# Heat Pipes:
# an Introductory Short Course

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# Outline

- Heat Pipes
  - generic description
  - uses
  - theory of operation
  - types of heat pipes
  - operating limits
  - off-nominal performance
  - performance testing
  - heat pipe selection
  - modeling

# Heat Pipe Generic Description

- A heat pipe is a two-phase device that takes advantage of boiling and condensation to move heat from one location to another (Lienhard, J. H., A Heat Transfer Textbook, Second Edition, Prentice-Hall, Englewood Cliffs, NJ, 1987)
- The definition of a heat pipe gets a little fuzzy from here - some things are called heat pipes that, in truth, are not

# Heat Pipe Generic Description

- A heat pipe uses *capillary forces* to move the liquid and vapor within the pipe
  - no mechanical pump
  - reflux boilers are not heat pipes
- A heat pipe can be temperature controlled
  - but not all temperature controlled capillary pumped devices are heat pipes, e.g. capillary pumped loops

# Heat Pipe Uses

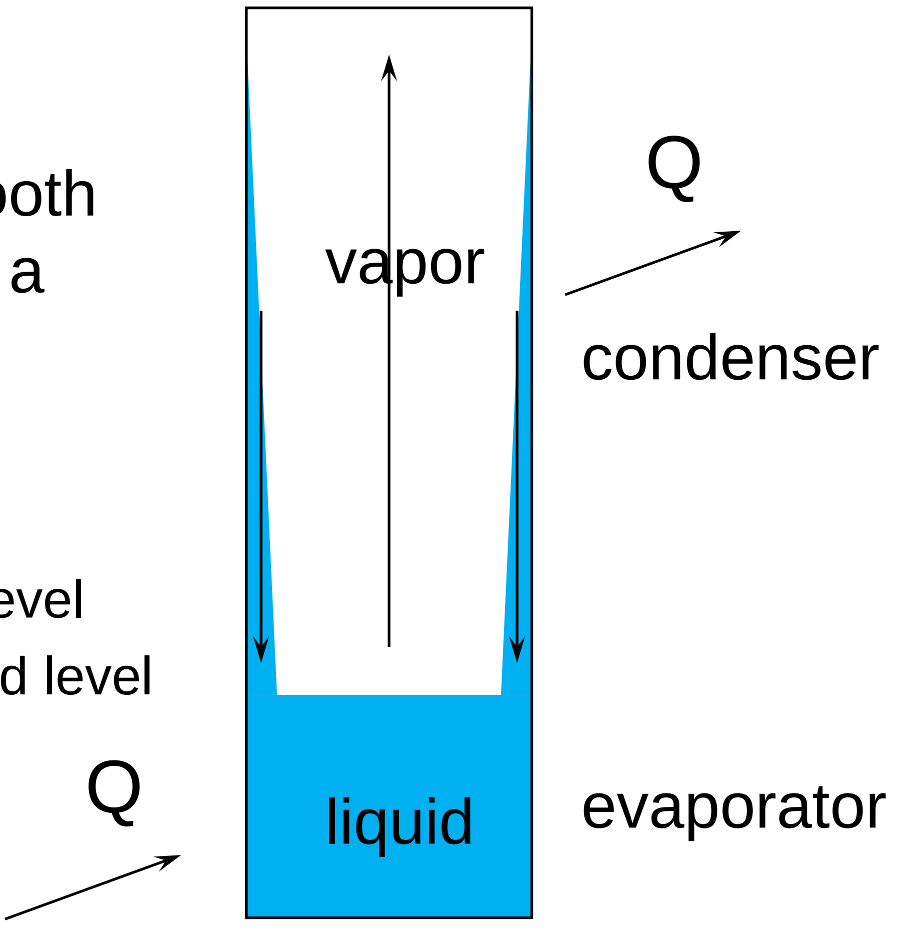
- Terrestrial uses
  - Alaska pipeline
  - “Heat pipe” condenser
- Space uses
  - robotic spacecraft
  - human spacecraft
    - Russian practice
    - US practice

# Heat Pipe Uses

- Because traditional heat pipes are low head devices they have few terrestrial uses
  - capillary forces are weak
    - generally on the order of a few psf
    - tenths of inches of capillary head
- Terrestrial uses of “heat pipes” tend to be reflux boilers

# Heat Pipe Uses

- Reflux Boilers
  - reflux boilers are smooth wall tubes containing a two-phase mixture
  - liquid transport is by gravity
    - evaporator at lowest level
    - condenser above liquid level



# Terrestrial Heat Pipe Uses

- Alyeska (Alaska) pipeline
  - the pipeline was built on permafrost, but there were concerns that the pipeline heat would melt the permafrost and it would be destabilized
  - reflux boilers were installed in the permafrost beneath the pipeline and exposed to air
  - the diode nature of the reflux boilers allows heat to flow from the permafrost to the air, but not in the opposite direction
  - the permafrost remains frozen

# Alyeska Pipeline

# Terrestrial Heat Pipe Uses

- Gillette, Wyoming “Heat Pipe” Condenser
  - Foster Wheeler Energy Corporation built a demonstration “heat pipe” air cooled steam condenser (mid-1970’s) to address the issue of air-cooled condenser freezing
  - vertical reflux boiler tubes were mounted with the evaporators (bottom) in a large steam duct and finned condensers in a forced airstream
  - the reflux boilers caused the steam to condense in the duct
  - modulation of the airflow was to be used to prevent freezing
  - concept was not commercialized owing to poor performance -- the reflux tube thermal resistance was unacceptable

# Space-Based Heat Pipe Uses

- Heat pipes are widely used for heat dissipation on robotic spacecraft (low power levels with short transport distances)
  - passive design allows use as heat transport devices without use of electrical power or vibration
  - literally thousands are flying today
  - traditional axial groove heat pipes have shown such a high reliability that they are used without hesitation
    - individual heat pipes are tested at vendor
    - integrated performance testing is usually not required

# Space-Based Heat Pipe Uses

- The Russian segment of ISS uses heat pipe radiators
  - legacy design from Mir
  - used mainly for protection from micrometeoroids and orbital debris (MMOD)
    - MMOD puncture means the loss of one of hundreds of heat pipes, not loss of a fluid loop
  - all Russian segment body mounted radiators use heat pipes to transport their heat from the external thermal control system to the radiator surface

# Space-Based Heat Pipe Uses

- Use of heat pipes on United States human spacecraft is limited
  - used on Skylab to ameliorate azimuthal temperature variations on crew module (solar inertial attitude)
  - limited exposure period of individual spacecraft has meant acceptable risk of MMOD puncture for low-Earth orbit spacecraft
  - the Orbiter used flow-through radiators
    - MMOD puncture was considered in mission planning

# Space-Based Heat Pipe Uses

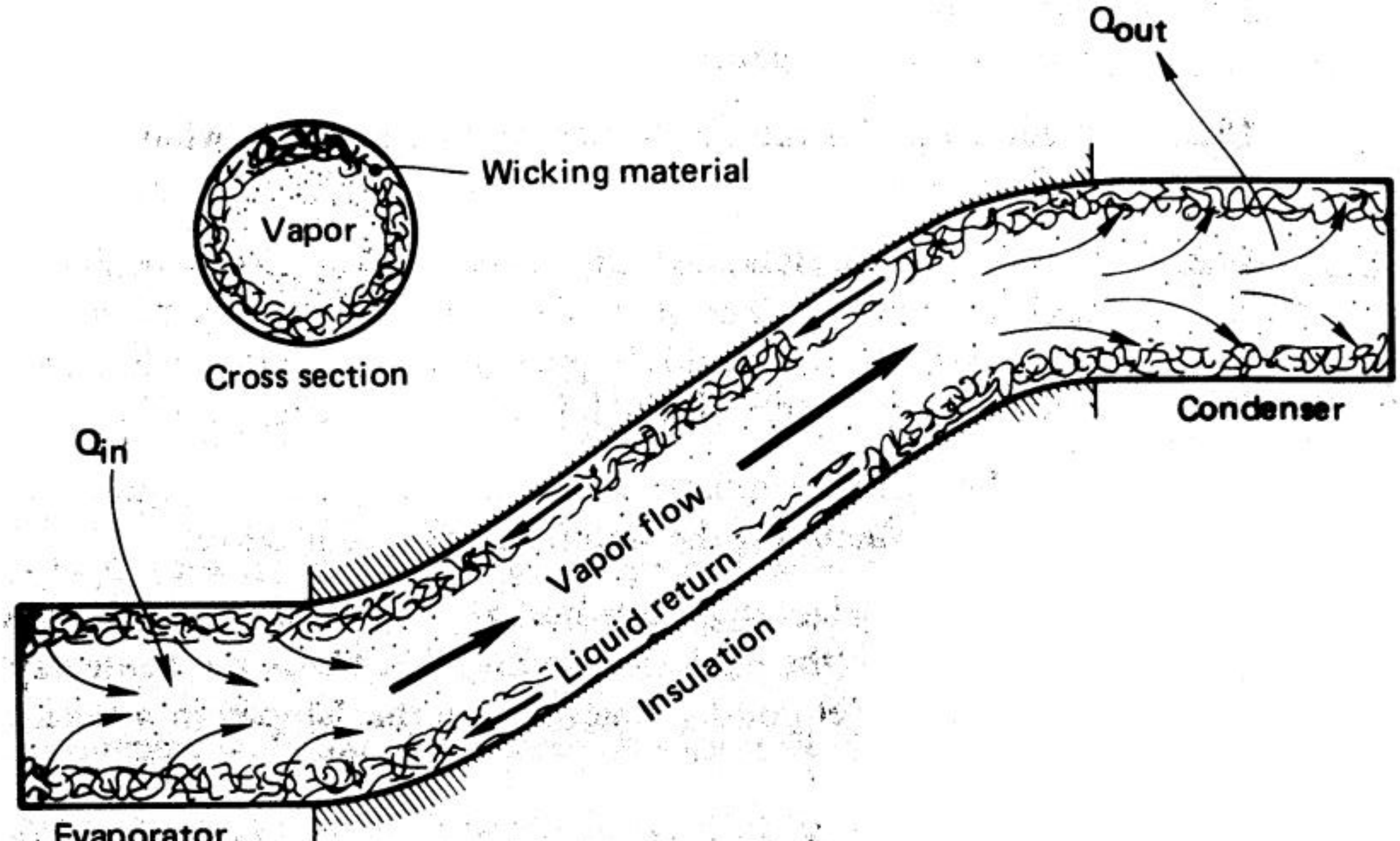
- US Segment of Space Station
  - Space Station originally designed with heat pipe radiators to guard against MMOD puncture
  - prior to the transition to ISS in early 1990s, design was changed to flow-through radiators
    - new MMOD models showed acceptable risk of radiator puncture over lifetime
      - radiators are orbital replacement units and can be replaced, unlike Russian ISS radiators
    - flow-through radiators were less expensive and reject more heat per unit area
      - they are warmer owing to the lack of heat pipe thermal resistance
      - 5°F decrease in radiator temperature yields 10% decrease in specific heat rejection for ISS environment

# Space-Based Heat Pipe Uses

- US Segment of Space Station
  - heat pipes are used for heat spreading and transport on Z1 and S0 truss segments that were launched before the active thermal control system
  - heat loads up to 250W on Z1
  - S0 truss contains a large radiator with spreader heat pipes
  - transport heat pipes bring small heat loads (on the order of 100 W) to the radiator

