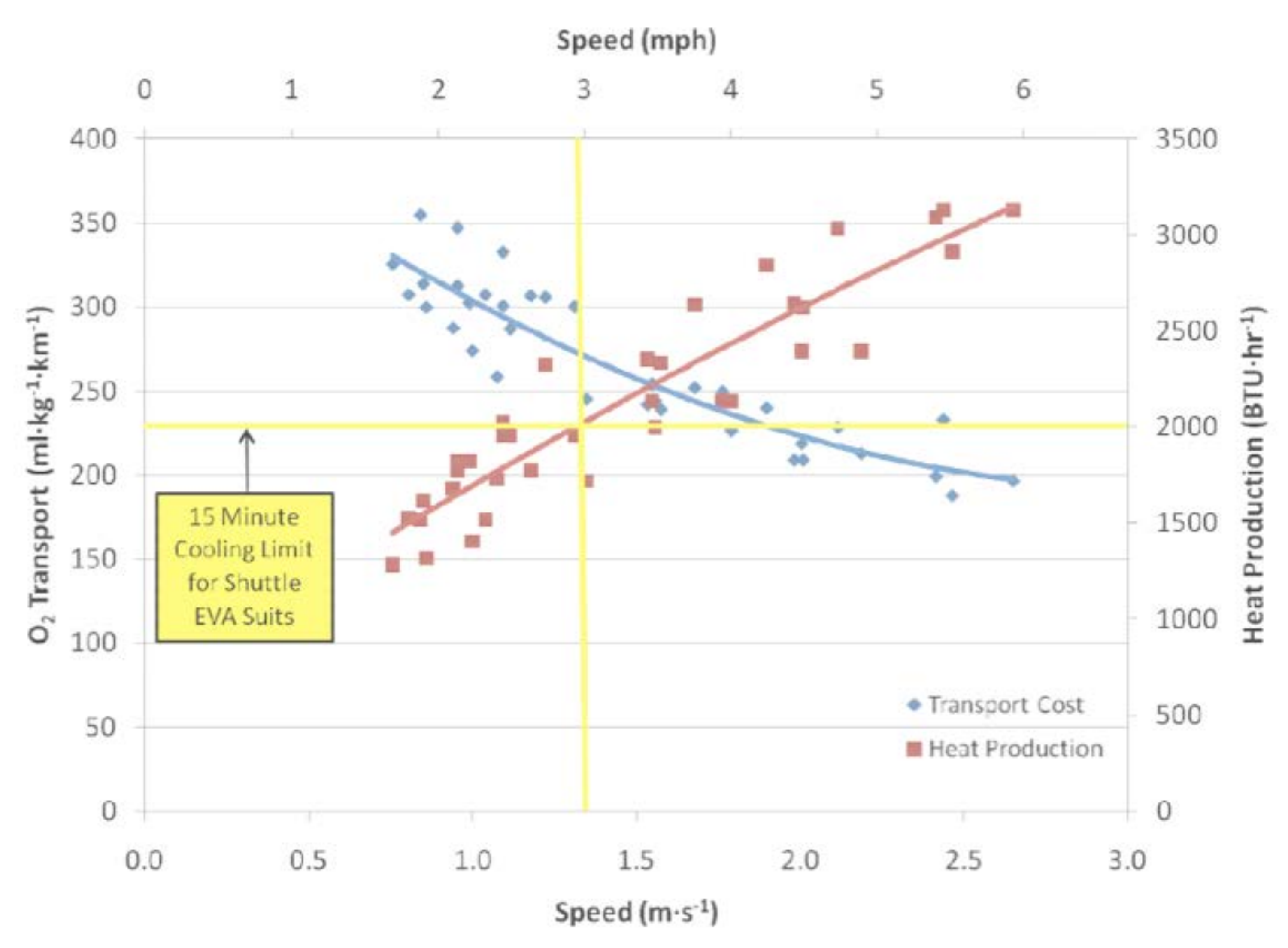
Thermal Analysis and Modeling

- Cooling humans
- Fundamentals of heat transfer
- Radiative equilibrium
- Surface properties
- Non-ideal effects
- Conduction
- Human thermal modeling
- Thermal system components

Cooling and Energy Use in Lunar Run



So What Is 2000 BTU?

- 2000 BTU/hr = 504 kcal/hr (or "calories" of food)
- Would heat 8 kg of water from body temperature (37°C) to boiling
- Water heat of vaporization = 2257 kJ/kg = 535 kcal/kg
- Need to convert 0.94 kg of water to vapor
- At 70°F, air can hold 1.15 lbs of water per KCFM
- Suit flow rate of 6 CFM = 3 gms/minute = 0.19 kg/hr = 100 kcal/hr = 397 BTU/hr

Sublimation to Dump Heat

- Flow water over porous plate exposed on other side to vacuum
  - Water passing through pores evaporates in vacuum - cools at rate of 535 kcal/kg
  - Plate reaches 0°C and water in pores freezes
  - Water at vacuum surface sublimates (solid—>gas)
    - Heat of melting 80 kcal/kg
    - Heat of vaporization 535 kcal/kg
    - Total heat of sublimation 615 kcal/kg
- Cooling 2000 BTU/hr (504 kcal/hr) requires 0.82 kg of water (ideal case)

Apollo PLSS Sublimator

Sublimator Schematic

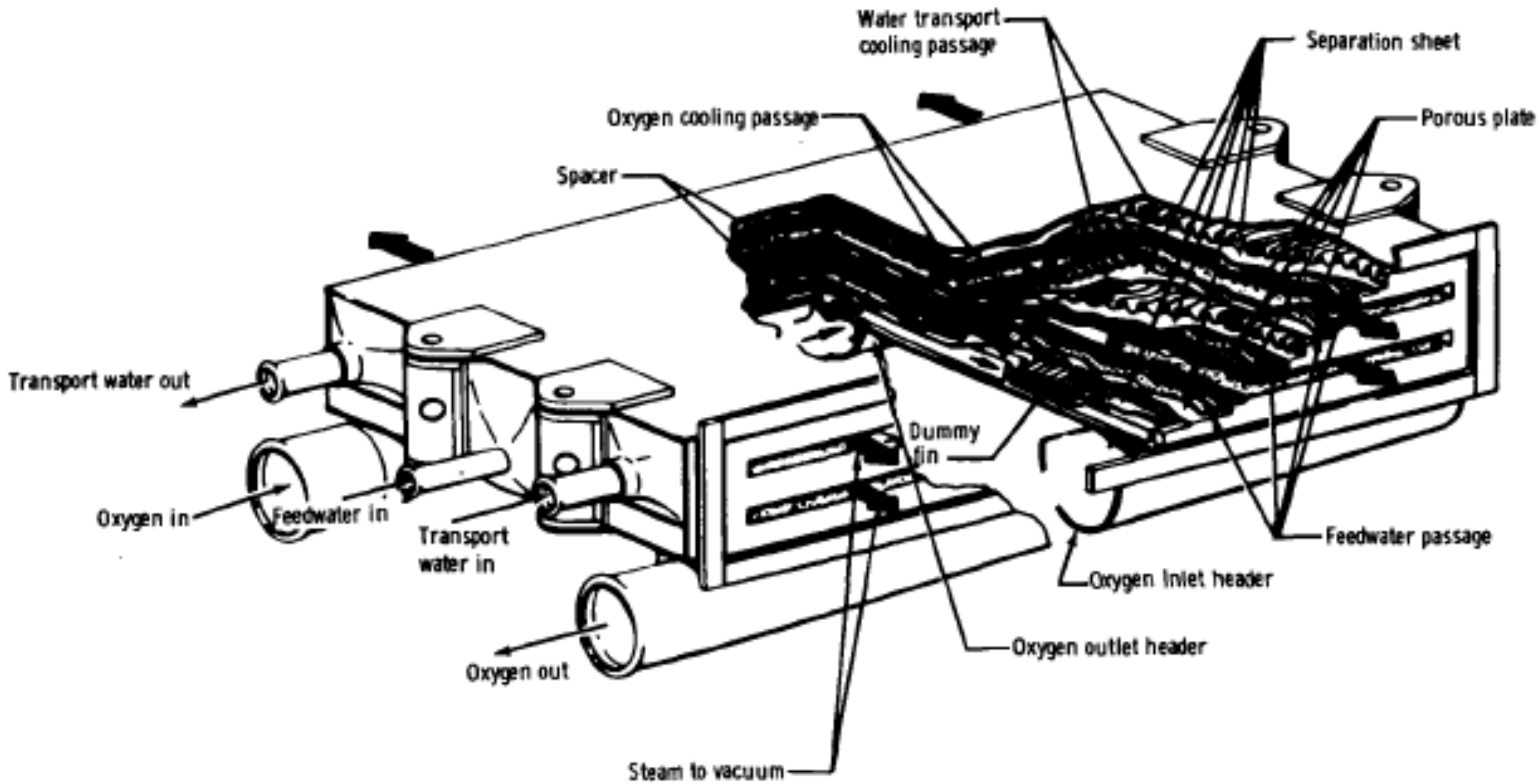
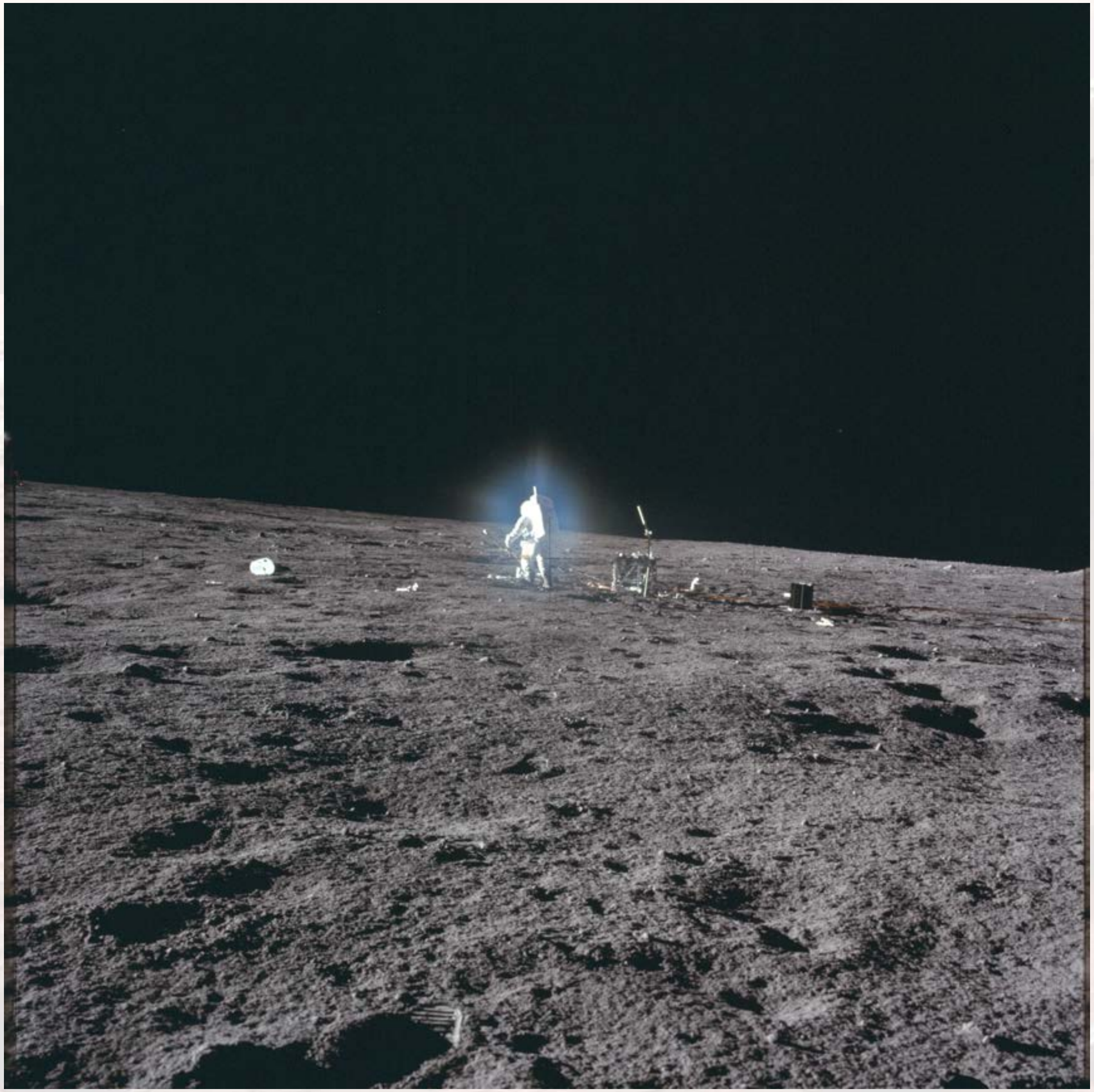


