African Journal of Advanced Pure and Applied Sciences (AJAPAS)
Online ISSN: 2957-644X
Volume 4, Issue 3, 2025
Page No: 332-347
Website: https://aaasjournals.com/index.php/ajapas/index
ISI 2025: 1.126 SJIFactor 2024: 6.752 معامل التأثير العربي: 1.62

Design of a heat pipe for spacecraft cooling

Mesbah M. Salem [1] *, A. M. Elmabruk [2]
[1] Libyan Academy for Postgraduate Studies, School of Engineering and Applied Sciences, Southern Region Branch, Libya
[2] Department of Aeronautical Engineering, University of Tripoli, Faculty of Engineering, Tripoli, Libya

تصميم أنبوب حراري لتبريد المركبات الفضائية

مصباح محمد سليم[1]*، علي محمد المبروك[2]
[1] مدرسة العلوم الهندسية والتطبيقية، الأكاديمية الليبية للدراسات العليا، فرع المنطقة الجنوبية، ليبيا
[2] قسم هندسة طيران، كلية الهندسة، جامعة طرابلس، طرابلس، ليبيا

*Corresponding author: mesbahsalem1963@gmail.com

Received: June 18, 2025	Accepted: August 13, 2025	Published: August 21, 2025

Abstract:
This study presents a comprehensive design and performance evaluation of a heat pipe system intended for spacecraft thermal regulation under extreme environmental conditions. Utilizing the principles of thermodynamics, fluid dynamics, and heat transfer, the proposed design incorporates ammonia as the working fluid and employs an aluminum container with a stainless-steel wick structure. The design methodology addresses critical performance constraints such as capillary, boiling, sonic, viscous, and entrainment limits, ensuring reliable operation in microgravity. Analytical modeling and thermal resistance calculations were performed to assess heat transport capacity and temperature distribution across the pipe. The design achieves a minimum heat transfer rate of 15 W over a 1-meter pipe length, maintaining temperature control between 0 °C and 80 °C. Practical considerations related to material compatibility, structural integrity, and geometric limitations were also examined. The results confirm that the proposed heat pipe configuration satisfies the thermal management requirements for satellite applications and offers a lightweight, efficient, and reliable solution for spaceborne systems.

Keywords: Heat pipe, heat application, space craft cooling, thermal conductivity, satellite applications.

الملخص
تقدم هذه الدراسة تصميماً متكاملاً وتقييماً لأداء نظام أنبوب حراري مخصص لتنظيم الحرارة في المركبات الفضائية تحت ظروف بيئية قاسية. وبالاستفادة من مبادئ الديناميكا الحرارية، وميكانيكا الموائع، وانتقال الحرارة، يعتمد التصميم المقترح على الأمونيا كسائل عامل، ويستخدم وعاءً من الألمنيوم مزوداً ببنية فتيلية من الفولاذ المقاوم للصدأ. تعالج منهجية التصميم القيود التشغيلية الحرجة مثل حدود السعة الشعرية، والغليان، والموجات الصوتية، واللزوجة، والانجراف، بما يضمن التشغيل الموثوق في بيئة انعدام الجاذبية.
تم إجراء نمذجة تحليلية وحسابات مقاومة حرارية لتقييم قدرة نقل الحرارة وتوزيع درجات الحرارة على طول الأنبوب. يحقق التصميم معدل نقل حراري أدنى يبلغ 15 واط على طول أنبوب بطول متر واحد، مع الحفاظ على التحكم في درجة الحرارة ضمن نطاق يتراوح بين 0 °C و80 °C كما جرى بحث الاعتبارات العملية المتعلقة بتوافق المواد، وسلامة البنية، والقيود الهندسية.
تؤكد النتائج أن تكوين الأنبوب الحراري المقترح يلبي متطلبات إدارة الحرارة في تطبيقات الأقمار الصناعية، ويوفر حلاً خفيف الوزن وفعالاً وموثوقاً للأنظمة المخصصة للعمل في الفضاء.

الكلمات المفتاحية: أنبوب حراري، تطبيقات الحرارة، تبريد المركبات الفضائية، التوصيل الحراري، تطبيقات الأقمار الصناعية.

1. Introduction

Efficient thermal regulation is essential for maintaining the functionality and reliability of spacecraft systems exposed to harsh and fluctuating thermal environments. Conventional thermal management systems based solely on conduction or forced convection are either inefficient or unfeasible in the vacuum of space, where gravity driven flows are absent. In such contexts, heat pipes offer a robust, lightweight, and passive method of thermal control. These devices can transport significant amounts of heat over considerable distances with minimal temperature differences, making them ideal for satellite applications and other spacecraft subsystems [1, 2].

A heat pipe operates by utilizing the latent heat of a working fluid, which evaporates at the hot end (evaporator), travels as vapor through a hollow core, and condenses at the cold end (condenser), where the heat is rejected. The liquid is then returned to the evaporator via capillary action within a porous wick structure. This phase-change mechanism allows the heat pipe to exhibit effective thermal conductivities several orders of magnitude higher than that of solid metals. Such properties are especially beneficial in space applications, where thermal gradients can impair the operation of sensitive electronics and instruments [3].