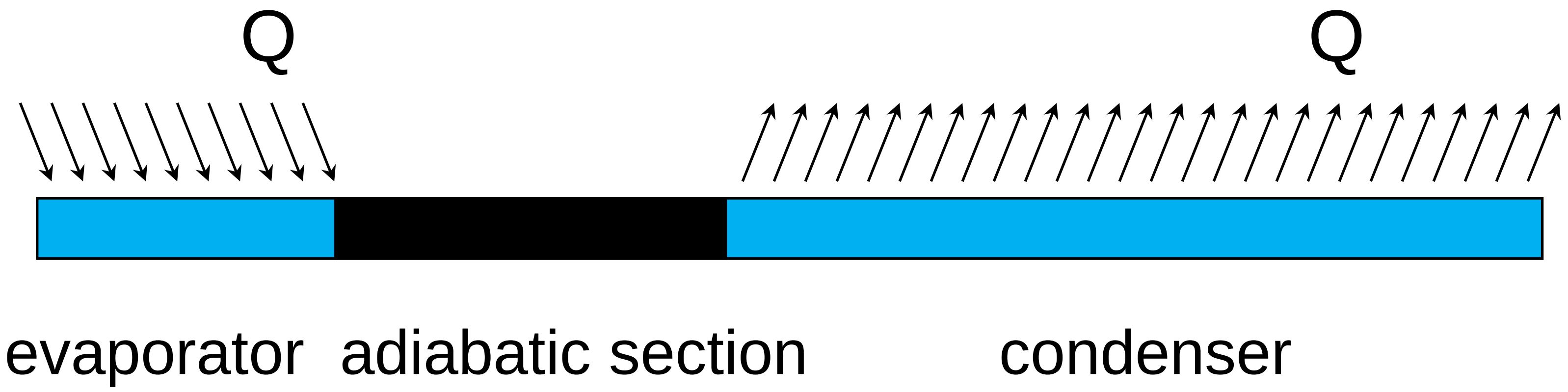
# Heat Pipe Theory of Operation

- Liquid evaporates in the heat pipe evaporator
- Vapor is transported through the vapor space by the evaporator/condenser pressure difference
- Vapor condenses in the condenser
- Liquid is transported through the liquid space by the capillary pressure difference
- Capillary pressure difference also drives vapor flow

