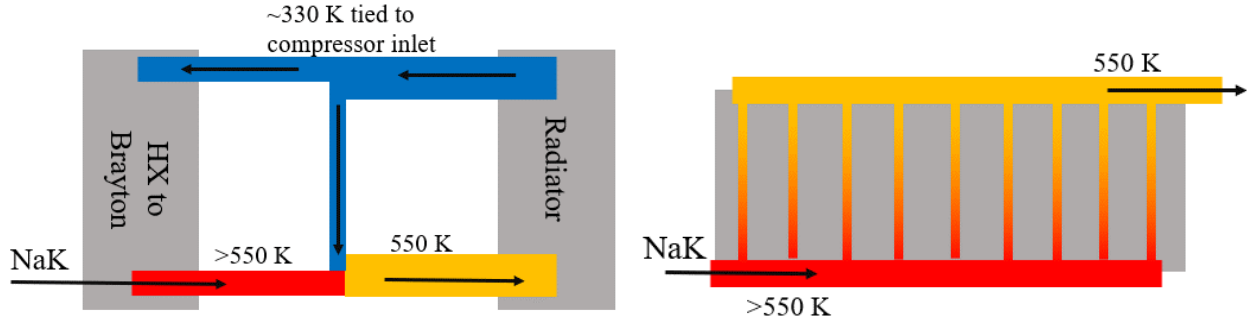


Fig. 9 Tap-off configuration (left) and pre-cooling configuration (right).

V. PHRS Trade Studies

A. Reference Case

The trade studies performed in the following section use the 1200 K PCIT, 2 MW_e reference case from Fig. 1. The major inputs and outputs from the PHRS mass model at this reference case are given in Table 1. In the following analyses, inputs such as the number of parallel loops, high inlet temperature method, number of wings, panel width, heat pipe spacing, and shield half-angle will be varied to determine their effect on the PHRS mass. Other variables will be held constant at the value defined in Table 1. The specific mass, α , refers to the mass per kilowatt of electricity generated by the Brayton cycle, for this reference case that is 2 MW.

Table 1 Major inputs (blue) and outputs (green) of 1200 K PCIT, 2 MW_e reference case.

Pumped fluid	Eutectic NaK	Number of heat pipes	9280
Heat pipes	Ti/H ₂ Oi	Average heat pipe length	1.2 m
Total heat load	3.76 MW _{th}	Average heat pipe diameter	2.4 cm
Inlet temperature	556 K	Average heat pipe power	400 W
Inlet pressure	556 K	Average heat pipe mass	0.6 kg

Outlet temperature	333 K	Pump specific mass	260 kg/kW _e
Outlet pressure	400 kPa	Fin effectiveness	0.91
Mass flow rate	18.4 kg/s	Panel thickness	1.5 mm
High inlet temperature method	Tap-off	Fluid α	1.0 kg/kW _e
Tap-off mass flow rate	18.9	Piping α	1.2 kg/kW _e
Number of wings	4	Heat pipe α	2.8 kg/kW _e
Number of parallel loops	30	Panels α	2.5 kg/kW _e
Area perfect	3280 m ²	Pump α	1.0 kg/kW _e
Area actual	3610 m ²	Accumulator α	0.3 kg/kW _e
Emissivity	0.9	Structure α	2.7 kg/kW _e
Sink temperature	4 K	Total PHRS α	11.5 kg/kW _e
Effective radiating temperature	409 K	Shield α (Not included in PHRS α)	2.2 kg/kW _e
Panel width	1 m	Fluid ρ	0.5 kg/m ²
Shield half-angle	10°	Piping ρ	0.7 kg/m ²
View Factor	0.73	Heat pipe ρ	1.6 kg/m ²
Panel length	5.7 m	Panels ρ	1.4 kg/m ²
Number of panels	80	Pump ρ	0.6 kg/m ²
Central boom length	70 m	Accumulator ρ	0.2 kg/m ²
Piping diameter	1.3 cm	Structure ρ	1.6 kg/m ²
Piping length per loop (max)	200 m	Total PHRS ρ	6.6 kg/m ²
Heat pipe spacing	20 cm		

