Development of a Thermal Radiator Optimization Tool with Alternate Coolants

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Introduction

- Development of an analytical tool in Microsoft Excel™/VBA to size and optimize pumped fluid loop thermal radiators with alternate fluids is presented.

- This effort was supported by the NASA Space Technology Mission Directorate (STMD) through the *Lunar/Mars Thermal Habitat Sensitivity Study* (2024) and the *Advanced Thermal Control for Exploration Study* (2026).

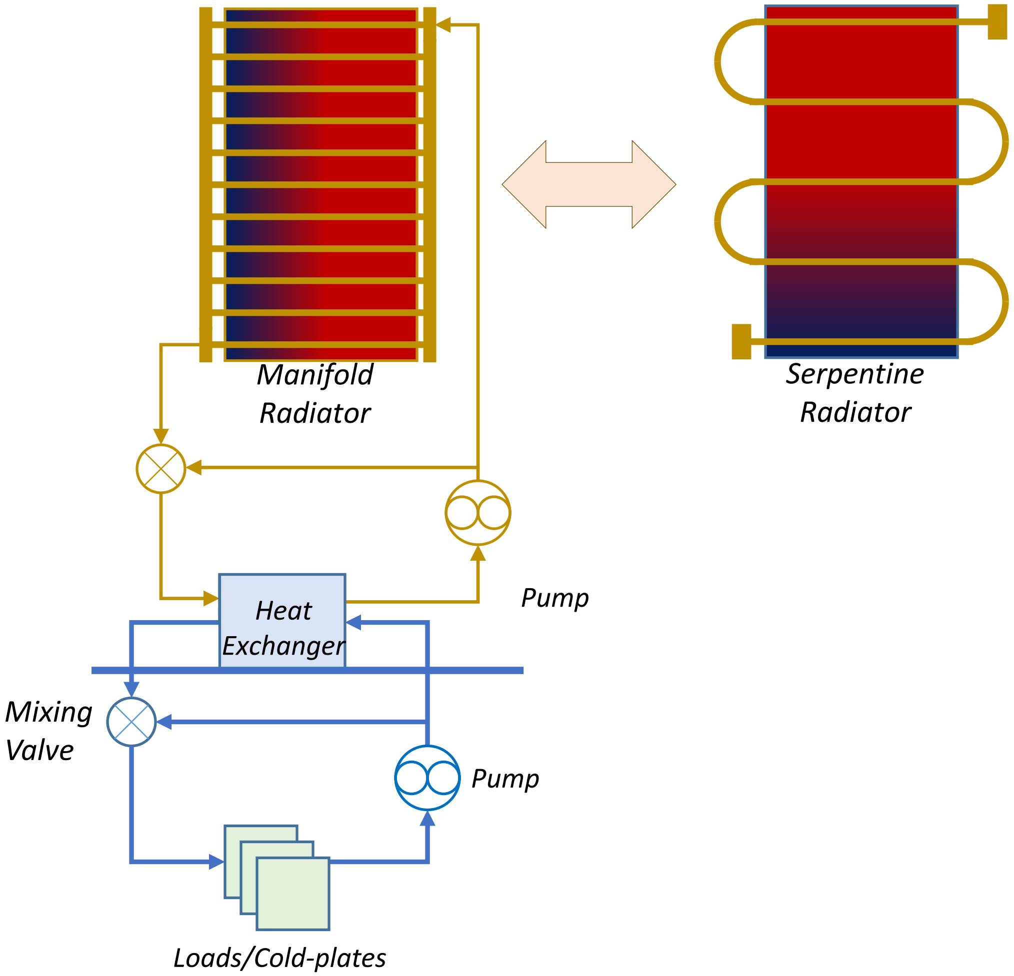
- Parametric results are presented in terms of Equivalent System Mass (ESM) to include radiator mass (subject to given inputs and boundary conditions) and the additional resources (mass and equivalent power) needed to pump fluid through the radiator.

- Sought to utilize more formulaic approaches for efficient computation to allow for greater parameterization of the radiator geometry, thermo-physical properties and fluids.

- Both manifold and serpentine flow radiators are included in the tool.

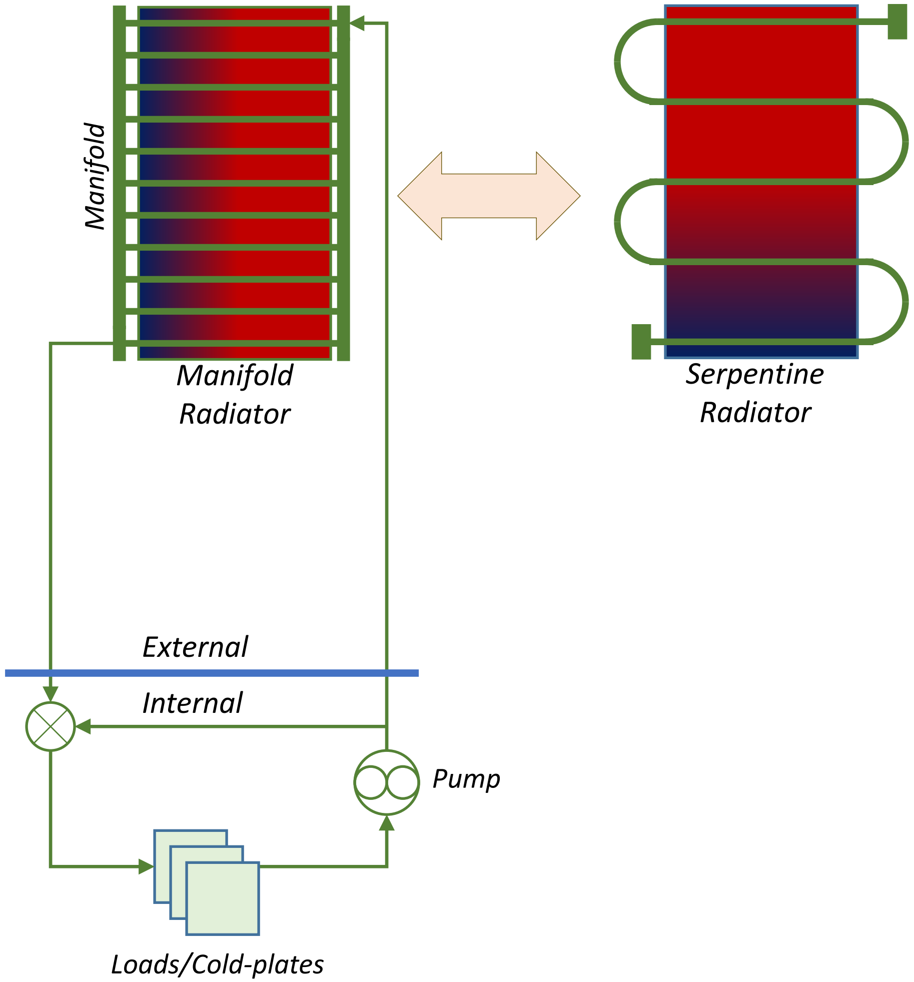
TCS Architecture

Single Phase Pumped Fluid Loop Options



Dual Loop Architecture

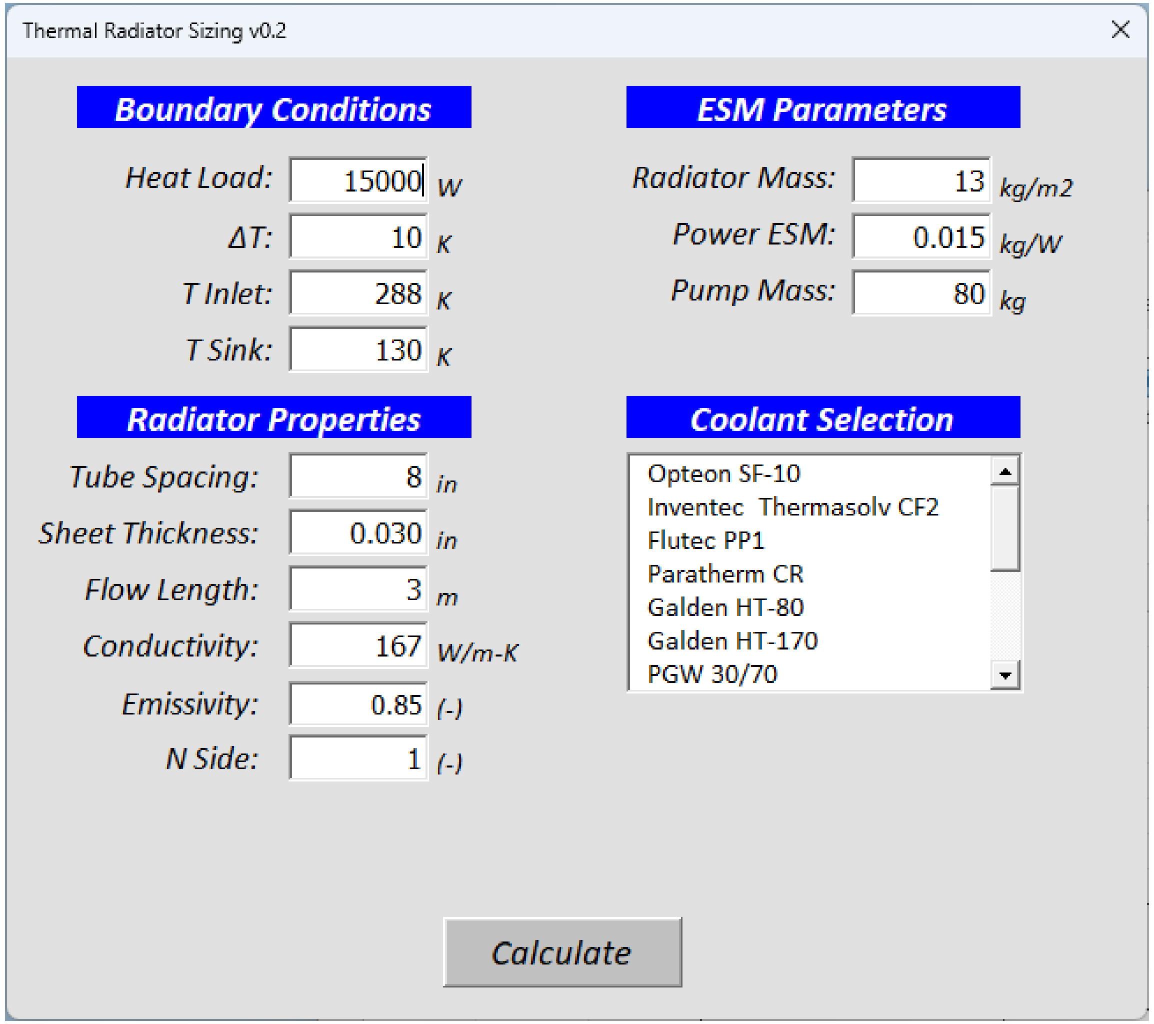
Different fluids internal and external. Allows for crew friendly internal and higher performing fluid externally.



Single Loop Architecture

Less complex and less resource intensive. Depends on availability of suitable fluid for both crew spaces and external environment.