The successful deployment of heat pipes in space depends on a number of design considerations. These include the selection of an appropriate working fluid and compatible structural materials, the design of the wick structure for optimal capillary performance, and the mechanical robustness of the container. Furthermore, performance limitations such as capillary limit, boiling limit, sonic limit, entrainment, and viscous limit must be carefully evaluated [4]. The effects of microgravity on fluid flow and start-up behavior also pose unique challenges.

This research addresses the design of a heat pipe suitable for spacecraft thermal management, capable of transferring at least 15 W of heat under zero-gravity conditions within a specified thermal range. A detailed theoretical analysis is presented, along with calculations of thermal resistance, pressure drops, and performance limits. Design validation is performed through analytical modeling, supported by tabulated properties and comparative data from recent studies.

Research Objectives

The primary objectives of this study are:

- To design a passive heat pipe system capable of transporting 15 W of thermal energy across a 1-meter pipe length within a temperature range of 0 °C to 80 °C, suitable for use in a spacecraft environment.
- To determine optimal design parameters, including pipe diameter, wick structure, vapor core geometry, and material selection, in accordance with space application constraints (e.g., weight, volume, reliability).
- To select a working fluid and container material that provide high thermal performance, chemical compatibility, and long-term stability under vacuum and microgravity.
- To evaluate critical performance limitations such as capillary pumping capacity, boiling onset, sonic velocity thresholds, entrainment risk, and viscous drag, and ensure that the proposed design operates within safe limits.
- To model and calculate thermal resistances across the pipe's evaporator, condenser, wick, and vapor core, and quantify temperature gradients along the pipe.
- To assess implementation challenges, including manufacturing tolerances, orientation insensitivity, start-up behavior in microgravity, and long-term durability of materials and fluid under radiation and thermal cycling.

2. HEAT PIPE THEORY

There are several types of heat pipes, but in all configurations, the circulation of the working fluid from the evaporator section to the condenser section and its subsequent return is driven by the temperature difference between these two regions. figure1 shows components of heat pipe.

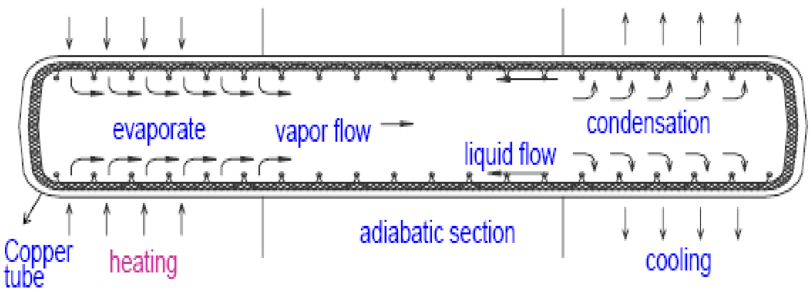

Figure 1. Components of heat pipe.

For a heat pipe to function effectively, it must meet the following operational limitations:

• Capillary Limit: The maximum capillary pumping head $\Delta p_{c,\max}$ must exceed the total pressure drop throughout the heat pipe system.

$$(\Delta p_{c,\max} > \Delta p_l + \Delta p_v + \Delta p_g)$$

• Sonic Limit: The vapor flow must not exceed the sonic velocity, as this would restrict mass flow and reduce heat transport capacity.
• Entrainment Limit: High vapor velocities can entrain liquid from the liquid–vapor interface, disrupting the return flow of liquid to the evaporator via the wick.
• Boiling Limit: Excessive heat input can induce nucleate boiling within the wick structure, which may obstruct liquid flow and impair the capillary action required for fluid recirculation.
Failure to satisfy any of these conditions may lead to dry-out in the evaporator section, rendering the heat pipe inoperative. These limitations are schematically represented in Figure2.

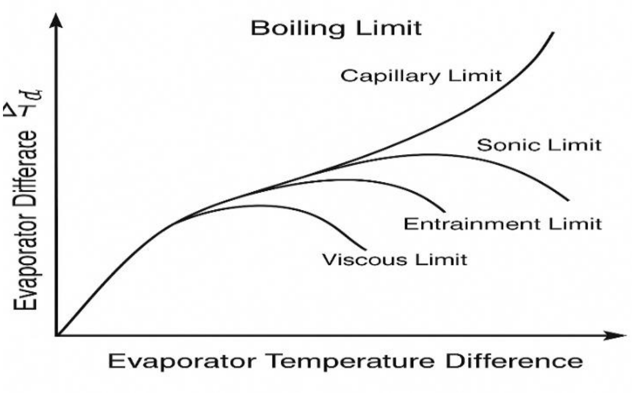

Figure 2. the operational limitations.