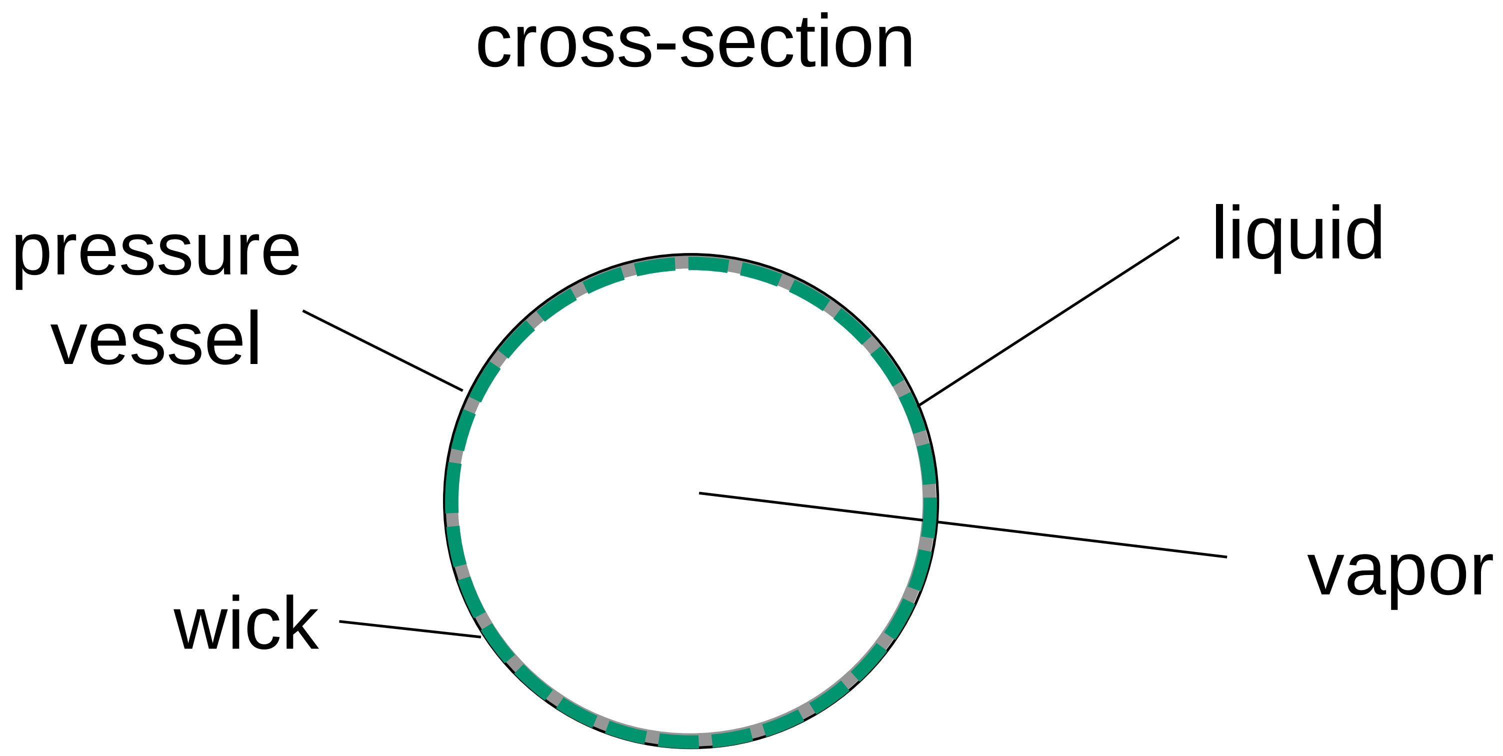


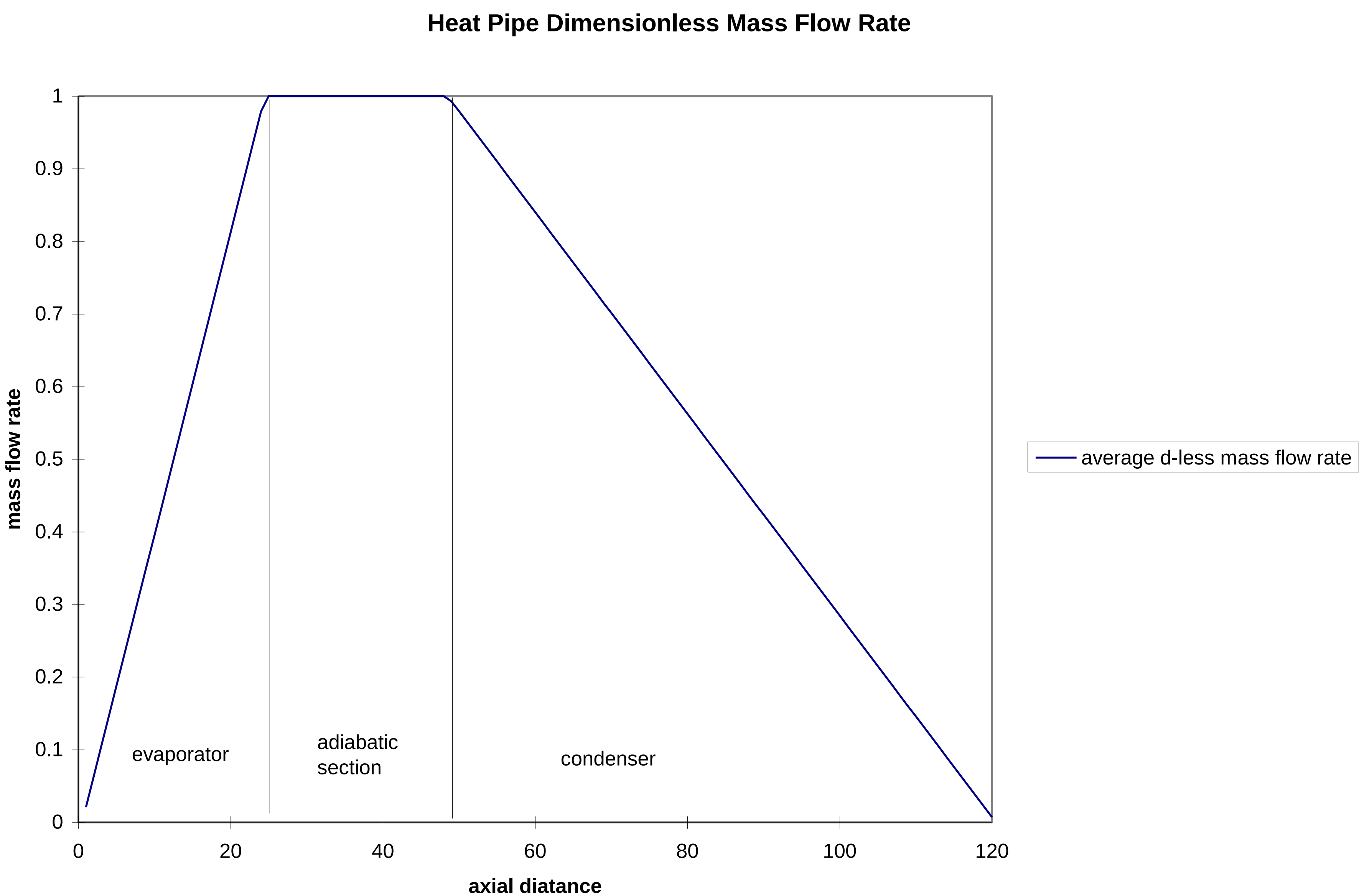

**A typical heat pipe configuration.**

# Theory of Operation

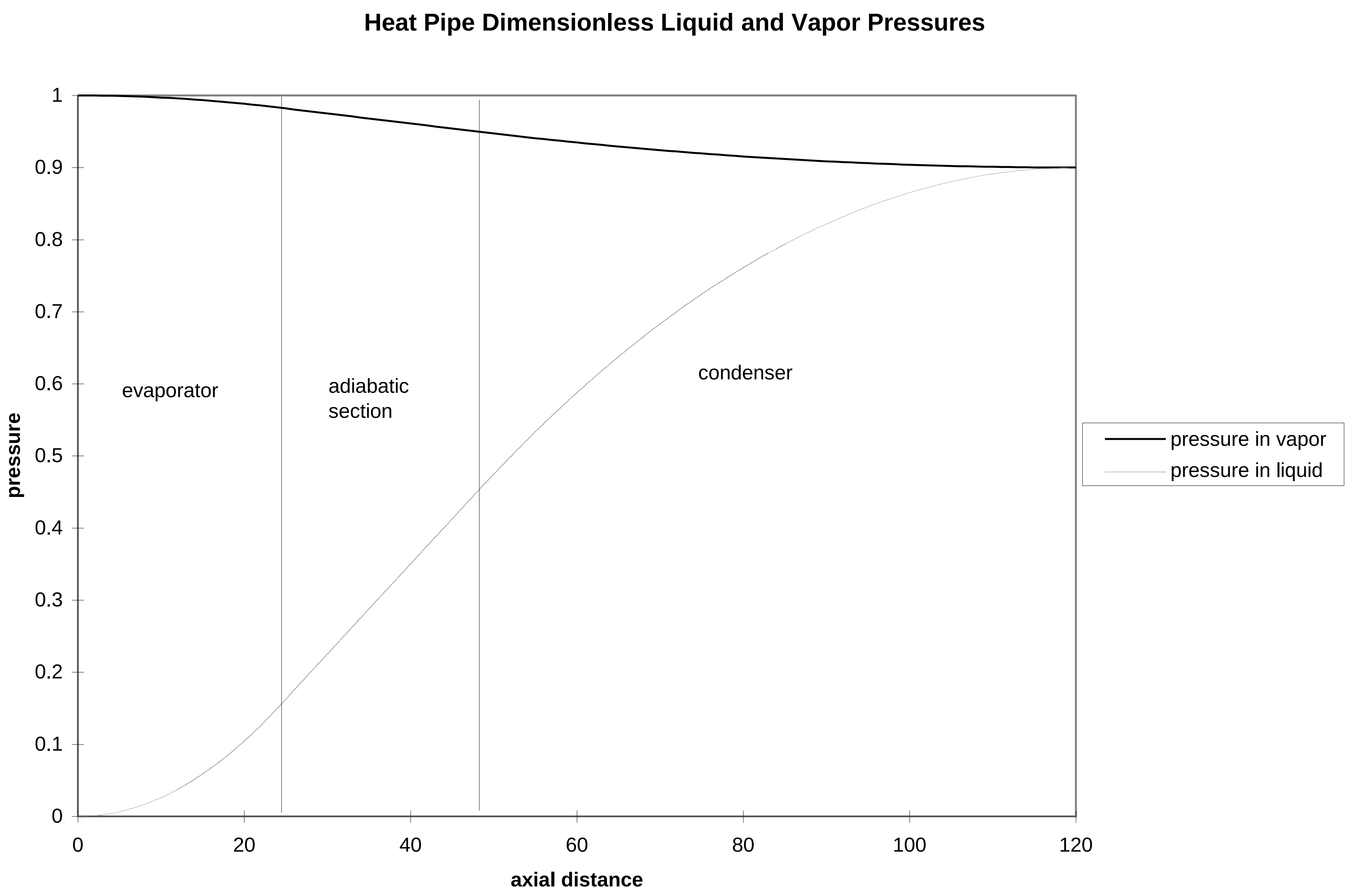


# Theory of Operation





Heat Pipe Dimensionless Mass Flow Rate
mass flow rate
1
0.9
0.8
0.7
0.6
0.5
0.4
0.3
0.2
0.1
0
evaporator
adiabatic
section
condenser
0
20
40
60
80
100
120
axial diatance
average d-less mass flow rate


Heat Pipe Dimensionless Liquid and Vapor Pressures
1
0.9
0.8
0.7
0.6
0.5
0.4
0.3
0.2
0.1
0
pressure
evaporator
adiabatic
section
condenser
pressure in vapor
pressure in liquid
0
20
40
60
80
100
120
axial distance

# Heat Pipe Types

- Constant conductance
- Variable conductance

- Screen wick
- Axial groove
- Loop Heat Pipes

# Constant Conductance vs. Variable Conductance Heat Pipes

- Constant conductance heat pipes (CCHPs)
  - are completely passive and have no independent means of temperature regulation
  - heat pipe temperature floats with the heat load and environment
  - conductance (or thermal resistance) is nearly constant
  - used in situations where payload temperature is not critical or is maintained by other means

# Constant Conductance vs. Variable Conductance Heat Pipes

- Variable conductance heat pipes (VCHPs)
  - include a non-condensable gas (NCG) reservoir
  - the presence of NCG blocks a portion of the heat pipe condenser
    - with no active control of the reservoir temperature, VCHPs naturally tend to moderate their temperature
      - at low heat pipe temperatures NCG volume expands reducing conductance
      - at high heat pipe temperatures NCG volume contracts increasing conductance
    - by controlling the temperature (pressure) of the reservoir and actively controlling the amount of NCG in the heat pipe, very tight temperature tolerances can be obtained

# Variable Conductance Heat Pipes

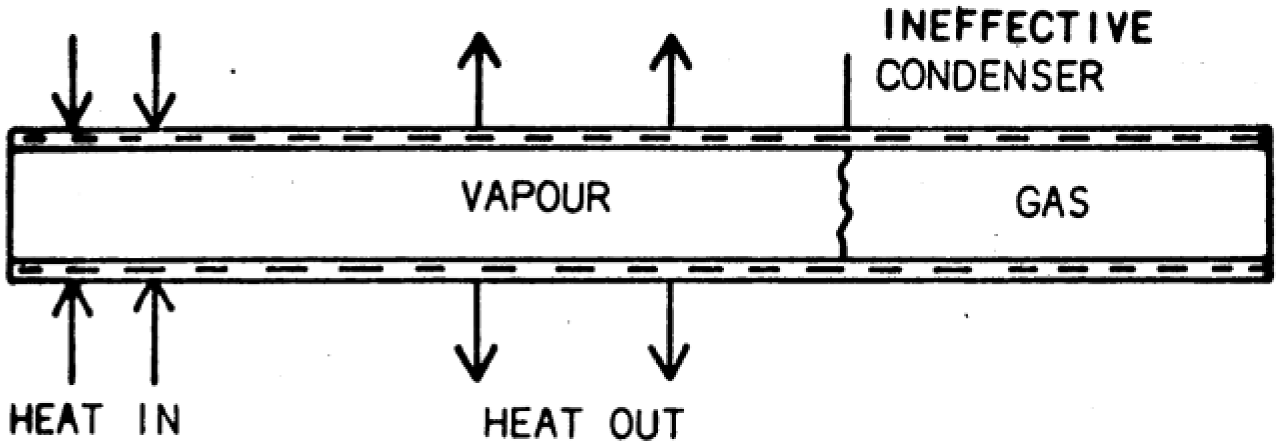



Fig.6.1 Equilibrium state of a gas-loaded heat pipe.

# Variable Conductance Heat Pipes

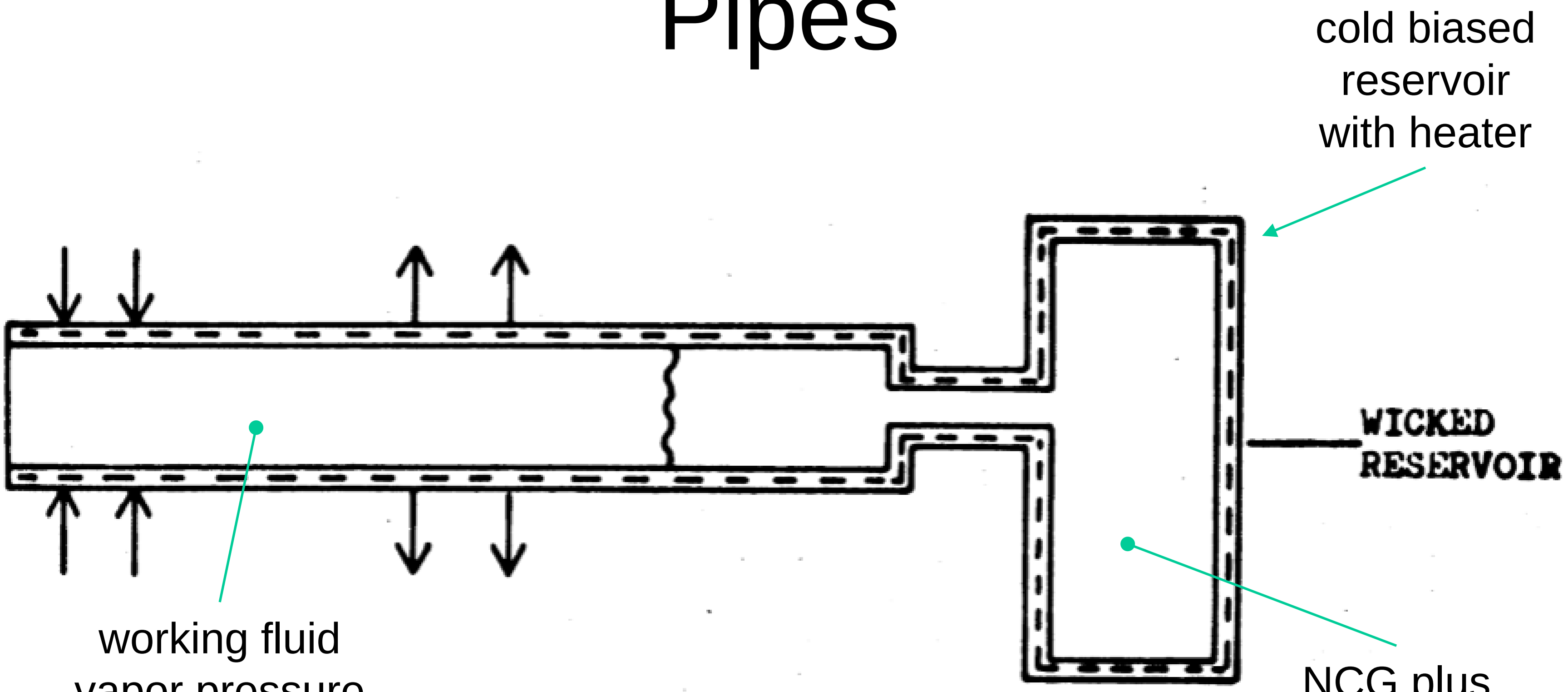
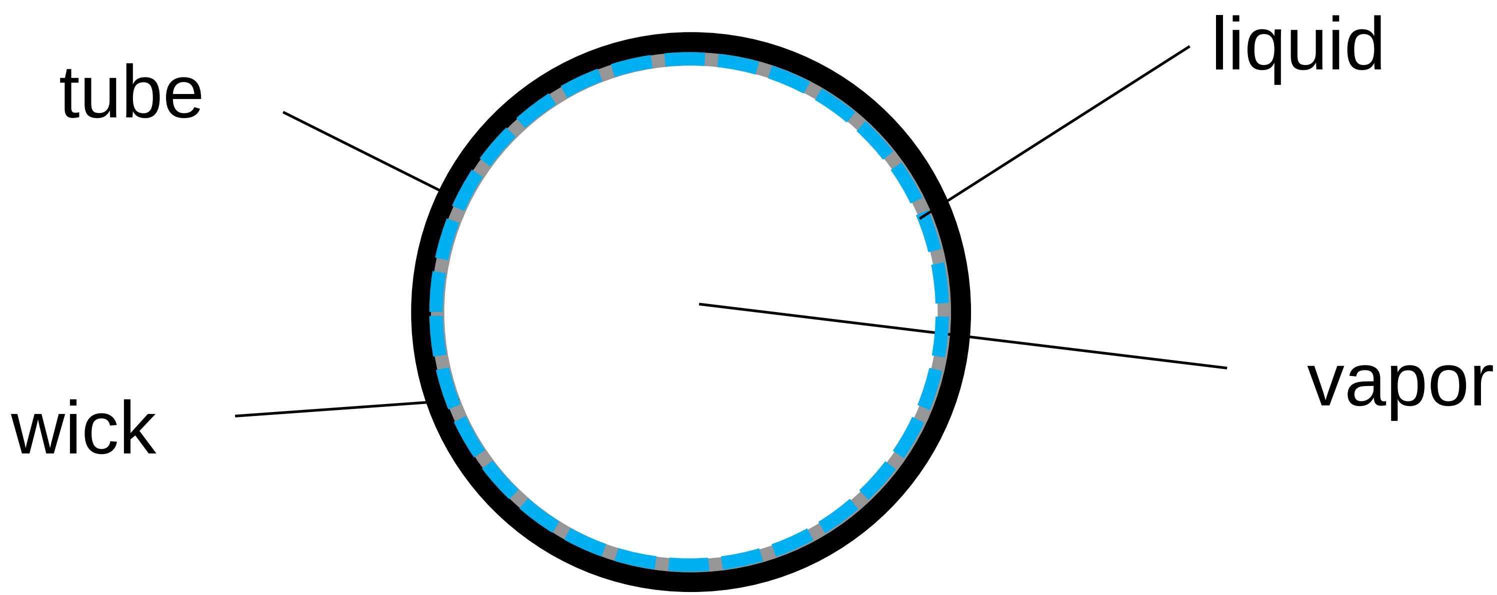



Fig.6.3 Cold wicked reservoir VCHP.