Radiator Thermal Model Inputs



- The top-level input form for the application is shown. The user would prescribe the overall heat rejection, the desired inlet-to-outlet delta temperature across the radiator, the inlet temperature and the boundary sink temperature.
- The number of panels may be specified for a serpentine radiator (at a lower level) but has no effect on the manifold radiator ESM.
- Geometry and thermo-physical/optical properties may also be specified for the radiator while ESM may also be parameterized versus tube spacing and face sheet thickness.
- The radiator area density, ESM power factor and pump mass are also prescribed for the computation.
- Internal calculations that account for the radiator areal density changes relative to tube spacing and face sheet thickness may be substituted for the constant value.

Basic Computational Flow

Inputs

$$Q, T_1, \Delta T, T_b, S, w, \varepsilon, L$$

Fin Effect Efficiency

$$\gamma = \frac{2kwL\sqrt{2G\left[\left(\frac{T_{avg}^5}{5} - T_{avg}T_b^4\right) - \left(\frac{T_m^5}{5} - T_m T_b^4\right)\right]}}{LSN_{side}\varepsilon\sigma\left(T_{avg}^4 - T_b^4\right)}$$

Definition

$$G = \frac{N_{side}\varepsilon\sigma}{kw}$$

Gnielinski Correlation

$$Nu = \frac{\frac{f}{8}(Re - 1000)Pr}{1 + 12.7\sqrt{\frac{f}{8}}\left(Pr^{2/3} - 1\right)}$$

$$hA_{tube} = (Nu)\pi kL$$

Iterate

$$0 = \frac{hA\Delta T_f}{LS} - \sigma\varepsilon\gamma(T_0, w, S)\left(\left(T_0 - \Delta T_f\right)^4 - T_b^4\right)$$

ΔT_f

Radiator Area

$$A = \frac{\left[ln\left(\sqrt{\left(\frac{T_1 + T_b}{T_1 - T_b}\right)\left(\frac{T_2 - T_b}{T_2 + T_b}\right)}\right) - \left(atan\left(\frac{T_2}{T_b}\right) - atan\left(\frac{T_1}{T_b}\right)\right)\right]}{N_{side}\gamma\left(T_{avg}, w, S\right)\varepsilon\sigma T_b^3}\frac{Q}{T_2 - T_1}$$

Tubes

$$N_{tube} = \frac{A}{LS}$$

Parametric

$$Re = \frac{\rho u D}{\mu}$$

Tube Diameter and Velocity

$$\frac{Q}{N_{tube}c_p\Delta T_{rad}} = \rho u\frac{\pi D^2}{4}$$

$$D = \frac{Q}{N_{tube}c_p\Delta T_{rad}}\left(\frac{4}{Re\mu\pi}\right)$$

$$u = \frac{Q}{N_{tube}c_p\Delta T_{rad}} \Big/ \rho\frac{\pi D^2}{4}$$

Delta Pressure

$$\Delta P = \frac{1}{2}f_f(Re)\rho u^2\frac{L}{D}$$

Pump Work

$$\dot{W} = \frac{\dot{V}\Delta P}{Efficiency}$$

Inputs

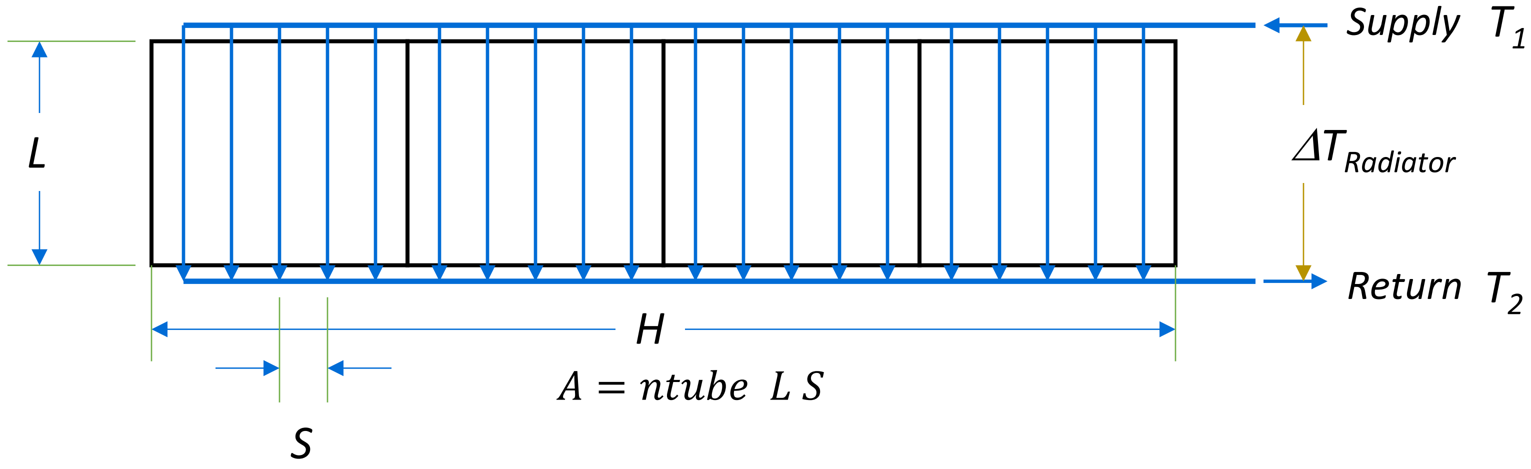
$$Areal\ Density, Power\ Factor, Pump\ Efficiency$$

Result

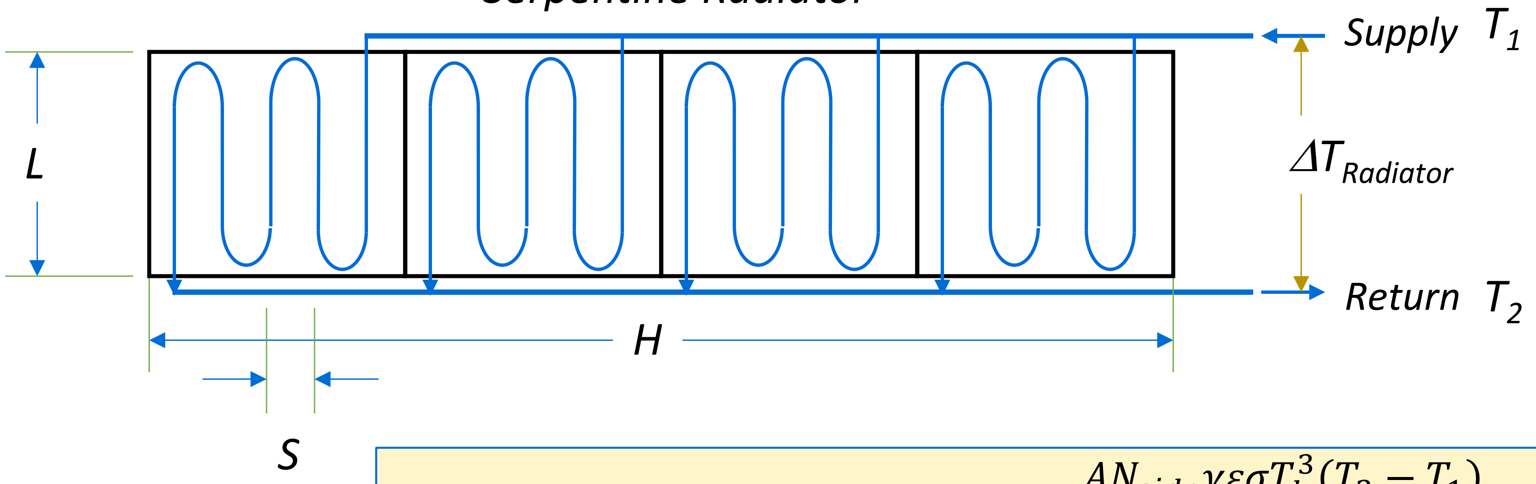
$$ESM = Mass_{Radiator} + Mass_{Pump} + Power_{Pump}\alpha$$

Radiator Thermal Model Description

Manifold Radiator



Serpentine Radiator

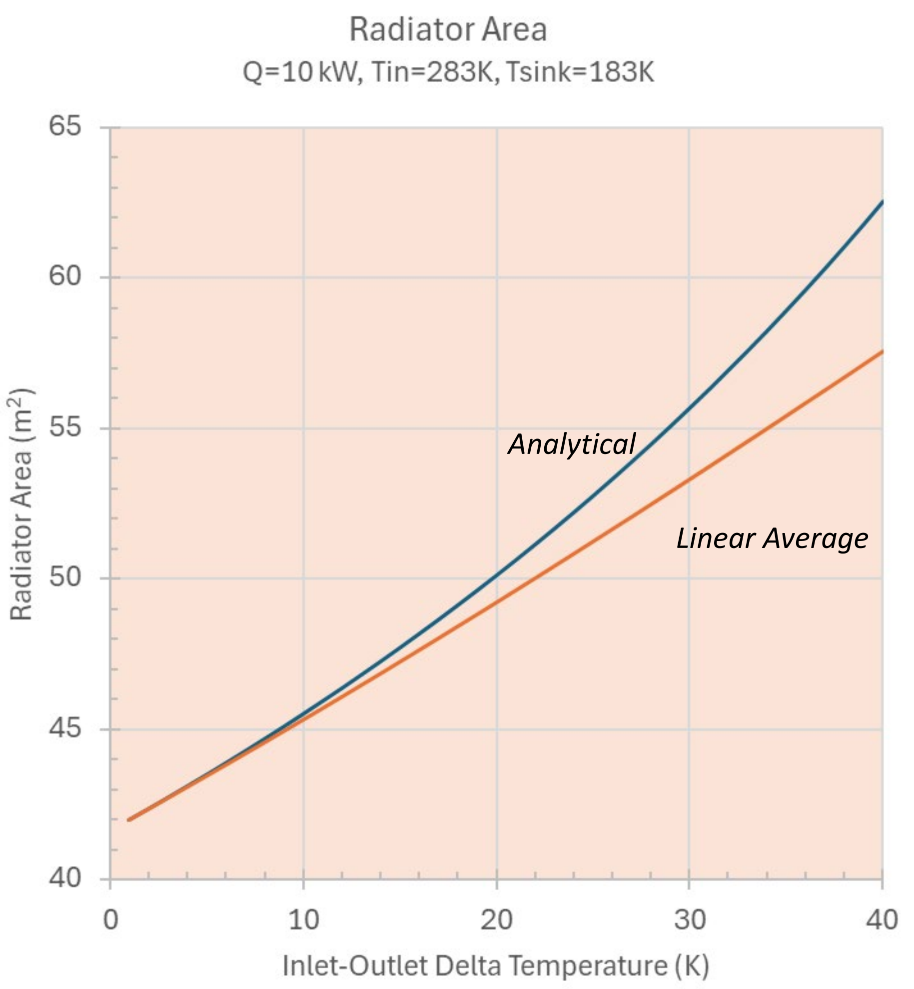


$$Q = \dot{m} c_p (T_2 - T_1) = \frac{A N_{side} \gamma \varepsilon \sigma T_b^3 (T_2 - T_1)}{\left[ln\left(\sqrt{\left(\frac{T_1 + T_b}{T_1 - T_b}\right)\left(\frac{T_2 - T_b}{T_2 + T_b}\right)} \right) - \left(atan\left(\frac{T_2}{T_b}\right) - atan\left(\frac{T_1}{T_b}\right) \right) \right]}$$