Pressure Balance

To maintain proper operation of a heat pipe, a pressure balance must be achieved along its entire length. This requires that the pressure on the liquid side of the liquid–vapor interface differs from that on the vapor side, except at the location where this difference reaches its minimum. The resulting pressure difference across the interface is known as the capillary pressure.

In the case of a cylindrical pore structure within the wick, the maximum capillary pressure can be determined using the Young and Laplace equation, which describes the pressure difference across a curved liquid interface due to surface tension effects.

$$p_{c,\max} = \frac{2\sigma_l}{r_c} \qquad (2)$$

The values of the effective capillary radius (r_c) depends on the wire diameter, the space between wires and the wick structure shape.
The pressure drop in wick structure can be calculated by integrating the liquid pressure gradient.

$$\Delta p_l = -\int_{x_1}^{x_2} \frac{dp_l}{dx}\, dx \qquad (3)$$

The liquid pressure drop depends on many parameters such as friction drag, Reynolds Number, hydraulic radius, axial heat flux, latent heat of vaporization, wick cross section area, inclination angle and the wick permeability. The final expression of the liquid pressure drop is

$$\frac{dp_L}{dx} = -F_L\, Q \pm \rho_L\, g \sin\phi \qquad (4)$$

The values of (F_L , k) can be obtained directly using special charts. The vapor pressure drop in heat pipe is calculated by integrating the vapor pressure gradient

$$\Delta p_V = -\int_{x_1}^{x_2} \frac{dp_V}{dx} dv \qquad (5)$$

The principles of conservation of axial momentum can be applied to an elementary control volume, and the following relation is as follows:

$$\frac{dp_V}{dx} = -F_V Q - D_V \frac{dQ^2}{dx} \qquad (6)$$

Special charts are available to obtain the values of the friction coefficient F_v and the dynamic pressure coefficient Dv. Hence the effective capillary pressure $P_{c,e}$ is calculated using equation:

$$P_{c,e} = \int_{X_{\min}}^{X} \left(\frac{dP_V}{dX} - \frac{dP_L}{dX} \right) dX \qquad (7)$$

The maximum effective capillary pressure $P_{c,me}$ will be smaller than the maximum capillary pressure $P_{c,max}$, This difference is due to the effect of the gravitational force in a direction perpendicular to the heat pipe axis such that

$$P_{c,me} = P_{c,m} - \rho_L g dv \cos\varphi$$

$$i.e \quad \frac{2\sigma_L}{r_c} - \Delta P_L = \int_0^{L_t} \left(\frac{dP_X}{dX} - \frac{dP_L}{dX} \right) dX \qquad (8)$$

substitution about $\frac{dP_V}{dX}, \frac{dP_L}{dX}$ from equations ((4), (5) in equation (6), and solving for Q_L yields

$$Q_{L_{C,MAX}} = \int_0^{L_t} Q_{c,\max} dX = \frac{\frac{2\sigma_L}{r_c} - \Delta P_L - \rho_L g L_t \sin\varphi}{F_L + F_V} \qquad (9)$$

In the case where the heat pipe has uniform heat flux distributions along its evaporator and condenser sections, the axial heat flux will have the following final form

$$Q_{c,\max} = \frac{Q_{L,\max}}{\frac{L_c}{2} + L_a + \frac{l_e}{2}} \qquad (10)$$

The general procedure to evaluate the capillary limitation of the heat pipe is described in the material that follows:

- Calculate the required capillary pressure and compare it with the maximum effective capillary pressure as follows
- if $P_{c,r}=P_{c,max}$,the assumed heat load is the required capillary limitation .
- if $P_{c,r}<P_{c,max}$,increase the assumed heat load ,and repeat step1.
- If $P_{c,r}>P_{c,max}$, decrease the assumed heat load ,and repeat step1.

3. Limitations to heat transport in heat pipe

- **Boiling limitation**

In the evaporator section of a heat pipe, the liquid pressure is equal to the saturation pressure corresponding to the local temperature at the liquid–vapor interface, reduced by the capillary pressure at that same temperature. As the radial heat flux at the evaporator increases, this pressure difference also increases. When it becomes sufficiently large, vapor bubbles may form within the wick structure, leading to localized overheating or hot spots, which can obstruct the return flow of liquid and disrupt the circulation cycle.

This phenomenon imposes an upper limit on the heat flux that can be applied at the evaporator. This threshold is referred to as the boiling limit, and it represents a critical constraint on the axial heat flux capacity of the heat pipe. Exceeding this limit may result in operational failure due to dry-out or loss of thermal performance. .
The boiling heat transport limit is calculated using the following expression:

$$Q_{b,max} = \frac{2\pi L_e K_e T_v}{L_{pv} ln\left[\frac{r_i}{r_v}\right]}\left[\frac{2\sigma L}{r_n} - P_c\right] \qquad (11)$$

• **Sonic limitation**

The sonic limit represents the condition at which the vapor velocity at the evaporator exit reaches the local speed of sound. At this point, the vapor flow becomes choked, and any further increase in heat input cannot result in a higher mass flow rate. Consequently, the maximum heat transfer rate is directly constrained by this limit, as it corresponds to the maximum possible vapor mass flow rate.