B. Series vs. Parallel Piping Configuration

As discussed, the piping layout within the PHRS array can have a huge effect on the mass of the heat rejection system. The piping layouts considered range from each panel in a wing connected in series to each panel connected in parallel. Fig. 10 shows the effect this assumption has on the PHRS. As expected, the pipe diameter decreases as the fluid mass flow rate is split into multiple parallel loops. This alone does not account for the decreased mass for larger numbers of parallel loops, since it simply means having more smaller pipes rather than fewer larger pipes. The mass savings for the fluid is correlated to the residence time of the fluid in the radiator. The significance of the residence time is that when multiplied by the mass flow rate, it determines the fluid mass in the PHRS. Although the residence time is a function of several variables such as the pipe radius, pipe length, and fluid velocity, the pipe length is the primary driver for the residence time decrease in Fig. 10. The total mass flow rate is the same even if it is broken up into multiple parallel paths; but in parallel, each flow path is shorter, not needing to visit every panel, and allowing for the average residence time in a fluid loop to decrease. Another way to view the residence time is that in the series case, the majority of the heat capacity of the NaK flowing through a particular panel is not used and is sent on to the next panel. As additional parallel loops are added, the mass flow through each panel is decreased and a larger fraction of the available fluid heat capacity is used in each panel. From a Lagrangian reference frame, the rate of heat transfer is increased for a given fluid volume entering the PHRS since it is divided to service multiple panels simultaneously. This decrease in residence times allows for many small pipes to be significantly lighter than few larger ones.

In addition to potential mass savings, having multiple parallel fluid loops has other benefits. During system start up and shut down, the effective radiator area should vary as a constant radiator area will result in more heat rejection than necessary thus yielding in either decreased system performance or a non-operative system. The independent parallel flow paths on the PHRS will allow for a convenient method for the transient start up and shut down of the system. Furthermore, the system will also be throttleable with the variable effective radiator area capability. Although this study has examined PHRS operation with the assumption that the heat pipes will operate without thawing, future work should examine system performance during early transient effects when the fluid inside the heat pipes is frozen.

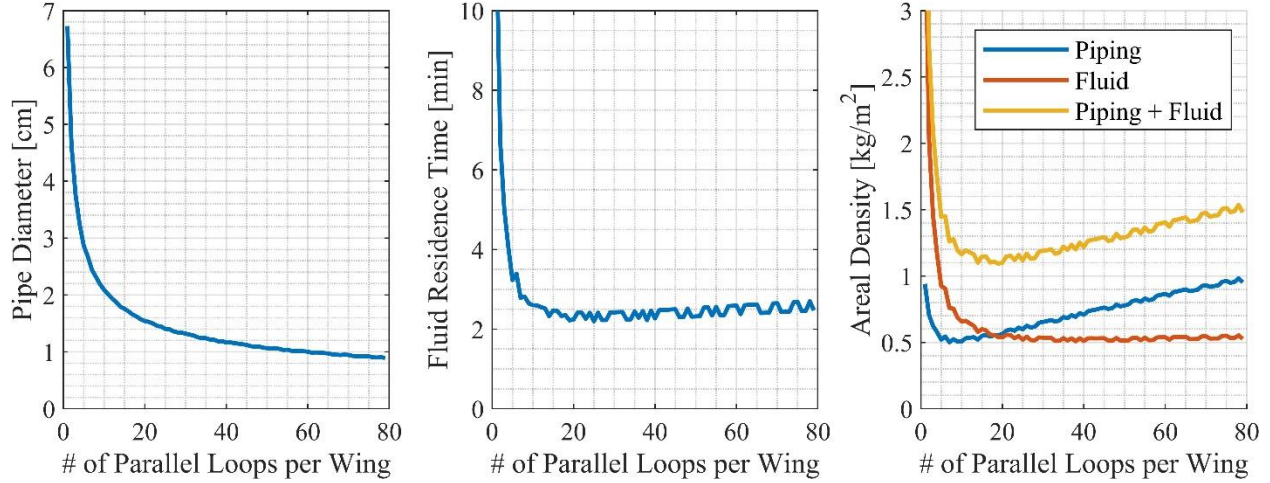


Fig. 10 Effect of the piping configuration on the PHRS.