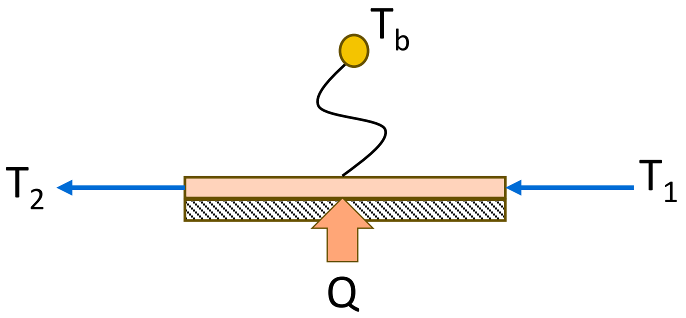
- An analytical solution for radiator performance, based upon the inlet and outlet temperatures, sink temperature and optical properties, is used in the tool.
- The radiator performance solution is derived in the backup slides.
- The radiator performance is scaled by a fin effect efficiency, γ.
- A computed film coefficient is deducted from the fluid supply and return temperatures to account for heat transfer from the fluid to tube wall (typically a 1-5% impact overall)

T_1	*Inlet Temp*
T_2	*Outlet Temp*
T_b	*Sink Temp*
γ	*Fin Efficiency*
N_{side}	*N Sides*

Non-Linear Temperature Distribution through the Radiator



- The analytical radiator solution provides realistic predictions for large ΔT's across the radiator where performance can deviate from linear behavior.



T_1	*Inlet Temp*
T_2	*Outlet Temp*
T_b	*Sink Temp*
γ	*Fin Efficiency*
N_{side}	*N Sides*

Analytical Solution

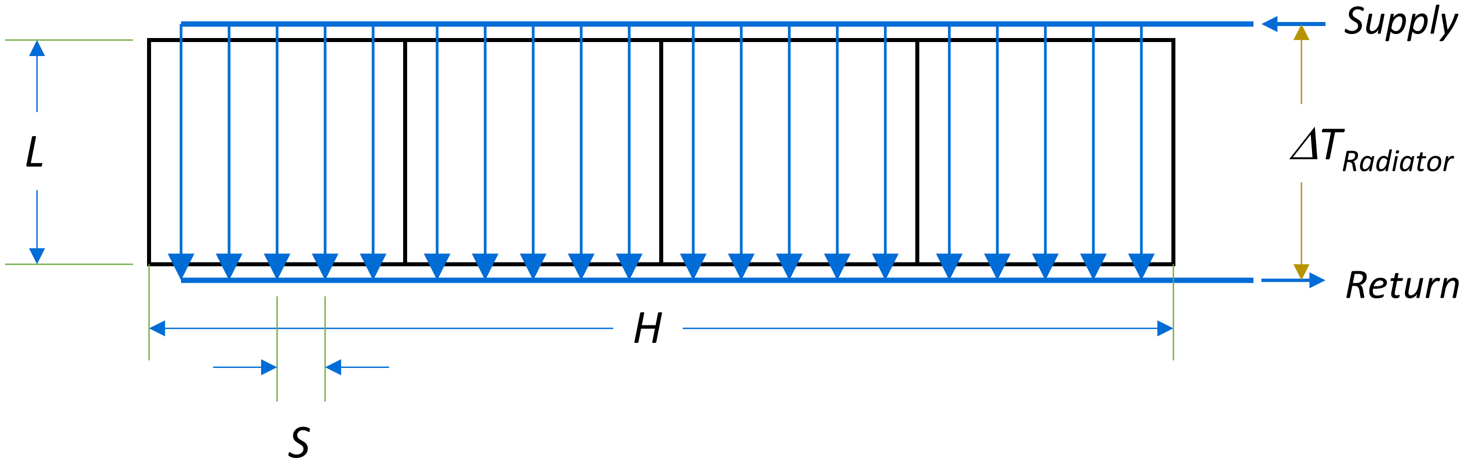
$$A = \frac{\left[ln\left(\sqrt{\left(\frac{T_1+T_b}{T_1-T_b}\right)\left(\frac{T_2-T_b}{T_2+T_b}\right)}\right) - \left(atan\left(\frac{T_2}{T_b}\right) - atan\left(\frac{T_1}{T_b}\right)\right)\right]}{N_{side}\gamma\varepsilon\sigma T_b^3}\frac{Q}{T_2-T_1}$$

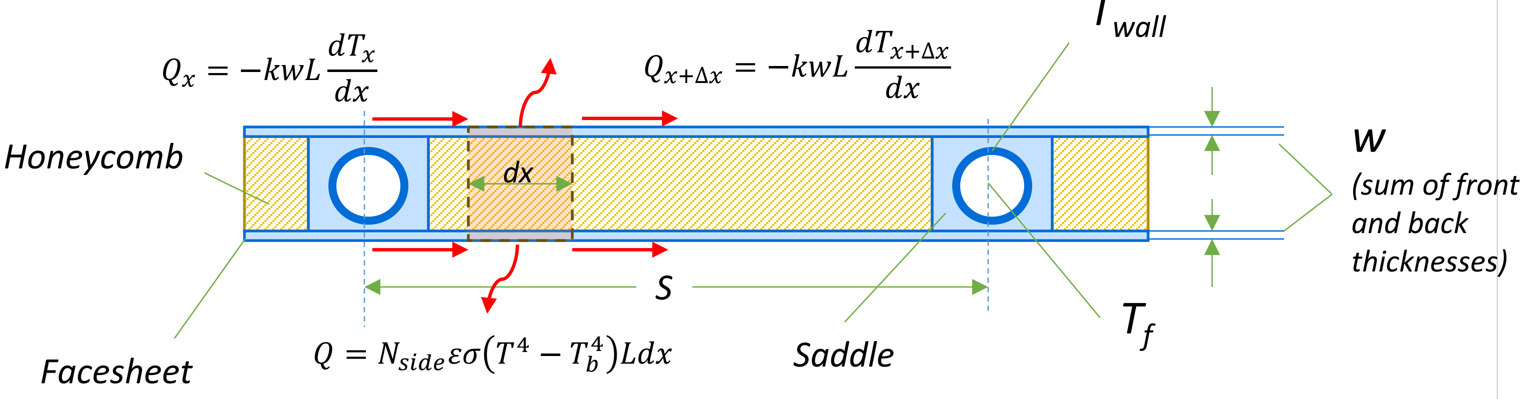
Linear Average

$$A = \frac{Q}{N_{side}\gamma\varepsilon\sigma\left[\frac{T_1^4+T_2^4}{2} - T_b^4\right]}$$