# Axial Wick Types - Screen Wicks

- Some earlier heat pipe designs used plain tubes with interior annular screen wicks



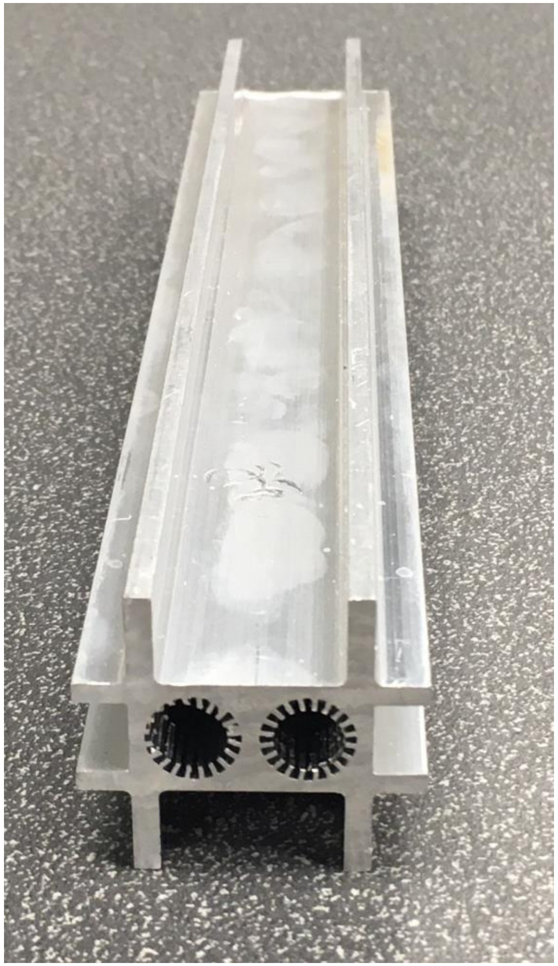
# Axial Wick Types - Screen Wicks

- Owing to their high liquid flow resistance, screen wick heat pipes typically have a low heat transport capacity
  - not commonly used in space applications today
  - axial groove heat pipes are most common

# Axial Wick Types - Extruded Wicks

Axial Groove

# Dual Axial Groove Heat Pipe Extrusion



# Axial Groove Heat Pipe Characteristics

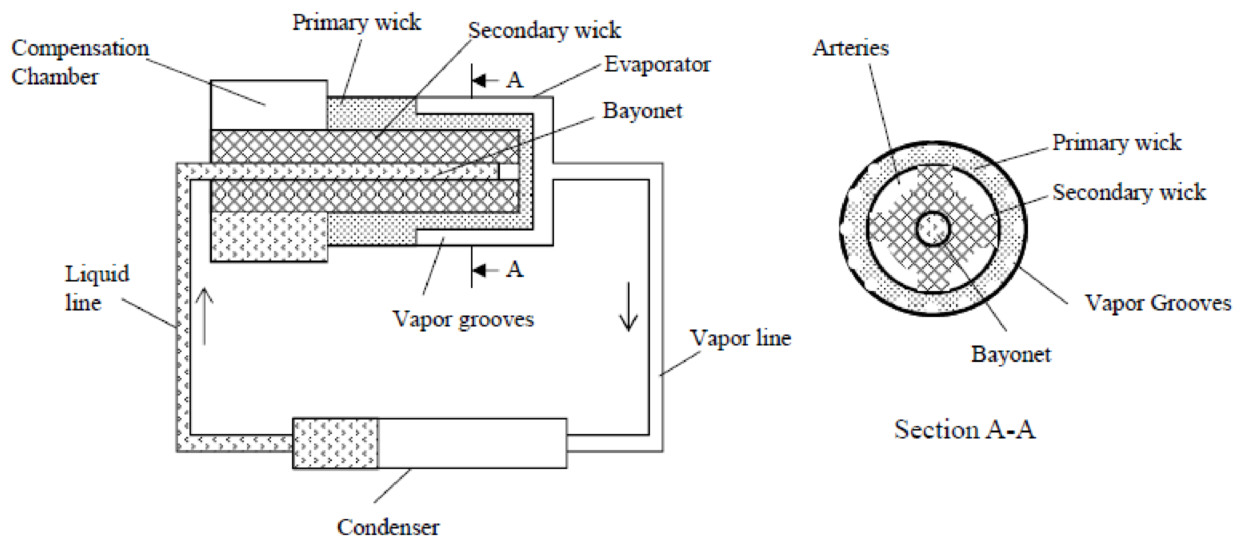
- Freezing is benign
  - small liquid charge
  - open liquid spaces
- Self-starting
- Simple physics

# Loop Heat Pipes

- A loop heat pipe is capable of transporting heat over long distances relative to a conventional heat pipe
- The evaporator and condenser sections are not integrated within a single tube, but are physically separated and connected by vapor and liquid transport lines.
- The transport lines can include flexible sections to facilitate heat transfer across a hinged interface.

# Loop Heat Pipes

- Primary wick with pore size on the order of 1 micron provides pumping power (vs. 1000 micron axial groove)
- Heat applied to the evaporator generates vapor at the outer surface of the primary wick
- The vapor transport line delivers this vapor to the condenser section
- A secondary wick structure connecting the evaporator and compensation chamber (CC) provides a path to move liquid from the CC to the evaporator when needed
- The compensation chamber temperature can be actively controlled by heaters to control the LHP setpoint – CC is naturally cold biased



# Loop Heat Pipes

- Have very high pumping head (5 psi or more) and are ground testable
- Compensation chamber temperature control allows simple control of setpoint
- Are replacing VCHPs in many situations
- Hundreds of loop heat pipes are flying
  - They are a time-tested design

# Loop Heat Pipe Characteristics

- Cannot be allowed to freeze – thin wall condenser tube is liable to bursting
- On startup
  - CC heater on to flood LHP plus evaporator
    - the liquid inventory prevents CC dryout
  - button heater initiates vapor formation
- Much more complex physics than axial groove heat pipes

# Axial Wick Types - Extruded Wicks

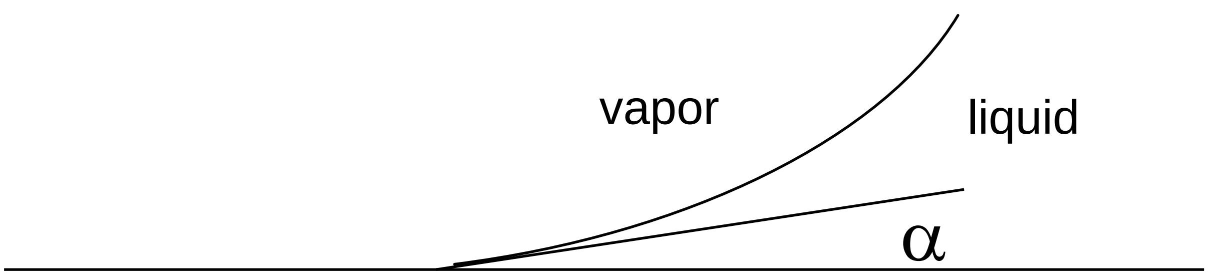
- Capillary pressure rise

$$\Delta p_{cap} = \sigma / r_{cap}$$

where $\Delta p_{cap}$ is the capillary pressure rise
$\sigma$ is the liquid surface tension
$r_{cap}$ is the capillary radius

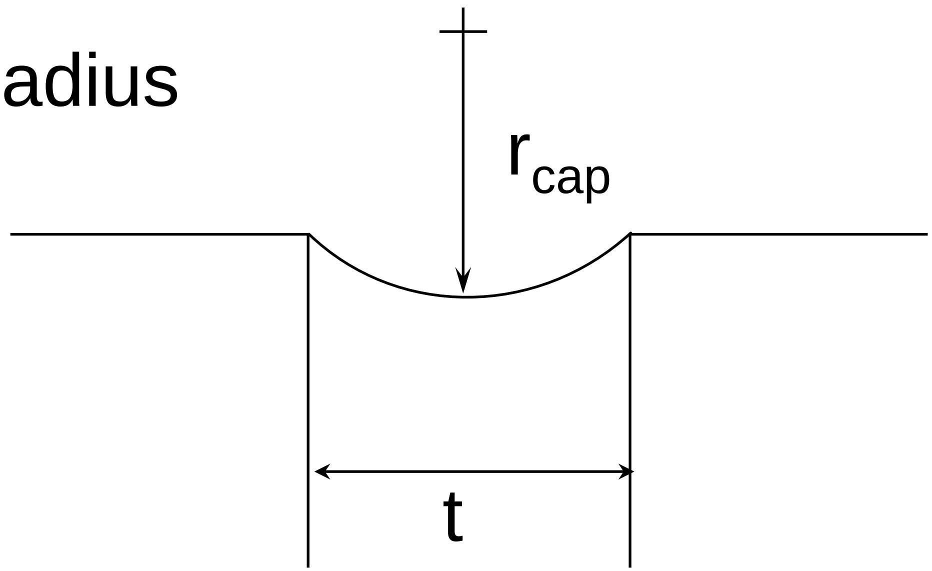
# Axial Wick Types - Extruded Wicks

- Contact angle
  - advancing contact angle, $\alpha_a$ - angle of a liquid advancing over a dry surface at infinitesimally small speed
  - receding contact angle, $\alpha_r$ - angle of a liquid receding over a dry surface at infinitesimally small speed



# Axial Wick Types - Extruded Wicks

- Capillary radius




- For a fully wetting fluid ($\alpha$=0) in a completely filled slot, the minimum value of $r_{cap}$ approaches t/2 and $\Delta p_{cap,max} = 2\sigma / t$

# Axial Wick Types - Extruded Wicks

- For a partially filled slot, the geometry is much more complicated, but...
- Axial grooves - fully wetting fluid
  - minimum partially filled capillary radius $< t/2$
  - partially dried out groove has **higher** capillary pressure rise than completely filled groove
  - this affects the ability of the heat pipe to rewet the evaporator under load after dryout (discussed later)

# Axial Heat Pipe Operating Limits

Steady-State Performance Limits

- Pressure drop limit
- Sonic limit
- Droplet entrainment limit
- Boiling limit

# Axial Heat Pipe Operating Limits

- To calculate the performance limit of a heat pipe
  - the pressure drop, boiling, sonic, and droplet entrainment limits are calculated
  - the lowest calculated heat load limit is taken as the heat pipe performance limit
    - space-based heat pipes are normally pressure drop limited
  - performance limit is a function of
    - heat pipe configuration
    - heat pipe working fluid
    - temperature
    - (possibly) tilt and gravity level

# Axial Pressure Drop Limit

- The pressure drop limit is reached when the combined pressure drop in the liquid and vapor is equal to the maximum pressure rise
  - maximum capillary pressure rise minus the gravity head
- Axial groove
  - acceleration and deceleration effects are minimal
  - laminar liquid frictional pressure drop dominates owing to small size of liquid passages
  - calculation is straightforward

# Sonic Limit

- If the vapor velocity in a heat pipe approaches the speed of sound, the flow will become choked
- Maximum vapor speed occurs at the evaporator exit
- Sonic velocity, a = $\sqrt{\gamma \text{ R T}}$
- where
  - $\gamma$ is the ratio of gas specific heats
  - R is the gas constant
  - T is the vapor temperature

# Entrainment Limit

- At high vapor velocities, droplets of liquid can be torn from the liquid surface and carried toward the condenser
- Once this occurs, the performance limit is reached
- The entrainment limit is usually taken to occur when the vapor Weber number is unity