C. High PHRS Inlet Temperature Trade

In Section IV, some methods for accommodating a PHRS inlet temperature greater than 550 K were suggested. Fig. 11 and Fig. 12 show how the resulting PHRS mass compares for each of these methods. Two different pre-cooled configurations were examined, one where all the pre-cooled panels were in series before splitting into separate loops in the rest of the PHRS and one where each parallel loop had its own section of pre-cooled panels. There are slightly fewer parallel loops possible in the series pre-cooled case since the pre-cooled panels subtract from the total number of heat pipe panels that can be put onto parallel loops. In the pre-cooled parallel case, there are half as many possible parallel loops because each panel must either have pumped NaK or heat pipes and when the maximum number of parallel loops are used, each loop has one of each type of panels.

In Fig. 11, the 1200 K PCIT reference case is examined. There are two lines representing the temperature-limited case, the first represents just the PHRS mass, and the second represents the PHRS mass as well as the additional mass accrued in the rest of the power conversion system due to operating the Brayton cycle away from the optimal PHRS inlet temperature. The tap-off case can be observed to have a slightly higher mass than the temperature limited case. This is because the tap-off case is radiating at the same temperature without the benefit of the more efficient cycle that comes with a lower power conversion sink temperature, and results in the tap-off configuration needing to reject more waste heat than the temperature limited case. The pre-cooled series case has a high mass for similar reasons as the series configuration in Fig. 10. The pre-cooled parallel loop case on the other hand has the lowest mass. In this method, the parallel loops reduce the mass flow through each pipe and prevent the pre-cooled pipes from being excessively large. Since for the maximum number of parallel fluid loops, each loop has one heat pipe panel and one NaK pipe panel, half of the PHRS is made up of pumped NaK pipes even though the PHRS inlet temperature is only slightly higher than 550 K for the 1200 K PCIT reference. The heat pipes being replaced are also the highest temperature and power and therefore the heaviest. Despite having a much lower mass, the pre-cooled parallel option has significant drawbacks. In particular, MMOD is a huge concern given that NaK pipes are now exposed on half the PHRS's area. A puncture in the NaK pipe can result in a significant performance decrease and additional shielding may be needed, negating the mass savings of the pre-cooled parallel case.

Fig. 12 shows the same comparison for a 1400 K PCIT reference cycle. The higher PCIT Brayton cycle is 10% more efficient, allowing for a smaller PHRS and a much lower mass overall. The primary difference is that there is much more variation between the cases, due to the optimum PHRS inlet temperature being much higher, at 625 K. Another significant difference is that the tap-off case has by far the highest mass. This is because both the inlet and outlet temperature to the PHRS is higher, and the mass flow through the PHRS will significantly increased since much more mass flow must be diverted to reach a temperature of 550 K.

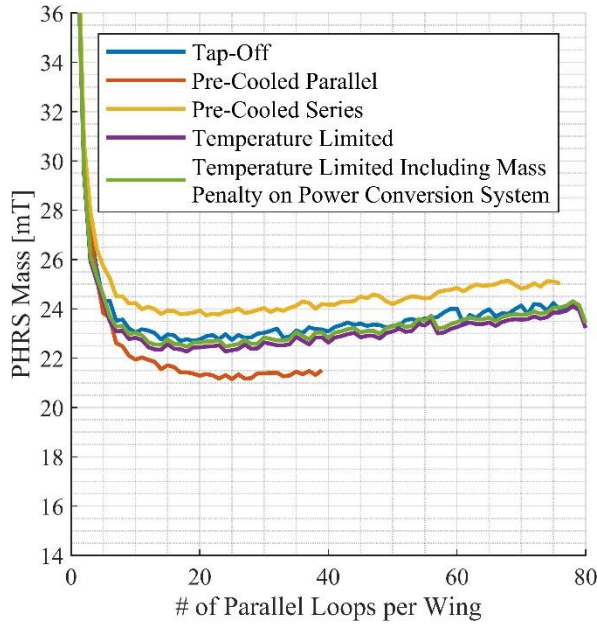


Fig. 11 High temperature inlet options for the 1200 K PCIT, 2 MW_e reference case.