Manifold Radiator Fin Effect Model Description

Manifold Radiator





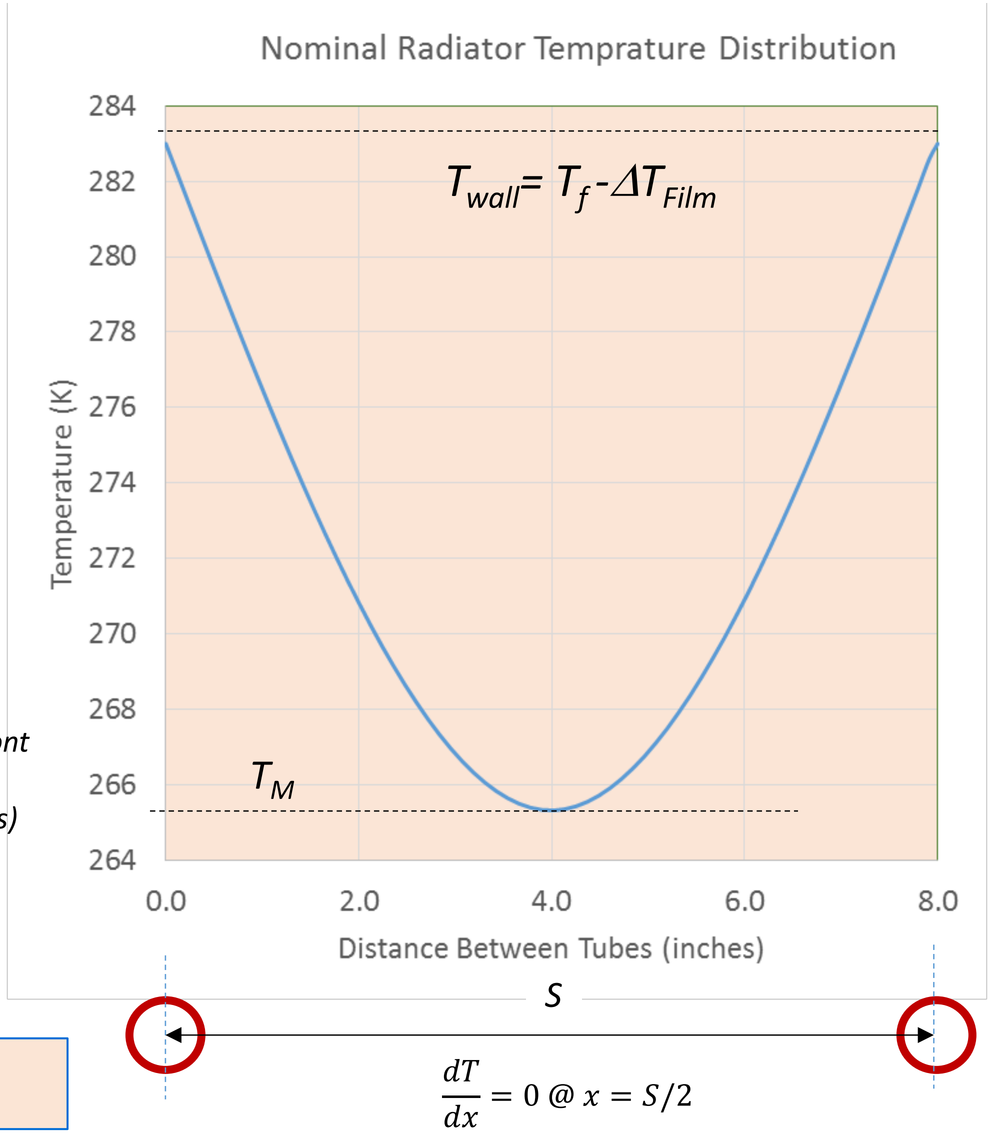
Energy Balance

$$Q_{x+\Delta x} - Q_x = N_{side}\varepsilon\sigma\left(T^4 - T_b^4\right)Ldx$$

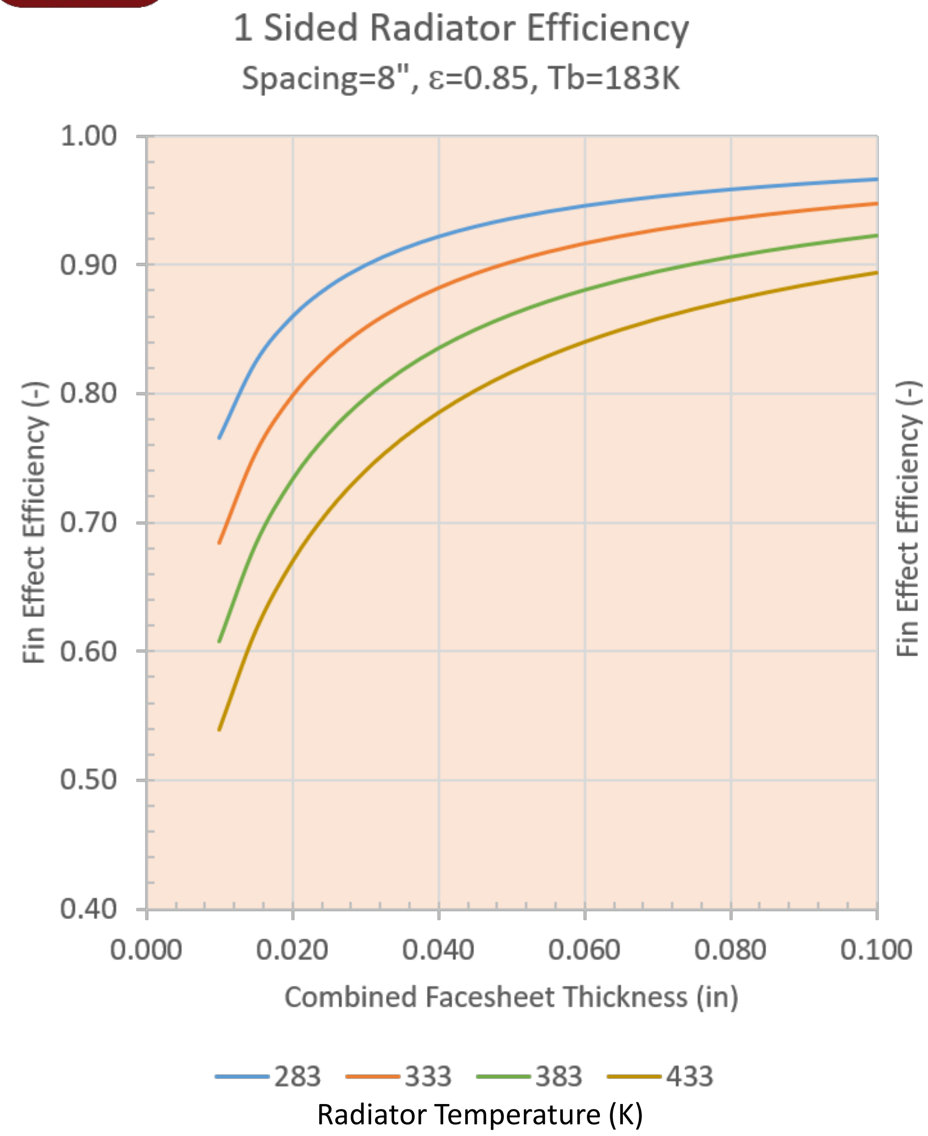
$$\frac{-kwL\frac{dT_{x+\Delta x}}{dx} - \left(-kwL\frac{dT_x}{dx}\right)}{Ldx} = N_{side}\varepsilon\sigma\left(T^4 - T_b^4\right)$$

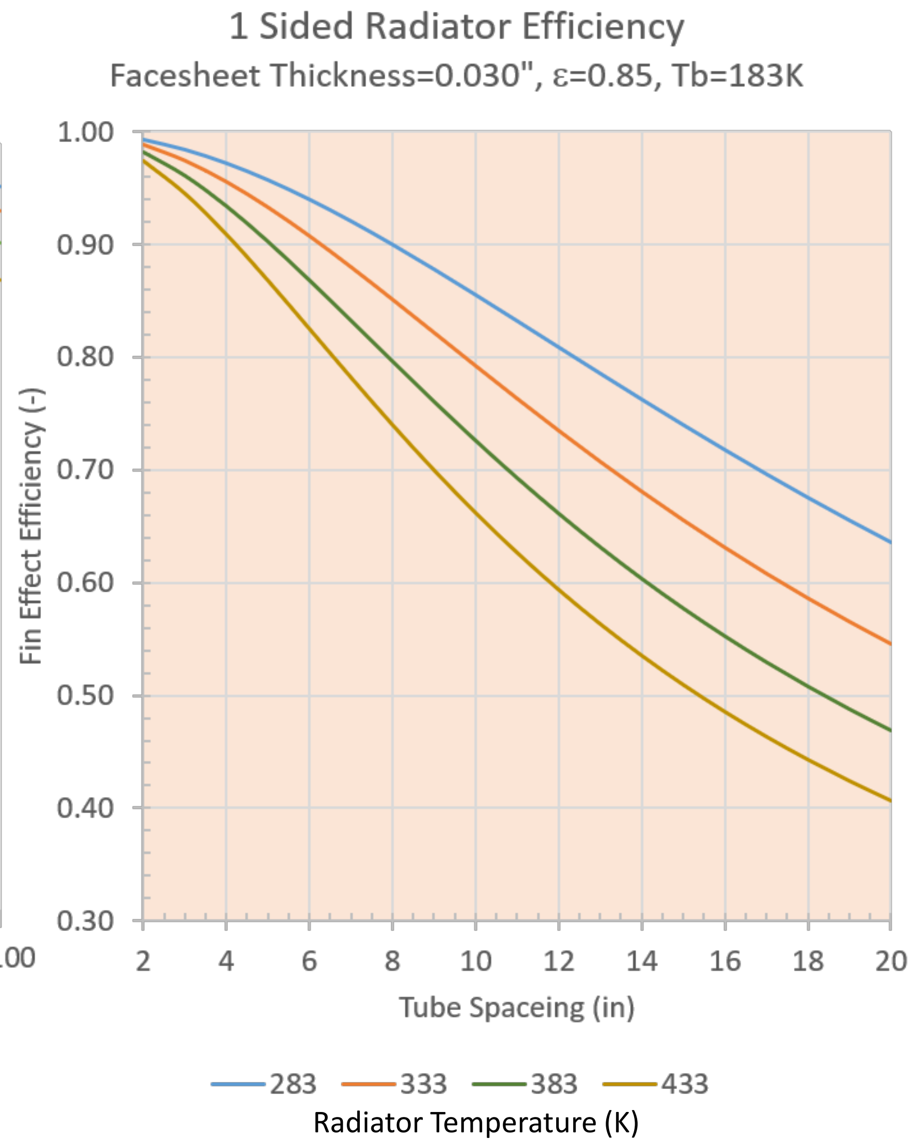
Governing Equation

$$\boxed{\frac{d^2T}{dx^2} = \frac{N_{side}\varepsilon\sigma}{kw}\left(T^4 - T_b^4\right)}$$



Thermal Radiator Fin Effect Efficiency



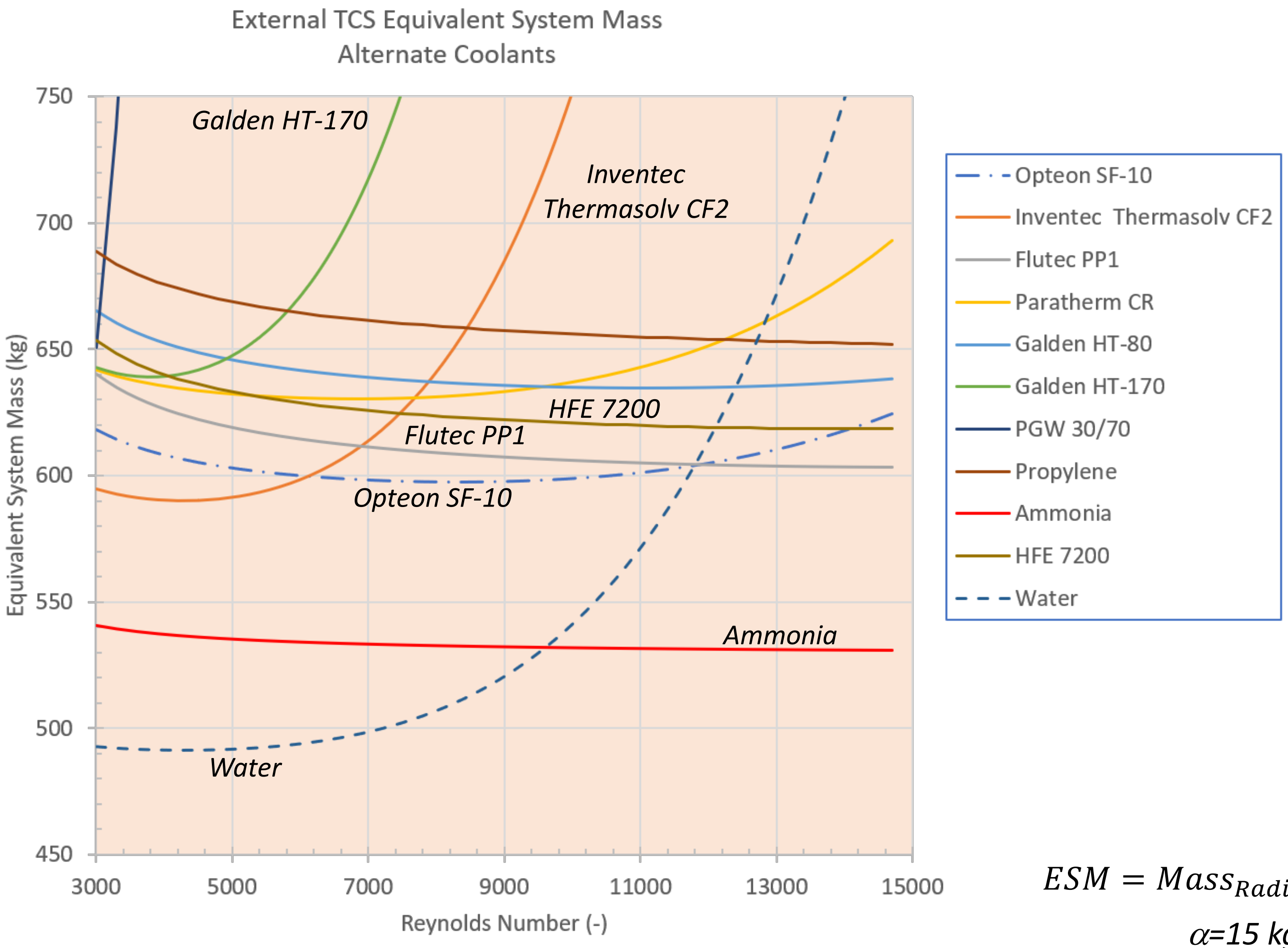


- The thermal radiator fin effect efficiency due face-sheet heat conduction and radiation between flow tubes can be estimated based upon tube spacing, face sheet thickness, optical properties and temperature.
- The fin effect efficiency is defined as the ratio of actual heat rejection to ideal rejection of an isothermal radiator cross-section.
- The radiator becomes less efficient with greater temperature and tube spacing.
- Derivation of the fin effect efficiency formulation is provided in the backup slides.