Figure 45. - Sectional view of the sublimator.

Sublimator Vapor "Cloud" on Moon



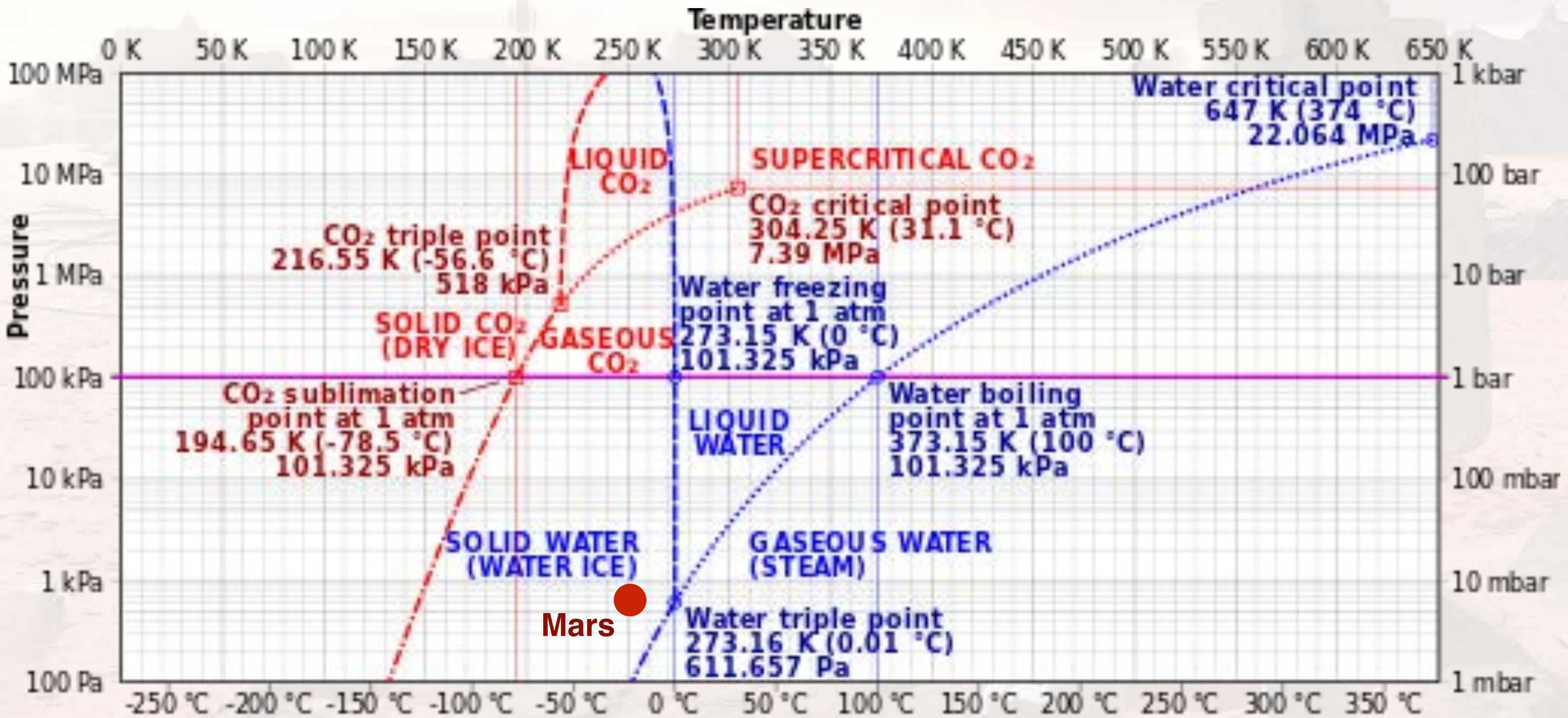
Past PLSS Thermal Capacities

- Apollo
  - 8 hrs @ 930 BTU/hr
  - 6 hrs @ 1200 BTU/hr
  - 5 hrs @ 1600 BTU/hr
- Shuttle/ISS EMU
  - 7 hrs @ 1000 BTU/hr (PLSS contains 3.9 kg of water - previous calculation predicts 2.87 kg)
- Advanced EMU (development)
  - 8 hrs @ 1200 BTU/hr

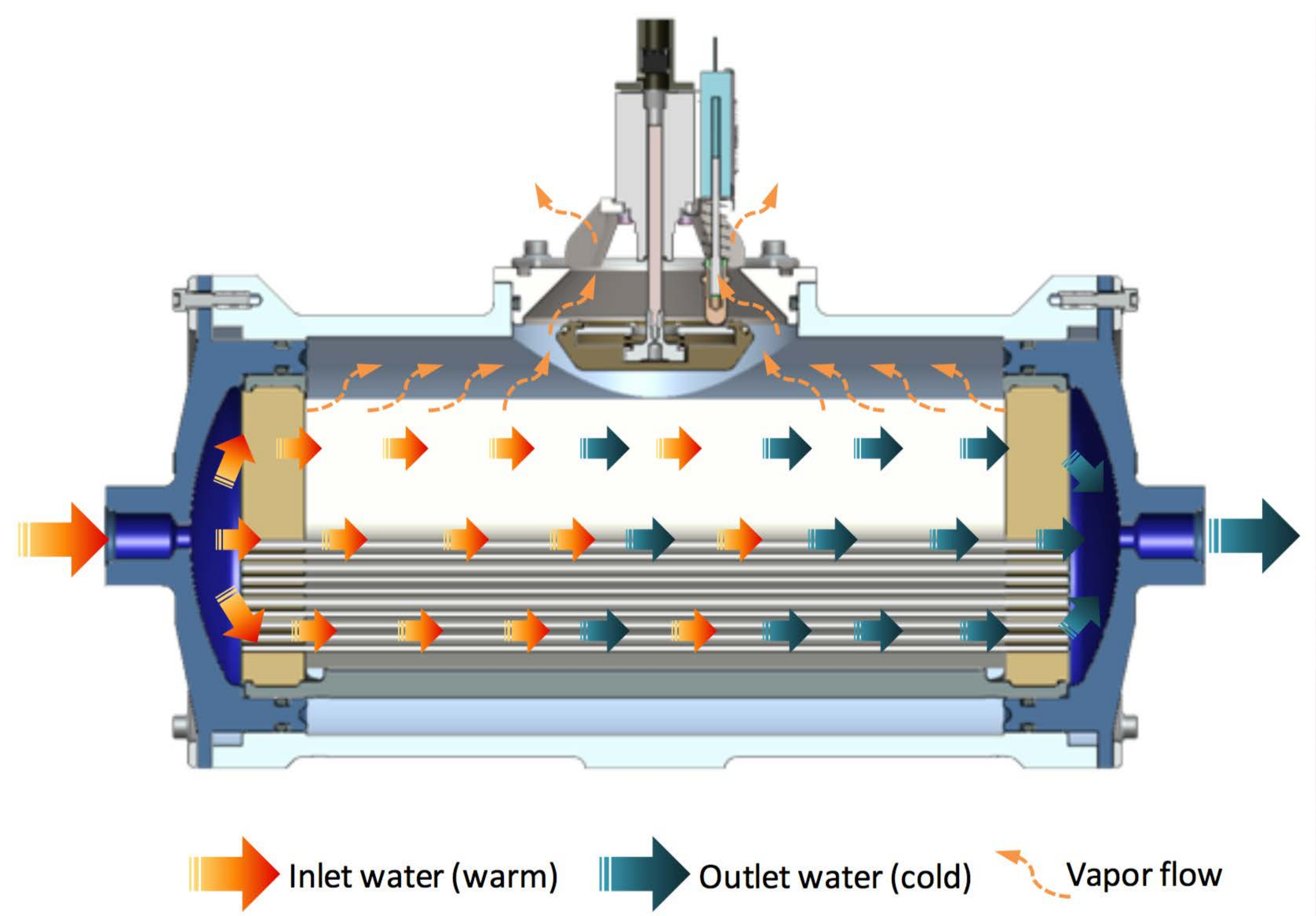
Issues with Sublimators

- Have to be “charged” - i.e., run water through long enough to freeze porous plate
- Susceptible to contamination damage (e.g., plugging pores in plate)
- Issues with liquid/gas separation (cooling both LCVG water and suit atmosphere)
- Only works in conditions below triple point of water