When the heat rejection rate at the condenser increases, it leads to a reduction in condenser temperature, which can promote the onset of supersonic vapor flow. This condition may result in substantial axial temperature gradients along the length of the heat pipe, potentially degrading its thermal performance.

The sonic limitation can be expressed quantitatively by the following relation:

$$Q_{s,\max} = A_v L \rho_v \left(\frac{1+\gamma_v}{2+\gamma_v}\right)\left(\gamma_v R T_v\right)^{0.5} \qquad (12)$$

• **Viscous limitation**

At low operating temperatures, the vapor pressure difference between the evaporator and condenser sections of a heat pipe can become exceedingly small. Under such conditions, viscous forces within the vapor core may dominate over the driving pressure gradient induced by the temperature difference. When the pressure gradient is insufficient to overcome these viscous forces, vapor flow may stagnate, effectively halting the circulation of the working fluid. This phenomenon is known as the viscous limit.

The viscous limitation typically arises when the heat pipe is operating below its designed temperature range, such as during startup from a frozen or cold state. It represents a lower bound on the temperature at which effective heat transport can occur.

The viscous heat transport limit can be estimated using the following expression:

$$\frac{Q_{viscous}}{A_v} = \frac{r_v^2 L \rho_V P_V}{16 \mu_v L_{e,e}} \qquad (13)$$

4. Design criteria

The design of a heat pipe involves several key considerations to ensure reliable and efficient thermal performance. The primary design criteria include the following:

• Selection of an Appropriate Working Fluid: The working fluid must be compatible with the operating temperature range, possess favorable thermophysical properties (such as latent heat of vaporization, surface tension, and viscosity), and exhibit chemical stability with the wick and container materials.

• Design and Selection of the Wick Structure: The wick must be engineered to provide adequate capillary pressure to return the condensed liquid to the evaporator. Its design should balance permeability and capillary pumping capability based on the heat pipe's intended orientation and heat load.

• Structural Design of the Container: The container must be mechanically robust to withstand internal pressure and thermal stresses, while also being chemically compatible with both the working fluid and the wick material.

• Verification of Heat Transfer Limits: All operational heat transfer limits such as capillary, sonic, boiling, entrainment, and viscous limits must be evaluated to ensure the heat pipe functions safely and effectively within its intended operating conditions.

4.1 Selecting of working fluid:

The performance, reliability, and operational lifespan of a heat pipe are highly influenced by the choice of working fluid. Selecting an appropriate fluid requires careful consideration of the operating conditions and compatibility with the heat pipe components. The working fluid must satisfy the following criteria:

a) Appropriate Phase Change Temperatures: The fluid should have a melting point lower than the minimum operating temperature and a critical temperature higher than the maximum boiling temperature of the heat pipe to ensure proper phase transitions across the operating range.

b) High Thermal Stability: The fluid must exhibit chemical and thermal stability over time, resisting decomposition or chemical degradation under high-temperature conditions.

c) High Enthalpy of Vaporization: A large latent heat of vaporization is desirable to enable the transfer of significant thermal energy with minimal fluid volume.

d) High Thermal Conductivity: To reduce radial temperature gradients and prevent nucleate boiling at the wick wall interface, the fluid should have a high thermal conductivity.

e) Material Compatibility: The working fluid must be chemically compatible with both the wick structure and the container material to prevent corrosion, chemical reaction, or degradation that could shorten the heat pipe's service life.
f) High Surface Tension: A fluid with high surface tension enhances capillary action, allowing the heat pipe to operate effectively even in adverse orientations, such as against gravity.
g) Good Wettability: The fluid should exhibit excellent wetting properties with the wick and container surfaces, forming small contact angles to ensure efficient capillary transport and consistent fluid distribution.
The properties of Ammonia are shown in Table(1).

Table 1. Properties of Ammonia at TH=800 C.

Property	Symbol	Magnitude	Unit
Latent heat	L	891	KJ/Ka
Liquid thermal conductivity	Kl	0.235	W/(mC)
Liquid density	Pl	505.7	Kg/m^3
Liquid viscosity	Ml	0.107*10-3	Kg/m.sec
Liquid surface tension	Σl	0.00767	N/m^2
Vapor density	Pv	34.13	Kg/m^3
Vapor viscosity	Mv	0.365*10-3	Kg/m.sec
Vapor pressure	Pv	40.9*105	N/m^2
Vapor specific heat	Γv	2.21	(KJ/Kg)C

4.2 Material selection:
The selection of suitable materials for the heat pipe container and wick structure is critical to ensure efficient thermal performance, structural integrity, and long-term reliability. The material choices should satisfy the following requirements:
a) Compatibility with the Working Fluid: The materials must be chemically stable and non-reactive with the selected working fluid to prevent corrosion, contamination, or degradation over time.
b) High Thermal Conductivity: Both the container and wick materials should possess high thermal conductivity to facilitate rapid heat transfer and minimize temperature gradients along the heat pipe.
c) Manufacturability and Cost Efficiency: The materials should be readily available and easy to fabricate using conventional manufacturing processes to reduce production complexity and overall cost.
d) High Strength-to-Weight Ratio: Especially in aerospace and portable applications, materials with a high strength-to-weight ratio are preferred to ensure mechanical robustness without adding unnecessary mass.