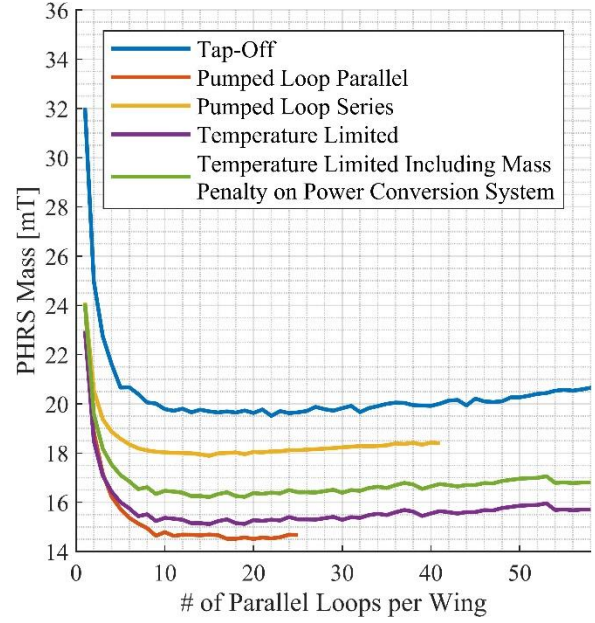


Fig. 12 High temperature inlet options for the 1400 K PCIT, 2 MW_e reference case.

4. All-Pumped-NaK Configuration

The pre-cooled parallel case in Fig. 11 had a very low mass, meaning that NaK pipes have a lower mass than the heat pipes. Given this result, another interesting configuration was if NaK pipes were used for the whole PHRS, and the heat pipes were eliminated altogether. While the mass of the all-pumped-NaK configuration was lower than the tap-off case, as seen in Fig. 13, it does not have the dramatic difference present between the tap-off and the pre-cooled parallel cases in Fig. 11. This is because the lower temperature heat pipes have much lower mass than their high temperature counterparts. The NaK pipes are lighter than high temperature heat pipes but not the low temperature heat pipes. When all the heat pipes are replaced with NaK pipes, it is not nearly as beneficial for the overall PHRS mass as just replacing the hotter heat pipes with NaK pipes. This, in combination with the MMOD concerns, makes the all-pumped-NaK configuration undesirable.

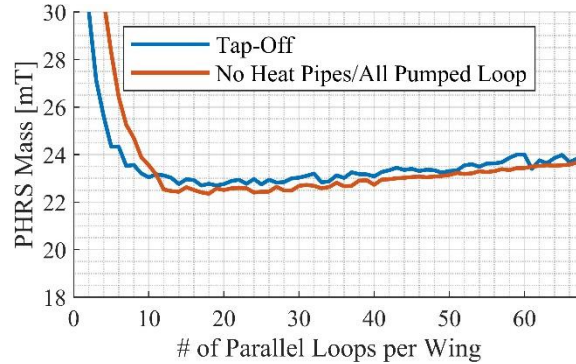


Fig. 13 Comparison to all pumped loop with no heat pipes (1200 K PCIT, 2 MW_e reference case).

D. Number of Wings

Changing the number of wings has two major effects on the PHRS, which are shown in Fig. 14. First, as more wings are added, the view factor decreases, and a larger PHRS is needed. Second, the ducting path decreases, resulting in a reduced areal density. These effects mostly cancel each other out in the overall PHRS mass, and the actual difference between the two, three, and four wing configurations is relatively small. The change in area effect wins out, resulting in a slightly lower mass for the two-wing configuration. Aside from mass there are several other factors that might influence the choice on number of wings, such as compactness and ease of folding up for launch.