Equivalent System Mass Parametric-Manifold

Manifold Radiator

Q=15 kW, Aluminum Face-sheets, S=8”, w=0.030”, Tb=130K, N Sides=2



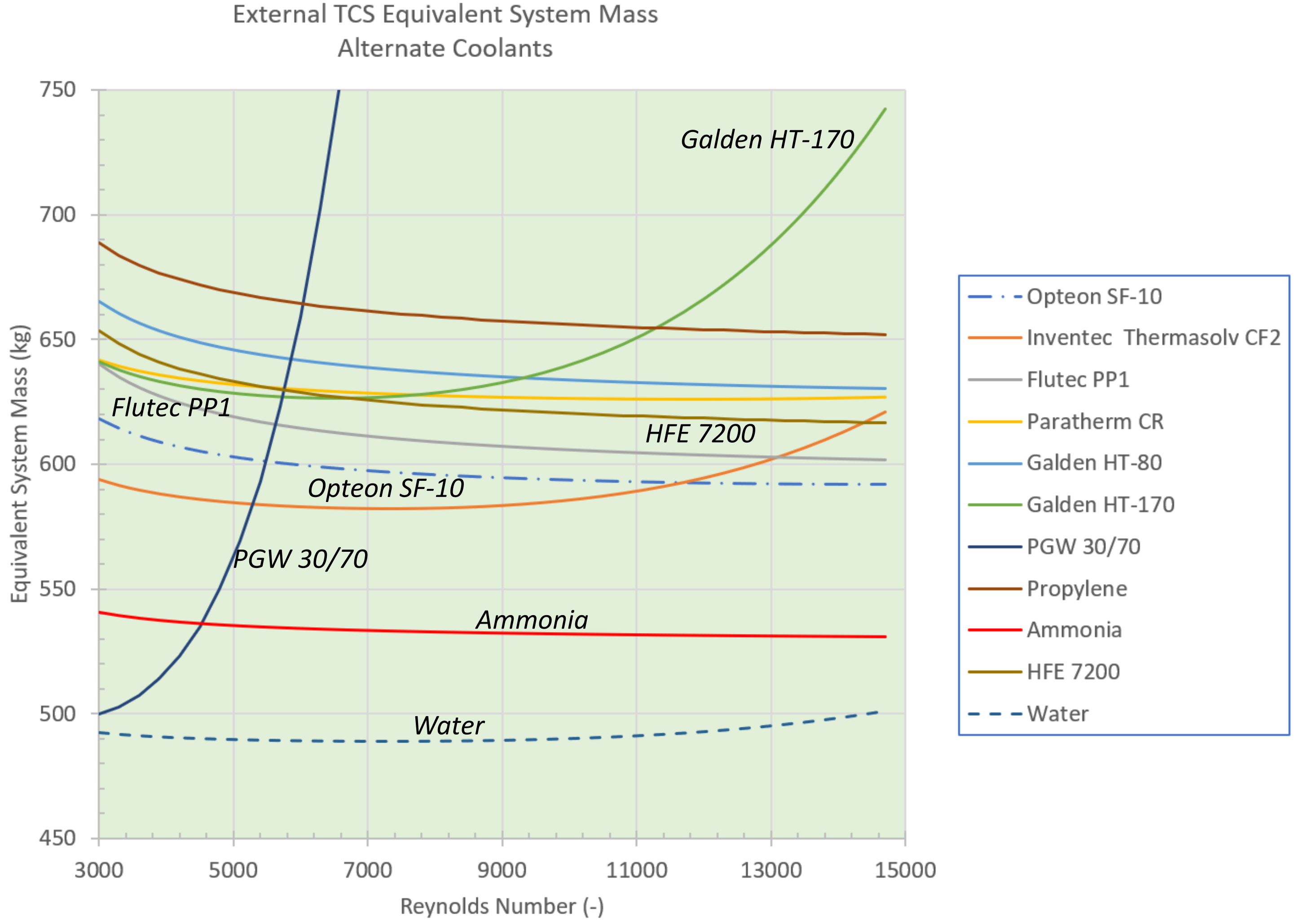
- The non-dimensional Reynold’s number provides a convenient independent variable to parameterize radiator performance for multiple fluids.
- The Equivalent System Mass for the total radiator area is derived for a prescribed radiator Q, inlet to outlet ΔT and geometry.
- Many fluids exhibit an optimum point over a Reynolds number range representing transitional to turbulent flow.
- Initially the radiator benefits from increased heat transfer with increasing Reynolds number.
- But the increase of ESM at high Reynolds is due to greater pump power to overcome pressure drop as the tube diameter shrinks.

$$ESM = Mass_{Radiator} + Mass_{Pump} + Power_{Pump}\alpha$$

α=15 kg/kW for Photovoltaic Power

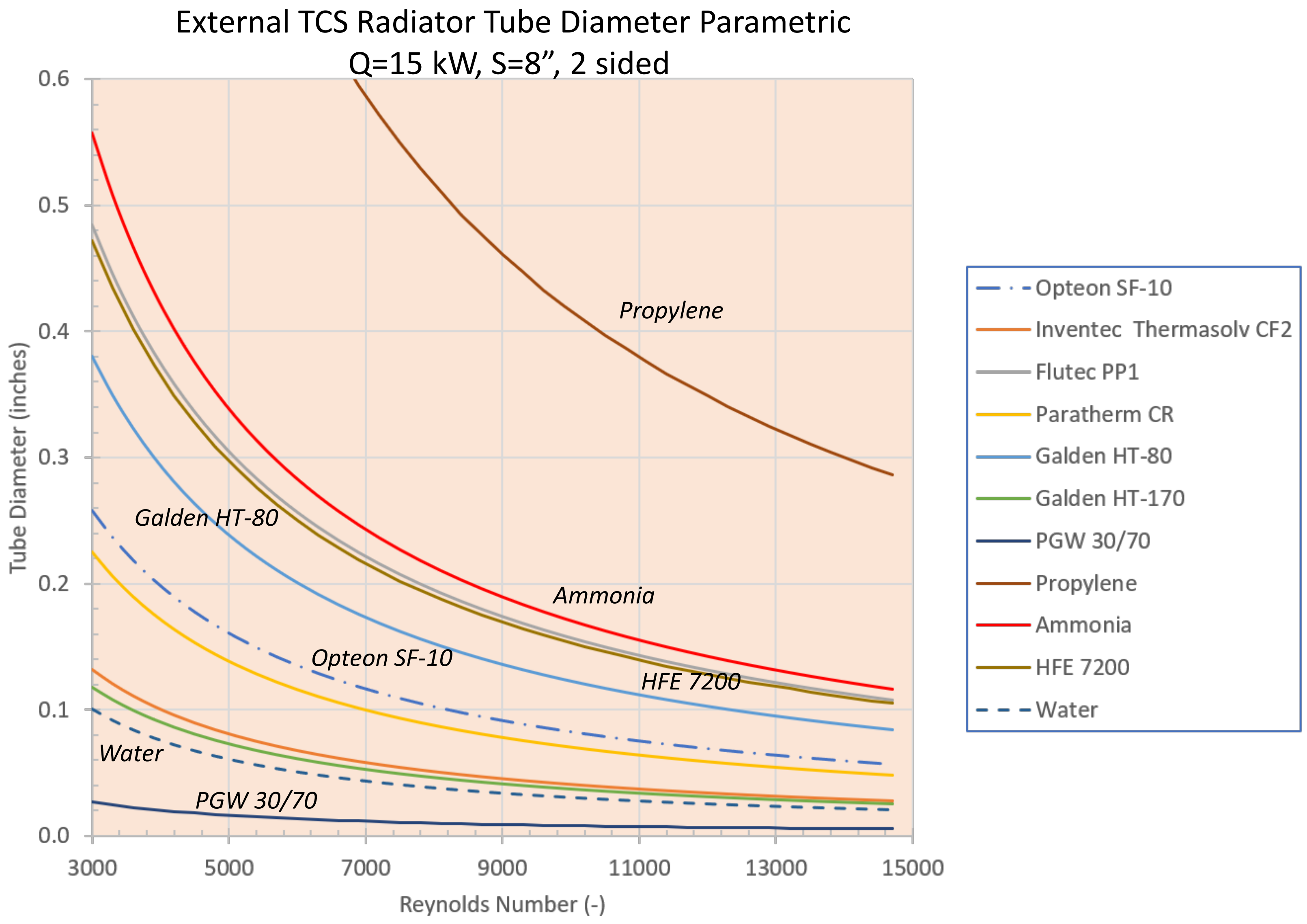
Equivalent System Mass Parametric-Serpentine

Serpentine Radiator
Q=15 kW, Aluminum Face-sheets, S=8”, w=0.030”, Tb=130K
N Sides=2, N Panel=6



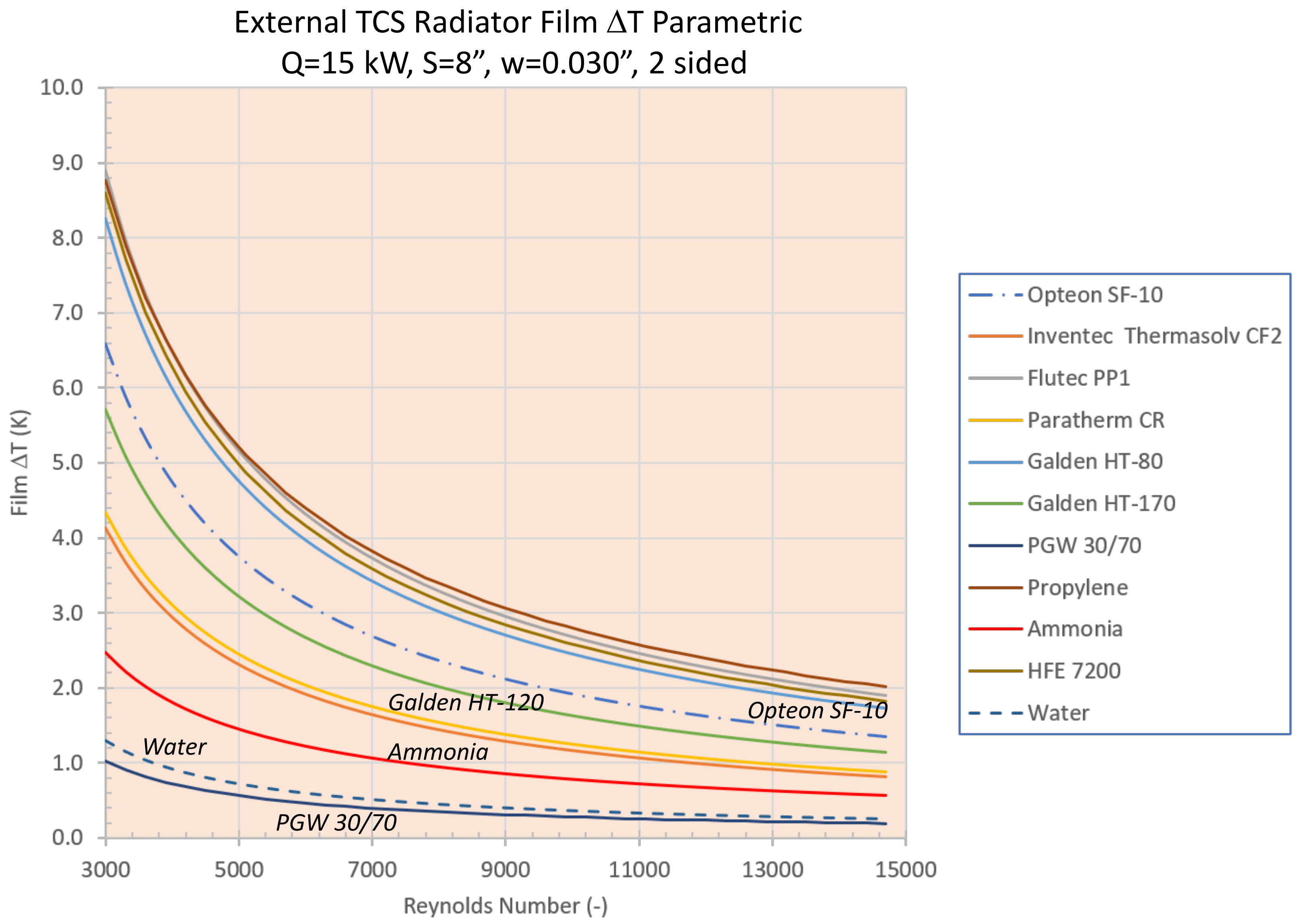
- The serpentine radiator topology allows for a much larger “sweet spot” for many coolants as well as the utilization of coolants not viable for manifold radiators.
- At low Reynolds number, PGW outperforms all of the coolants except water.
- The tubing diameter increase for serpentine over manifold radiators may be enough to mitigate pressure drop despite the longer flow path through the serpentine loop (L/D).
- The results shown are for a radiator subdivided into 6 panels but the overall ESM may decrease proportionally to the number of radiator panels.
- Fewer panels could be accomplished with larger panels (but the same overall area) or smaller panels jumpered to perform as larger panels.