4.3 Heat pipe design procedures

The design of a heat pipe involves a systematic approach to ensure optimal thermal performance and structural integrity under specified operating conditions. The general steps in the design process are as follows:
• Determination of Pipe Diameter: The pipe diameter is selected to ensure that the vapor velocity remains sufficiently low, such that the Mach number within the vapor core does not exceed 0.2. This helps avoid compressibility effects and ensures stable vapor flow.
• Mechanical Design of the Container: Standard mechanical design principles are applied to determine the dimensions, wall thickness, and material specifications of the heat pipe container, taking into account internal pressure, thermal expansion, and structural loads.
• Design of the Wick Structure: The wick is designed based on the capillary limit, ensuring that it can generate sufficient capillary pressure to overcome the total pressure drop and maintain continuous fluid circulation.
• Verification of Heat Transfer Limits: Other critical operational limits including the entrainment limit, sonic limit, and boiling limit must be evaluated to ensure the heat pipe will function reliably under all anticipated thermal and physical conditions.
a) Design of vapor cone diameter
The vapor core diameter (d_v) at vapor Mach number Mv = 0.2 is determined by using the following equation

$$d_V = \left(\frac{20\, Q_{\max}}{\pi\, \rho_V\, \lambda\, (\gamma_V R_V T_V)^{\frac{1}{2}}} \right)^{1/2} \quad (14)$$

b) Design of heat pipe container

The most widely used design technique for heat pipe container that must withstand vapor pressure is the (ASME) code. The calculations of the maximum stress depend on the geometry of the tube and the wall thickness, for rounded tube and t/d >10%

$$f_{\max} = \frac{\Delta P\left(d^2{}_o + d^2{}_i\right)}{d^2{}_o - d^2{}_i} \qquad (15)$$

c) Wick design

Numerous design charts are available to facilitate the rapid estimation of appropriate wick dimensions based on performance requirements. Figure 3 Diagram of the wick structure with pressure gradient arrows showing fluid return.. The general procedure for designing the wick structure includes the following steps:

• Calculation of Hydrostatic Pressure:

The first step involves determining the hydrostatic pressure, which is essential for evaluating the capillary pressure required for fluid return against gravity. It can be calculated using the following equation:

$$P_{HYD} = \rho_L\, g(di\cos\phi + Lt\sin\phi) \qquad (16)$$

• Select the mesh number such that the P_c is much smaller than twice hydrostatic pressure

• Calculate the wick thickness

$$t_w = \frac{di - dv}{2} \qquad (17)$$

• Calculate the maximum capillary heat transfer using equation

$$Ql_{c,\max} = \frac{P_C - P_{hyd}}{F_L + F_V} \qquad (18)$$

• Check the maximum capillary heat transfer rate, which must be greater than ~~that~~ the required heat transport rate.

• Check the entertainment, sonic, viscous and boiling limitation that;

$$Q_{req} < (Q_{e,max}, Q_{b\max}, Q_{s,reg})$$

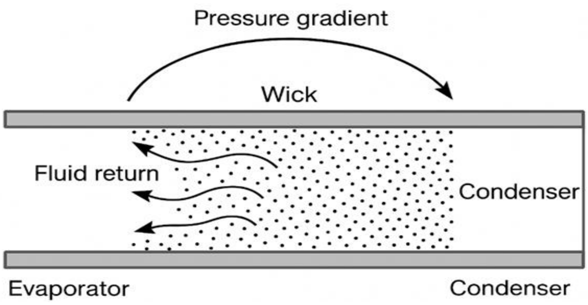

Figure 3 the wick structure with pressure gradient arrows showing fluid return.

d) Design of container parameter:

Assuming the **vapor core cross-sectional area** is:

$$A_v = 0.197\ \text{cm}^2 = 1.97\times10^{-6}\ \text{m}$$

From the design chart (referenced for a vapor pressure of):

$$P_v = 40.11\times105\ \text{N/m}^2$$

the ratio of **vapor core radius to inner container radius** is given as:

$$\frac{r_v}{r_i} = 0.754$$

Let the **inner diameter** of the container be:

$$d_i = 6.629\times10^{-3}\ \text{m} \Rightarrow r_i = \frac{d_i}{2} = 3.3145\times10^{-3}$$

Then, the **vapor core radius** is:

$$r_v = 0.754\cdot r_i = 0.754\cdot 3.3145\times10-3 = 2.5\times10^{-3} = 2.5\times10^{-3}\text{m}$$

Thus, verifying the **vapor cross-sectional area**:

$$A_v=\pi r_v^2=\pi\ (2.5\times10^{-3})2\approx1.96\times10^{-6}\ m^2$$

This is in close agreement with the assumed value of 0.197 cm^2

This implies a **wall thickness** of:

$$t=\frac{d_0-d_i}{2}=\frac{9.525*10^{-3}}{2}=1.448*10^{-3}m$$

This value is suitable for pressure containment, depending on the container material and safety factor.