Phase Curves for CO_2 and H_2O



Suit Membrane Water Evaporator



Ref: Bue et.al., "Long-Duration Testing of a Spacesuit Water Membrane Evaporator Prototype" 42nd ICES, AIAA 2012-3459

SWME Fundamentals

- 14,900 hollow fibers carrying water for cooling
- Hydrophobic fiber material rejects water, but allows water vapor to pass
- Chamber pressure is controlled with a back-pressure regulator
- Rate of evaporation (and rate of cooling) controlled by internal pressure in chamber
- Evaporated vapor purged through regulator to ambient
- Evaporation provides 87% of sublimation cooling

Classical Methods of Heat Transfer

- Convection
  - Heat transferred to cooler surrounding gas, which creates currents to remove hot gas and supply new cool gas
  - Don't (in general) have surrounding gas or gravity for convective currents
- Conduction
  - Direct heat transfer between touching components
  - Primary heat flow mechanism internal to vehicle
- Radiation
  - Heat transferred by infrared radiation
  - Only mechanism for dumping heat external to vehicle

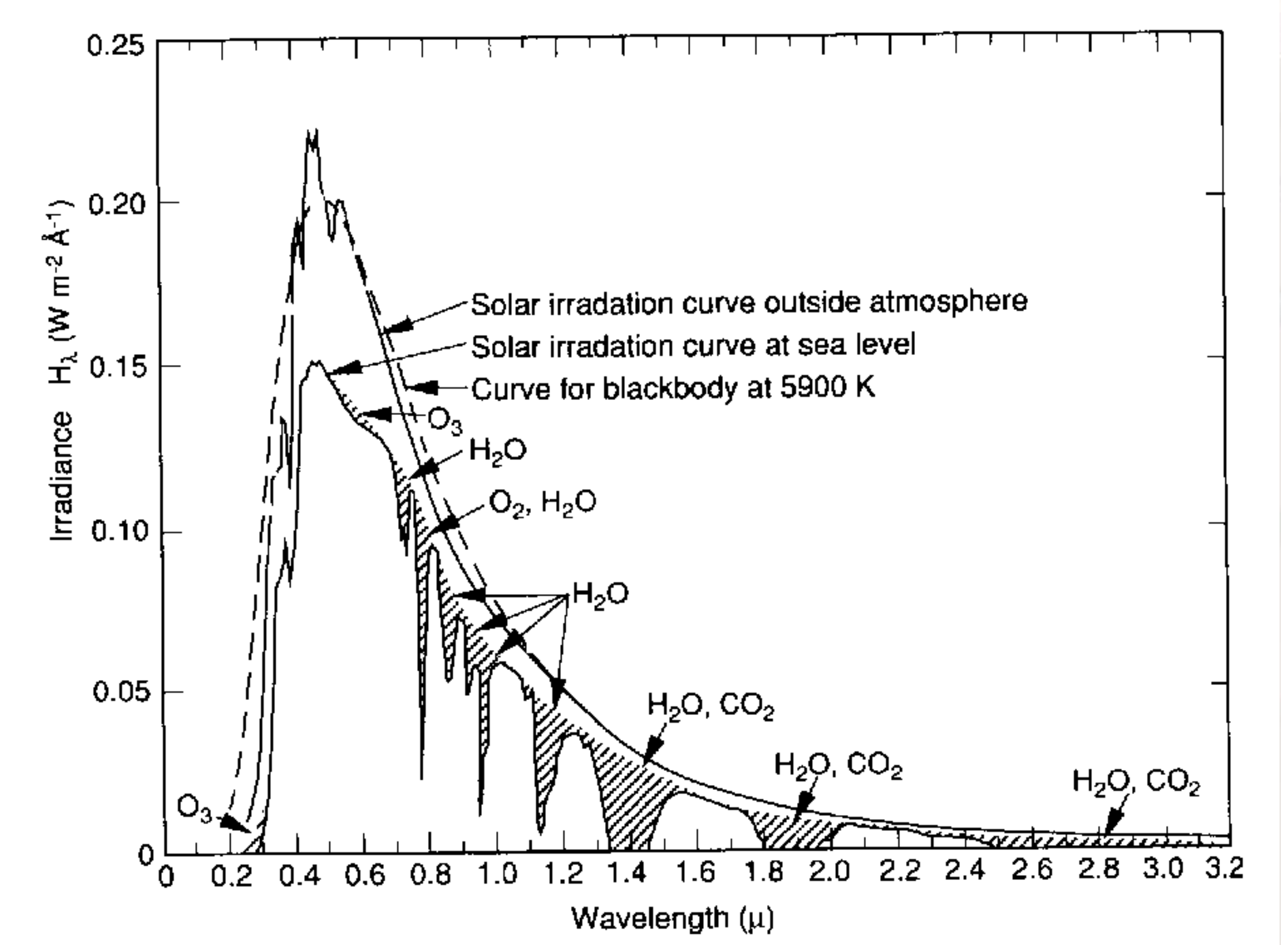
Ideal Radiative Heat Transfer

Planck's equation gives energy emitted in a specific frequency by a black body as a function of temperature

$$e_{\lambda b} = \frac{2\pi h C_0^2}{\lambda^5 \left[\exp\left(\frac{-hC_0}{\lambda k T} \right) - 1 \right]}$$

(Don't worry, we won't actually use this equation for anything…)

The Solar Spectrum



Ref: V. L. Pisacane and R. C. Moore, Fundamentals of Space Systems Oxford University Press, 1994

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- Stefan-Boltzmann equation integrates Planck's equation over entire spectrum

$$P_{rad} = \sigma T^4$$

$$\sigma = 5.67x10^{-8} \frac{W}{m^2 \, {}^\circ K^4}$$

("Stefan-Boltzmann Constant")

Thermodynamic Equilibrium

- First Law of Thermodynamics

$$Q - W = \frac{dU}{dt}$$

heat in -heat out = work done internally

- Heat in = incident energy absorbed
- Heat out = radiated energy
- Work done internally = internal power used (negative work in this sense - adds to total heat in the system)

Radiative Equilibrium Temperature

- Assume a spherical black body of radius r
- Heat in due to intercepted solar flux

$$Q_{in} = I_s \pi r^2$$

- Heat out due to radiation (from total surface area)