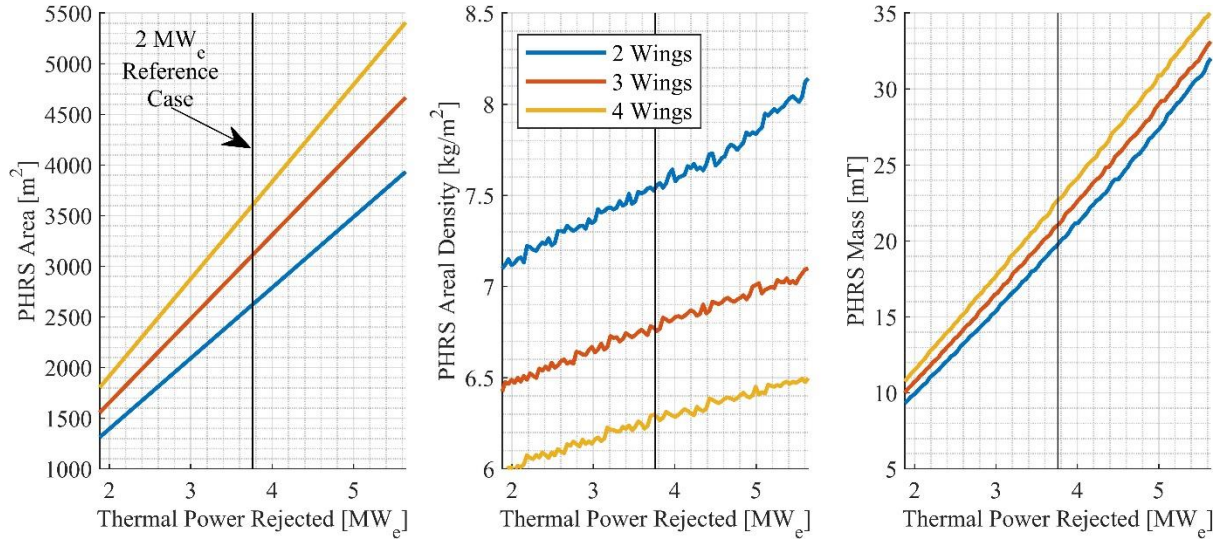


Fig. 14 Number of wings trade.

E. Half-Angle of Shadow Shield

The PHRS array on NEP vehicles is typically arranged in a triangular shape so they can be protected from the fission reactor's radiation behind a conical shadow shield. The shield is a truncated cone and the radii of base and top of this shape need to increase in order to produce a larger half-angle of radiation protection. This means that the shield mass increases quadratically with shield angle. There are other shield geometries that would likely be lighter for larger half-angles, however examination of these is left to future work. Fig. 15 shows both the conical shield and PHRS mass. The PHRS mass can be slightly reduced with a larger shield half-angle, due to larger view factor and shorter ducting path. This is significantly outweighed by the huge shield mass increase; meaning that is a larger half-angle is used a different shield geometry needs to be designed.

F. Heat Pipe Spacing

The temperature drop between the heat pipes is set by increasing the real radiator area by 10% over an ideal radiator with perfectly conductive panels. Maintaining this temperature drop when the heat pipe spacing is increased requires thicker panels in order to conduct the heat further. Fig. 16 shows how the area density of the panels and heat pipes varies with heat pipe spacing. As the heat pipe spacing increases, the power each heat pipe needs to transfer increases, which requires larger heat pipes. This is offset by needing far fewer heat pipes total, resulting in a decrease in the heat pipe areal density. The panel areal density is flat initially due to the implementation of a minimum panel thickness of 0.75 mm. At higher spacing the required thickness increases much past this value, driving up the panel areal density. For this given set of assumptions, there is an optimal heat pipe spacing of approximately 15 cm.