Manifold Radiator Tube Diameter



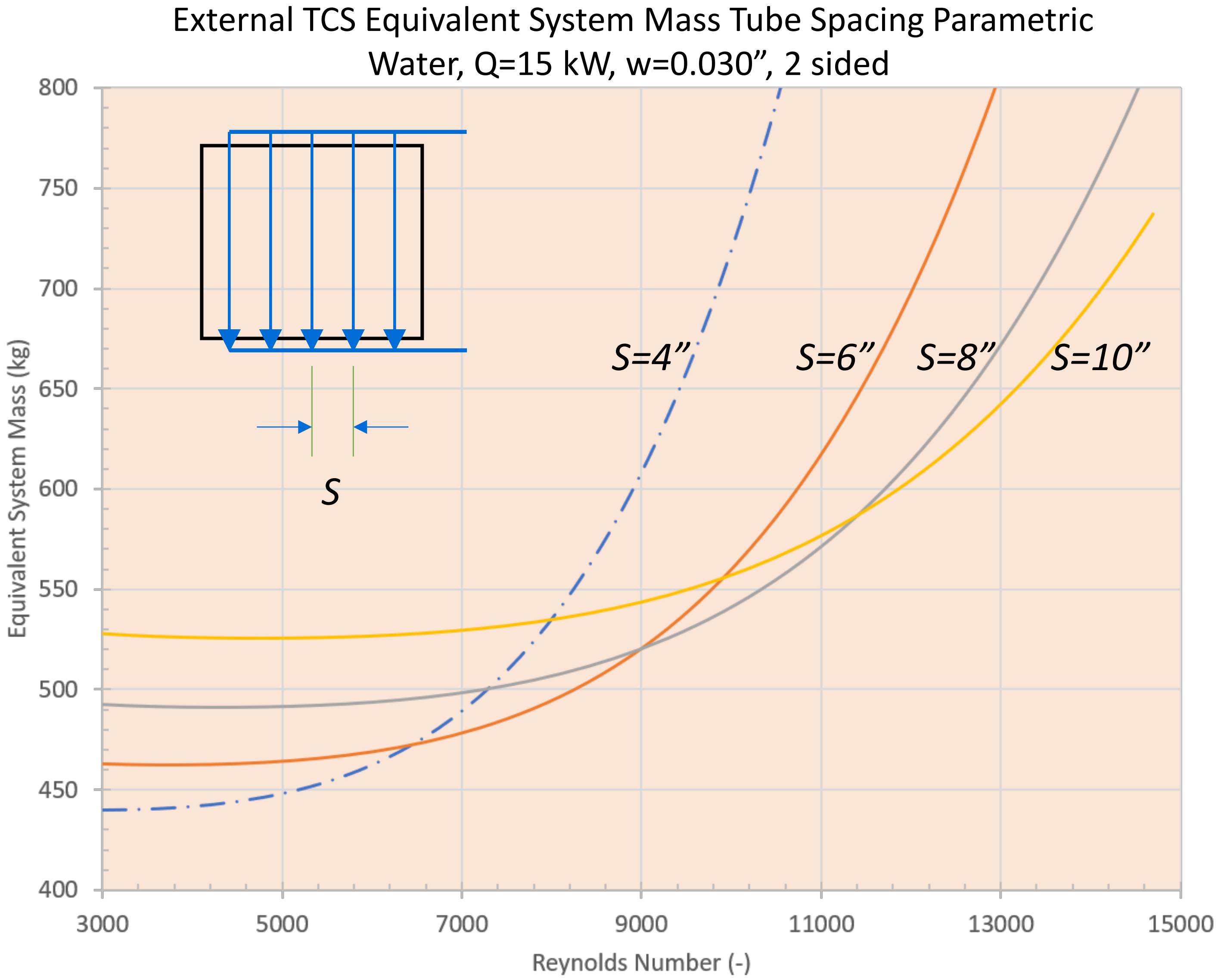
- A parametric of manifold tube diameter to achieve the given Reynolds number is shown.
- The mass flow rate is different amongst all of the fluids shown (due to specific heat) but higher viscosity fluids may require smaller tube diameters to maintain flow above Re=3000.
- Propylene and Ammonia have the lowest viscosities of all the fluids in the database.
- For a manifold radiator it may be difficult for a high viscosity/high specific heat fluid like PGW to maintain a high Reynolds number due to the combined effect.

Manifold Radiator Film ΔT Parametric



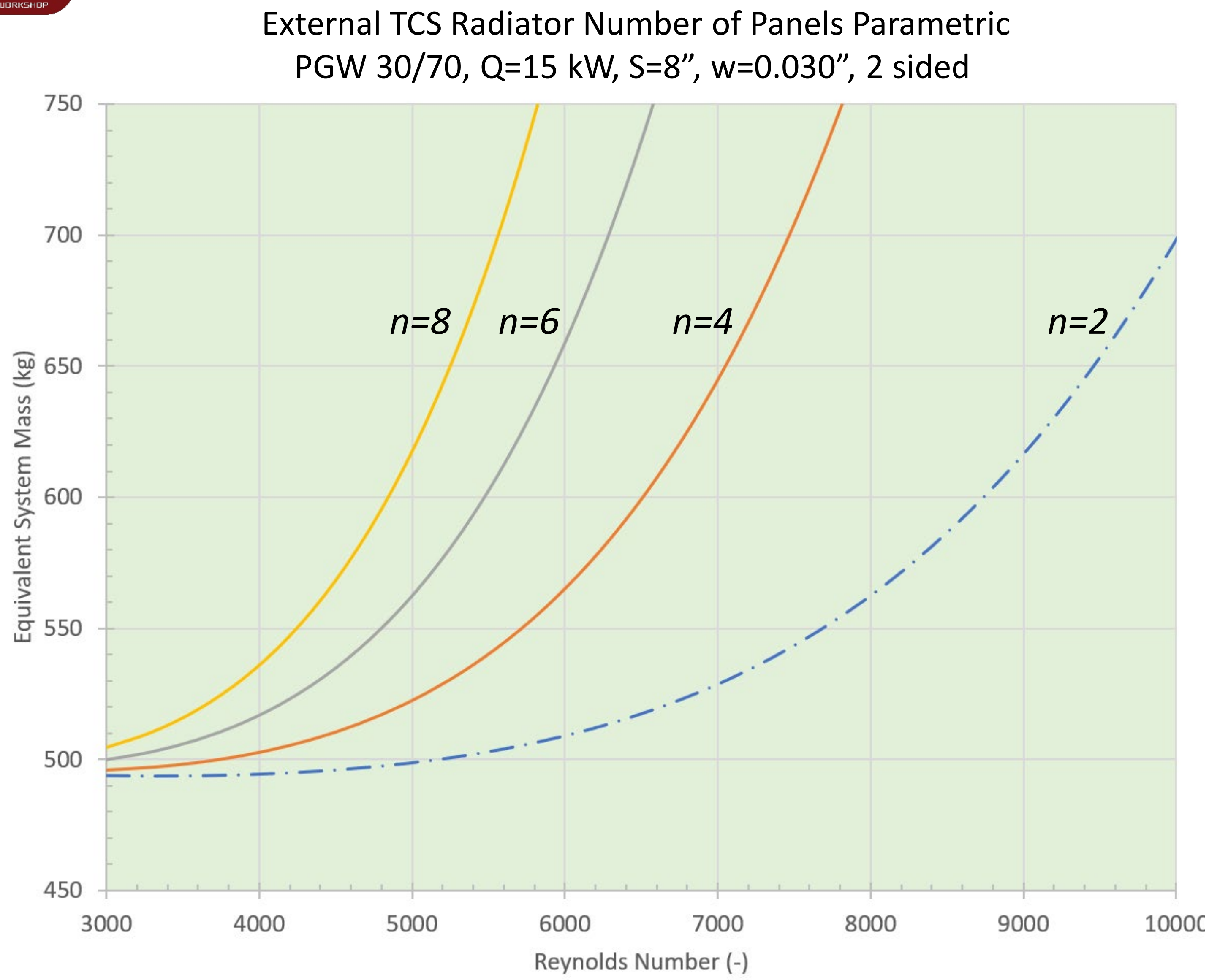
- A parametric of manifold tube film delta temperature (between the fluid and wall) is shown.
- The fluids with the greatest thermal conductivity (Water, PGW, Ammonia) provide the lowest film delta temperatures.
- Delta film temperatures are somewhat high at low Reynolds number due to the reduced heat transfer coefficients.
- High film delta temperatures, greater than 2-3K, may begin to significantly affect habitat radiator performance (i.e., average rejection temperature < 283K).