$$Q_{out} = 4\pi r^2 \sigma T^4$$

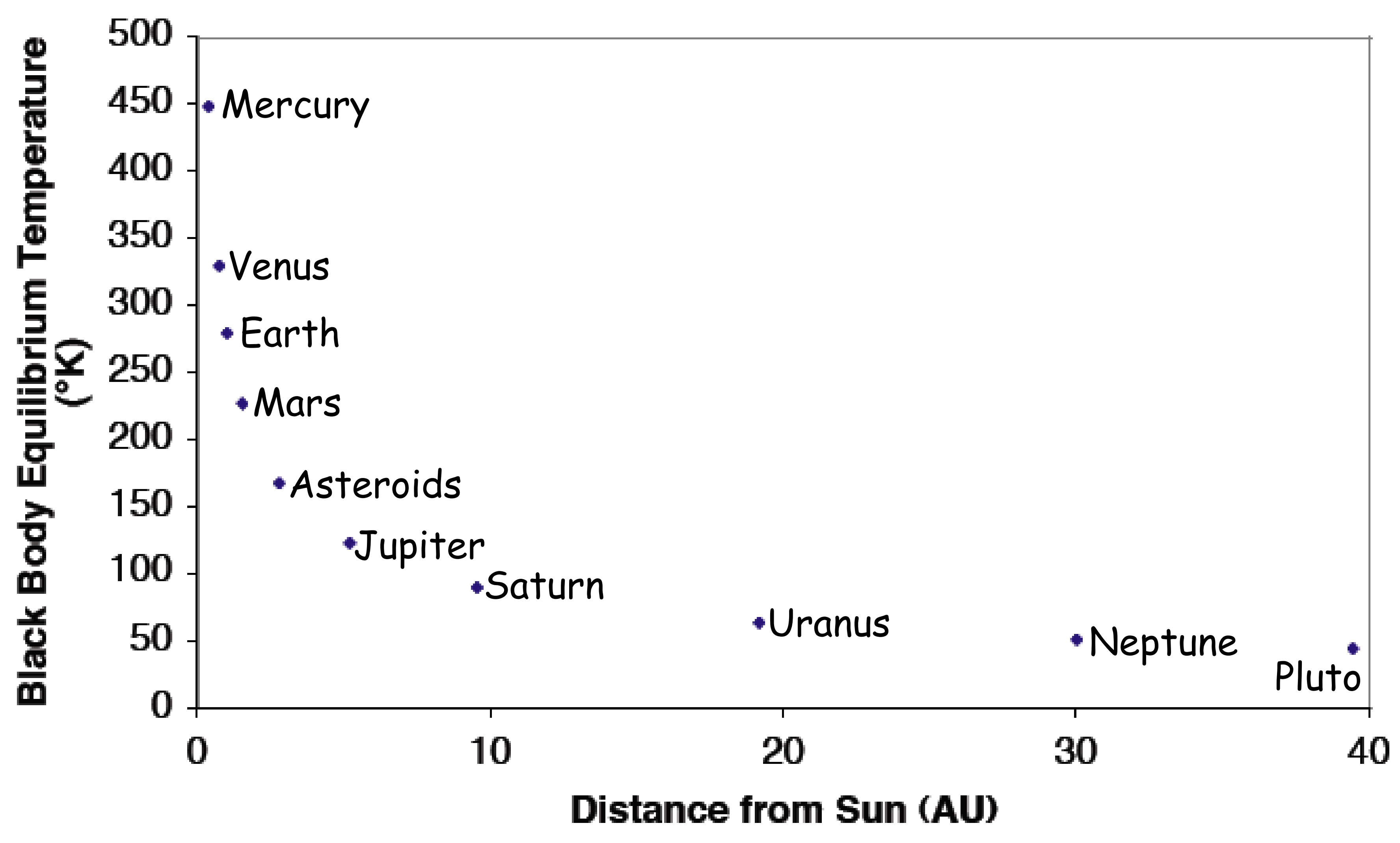
- For equilibrium, set equal

$$I_s \pi r^2 = 4\pi r^2 \sigma T^4 \Rightarrow I_s = 4\sigma T^4$$

- 1 AU: I_s=1394 W/m^2; T_{eq}=280°K

$$T_{eq} = \left(\frac{I_s}{4\sigma} \right)^{1/4}$$

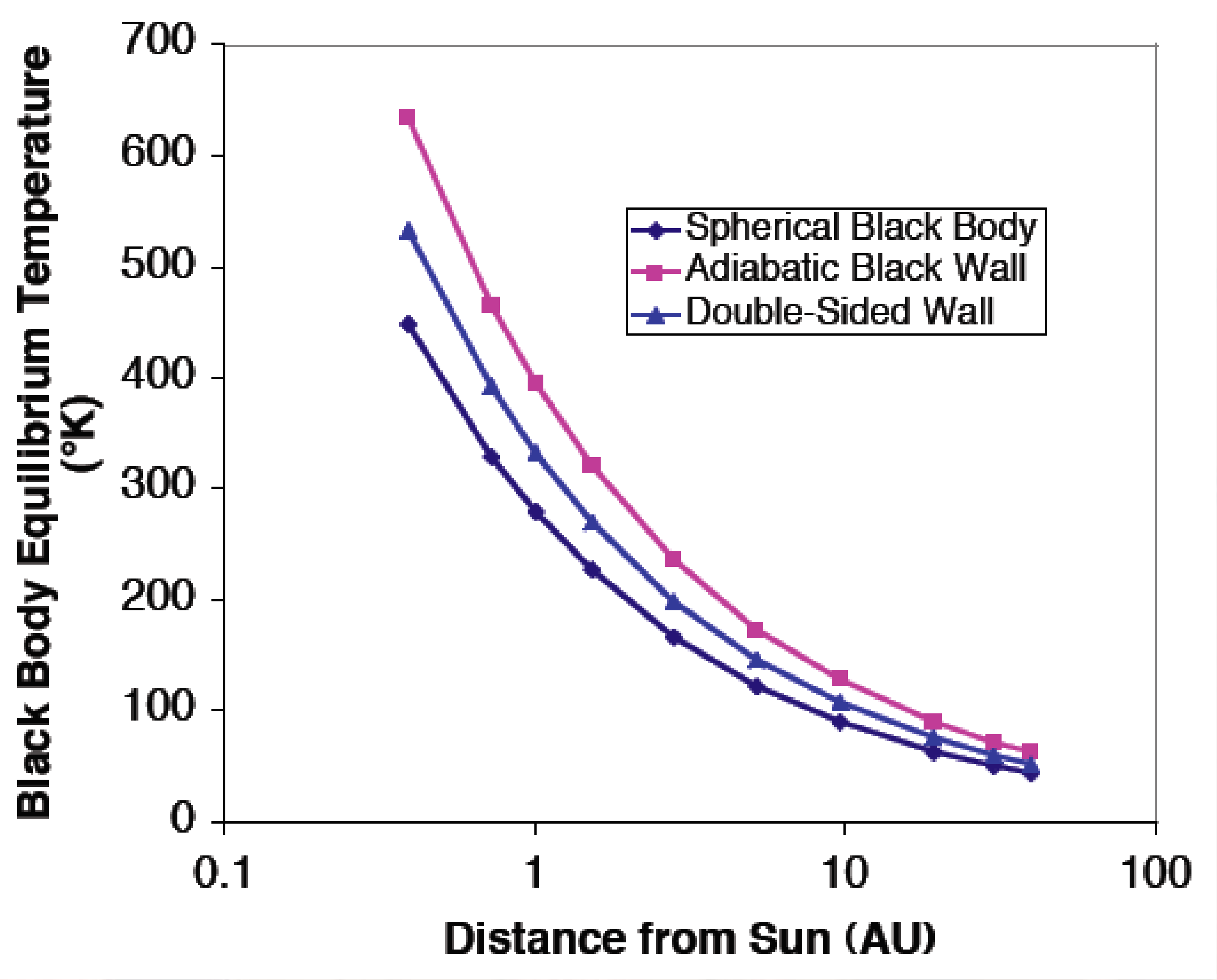
Effect of Distance on Equilibrium



Shape and Radiative Equilibrium

- A shape absorbs energy only via illuminated faces
- A shape radiates energy via all surface area
- Basic assumption made is that black bodies are intrinsically isothermal (perfect and instantaneous conduction of heat internally to all faces)

Effect of Shape on Black Body Temps



Incident Radiation on Non-Ideal Bodies

Kirchkoff's Law for total incident energy flux on solid bodies:

$$Q_{Incident} = Q_{absorbed} + Q_{reflected} + Q_{transmitted}$$

$$\frac{Q_{absorbed}}{Q_{Incident}} + \frac{Q_{reflected}}{Q_{Incident}} + \frac{Q_{transmitted}}{Q_{Incident}} = 1$$

$$\alpha \equiv \frac{Q_{absorbed}}{Q_{Incident}}; \quad \rho \equiv \frac{Q_{reflected}}{Q_{Incident}}; \quad \tau \equiv \frac{Q_{transmitted}}{Q_{Incident}}$$

where

- α =absorptance (or absorptivity)
- ρ =reflectance (or reflectivity)
- τ =transmittance (or transmissivity)

Non-Ideal Radiative Equilibrium Temp

- Assume a spherical black body of radius r
- Heat in due to intercepted solar flux

$$Q_{in} = I_s \alpha \pi r^2$$

- Heat out due to radiation (from total surface area)

$$Q_{out} = 4\pi r^2 \varepsilon \sigma T^4$$

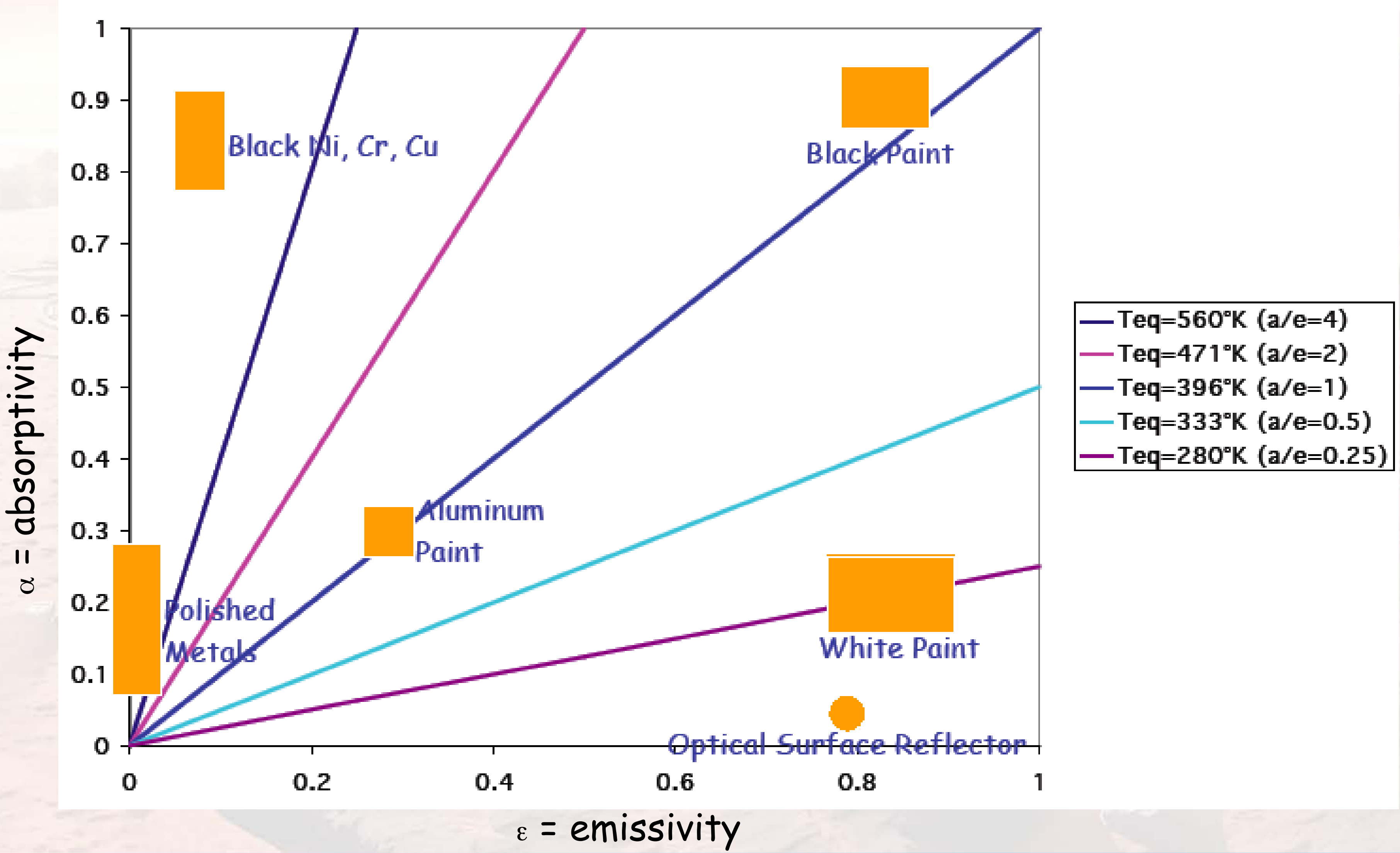
(ε = "emissivity" - efficiency of surface at radiating heat)