# Entrainment Limit

- Vapor Weber number, $We_g$
  - ratio of inertial forces to surface tension forces

$$We_g = \frac{\rho_g u_g^2 t}{\sigma}$$

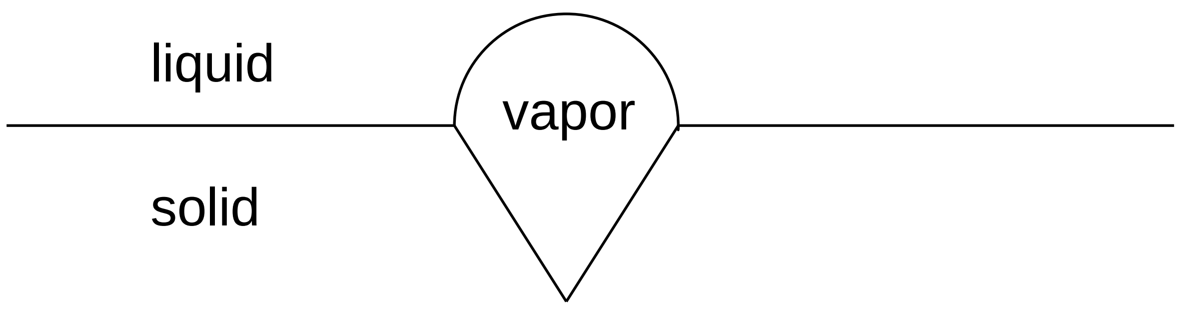
- where
  - $\rho_g$ is the vapor density
  - $u_g$ is the vapor velocity
  - t is a characteristic liquid dimension (normally the groove width)
  - $\sigma$ is the surface tension

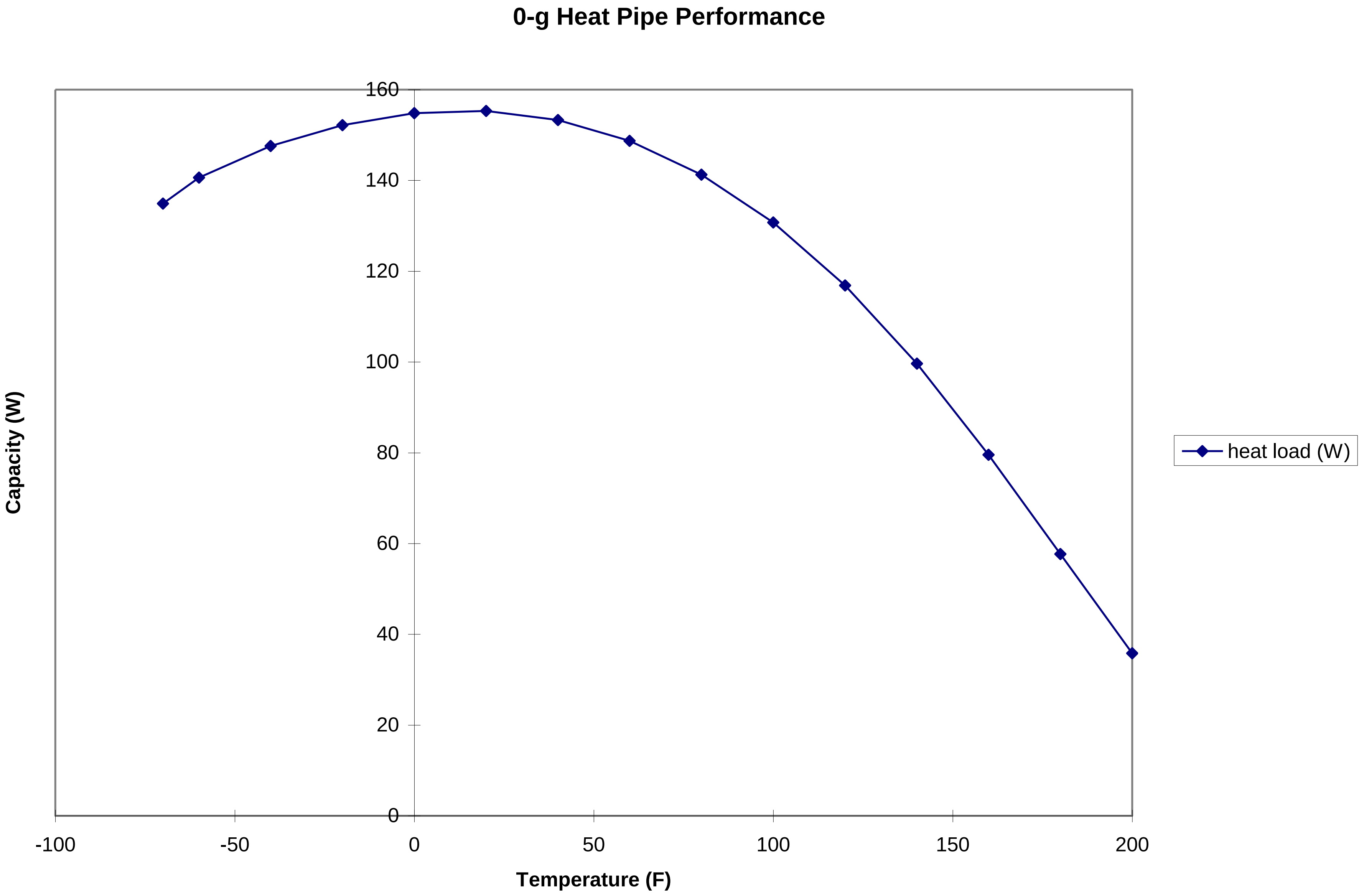
# Boiling Limit

- Because the evaporator heat input must pass through a liquid layer, the possibility of boiling exists
- If the liquid boils, the heat pipe will dry out locally and will cease to operate
- The boiling limit normally results in a limitation on the evaporator flux density

# Boiling Limit

- The evaporator flux density must be maintained at a low enough level that a nominal surface imperfection cannot become an active nucleation site
  - the superheat at the liquid/solid interface cannot exceed $\Delta T_{superheat} = (2\sigma/r_b)(dT/dp)_{sat'n}$
    - where $r_b$ is the cavity radius
      $(dT/dp)_{sat'n}$ is the slope of the saturation line




0-g Heat Pipe Performance
Capacity (W)
Temperature (F)
heat load (W)
160
140
120
100
80
60
40
20
0
-100
-50
0
50
100
150
200

# Off-Nominal Performance

- Freeze/thaw
- Startup
- Dryout
- Recovery from dryout

# Freeze/Thaw

- Axial groove heat pipes can be frozen and thawed at will owing to their small liquid charge and small open liquid channels

# Freeze/Thaw

- If a heat pipe freezes on-orbit, the evaporator will be completely dry
  - low condenser temperature freezes condensed vapor
  - evaporator normally attached to warm mass that provides heat for evaporation (even in the absence of input load)
  - evaporator empties eventually -- all working fluid frozen in condenser

# Freeze/Thaw

- To thaw a frozen heat pipe on-orbit, the condenser temperature must be raised above the working fluid freezing point
- The heat pipe cannot be thawed from the evaporator side (it’s dry)
- Normally radiator heaters are used

# Startup

- For heat pipes that cool an electrically heated payload, startup is straightforward
- If the heat pipe is not frozen, it will react immediately to the heat load
- As long as the applied load is lower than the transport capacity, the heat pipe will start up and operate successfully

# Startup

- For heat pipes that cool liquid loops, startup is more complex
- When the fluid loop places load on a cold heat pipe, it is a constant temperature load (rather than a constant heat load) which can result in very high transient loads
- The high transient loads can easily deprime the heat pipe
  - evaporator liquid evaporates faster than it can be replaced
- Axial groove heat pipe will reprime against the load

# Dryout

- Dryout occurs when the heat pipe transport capacity has been exceeded
  - evaporator load beyond the transport capability of the heat pipe
  - acceleration adds an unfavorable head term that drops capillary transport capacity below the applied load

# Recovery from Dryout

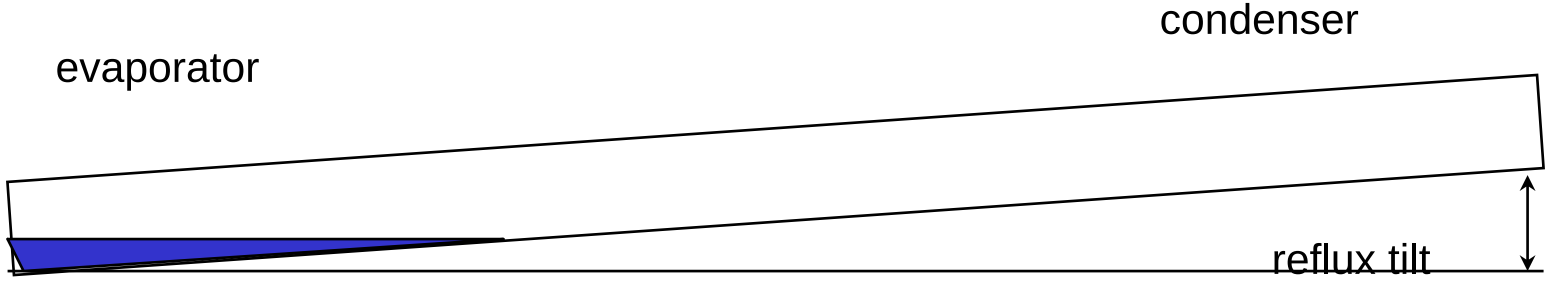
- If the evaporator heat load is maintained at a level higher than the heat pipe can transport, a runaway dryout will occur
  - the evaporator temperature will rise without limit
- If the high heat load (or low transport capacity) is a transient effect, the heat pipe will reprime provided that
  - the heat pipe structural limits are not exceeded
  - the transport limit with a partially dried out groove is higher than the applied load

# Performance Testing

- Reflux tilt
- Adverse tilt
- Heat pipe charging

# Reflux Tilt

- Heat pipes are often placed in reflux tilt in system level ground tests
- The heat pipe acts as a reflux boiler and is isothermal, but this gives no indication of whether
  - the heat pipe performance meets specification
  - the heat pipe charge is sufficient