Manifold Radiator Tube Spacing Parametric



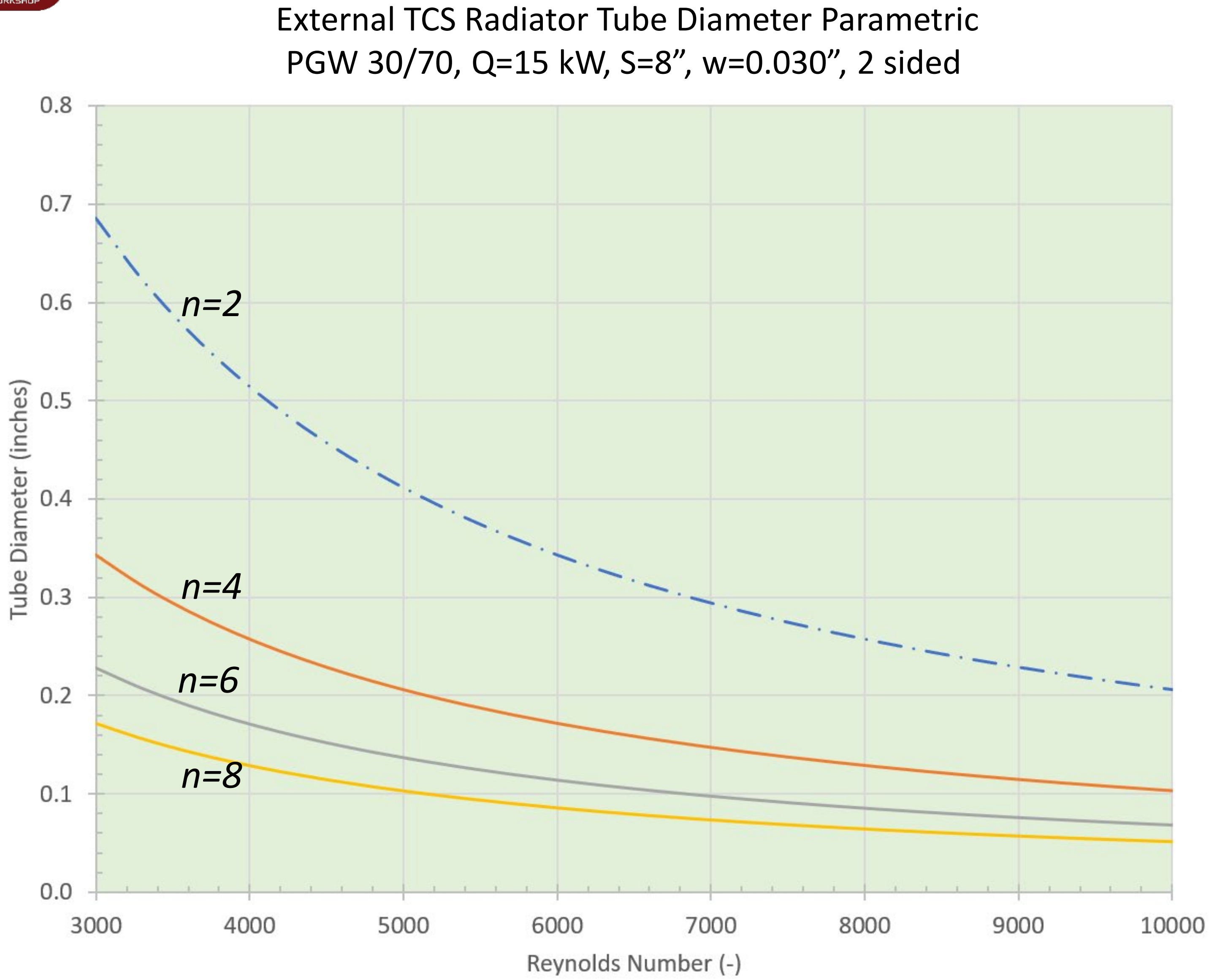
- A parametric showing manifold tube spacing to achieve the given Reynolds number is shown for water.
- Decreasing the number of tubes (i.e., increasing spacing) delays the onset of the ESM pump resource effect due to an increased diameter and lower velocity per tube.
- Primary rationale for the lower radiator ESM at low Reynold's number with more tubes may be the increase in efficiency.
- Assumes that the saddle mass is reduced with more tubes. If not the case, the benefit of more tubes at low Reynolds could be cancelled out.

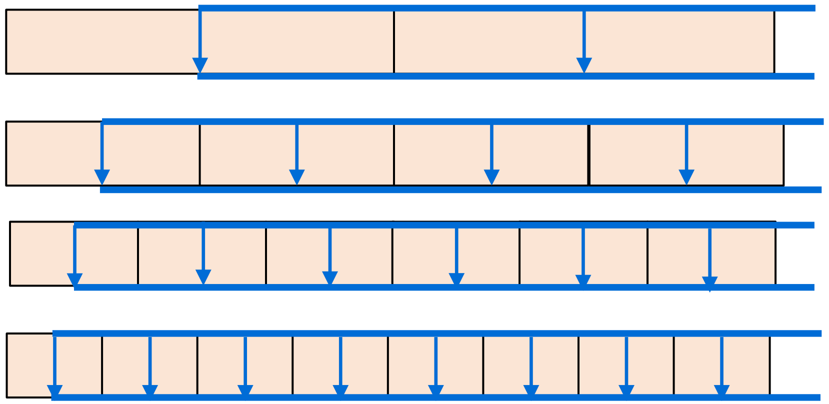
Serpentine Radiator Number of Panels Parametric



- A parametric showing serpentine ESM versus number of panels is presented for PGW.
- Reducing the number of panels significantly delays the onset of the ESM pump resource penalty for PGW creating a larger “sweet spot” over much of the Reynolds number space.
- This increases the flow rate per panel but is offset by the attendant tube diameter increase resulting from fewer panels.

Serpentine Radiator Tube Diameter Parametric





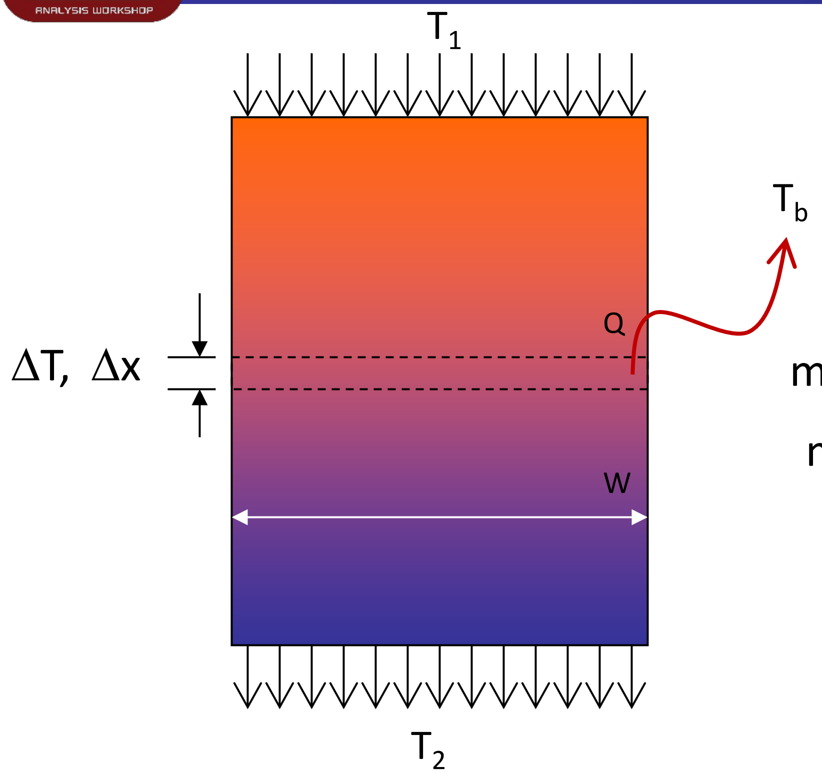
- But are the tube diameters associated with the panel decrease too large for serpentine radiators?
- It appears that above a moderate Reynolds number threshold that the tube diameter becomes very manageable.

Summary, Future Plans and Acknowledgements

- A thermal radiator modeling tool is developed to optimize the design of a radiator based on capacity, rejection temperature, geometry, topology and coolant.
- The non-dimensional Reynolds number provides a convenient independent variable upon which to parameterize radiator performance and design features.
- Serpentine radiators may offer advantages for high viscosity fluids.
- Future plans might include correlation of the modeling tool to detailed model results or empirical data.
- Temperature dependent coolants would also be beneficial to consider applications outside typical habitat temperature limits, like NSFP or ISRU.
- Special thanks to the NASA Space Technology Mission Directorate (STMD) and numerous co-workers and colleagues who have supported this effort through two separate studies.

Backup Slides

Manifold Radiator Equation Derivation



$$mcp\ \Delta T = \varepsilon\sigma(T^4 - Tb^4)W\Delta x$$

$$mcp\ dT = \varepsilon\sigma(T^4 - Tb^4)dA$$

Energy balance over a strip of the radiator

$$\int_{T1}^{T2} \frac{1}{T^4 - Tb^4}\, dT = \int_{0}^{A} \frac{\varepsilon\sigma}{mcp}\, dA$$

$$\int_{T1}^{T2} \frac{1}{T^4 - Tb^4}\, dT$$

Integrate over the entire area and simplify…

$$\frac{-1}{4}\cdot\frac{-\ln(-Tb + T2) + \ln(Tb + T2) + 2\cdot \operatorname{atan}\left(\frac{T2}{Tb}\right)}{Tb^3} + \frac{1}{4}\cdot\frac{-\ln(-Tb + T1) + \ln(Tb + T1) + 2\cdot \operatorname{atan}\left(\frac{T1}{Tb}\right)}{Tb^3}$$

$$\frac{-1}{4\cdot Tb^3}\cdot\left[\ln\left[\frac{(T2 + Tb)}{(T2 - Tb)}\right] + 2\cdot\operatorname{atan}\left(\frac{T2}{Tb}\right)\right] + \frac{1}{4\cdot Tb^3}\cdot\left(\ln\left(\frac{T1 + Tb}{T1 - Tb}\right) + 2\cdot\operatorname{atan}\left(\frac{T1}{Tb}\right)\right)$$

$$\frac{e\cdot s\cdot A}{m\cdot cp} = \frac{1}{2\cdot Tb^3}\cdot\left[\ln\left(\sqrt{\frac{T1 + Tb}{T1 - Tb}\cdot\frac{T2 - Tb}{T2 + Tb}}\right) - \left(\operatorname{atan}\left(\frac{T2}{Tb}\right) - \operatorname{atan}\left(\frac{T1}{Tb}\right)\right)\right]$$

$$A = \frac{1}{2\cdot Tb^3}\cdot\left[\ln\left(\sqrt{\frac{T1 + Tb}{T1 - Tb}\cdot\frac{T2 - Tb}{T2 + Tb}}\right) - \left(\operatorname{atan}\left(\frac{T2}{Tb}\right) - \operatorname{atan}\left(\frac{T1}{Tb}\right)\right)\right]\cdot\frac{mcp}{\varepsilon\sigma}$$

Tb > 0!

or $Q=mcp(T_2-T_1)$

$$A = \frac{1}{2\cdot Tb^3}\cdot\frac{1}{\varepsilon\sigma}\cdot\left[\ln\left(\sqrt{\frac{T1 + Tb}{T1 - Tb}\cdot\frac{T2 - Tb}{T2 + Tb}}\right) - \left(\operatorname{atan}\left(\frac{T2}{Tb}\right) - \operatorname{atan}\left(\frac{T1}{Tb}\right)\right)\right]\cdot\frac{Q}{T2 - T1}$$

There are six variables and two equations (five unknowns). Use $Q=mcp(T_2-T_1)$ to obtain desired final form of the area equation.