- For equilibrium, set equal

$$I_s \alpha \pi r^2 = 4\pi r^2 \varepsilon \sigma T^4 \Rightarrow I_s = 4 \frac{\varepsilon}{\alpha} \sigma T^4$$

$$T_{eq} = \left(\frac{\alpha}{\varepsilon} \frac{I_s}{4\sigma} \right)^{1/4}$$

Effect of Surface Coating on Temperature



Non-Ideal Radiative Heat Transfer

- Full form of the Stefan-Boltzmann equation

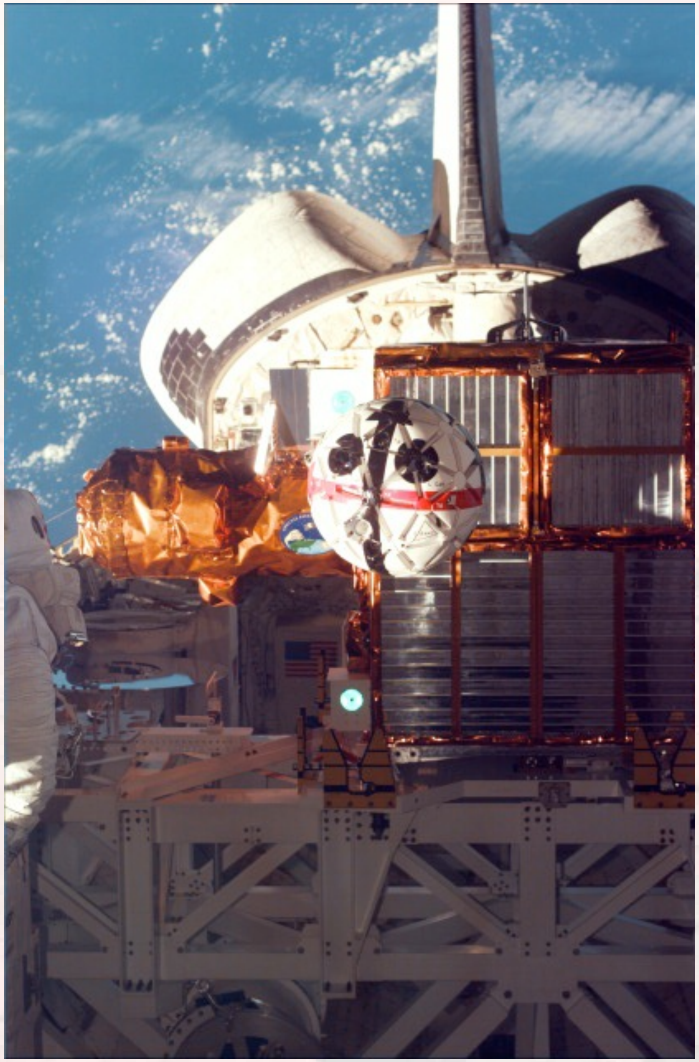
$$P_{rad} = \varepsilon \sigma A \left(T^4 - T_{env}^4\right)$$

where T_{env}=environmental temperature (=4°K for space)

- Also take into account power used internally

$$I_s \alpha A_s + P_{int} = \varepsilon \sigma A_{rad} \left(T^4 - T_{env}^4\right)$$

Example: AERCam/SPRINT



- 30 cm diameter sphere
- α=0.2; ε=0.8
- P_{int}=200W
- T_{env}=280°K (cargo bay below; Earth above)
- Analysis cases:
  - Free space w/o sun
  - Free space w/ sun
  - Earth orbit w/o sun
  - Earth orbit w/ sun

AERCam/SPRINT Analysis (Free Space)

- A_s=0.0707 m²; A_{rad}=0.2827 m²
- Free space, no sun

$$P_{int} = \varepsilon \sigma A_{rad} T^4 \Rightarrow T = \left(\frac{200W}{0.8\left(5.67 \times 10^{-8} \frac{W}{m^2 {}^\circ K^4}\right)\left(0.2827 m^2\right)} \right)^{1/4} = 354^\circ K$$

AERCam/SPRINT Analysis (Free Space)

- A_s=0.0707 m^2; A_{rad}=0.2827 m^2
- Free space with sun

$$I_s \alpha A_s + P_{\text{int}} = \varepsilon \sigma A_{rad} T^4 \Rightarrow T = \left(\frac{I_s \alpha A_s + P_{\text{int}}}{\varepsilon \sigma A_{rad}} \right)^{1/4} = 362°K$$

AERCam/SPRINT (LEO Cargo Bay)

- T_{env}=280°K
- LEO cargo bay, no sun

$$P_{\text{int}} = \varepsilon\sigma A_{rad}\left(T^4 - T_{env}^4\right) \Rightarrow T = \left(\frac{200W}{0.8\left(5.67\times10^{-8}\frac{W}{m^2{}^\circ K^4}\right)\left(0.2827\,m^2\right)} + (280^\circ K)^4\right)^{1/4} = 384^\circ K$$

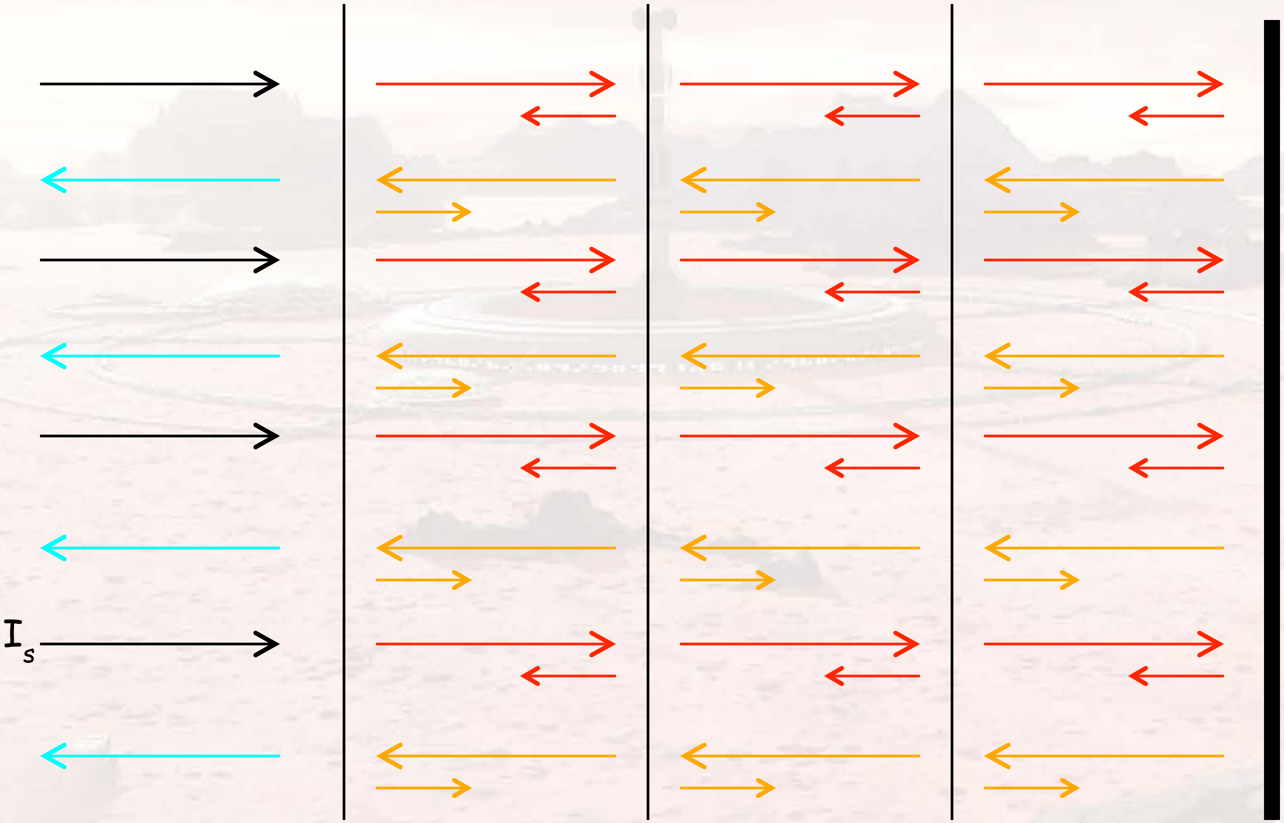
- LEO cargo bay with sun

$$I_s\alpha\, A_s + P_{\text{int}} = \varepsilon\sigma A_{rad}\left(T^4 - T_{env}^4\right) \Rightarrow T = \left(\frac{I_s\alpha\, A_s + P_{\text{int}}}{\varepsilon\sigma A_{rad}} + T_{env}^4\right)^{1/4} = 391^\circ K$$