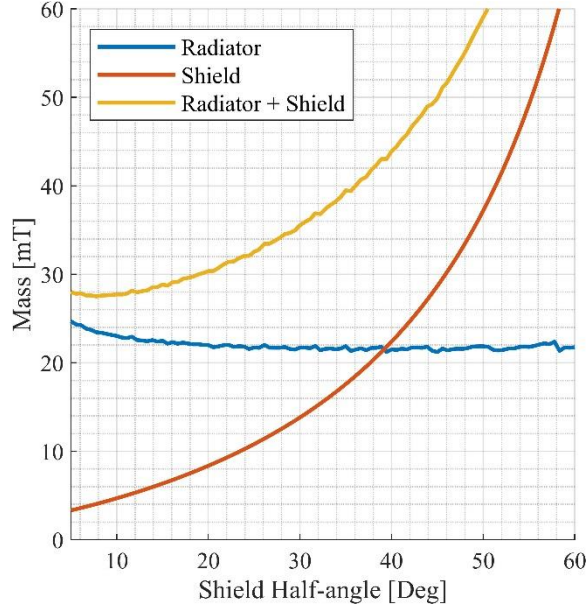


Fig. 15 Shadow shield half-angle trade (1200 K PCIT, 2 MW_e reference case).

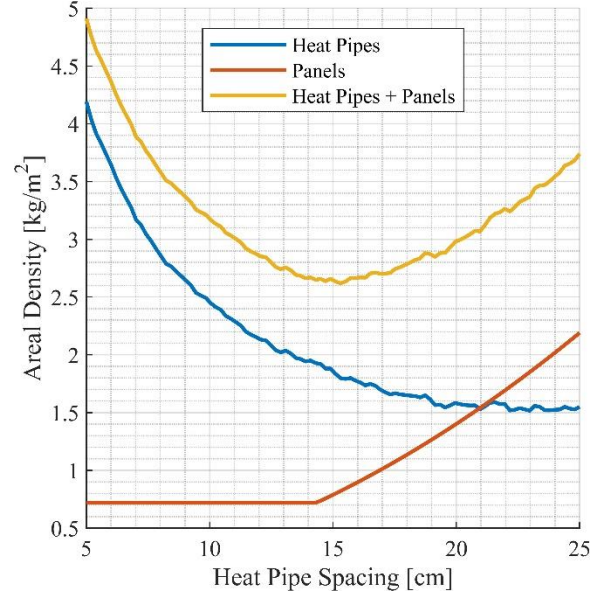


Fig. 16 Heat pipe spacing trade (1200 K PCIT, 2 MW_e reference case).

G. Length of Heat Pipe Condenser and Width of Panels

The width of the panels corresponds to the length of the heat pipe condenser. Two values for this parameter, 0.75 and 1 meter, are compared in Fig. 17. The average heat pipe mass is much smaller for the shorter heat pipes, but many more heat pipes are needed in the full PHRS. Overall, the shorter heat pipes resulted in a larger areal density as the increase in number of heat pipes proved to be the driving effect in the heat pipe areal density.