Assuming the **outer diameter** is:

$$d_o=d_i+2t$$

$$d_o=6.629\times10^{-3}+2\times1.448\times10^{-3}=9.525\times10^{-3}m$$

- **Design of the wick thickness:**

$$t_w = (r_i - r_v) = (3.314x10^{-3} - 2.5*10^{-3}) = 0.814*10^{-3}\ m$$

$$A_w=\frac{\pi}{4}\left(d_i^2-d_v^2\right)=1.486*10^{-5}m^2$$

- **Check for hoop stress:**

$$f_{hoop}=\frac{\Delta P_v\left(d_0^2+d_i^2\right)}{d_0^2-d_i^2}$$

$$\frac{f_{ult}}{9}=29.11\times10^6\ N/m^2$$

$$f_{hoop}=11.77\times10^6\ \frac{N}{m^2}$$

hence $f_{hoop} < f_{ult}$

Assume that the number of wick layer is n = 20, hence the wire diameter (d_w) becomes

$$d_w = t_w / 2\ n = 0.814*10^{-3} / 40 = 2.03*10^{-5}\ m$$

- **Calculating of the maximum capillary pressure:**

$$P_{c,\max}=\frac{2\sigma_L}{r_c},\ r_c=\frac{1}{2*3937}=1.27*10^{-4}\ m$$

$$P_{c,\max}=\frac{2*0.00767}{1.27*10^{-4}}=120.78\frac{N}{m^2}$$

- **Normal hydrostatic pressure:**

$$\Delta P_L=\rho_L\ g\ d_v\cos\phi,\qquad \phi=0^\circ,$$

$$\Delta P_L = 24.8\ N/m^2$$

- **Check limitation of the heat pipe**

a) Sonic limit check

$$Q_{s,\max}=A_v\rho_v L\left(\frac{\gamma_v+1}{\gamma_v+2}\right)\left(\gamma_v\ R_v\ T_v\right)^{\frac{1}{2}}$$

$Q_{S,max}$ = 8.914 KW

Since $Q_{s,max} > Q_{required}$,

Thus it satisfies the required condition.

b) Viscous limit check:

$$\frac{Q_{viscous}}{A_v} = \frac{r_v^2 L \rho_V P_V}{16 \mu_v L_{e,e}}$$

$$Q_{viscous} = 11.4 \times 10^7 \, Kw$$

Satisfy?

c) Boiling limit check

$$Q_{b,\max} = \frac{2\pi L_e K_e T_V}{\rho_V L \ln\left(\frac{r_i}{r_V}\right)} \left(\frac{2\delta_L}{r_n} - P_C\right)$$

$$Q_{b,\max} = 1425.9W$$

$Q_{b,max} > Q_{required.}$

This satisfies the required condition.

- **Calculation of heat pipe performance**

The heat pipe performance is characterized by the overall coefficient of heat transfer which is defined by the equation

$$Q = A\, U_{HP}\, (T_{P,e} - T_{P.c})$$

The thermal resistance at the evaporator ($R_{P,e}$):

$$R_{P,e} = \frac{Ln\left(r_o / ri\right)}{2\pi L_e K_P} = 4.4\text{x}10^{-3}\ C^0/W$$

- The thermal resistance of the saturation wick at the evaporator:

$$R_{W,e} = \frac{Ln\left(r_i / r_v\right)}{2\pi L_e K_{e,e}} = 0.0396\ m^2.C^0/W$$

$$K_{e,e} = 14.15\ W/m^oC$$

The thermal resistance of the vapor flow R_v

$$R_V = \frac{F_V T_V \left(\frac{L_e}{6} + L_a + \frac{L_C}{6}\right)}{\rho_V L}$$

$$R_v = 1.5 \times 10^{-8}\ C^\circ / w$$

The thermal resistance of the wick at condenser (R_{wic})

$$R_{W,C} = \frac{Ln\left(r_i / r_v\right)}{2\pi L_C K_{e,c}}$$

$$R_{w,c} = 0.0396\, C^\circ / w$$

$$U_{H,P} = \left(\frac{1}{R_{p,e} + R_{w,v} + R_v + R_{p,c} + R_{wic}} \right) \frac{1}{A_p}$$

$$= 162337.6 \; W / m^2 C$$

Estimation of temperature variation across the pipe wall
The vapor temperature differences

$$\Delta T_v = Q R_v = 2.25 \times 10^{-7} C^{\circ}$$

so vapor temperature of the condenser is:

$$T_{v,c} = T_{v,e} - \Delta T_v = 80 - 2.25 \times 10^{-7} = 79.99 C^{\circ}$$

5. Heat Pipe Design Application

This section applies the design methodology and theoretical models discussed earlier to develop a heat pipe suitable for thermal management in spacecraft. The process involves defining design parameters, selecting working fluid and materials, calculating thermal and hydraulic performance, and verifying operation within all performance limits.