Radiative Insulation

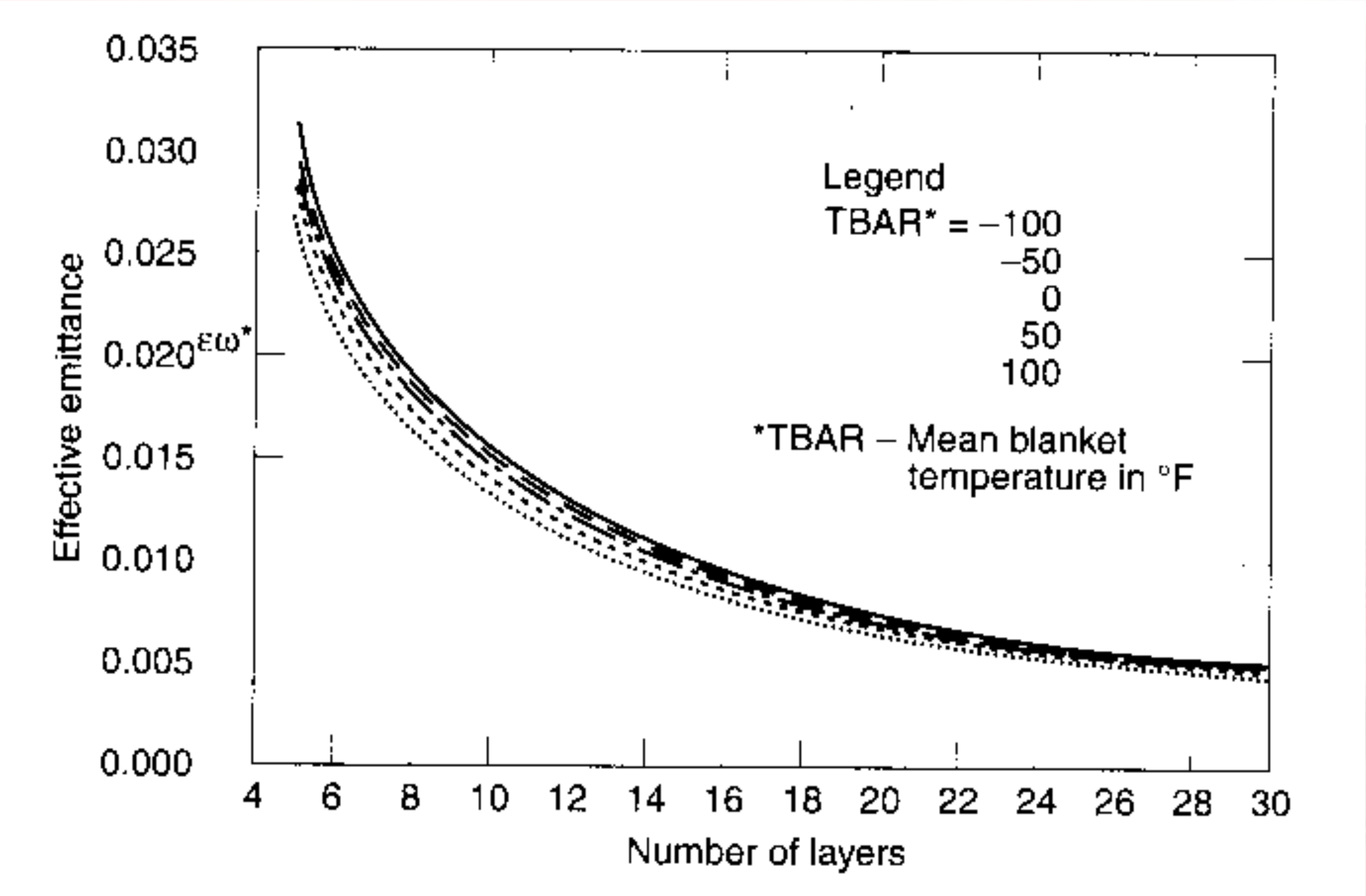
- Thin sheet (mylar/kapton with surface coatings) used to isolate panel from solar flux
- Panel reaches equilibrium with radiation from sheet and from itself reflected from sheet
- Sheet reaches equilibrium with radiation from sun and panel, and from itself reflected off panel

Multi-Layer Insulation (MLI)



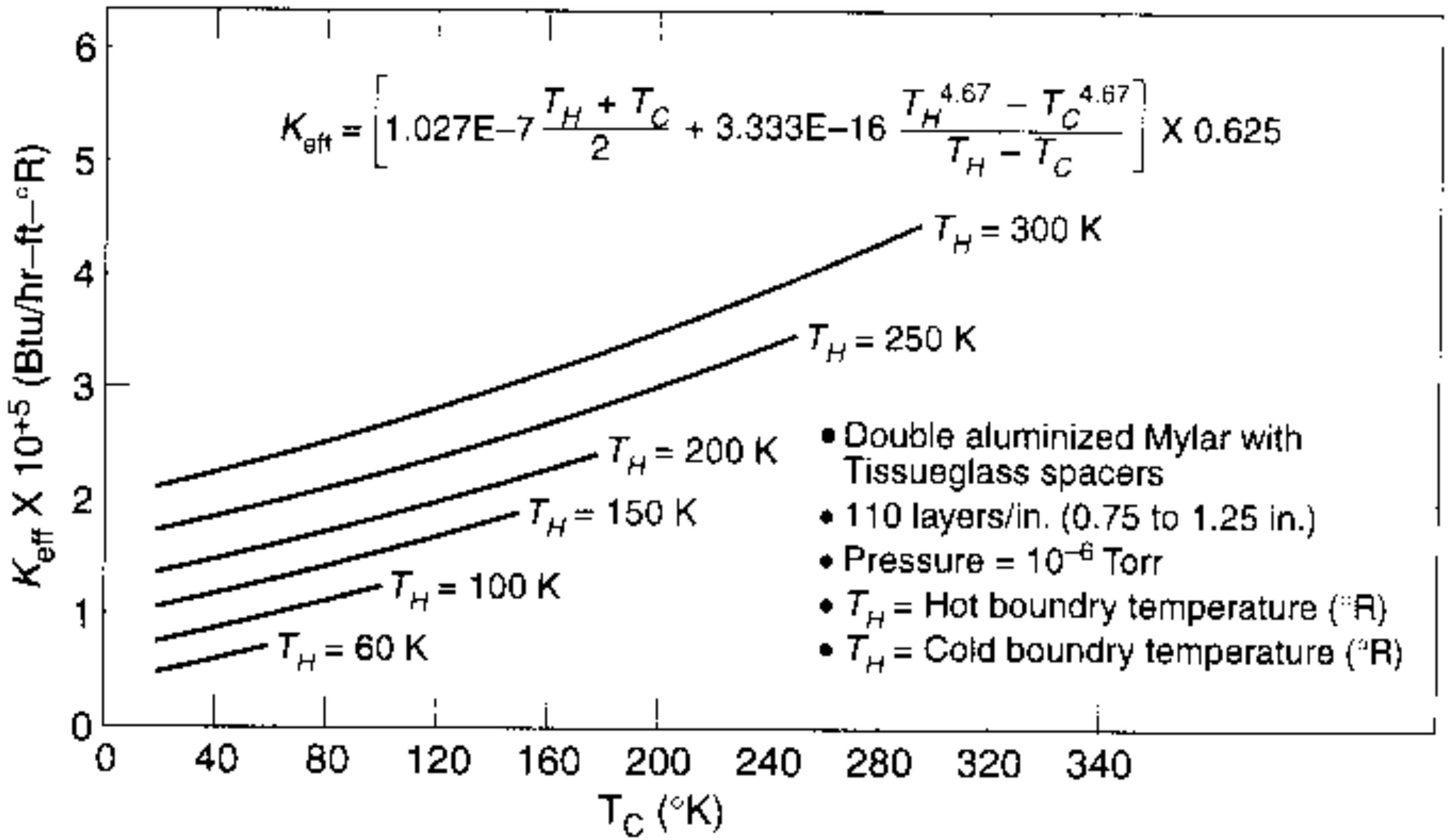
- Multiple insulation layers to cut down on radiative transfer
- Gets computationally intensive quickly
- Highly effective means of insulation
- Biggest problem is existence of conductive leak paths (physical connections to insulated components)

Emissivity Variation with MLI Layers



Ref: D. G. Gilmore, ed., Spacecraft Thermal Control Handbook AIAA, 2002

MLI Thermal Conductivity



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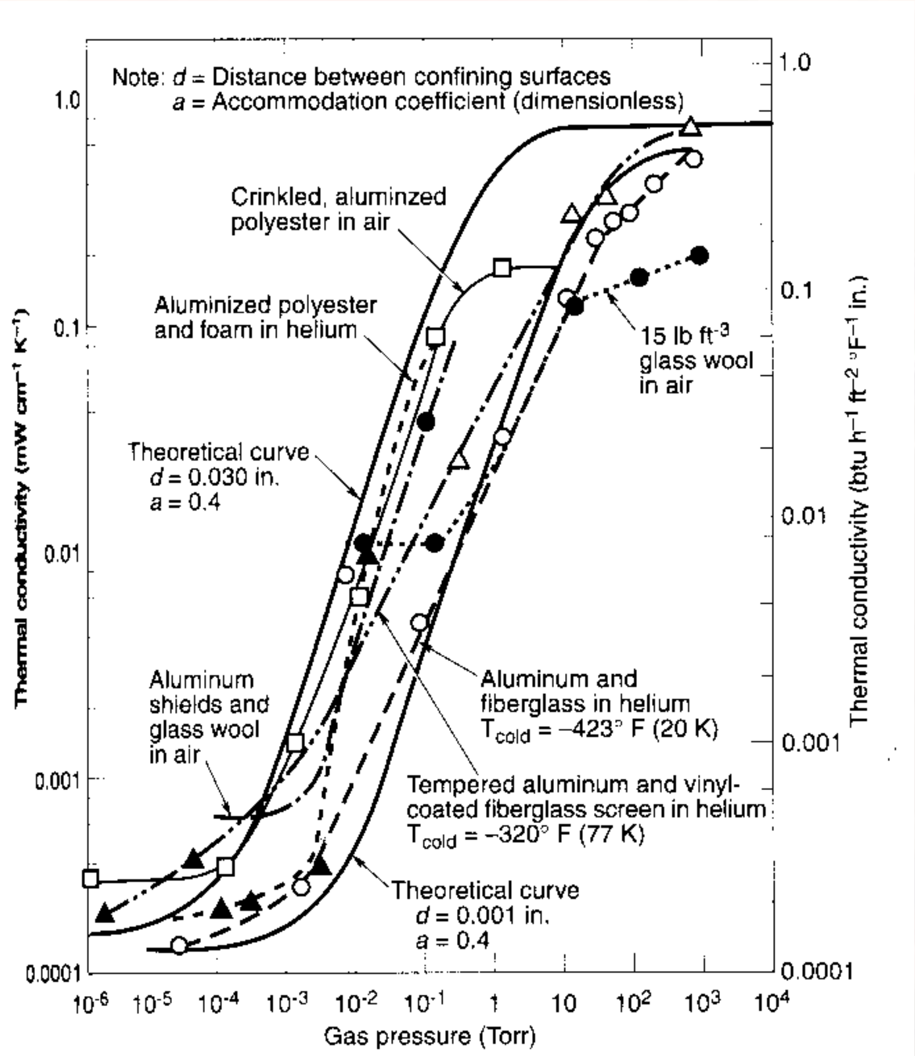
Effect of Ambient Pressure on MLI



Ref: D. G. Gilmore, ed., Spacecraft Thermal Control Handbook AIAA, 2002

1D Conduction

- Basic law of one-dimensional heat conduction (Fourier 1822)

$$Q = -KA\frac{dT}{dx}$$

where

K=thermal conductivity (W/m°K)

A=area

dT/dx=thermal gradient

3D Conduction

General differential equation for heat flow in a solid

$$\nabla^2 T(\vec{r},t) + \frac{g(\vec{r},t)}{K} = \frac{\rho c}{K} \frac{\partial T(\vec{r},t)}{\partial t}$$

where

g(r,t)=internally generated heat

ρ=density (kg/m^3)

c=specific heat (J/kg°K)