# Adverse Tilt

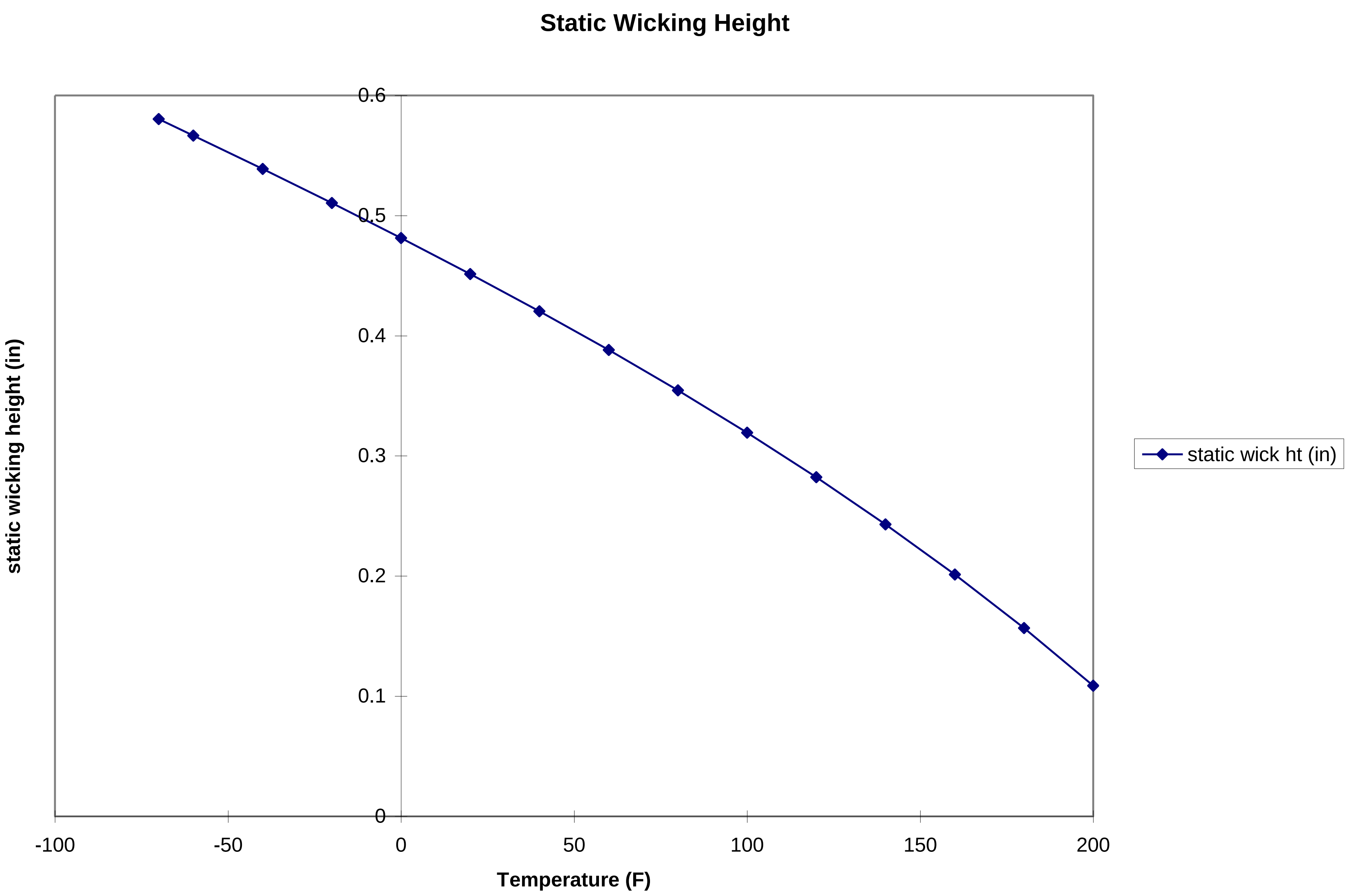
- Heat pipes for 0-g use are individually performance tested using adverse tilt
- The maximum heat pipe capillary head can be expressed in terms of a 1-g head rise

$$\Delta p_{max} = 2\cos(\alpha)\,\sigma/t = \rho\, g\, h$$

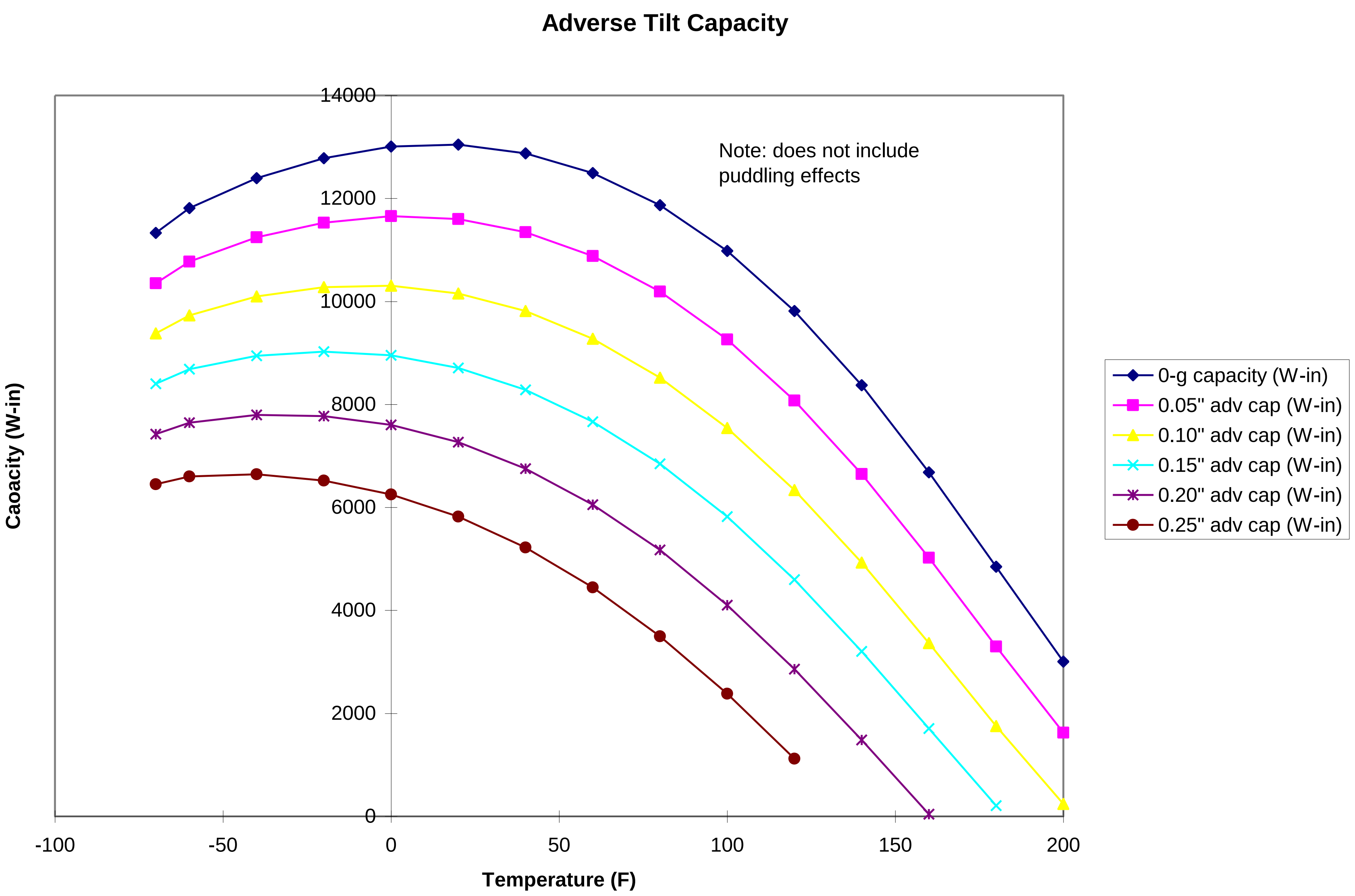
- where
  - $\alpha$ is the liquid/solid contact angle
  - $\rho$ is the liquid density
  - g is gravitational acceleration
  - h is static wicking height

# Adverse Tilt

- Or $h = 2\cos(\alpha)\,\sigma / (t\,\rho\,g)$
- The adverse tilt results in a loss of effective capillary pumping capacity
  - in a heat pipe with a static wicking height of 1/2”, a 1/4” adverse tilt results in a 50% decrease in pumping capacity
- If the transport/capillary pumping relationship is linear, the loss of capacity is proportional to the ratio of the adverse tilt and static wicking height (without accounting for puddle effects)

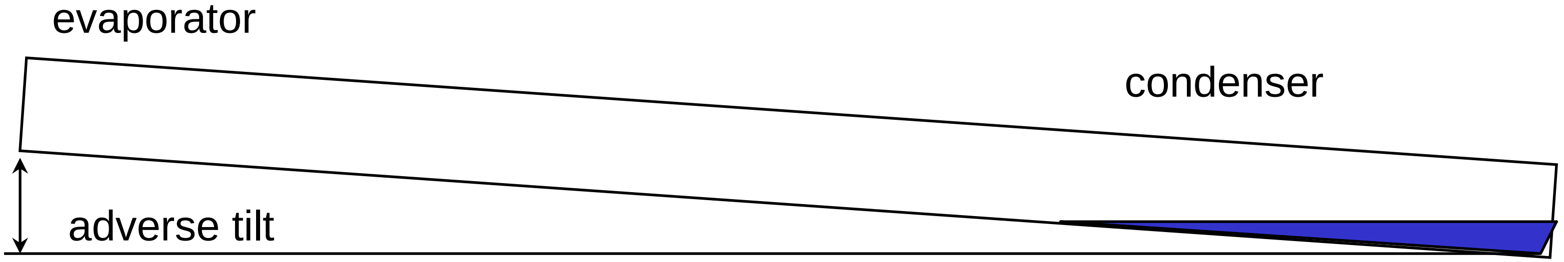


Static Wicking Height
static wicking height (in)
0.6
0.5
0.4
0.3
0.2
0.1
0
-100
-50
0
50
100
150
200
Temperature (F)
static wick ht (in)


Adverse Tilt Capacity
Note: does not include puddling effects
Caoacity (W-in)
Temperature (F)
14000
12000
10000
8000
6000
4000
2000
0
-100
-50
0
50
100
150
200
0-g capacity (W-in)
0.05" adv cap (W-in)
0.10" adv cap (W-in)
0.15" adv cap (W-in)
0.20" adv cap (W-in)
0.25" adv cap (W-in)

# Adverse Tilt

- Adverse tilt puddling in the condenser end aids performance by decreasing the effective adverse tilt



# Heat Pipe Charge

- Heat pipes are charged with enough liquid to fill the liquid grooves at the minimum operating temperature (plus a lagniappe)
- Owing to the liquid expansion that occurs at higher temperatures there is excess liquid in the heat pipe at most times
  - ammonia expands by 30% from -100 to 140°F
- The excess liquid puddles in the condenser in 1-g and forms a plug in the condenser end in 0-g

# Heat Pipe Selection

- Fluid choices
- Extrusion selection
- Sizing criteria
- NCG Generation
- On-orbit failure modes
- Modeling

# Fluid Choices

- The ideal working fluid combines
  - high latent heat of vaporization
  - high surface tension
  - good wetting ability
  - low viscosity
  - high liquid thermal conductivity
  - high vapor density
  - low freezing point
- The fluid selected for a heat pipe application is typically driven by the heat pipe operating temperature range

# Fluid Choices

- We can define a fluid figure of merit
  - $\Delta p_{cap} \propto \sigma$
  - $\Delta p_{viscous} \propto (1/Re)\, \rho_f V_f^2 = \{\mu_f /(\rho_f V_f d)\}\, \rho_f V^2 \propto \mu_f V_f$
  - $V_f \propto \dot{m}/\rho_f$
  - $\dot{m} \propto 1/\, h_{fg}$
  - $\Delta p_{viscous} \propto \mu_f /(\rho_f\, h_{fg})$
  - fluid figure of merit – higher is better
  - $\Delta p_{cap}/\Delta p_{viscous} \propto \rho_f\, h_{fg}\, \sigma/\mu$ (units of $W/m^2$)