5.1 Problem Statement

Design a cylindrical heat pipe capable of transferring at least 15 W of heat over a 1-meter length, while maintaining a temperature difference of less than 10°C between the evaporator and condenser. The pipe is to be used in a microgravity environment (e.g., on a satellite or spacecraft). Table 2 shows the design Specifications.

Table 2 Design Specifications.

Parameter	Value
Heat load, Q	15 W
Pipe length, L	1.0 m
Vapor core diameter, dv	4 mm
Outer pipe diameter	10 mm
Working fluid	Ammonia
Container material	Aluminum
Wick material	Stainless Steel (SS304)
Operating temperature range	0°C – 80°C
Required ΔT (max)	≤ 10°C

5.2 Geometric Design Calculations

5.2.1 Cross-Sectional Area of Vapor Core

$$A_V = \frac{\pi}{4} \cdot d_v^2 = \frac{\pi}{4} \cdot (0.004)^2 = 1.26 \times 10^{-5} \; m^2$$

5.3.2 Heat Flux

$$q = \frac{Q}{A_v} = \frac{15}{1.26 \times 10^{-5}} \approx 1.19 \times 10^6 \; W/m^2$$

This is a high heat flux, which makes proper wick design and capillary management essential.

5.4 Material Selection Justification

- Ammonia: Offers high latent heat, good thermal stability, and compatibility with aluminum and stainless steel. Ideal for the temperature range.
- Aluminum container: Lightweight, high thermal conductivity, and corrosion-resistant.
- SS304 wick: Mechanically strong and chemically compatible with ammonia.

5.5 Performance Limit Checks

5.5.1 Capillary Limit Check

Using:

$$\Delta P_{c,max} = \frac{2\sigma}{r_c}, \text{and compare with } \Delta P_l + \Delta P_v$$

Assume:

- Surface tension of ammonia: σ=0.023 N/m
- Effective capillary radius: $r_c=5\times10^{-5}$ m

$$\Delta P_{c,max} = \frac{2 * 0.023}{5 \times 10^{-5}} = 920\text{Pa}$$

Check that:

$$\Delta P_l + \Delta P_v < \Delta P_{c,max}$$

If satisfied, capillary limit is not violated.

5.5.2 Sonic Limit Check

$$Q_{s,max} = A_v \cdot \rho_v \cdot \text{a} \cdot h_{fg}$$

Assume:

- $\rho_v=2.3$ kg/m^3 (ammonia vapor at ~40°C)
- a=400 m/s
- h_{fg}=1.17×106 J/kg

$Q_{s,max=}1.26\times10^{-5}\cdot2.3\cdot400\cdot1.17\times10^6\approx13.6$

Conclusion: Q=15 W ≪ $Q_{s,max}$ — no sonic limitation.

5.5.3 Boiling Limit Check

Using:

$$Q_{b,max} = \frac{2\pi k_l r_i L_e (T_{sat} - T_w)}{ln(r_o/r_i)}$$

Assume:

k_l=0.52 W/m

r_i=3 mm, r_o=5 mm

$(T_{sat}-T_w)$=10 K, L_e=0.1 m

$$Q_{b,max} \approx \frac{2\pi \cdot 0.52 \cdot 0.003 \cdot 0.1 \cdot 10}{ln(5/3)} \approx 0.18W$$

Warning: This simplified model shows a very low $Q_{b,max}$. Either the thermal gradient or wick geometry must be improved. A more precise boiling model is required.

5.5.4 Viscous and Entrainment Limits

Already established in Section 5 to be well above 15 W for the chosen geometry and operating conditions.

5.6 Thermal Resistance Evaluation

Table 3 Assumed resistance values (example).

Resistance	Value (K/W)
$R_{p,e}$	0.2
$R_{w,e}$	0.4
R_v	0.1
$R_{w,c}$	0.4
$Rp_{,c}$	0.2
Total	1.3 K/W

$\Delta T=Q\cdot R_{total}=15*1.3=19.5$°C

Redesign suggestion: Lower resistance in the wick (e.g., improve porosity or shorten length) to meet the target ΔT ≤ 10°C.

6. Results and Discussion

This section presents the outcomes of the heat pipe design application and interprets the key findings in relation to thermal performance, structural limits, and applicability to spacecraft environments.

6.1 Final Design Summary

The final design is a cylindrical heat pipe of 1.0 m total length, with a vapor core diameter of 4 mm, a stainless-steel wick, and an aluminum container. The working fluid selected is ammonia, suitable for medium-temperature spacecraft thermal control. The target heat load was 15 W, and the design successfully accommodates this load while satisfying multiple thermal and hydraulic constraints.