K/ρc=thermal diffusivity

Simple Analytical Conduction Model

- Heat flowing from (i-1) into (i)

…	T_{i-1}	T_i	T_{i+1}	…

$$Q_{in} = -KA\frac{T_i - T_{i-1}}{\Delta x}$$

- Heat flowing from (i) into (i+1)

$$Q_{out} = -KA\frac{T_{i+1} - T_i}{\Delta x}$$

- Heat remaining in cell

$$Q_{out} - Q_{in} = \frac{\rho c}{K}\frac{T_i(j+1) - T_i(j)}{\Delta t}$$

Finite Difference Formulation

- Time-marching solution

$$T_i^{n+1} = T_i^n + d(T_{i+1}^n - 2T_i^n + T_{i-1}^n)$$

where

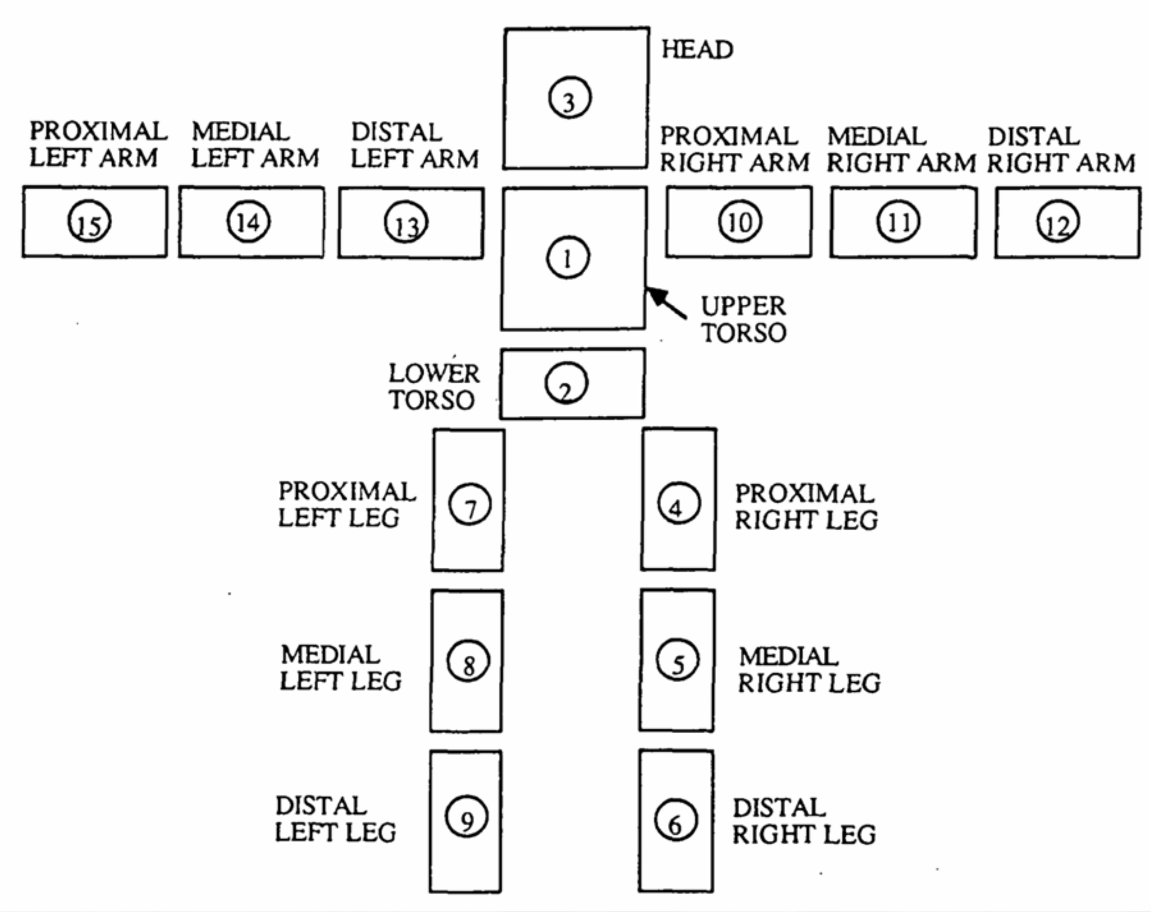
$$d = \frac{\alpha \Delta t}{\Delta x^2} \qquad \alpha = \frac{k}{\rho C_v} = \text{thermal diffusivity}$$

- For solution stability,

$$\Delta t < \frac{\Delta x^2}{2\alpha}$$

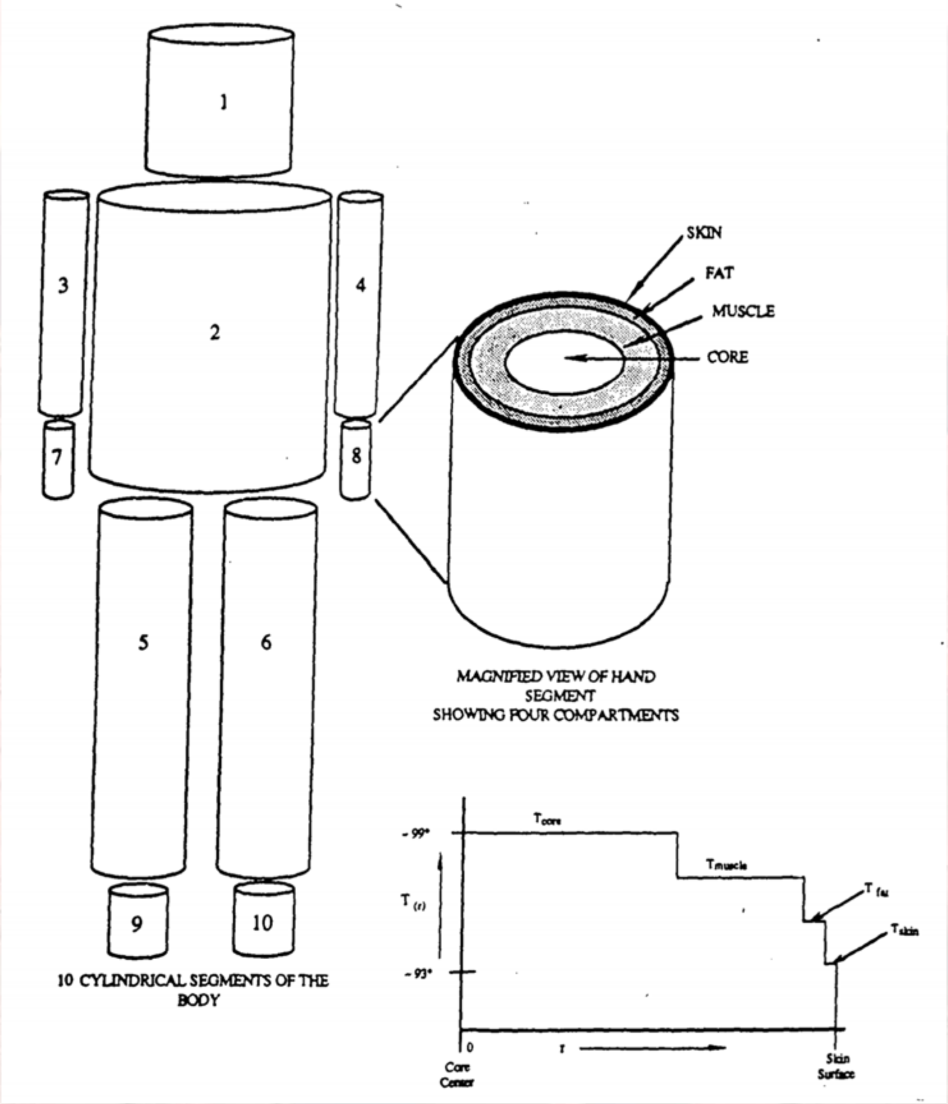
Human Thermal Model (Wissler)

- 15 elements per body
- 15 nodes per element
- Additional elements: e.g., skin, sweat, air circulation
- ~300 nodes in full model



Human Thermal Model (METMAN)

- 10 element in body
- 4 nodes per element
  - Skin
  - Fat
  - Muscle
  - Core
- Blood is separate node
- 41 nodes total



Energy Balance in Each Node

$$Q_{st} = Q_m - Q_c - Q_r - Q_e - Q_k - Q_{resp} - Q_{LCG} = mc_P \frac{\partial T}{\partial t}$$

- Q_{st} - heat rate saved into tissue
- Q_m - heat rate due to internal metabolism
- Q_c - heat rate due to surface convection
- Q_r - heat rate due to radiative losses
- Q_e - heat rate due to evaporation
- Q_k - heat rate due to conduction to other nodes
- Q_{resp} - heat rate due to respiratory cooling
- Q_{LCG} - heat rate due to liquid cooling garment

41-Node Heat Flow Equations (1)

Core layer:

$$M_c C p_c \frac{dT_c}{dt} = \dot{m}_{b \to c} C p_b (T_b - T_c) + G_{m \leftrightarrow c}(T_m - T_c)$$

$$+ \dot{Q}_{\text{met}} - \dot{Q}_{\text{resp}}$$

Muscle layer:

$$M_m C p_m \frac{dT_m}{dt} = \dot{m}_{b \to m} C p_b (T_b - T_m) + G_{c \leftrightarrow m}(T_c - T_m)$$

$$+ G_{f \leftrightarrow m}(T_f - T_m) + \dot{Q}_{\text{met}} - \dot{Q}_{\text{shiv}} - \dot{Q}_{\text{resp}}$$

from Campbell, French, Nair, and Miles, “Thermal Analysis and Design of an Advanced Space Suit” J. Thermophysics and Heat Transfer, v.14 n.2, April-June 2000

41-Node Heat Flow Equations (2)

Fat layer:

$$M_f C p_f \frac{\mathrm{d}T_f}{\mathrm{d}t} = \dot{m}_{b \to f} C p_b (T_b - T_f) + G_{m \leftrightarrow f}(T_m - T_f)$$

$$+ G_{s \leftrightarrow f}(T_s - T_f) + \dot{Q}_{\mathrm{met}} - \dot{Q}_{\mathrm{resp}}$$

Skin layer:

$$M_s C p_s \frac{\mathrm{d}T_s}{\mathrm{d}t} = \dot{m}_{b \to s} C p_b (T_b - T_s) + G_{f \leftrightarrow s}(T_f - T_s)$$