# Typical Fluid Choices

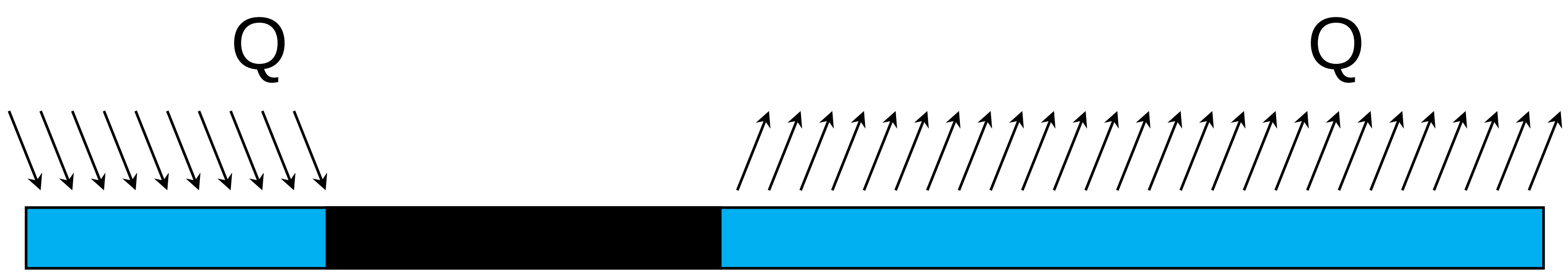
- Water – room temperature non-toxic applications
  - low toxicity
  - low vapor pressure
  - high surface tension
  - high freezing point
- Ammonia – room temperature external applications
  - high vapor pressure
  - high surface tension
  - moderate freezing point
- Propylene – low sink temperature applicaitons
  - high vapor pressure
  - low surface tension
  - low freezing point

# Heat Pipe Selection

- Select the fluid
- Decide whether CCHP or VCHP is required
  - CCHPs are typically used if the operating temperature of the heat pipe isn't critical
  - VCHPs are typically used if the heat pipe must be maintained within a narrow temperature range
- Check the transport capacity

heat load x (evaporator length/2 + adiabatic length + condenser length/2)

  - axial groove heat pipes come in transport capacities of as high as 50,000 W-in



# Heat Pipe Type Selection

- Obtain information from the manufacturer
  - transport capacity
  - heat input limit
  - static wicking height
  - overall UA
- Normal selection criteria -- usually applied to the case of one heat pipe out
  - 100% margin on heat pipe transport at maximum axial acceleration
  - sufficient margin on boiling limit (usually not an issue)

# NCG Generation

- All heat pipes contain some level of non-condensable gases (NCGs)
- NCGs block off sections of the condenser, reducing the effectiveness of the heat pipe
  - CCHPs should generally contain no more than ~100 ppm NCGs
  - VCHPs contain much more NCG by nature
- NCGS are formed when the working fluid reacts with the heat pipe material
  - NCG generation must be kept to a minimum

# NCG Generation

- Heat pipe manufacturers passivate their heat pipes prior to charging
- Passivation is a proprietary technique, but usually consists of repeatedly filling and operating the heat pipe prior to the final fill
- Goal of passivation is to form a oxide stable layer on the heat pipe wall that prevents additional NCG generation
- If the heat pipe is bent after the final fill, the protective layer can flake off and NCGs can form

# On-Orbit Heat Pipe Failure Modes

- Loss of working fluid
  - owing to poor workmanship or pressure vessel failure
    - very rare -- heat pipe manufacturers say that the chances of this are much less than 1%
  - owing to MMOD penetration
    - must be accounted for in design
- NCG generation owing to damage to the protective internal layer
  - launch vibration or poor handling
  - this is the most dangerous failure because it is likely to occur on all the heat pipes in the system

# Heat Pipe Modeling

- When CCHPs are included in thermal math models they can usually be modeled simply
  - use heat pipe overall UA as a thermal resistance
    - the end-to-end temperature drop in typical CCHPs is on the order of 5°F at nominal load
    - in VCHPs UA is a complex function of several variables
  - include flags to show if
    - transport limit is exceeded
    - heat flux input limit is exceeded

# Heat Pipe Modeling

- If axial acceleration components are a significant fraction of the static wicking height, the transport capability should be de-rated accordingly
- If transient startup, dryout, and rewetting are to be included, the heat pipe must be modeled in excruciating detail

# Conclusion

- Heat pipes are passive heat transport devices that find a wide range of uses in spacecraft
- If selected, analyzed, and used correctly, they can be of great benefit

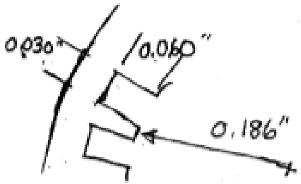
# Backup

# Axial Groove Heat Pipe Sample Calculation

Consider an axial groove aluminum extrusion.

vapor passage ID = 0.392 inches

25 liquid passages each 0.030" wide 0.060" tall




The heat pipe is 12 ft long and is charged with $NH_3$

evaporator is 2 ft long and is uniformly heated

adiabatic section is 2 ft long

condenser is 8 ft long and is uniformly cooled

what is heat pipe performance at 80°F?

Ammonia at 80°F

$\rho_f$ = 37.486 lbm/ft$^3$

$\rho_g$ = 0.5106 lbm/ft$^3$

$\mu_f$ = 0.341 lbm/ft hr

$\mu_g$ = 0.0277 lbm/ft hr

$h_{fg}$ = 498.8 BTU/lbm

$\sigma$ = 0.00138 lbf/ft

$k_f$ = 0.276 BTU/hr ft °F

$\left.\dfrac{dp}{dT}\right|_{sat} = 2.585 \dfrac{psi}{°F}$

$a_s$ = 1320 ft/s

1) what is the maximum capillary head rise?

ammonia and aluminum has a near zero contact angle, so

$$\Delta P_{cap} = \frac{2\sigma}{t} = 2 \times 0.00138 \frac{lbf}{ft} \times \frac{12}{0.030 ft} = \underline{\underline{1.104 \frac{lbf}{ft^2}}}$$

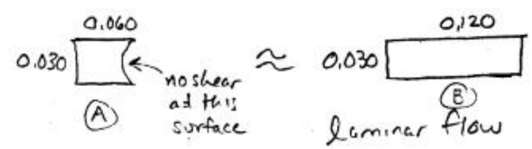
$$\Delta P_{cap} = \rho g h \qquad h = \frac{\Delta P_{cap}}{\rho g}$$

$$h = 1.104 \frac{lbf}{ft^2} \times \frac{ft^3}{37.486\ lbm} \times \frac{s^2}{32.17\ ft} \times \frac{32.17\ lbm\,ft}{lbf\ s^2}$$

$$= 0.0295 ft = \underline{\underline{0.353 in}} \quad \text{(static wicking height)}$$

2) what is the pressure drop limit?

consider the liquid channel first – assume laminar flow




4:1 aspect ratio

$$f = \frac{72.93}{Re_{d_{hyd}}}$$

Convective Heat Transfer
L. C. Burmeister, Wiley, 1983

$$\frac{dp}{dz} = -f \frac{1}{d_{hyd}} \frac{1}{2} \rho V^2$$

$$Re_{d_{hyd}} = \frac{\rho V d_{hyd}}{\mu}$$

$$\frac{dp}{dz} = \frac{-72.93\,\mu}{\cancel{\rho}\,\cancel{V} d_{hyd}} \frac{1}{d_{hyd}} \frac{1}{2} \cancel{\rho} V^{\cancel{2}} = \frac{-72.93}{2} \frac{\mu V}{d_{hyd}^2}$$

$$V = \frac{\dot{m}}{\rho A} \qquad \underline{\underline{\frac{dp}{dz} = \frac{-72.93}{2} \frac{\mu}{\rho} \frac{1}{d_{hyd}^2} \frac{\dot{m}}{A}}}$$

for A

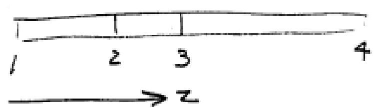
$$d_{hyd} = \frac{4A}{P} = \frac{4 \times 0.030 \times 0.060}{2 \times 0.060 + 0.030} = 0.048 \text{ inches}$$

for B

$$d_{hyd} = \frac{4A}{P} = \frac{4 \times 0.030 \times 0.120}{2 \times (0.030 + 0.120)} = 0.048 \text{ inches}$$

same result.

liquid channel $\Delta p$




adiabatic section

$$\frac{dp}{dz} = \frac{-72.93}{2} \frac{\mu}{\rho} \frac{1}{d_{hyd}^2} \frac{\dot{m}}{A}$$

$$\int_2^3 dp = \int_2^3 \frac{-72.93}{2} \frac{\mu}{\rho} \frac{1}{d_{hyd}^2} \frac{\dot{m}}{A}\, dz$$

$$p_3 - p_2 = \frac{-72.93}{2} \frac{\mu}{\rho} \frac{1}{d_{hyd}^2} \frac{\dot{m}}{A} (z_3 - z_2)$$

$$\dot{m} = \frac{Q}{n\, h_{fg}}$$

$Q$ = evaporator heat input

$n$ = # liquid channels

$$p_2 - p_3 = \frac{72.93}{2} \frac{\mu}{\rho} \frac{1}{d_{hyd}^2} \frac{1}{A} \frac{Q}{n h_{fg}} (z_2 - z_3)$$

evaporator

uniform heat input.

$$\frac{dp}{dz} = \frac{-72.93}{2} \frac{\mu}{\rho} \frac{1}{d_{hyd}^2} \frac{\dot{m}}{A} \qquad \dot{m} = \frac{Q}{n h_{fg}} \frac{z}{z_2}$$

$$\int_1^2 dp = \int_1^2 \frac{-72.93}{2} \frac{\mu}{\rho} \frac{1}{d_{hyd}^2} \frac{1}{A} \frac{Q}{n h_{fg}} \frac{z}{z_2}\, dz$$

$$p_2 - p_1 = \frac{-72.93}{2} \frac{\mu}{\rho} \frac{1}{d_{hyd}^2} \frac{1}{A} \frac{Q}{n h_{fg}} \frac{1}{z_2} \left.\frac{z^2}{2}\right|_1^2 \qquad z_1 = 0$$

$$p_2 - p_1 = \frac{-72.93}{2} \frac{\mu}{\rho} \frac{1}{d_{hyd}^2} \frac{1}{A} \frac{Q}{n h_{fg}} \frac{1}{z_2} \frac{z_2^2}{2}$$

$$p_1 - p_2 = \frac{72.93}{2} \frac{\mu}{\rho} \frac{1}{d_{hyd}^2} \frac{1}{A} \frac{Q}{n h_{fg}} \frac{z_2}{2}$$

half of the pressure drop per unit length of adiabatic section

Condenser

$$\frac{dp}{dz} = \frac{-72.93}{2}\frac{\mu}{\rho}\frac{1}{d_{hyd}^2}\frac{\dot m}{A} \qquad \dot m = \frac{Q}{nh_{fg}}\frac{z_4-z}{z_4-z_3}$$

$$\int_3^4 dp = \int_3^4 \frac{-72.93}{2}\frac{\mu}{\rho}\frac{1}{d_{hyd}^2}\frac{1}{A}\frac{Q}{nh_{fg}}\frac{(z_4-z)}{z_4-z_3}$$

$$P_4-P_3 = -\frac{72.93}{2}\frac{\mu}{\rho}\frac{1}{d_{hyd}^2}\frac{1}{A}\frac{Q}{nh_{fg}}\left.\frac{z_4 z-\frac{z^2}{2}}{z_4-z_3}\right|_3^4$$

$$P_4-P_3 = \frac{-72.93}{2}\frac{\mu}{\rho}\frac{1}{d_{hyd}^2}\frac{1}{A}\frac{Q}{nh_{fg}}\frac{z_4^2-z_4^2/2-z_3z_4+z_3^2/2}{z_4-z_3}$$

$$P_3-P_4 = \frac{72.93}{2}\frac{\mu}{\rho}\frac{1}{d_{hyd}^2}\frac{1}{A}\frac{Q}{nh_{fg}}\frac{\frac{1}{2}(z_4^2-2z_3z_4+z_3^2)}{z_4-z_3}$$

$$P_3-P_4 = \frac{72.93}{2}\frac{\mu}{\rho}\frac{1}{d_{hyd}^2}\frac{1}{A}\frac{Q}{nh_{fg}}\frac{1}{2}\frac{(z_4-z_3)^{\cancel{2}}}{\cancel{z_4-z_3}}$$

$$P_3-P_4 = \frac{72.93}{2}\frac{\mu}{\rho}\frac{1}{d_{hyd}^2}\frac{1}{A}\frac{Q}{nh_{fg}}\frac{1}{2}(z_4-z_3)$$

half of the pressure drop per unit length in adiabatic section

$$\Delta P_{tot} = P_1-P_2+P_2-P_3+P_3-P_4$$

$$\Delta P_{tot,f} = \frac{72.93}{2}\frac{\mu_f}{\rho_f}\frac{1}{d_{hyd}^2}\frac{1}{A}\frac{Q}{nh_{fg}}\left(\frac{z_2-z_1}{2}+z_3-z_2+\frac{z_4-z_3}{2}\right)$$

liquid properties

vapor passage: assume laminar flow

$$\frac{dp}{dz} = -\frac{64}{Re_d}\frac{1}{dz}\rho V^2$$

$$= \frac{-64}{2}\frac{\mu}{\cancel{\rho}\cancel{V}d}\frac{1}{d}\cancel{\rho}V^{\cancel{2}}$$

$$= \frac{-64}{2}\frac{\mu}{d^2}V \qquad V = \frac{\dot m}{\rho A}$$

$$= \frac{-64}{2}\frac{\mu}{d^2}\frac{\dot m}{\rho A}$$

$$\frac{dp}{dz} = \frac{-64}{2}\frac{\mu}{\rho}\frac{1}{d^2}\frac{\dot m}{A}$$

It can be shown, similarly to liquid passage calc.