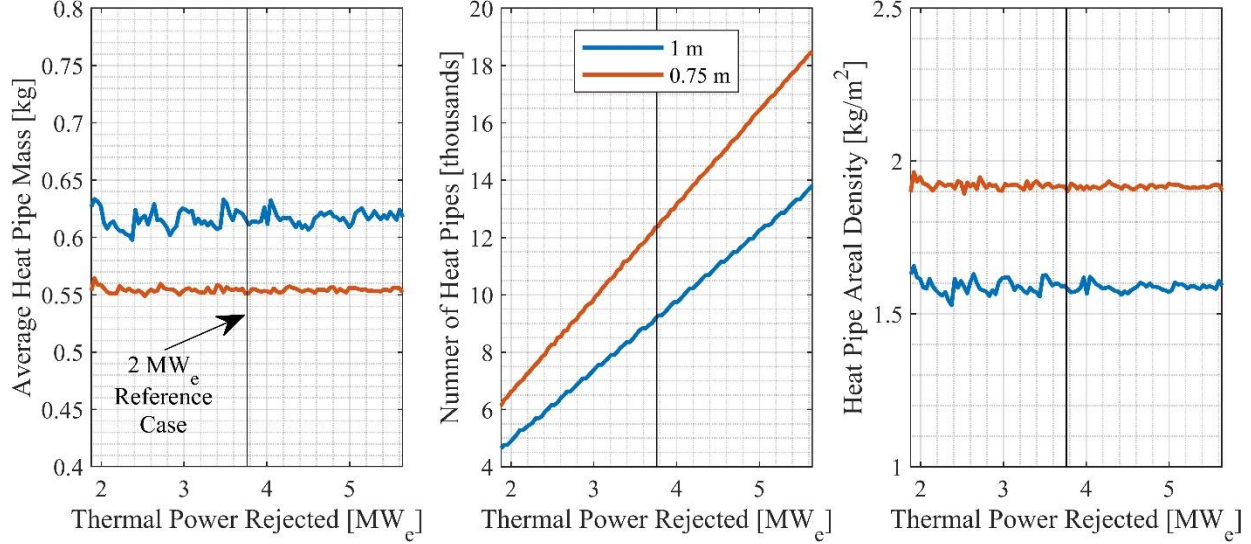


Fig. 17 Panel width trade.

H. Heat Pipe Discretization

In order to avoid having thousands of unique heat pipe geometries, the heat pipe masses were calculated at discrete temperatures and power levels. Since heat pipes operating between these points were rounded up to the next highest heat pipe geometry, the spacing of this discretization becomes an important variable in determining the total heat pipe mass. Table 2 shows the resulting areal densities for different numbers of unique heat pipe geometries. The heat pipe average mass increases when fewer unique geometries are used. This is because the gaps between the discrete points

are larger so the heat pipes will need to be rounded further to get to the next geometry. This results in more heat pipes being set at the higher temperature and power levels, which are heavier.

Table 2 Heat pipe discretization.

# Temperatures	# Powers	Average heat pipe mass
5	8	0.70 kg
5	4	0.74 kg
5	2	1.00 kg
3	8	0.86 kg
3	4	0.94 kg
3	2	1.54 kg

VI. Conclusion

There are numerous variables and assumptions that effect the mass of the PHRS. Many of those with a significant impact are examined and the results provide a representative cross section of the design space available to the PHRS. The goal of these trade studies is to demonstrate the importance of many of these assumptions, and to guide the NEP technology development effort.

Explicitly modeling the heat pipe physics and parallel loop piping layout has allowed the uncertainty in the mass of the heat rejection system to be significantly reduced. In addition, important design considerations were uncovered. One such finding is that for a multi-megawatt class vehicle capable of a crewed Mars mission, many parallel loops must be considered in order to reduce the mass of the fluid and piping. This was not necessarily the case for prior work such as JIMO and SP-100 which considered much lower power NEP vehicles. Another consideration is the PHRS temperature; conceptual NEP vehicle designs often assume that a higher temperature radiator is desirable due to the significant area reductions provided by radiative power being proportional to the difference between the temperatures raised to the fourth power according to Planck's radiation law. While these area and mass reductions can be realized, this design choice is more than just a mass trade. The technological hurdles of designing a PHRS capable of operating at this temperature are not trivial. Heat pipes in this temperature range are at a low TRL or otherwise have corrosion or degradation concerns and avoiding heat pipes entirely at this temperature presents its own drawbacks. These issues need to be considered when designing a PHRS for high temperature.

The PHRS mass model presented here is intended to demonstrate trends within the design space. While the general trends shown are expected to hold irrespective of realistic technology substitutions, the absolute values will vary as technology development continues and the model is refined. Several design considerations, such as having a panels and heat pipes on both sides of a central NaK flow, alternative reactor shield geometries, freezing/thawing of the working fluids, and albedo/radiator orientation considerations have yet to be examined. Due to this and the current low TRL of involved technologies, there is still significant uncertainty present in the model. The presented trade studies are not intended to prescribe a particular design choice, but rather to illustrate the effect of particular assumptions. As the technology maturation effort continues and prototypes are built developed, the model can be further improved.

Acknowledgments

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