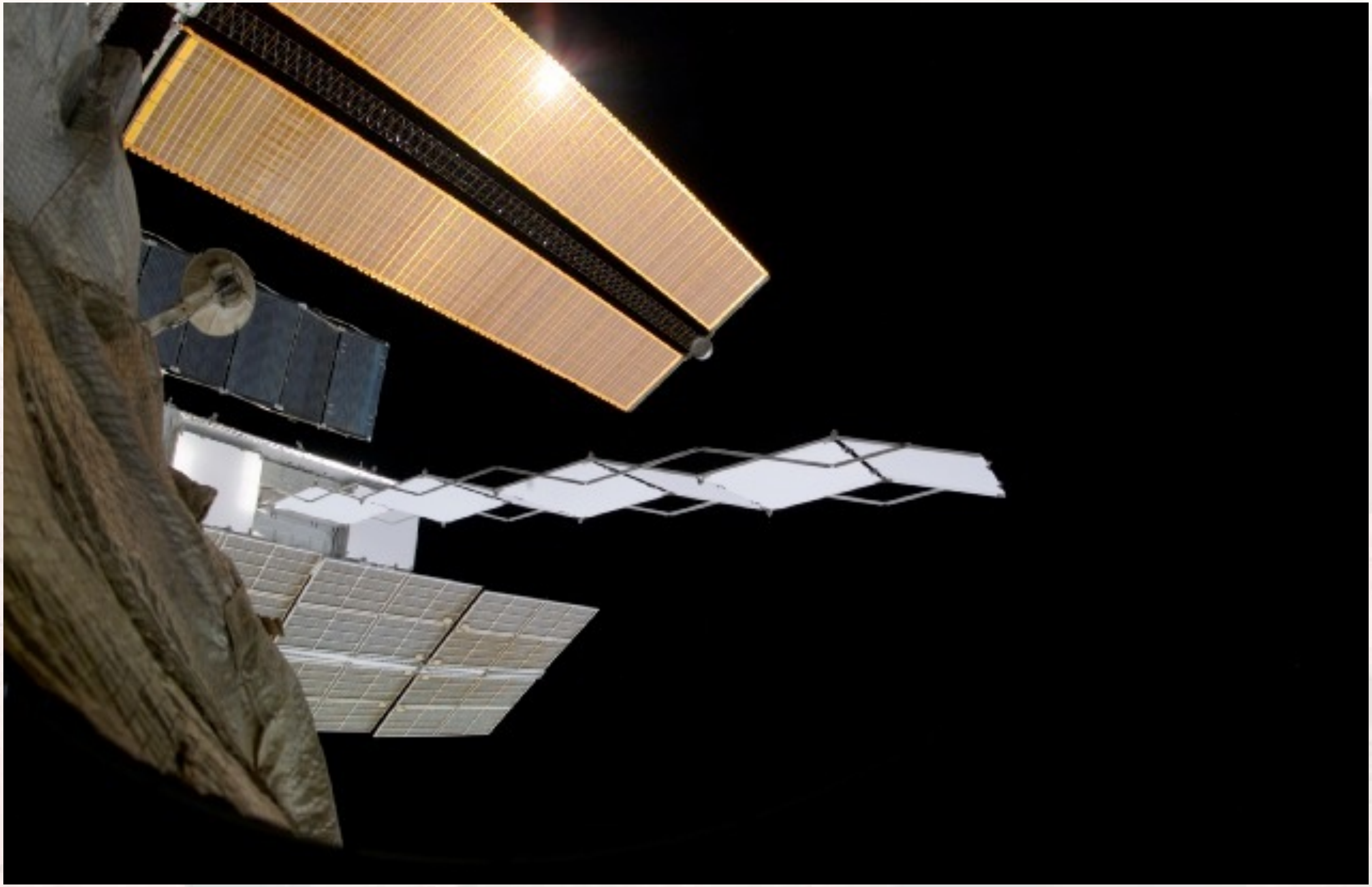
$$+ \dot{Q}_{\mathrm{met}} - \dot{Q}_{\mathrm{lat}} - \dot{Q}_{\mathrm{LCG}} - \dot{Q}_{\mathrm{VG}} - \dot{Q}_{\mathrm{suit}}$$

Blood pool:

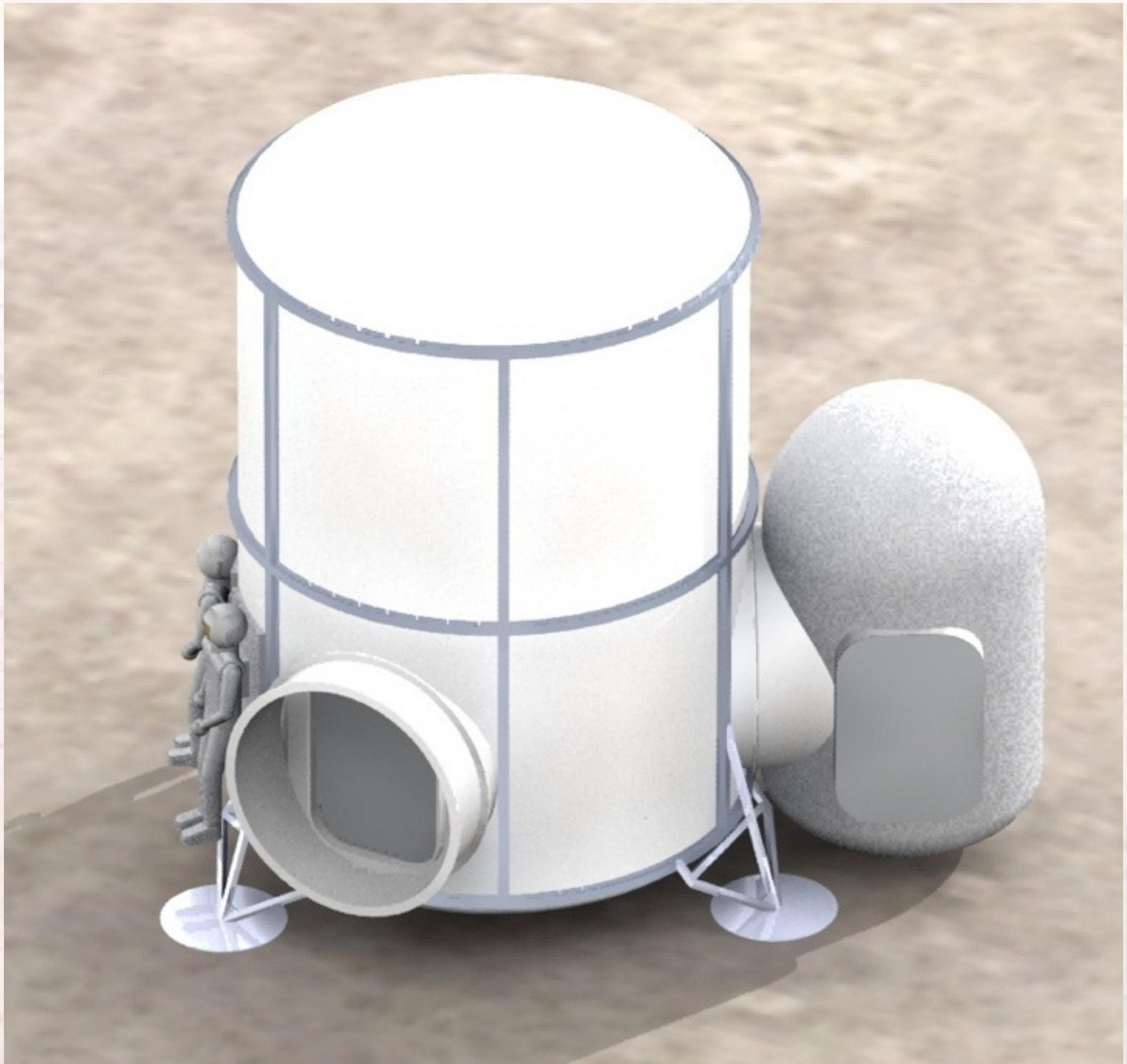
$$M_b C p_b \frac{\mathrm{d}T_b}{\mathrm{d}t} = C p_b \sum_{i=1}^{10} \sum_{j=1}^{4} \dot{m}_{b \to i,j}(T_{i,j} - T_b)$$

from Campbell, French, Nair, and Miles, “Thermal Analysis and Design of an Advanced Space Suit” J. Thermophysics and Heat Transfer, v.14 n.2, April-June 2000

ISS Radiator Assembly



Case Study: ECLIPSE Thermal Analysis



- Developed by UMd SSL for NASA ESMD
- Minimum functional habitat element for lunar outpost
- Radiator area - upper dome and six upper cylindrical panels

ECLIPSE Heat Sources

- Solar heat load (modeling habitat as right circular cylinder)

$$A_{illuminated} = \ell d \sin\beta + \frac{1}{4}\pi d^2 \cos\beta$$

$$Q_{solar} = A_{illuminated} \alpha I_s$$

- Electrical power load = 4191 W
- Metabolic work load (4 crew) = 464 W

Thermal Modeling for Lunar Surface

- Assume upper dome radiates only to deep space
- Assume side panels radiate half to deep space and half to lunar surface
- Assume (conservatively) that lunar surface radiates as a black body

$$Q_{internal} + Q_{solar} = \epsilon\sigma \left[A_{dome} T_{rad}^4 + n_{rad} A_{panel} \left(T_{rad}^4 - \frac{1}{2} T_{moon}^4 \right) \right]$$

$$T_{rad} = \left[\frac{1}{A_{dome} + n_{rad} A_{panel}} \left(\frac{Q_{internal} + Q_{solar}}{\epsilon\sigma} + \frac{1}{2} n_{rad} A_{wall} T_{moon}^4 \right) \right]^{\frac{1}{4}}$$

ECLIPSE Thermal Results

Case	Solar Angle (deg)	Lunar Surface Temp (K)	Active Wall Panels	Radiator Temp (K)
Polar Outpost Day	88	180	3	283
Local Midnight	N/A	120	1	285
Typical Mid-latitude	45	215	4	287
Equatorial Noon	0	380	6†	290

†Radiator geometry modified to reduce total lunar surface exposure