Doesn't account for film coefficient on radiator tube wall or fin effect efficiency but factors are added in the final formulation.

A	radiator area
Tb	sink temp
T1	inlet temp
T2	outlet temp
Q	total heat radiated
mcp	flow rate x specific heat

Fin Effect Solution – Find Tm

- If T_m can be found, then a simplified Taylor series can be used to find the temperature distribution (and ultimately the fin effect efficiency) between radiator tubes.
- We know that all the odd numbered derivatives in the Taylor series will be zero at $T=T_m$.
- Expanding a 4th order Taylor series about x=S/2:

$$T_O = T_M + \cancelto{0}{\frac{S}{2}\frac{dT_M}{dx}} + \frac{\left(\frac{S}{2}\right)^2}{2!}\frac{d^2T_M}{dx^2} + \cancelto{0}{\frac{\left(\frac{S}{2}\right)^3}{3!}\frac{d^3T_M}{dx^3}} + \frac{\left(\frac{S}{2}\right)^4}{4!}\frac{d^4T_M}{dx^4}$$

$$\frac{dT_M}{dx} = 0$$

$$\frac{d^2T_M}{dx^2} = \frac{N_{side}\varepsilon\sigma}{kw}\left(T_M^4 - T_b^4\right)$$

$$\frac{d^3T_M}{dx^3} = \frac{N_{side}\varepsilon\sigma}{kw}\left(4T_M^3\cancelto{0}{\frac{dT_M}{dx}}\right) = 0$$

$$\frac{d^4T_M}{dx^4} = \frac{N_{side}\varepsilon\sigma}{kw}\left(12T_M^2\cancelto{0}{\frac{dT_M}{dx}} + 4T_M^3\frac{d^2T_M}{dx^2}\right)$$

$$\boxed{T_O = T_M + \frac{S^2}{8}\frac{d^2T_M}{dx^2} + \frac{S^4}{384}\frac{d^4T_M}{dx^4}}$$

- T_o is known so it is possible to solve this equation for T_m.

Fin Effect Solution – Find Tm

- Let's simplify the equation:

$$\boxed{G = \frac{N_{side}\varepsilon\sigma}{kw}}$$

$$\frac{d^2T_M}{dx^2} = G\left(T_M^4 - T_b^4\right)$$

$$\frac{d^4T_M}{dx^4} = G^2\,4T_M^3\left(T_M^4 - T_b^4\right)$$

$$T_O = T_M + \frac{S^2}{8}\frac{d^2T_M}{dx^2} + \frac{S^4}{384}\frac{d^4T_M}{dx^4}$$

$$T_O = T_M + \frac{S^2}{8}G\left(T_M^4 - T_b^4\right) + \frac{S^4}{384}G^2\,4T_M^3\left(T_M^4 - T_b^4\right)$$

$$\boxed{T_O = T_M + G\left(T_M^4 - T_b^4\right)\left(\frac{S^2}{8} + \frac{S^4}{96}GT_M^3\right)}$$

- Solving the following equation for f(x)=0 will yield T_m.
- This can be accomplished via Newton's method.

$$\boxed{f(x) = x - T_O + G\left(x^4 - T_b^4\right)\left(\frac{S^2}{8} + \frac{S^4}{96}Gx^3\right)}$$

$$\boxed{f'(x) = 1 + G4x^3\left(\frac{S^2}{8} + \frac{S^4}{96}Gx^4\right) + G\left(x^4 - T_b^4\right)\left(\frac{S^4}{32}Gx^2\right)}$$

$$\boxed{x_{n+1} = x_n - \frac{f(x_n)}{f'(x_n)}}$$

Fin Effect Efficiency Derivation

$$P = \left(\frac{dT}{dx}\right)^2$$

$$\frac{dP}{dx} = 2\frac{dT}{dx}\frac{d^2T}{dx^2}$$

$$2\frac{dT}{dx}\frac{d^2T}{dx^2} = 2\frac{dT}{dx}G\left(T^4 - T_b^4\right)$$

Substitution trick by Professor Francis Wessling at UAH on similar problem.

$$\frac{dP}{dx} = 2\frac{dT}{dx}G\left(T^4 - T_b^4\right)$$

$$dP = 2G\left(T^4 - T_b^4\right)dT$$

$$\int_0^P dP = 2G\int_{T_m}^{T_o}\left(T^4 - T_b^4\right)dT$$

$$P = \left(\frac{dT}{dx}\right)^2 = 2G\left[\left(\frac{T_o^5}{5} - T_oT_b^4\right) - \left(\frac{T_m^5}{5} - T_mT_b^4\right)\right]$$

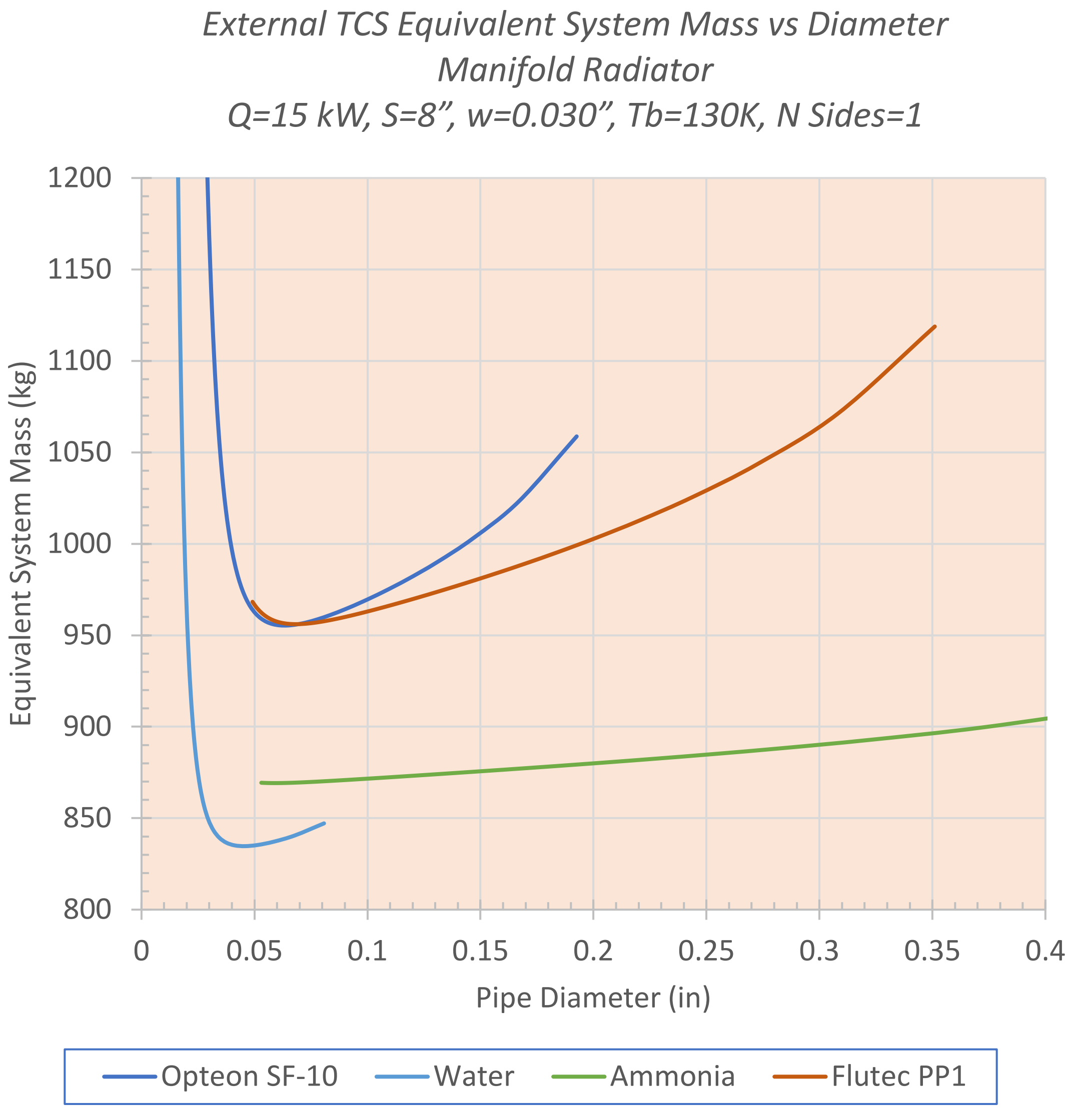
$$Q_{Reject} = -2kwL\frac{dT}{dx}$$

$$Q_{Reject} = 2kwL\sqrt{2G\left[\left(\frac{T_o^5}{5} - T_oT_b^4\right) - \left(\frac{T_m^5}{5} - T_mT_b^4\right)\right]}$$

$$Q_{Ideal} = LSN_{side}\varepsilon\sigma\left(T^4 - T_b^4\right)$$

$$\boxed{Efficiency = \frac{Q_{Reject}}{Q_{Ideal}}}$$

ESM Parametric vs Pipe Diameter



- Results aren’t normalized for the radiator ESM vs pipe diameter.
- The diameter range for each fluid reflects a suitable Reynolds number range which is different for each fluid.
- Mass flow through the radiator correlates strongly to specific heat for each fluid.
- Lower pipe diameters correspond to greater Reynolds number.

Fluid Property Database

Coolant	Chemistry	Warm* Viscosity	Density	Specific Heat	Thermal Conductivity	Pour/Freeze Point	Vapor Pressure	Prandtl #
(-)	(-)	Pa-s	kg/m^3	J/kg-K	W/m-K	^oC	kPa	(-)
Opteon SF-10	methoxytridecafluoroheptene	0.00112	1580	1240	0.077	-90	2.9	18.0
Inventec Thermasolv CF2	perfluoro-alcene	**0.00239**	1770	1177	0.115	-110	3.0	24.5
Flutec PP1	perfluorohexane (C6F14)	0.00066	1718	1090	0.065	-90	29.4	11.1
Paratherm CR	diethylbenzene	0.00083	833	1968	0.140	-110	3.7	11.7
Galden HT-80	perfluorinated polyether	0.00096	1690	963	0.065	-110	14.0	14.2
Galden HT-170	perfluorinated polyether	**0.00319**	1770	963	0.065	-97	0.1	47.3
PGW 30/70	H_2O+PG	**0.00362**	1031	3965	0.440	-13		32.6
Propylene	C_3H_6	0.00011	614	2424	0.113	-185	104.0	2.4
Ammonia	NH_3	0.00014	601	4683	0.561	-78	1000.0	1.2
HFE 7200	ethoxy-nonafluorobutane	0.00061	1430	1214	0.068	-138	14.5	10.9
Water	H_2O	0.00091	1000	4184	0.607	0	3.2	6.3