$$\dot m_{evap} = \frac{Q}{h_{fg}}\frac{z}{z_2} \qquad \dot m_{adiabatic} = \frac{Q}{h_{fg}} \qquad \dot m_{cond} = \frac{Q}{h_{fg}}\frac{z_4-z}{z_4-z_3}$$

and

$$\Delta P_{tot,g} = \frac{64}{2}\frac{\mu_g}{\rho_g}\frac{1}{d^2}\frac{1}{A}\frac{Q}{h_{fg}}\left(\frac{z_2-z_1}{2}+z_3-z_2+\frac{z_4-z_2}{2}\right)$$

Sample calculation

$Q = 100\ W$

$$\Delta P_{tot,g} = \frac{64}{2} \frac{\mu_g}{\rho_g} \frac{1}{d^2} \frac{1}{A} \frac{Q}{h_{fg}} \left( \frac{l_{evap}}{2} + l_{adiabatic} + \frac{l_{cond}}{2} \right)$$

$$= \frac{64}{2}\ 0.0277 \frac{\cancel{lbm}}{\cancel{ft}\,\cancel{hr}} \times \frac{\cancel{ft^3}}{0.5106\,\cancel{lbm}} \times \frac{144}{0.392^2 \cancel{ft^2}} \times \frac{4 \times 144}{\pi \times 0.392^2 \cancel{ft^2}}$$

$$\times 100\cancel{W} \times \frac{3.4121\ \cancel{BTU}}{\cancel{W}\ \cancel{hr}} \times \frac{\cancel{lbm}}{498.8\ \cancel{BTU}} \times \underbrace{\left(\frac{2}{2} + 2 + \frac{8}{2}\right)\cancel{ft}}_{7\,ft}$$

$$\times \frac{lbf\ \cancel{s^2}}{32.17\ lbm\,ft} \times \frac{\cancel{hr^2}}{3600^2\ \cancel{s^2}} = 0.0222\ \frac{lbf}{ft^2}$$

$$\Delta P_{tot,f} = \frac{72.93}{2} \frac{\mu_f}{\rho_f} \frac{1}{d^2_{hyd}} \frac{1}{A} \frac{Q}{n h_{fg}} \left( \frac{l_{evap}}{2} + l_{adiabatic} + \frac{l_{cond}}{2} \right)$$

$$= \frac{72.93}{2} \times 0.341 \frac{lbm}{ft\,hr} \times \frac{ft^3}{37.486\,lbm} \times \frac{144}{0.048^2 ft^2} \times \frac{144}{0.030 \times 0.060 ft^2}$$

$$\times 100W \times 3.4121 \frac{BTU}{W\,hr} \times \frac{1}{25} \times \frac{lbm}{498.8\,BTU} \times 7ft$$

$$\times \frac{lbf\,s^2}{32.17\,lbm\,ft} \times \frac{hr^2}{3600^2 s^2} = 0.762\ \frac{lbf}{ft^2}$$

$$\Delta P_{tot} = \Delta P_{tot,f} + \Delta P_{tot,g} = 0.762 \frac{lbf}{ft^2} + 0.022 \frac{lbf}{ft^2}$$

$$= 0.784 \frac{lbf}{ft^2}$$

at 100W $\quad \Delta P_{tot} = 0.784 \frac{lbf}{ft^2}$

$\Delta P_{cap} = 1.104 \frac{lbf}{ft^2}$

$$Q_{max} = \Delta P_{cap} \frac{Q}{\Delta P_{tot}} = 1.104 \frac{lbf}{ft^2} \times \frac{100W}{0.784 \frac{lbf}{ft^2}} = \underline{\underline{141\,W}}$$

Heat pipe capability is 141 W based on pressure drop limit

Q: was it correct to ignore condensation pressure recovery?

Maximum recovery in condenser is velocity head at entrance

$\Delta P_{recovery} = \frac{1}{2} \rho_g V_g^2$ at condenser inlet

$$V_g = \frac{\dot m}{\rho_g A} = \frac{Q}{h_{fg} \rho_g A}$$

$$= 141\,W \times 3.4121 \frac{BTU}{W\,hr} \times \frac{lbm}{498.8\,BTU} \times \frac{ft^3}{0.5106\,lbm} \times \frac{4 \times 144}{\pi \times 0.392^2 ft^2}$$

$$= 2254\ ft/hr = 0.626\ ft/s$$

$$\Delta P_{recovery} = \frac{1}{2} \rho_g V_g^2 = \frac{1}{2} \times 0.5106 \frac{lbm}{ft^3} \times 0.626^2 \frac{ft^2}{s^2} \times \frac{lbf\,s^2}{32.17\,lbm\,ft}$$

$$= 0.0031 \frac{lbf}{ft^2}$$

1/10 of vapor $\Delta P$

$\frac{1}{300}$ of total $\Delta P$

## Sonic Limit

$V_g = 0.626 \text{ ft/s}$ $\qquad$ $a = 1320 \text{ ft/s}$

well below sonic limit

## Droplet Entrainment Limit

$$We_g = \frac{\rho_g V_g^2 d}{\sigma} \qquad d = 0.030 \text{ in}$$

$$We_g = 0.571 \frac{\cancel{lbm}}{\cancel{ft^3}} \times 0.626^2 \frac{\cancel{ft^2}}{\cancel{s^2}} \times \frac{0.030 \cancel{ft}}{12} \times \frac{\cancel{ft}}{0.00138 \cancel{lbf}} \times \frac{\cancel{lbf}\,\cancel{s^2}}{32.17 \cancel{lbm}\,\cancel{ft}}$$

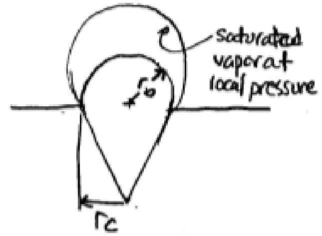
$$= 0.0126 \ll 1$$

no entrainment expected

## Boiling Limit

Boiling occurs at a surface cavity when

$r_b \geq r_c$ - bubble grows without limit




$$\Delta p_{cap} = \frac{F}{A} = \frac{2\pi r_b \sigma}{\pi r_b^2} = \frac{2\sigma}{r_b}$$

$$P_b = P_\infty + \frac{2\sigma}{r_b}$$

larger bubbles have lower capillary pressure

Bubble pressure is driven by local superheat

$$P_b = P_v = P_\infty + \Delta T_{supht} \left.\frac{dp}{dT}\right|_{sat}$$

$$\cancel{P_\infty} + \frac{2\sigma}{r_b} = \cancel{P_\infty} + \Delta T_{supht} \left.\frac{dp}{dT}\right|_{sat}$$

$$\Delta T_{supht} = \frac{2\sigma}{r_b} \left.\frac{dT}{dp}\right|_{sat}$$

at onset of boiling $\quad r_b = r_c$

$$\Delta T_{supht} = \frac{2\sigma}{r_c} \left.\frac{dT}{dp}\right|_{sat}$$

typical surface finishes have cavities on the order of 5 μm diameter $r_c = 2.5\ \mu m$

$$\Delta T_{supht} = \frac{2\sigma}{r_c}\left.\frac{dT}{dp}\right|_{sat} = 2\times 0.001381\frac{lbf}{ft}\times\frac{1}{2.5\times10^{-6}m}\times\frac{in^2\,°F}{2.585\,lbf}$$

$$\times\, 0.3048\frac{m}{ft}\times\frac{ft^2}{144\,in^2} = \underline{0.90°F}$$ suprht temp for ONB

1st order estimate of wall superheat

$T = T_{sat}(P_\infty)$

0.060" liquid $NH_3$

aluminum

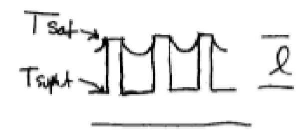
place where boiling would occur

$$Q = KA\frac{\Delta T}{\Delta x} \qquad A = \pi d_i\, l_{evap}$$

$$\Delta T = \frac{Q\Delta x}{KA} = 141W\times\frac{3.4121\,BTU}{W\,hr}\times\frac{12}{\pi\times0.392\,ft}\times\frac{1}{2ft}\times\frac{0.060ft}{12}\times\frac{hr\,ft\,°F}{0.276\,BTU}$$

$= 42.4°F \quad >> \quad 0.90°F$ ONB temp

In principle, $\frac{1}{50}$th of predicted $\Delta T$ would cause boiling in grooves. Will this occur? The fins help out




fins are 0.020" thick
0.060" long.

consider heat flow through the fins

25 fins each 0.020 in × 24" wide 12.0 in² total area

$$\Delta T = \frac{Ql}{KA} = 141W\times\frac{3.4121\,BTU}{W\,hr}\times\frac{0.060\,ft}{12}\times\frac{hr\,ft\,°F}{96.7\,BTU}\times\frac{144}{12\,ft^2}$$

← 6061-T6 Al.

$= 0.30°F$ which is less than the critical temperature.

A more detailed model would give a more accurate answer, but it looks as if we are OK.

Reynolds number check - is it laminar?

Vapor $$Re_g = \frac{\rho_g V_g d}{\mu} = 0.5106\frac{lbm}{ft^3}\times 2254\frac{ft}{hr}\times\frac{0.392\,ft}{12}\times\frac{ft\,hr}{0.0277\,lbm}$$

$= 1357$ laminar ✓

Liquid $$V_f = \frac{\dot m}{\rho_f A_{xs}} = \frac{\dot m}{\rho_f A_n} = \frac{Q}{h_{fg}\rho_f A_n}$$

$$= 141W\times 3.4121\frac{BTU}{W\,hr}\times\frac{lbm}{498.8\,BTU}\times\frac{ft^3}{37.486\,lbm}\times\frac{144}{0.030\times0.060\,ft^2}\times\frac{1}{25} = 82\frac{ft}{hr}$$

$$Re_f = \frac{\rho_f V_f d_{hyd}}{\mu} = 37.486\frac{lbm}{ft^3}\times 82\frac{ft}{hr}\times\frac{0.048\,ft}{12}\times\frac{ft\,hr}{0.341\,lbm}$$

$= 36$ laminar ✓