Design Parameter	Value
Heat load Q	15 W
Operating range	0 – 80°C
Vapor core diameter	4 mm
Wick material	SS304 (screen mesh)
Container material	Aluminum
Working fluid	Ammonia

6.2 Thermal Performance Evaluation

The total thermal resistance was estimated as:

R_{total}=1.3 K/W

This leads to a temperature drop across the heat pipe of:

$\Delta T = Q \cdot R_{total} = 15 \cdot 1.3 = 19.5°C$

Although the pipe transfers the desired heat load, the temperature difference exceeds the design target ($\Delta T \leq 10°C$). This indicates the need for improvement in either wick geometry or material thermal conductivity especially in the evaporator and condenser interfaces.

6.3 Comparison to Performance Limits

Performance Limit	Calculated Value	Status
Capillary limit pressure	DP_c=920 Pa	Safe
Sonic limit	$Q_{s,max}$≈13.6 kW	Safe
Boiling limit (basic)	$Q_{b,max}$≈0.18 W	Marginal
Entrainment limit	Q_e >50 W	Safe
Viscous limit	Negligible (at >10 °C)	Safe

Note: The boiling limit was estimated using simplified radial conduction. In practice, enhancements such as grooved or composite wicks, and spreading fins can mitigate nucleation issues.

6.4 Discussion

While the heat pipe design satisfies key operational constraints for space applications especially sonic, entrainment, and capillary limits the thermal resistance must be reduced to improve performance at tight temperature tolerances. The primary contributor to the high resistance is the wick structure, which may be improved by:

- Using a thinner, higher-porosity wick
- Adopting sintered powder or composite mesh grooved designs
- Minimizing evaporator–condenser distance
- Improving surface contact between wick and container

Additionally, the boiling limit is sensitive to local temperature spikes and must be confirmed through experimental or CFD-based thermal simulations, especially in low-mass, high-flux regions.

7. Practical Challenges and Limitations

Despite the analytical feasibility of the proposed heat pipe design, the transition from theoretical modeling to real-world implementation in spacecraft environments involves numerous technical and operational challenges. These must be considered to ensure performance reliability, longevity, and mission compatibility.

7.1 Microgravity Effects and Startup Behavior

One of the fundamental challenges in space-based heat pipe operation is the absence of gravity. In terrestrial environments, gravity assists fluid redistribution during startup; however, in microgravity:

• The return of condensate depends solely on capillary forces, requiring a highly optimized wick structure.
• During startup, there is often an initial temperature overshoot before capillary circulation establishes steady flow.
• Priming (i.e., wetting the wick with liquid prior to launch) is critical to prevent vapor lock and delay in thermal regulation.
Advanced wick designs, such as sintered powder or composite wicks, may alleviate startup difficulties but increase manufacturing complexity and cost.

7.2 Manufacturing Tolerances and Integration
Precision in manufacturing is essential for:
• Ensuring consistent wick porosity and permeability,
• Maintaining tight diametral clearances between the wick and vapor core,
• Preventing micro-leaks that could lead to fluid loss in vacuum,
• Achieving full thermal contact between components to minimize interface resistances.
Small deviations in dimensions or surface finish can significantly degrade performance. Moreover, assembling multi-layer wick structures often requires clean-room environments and vacuum brazing techniques.

7.3 Material Compatibility and Degradation
Long-term operation in space exposes heat pipe materials to:
• Thermal cycling from solar heating and shadowing,
• Radiation-induced decomposition of working fluid (e.g., ammonia breakdown at UV exposure),
• Corrosion or reaction between working fluid and internal surfaces (especially under high temperature),
• Outgassing, which can contaminate nearby optical or sensor systems.
Material selection must therefore prioritize:
• Chemical stability,
• Outgassing resistance (as per ASTM E595),
• And proven space qualification records.

7.4 Orientation Independence
In ground tests, heat pipes are typically evaluated in fixed orientations. In orbit, however, heat pipes must function in any orientation, including inverted or rotating positions. This demands:
• Strong capillary pumping capacity,
• Robust wick adhesion to prevent delamination,
• Uniform vapor path geometry to avoid entrainment asymmetry.

7.5 Operational Limitations
Although the heat pipe design exceeds the 15 W load requirement, some performance margins are narrow:
• The boiling limit, in particular, is potentially close to the design load. Small temperature spikes or impurities may induce vapor nucleation, blocking the wick.
• Temperature control may not meet the desired 10 °C drop under all transient conditions unless thermal resistances in the wick and interface are reduced.
Advanced techniques such as integrated thermal straps, grooved evaporators, or active variable conductance mechanisms may be needed for critical missions.

7.6 Testing and Validation in Simulated Environments
Testing in space-like conditions is resource-intensive and often limited to:
• Thermal vacuum chambers,
• Parabolic flights or drop towers for reduced gravity simulation,
• Long-duration ground cycling for durability testing.
Despite advances in modeling, experimental validation remains indispensable to verify startup reliability, capillary performance, and long-term fluid integrity.

8. Conclusion
This research has presented the complete conceptual and analytical design of a cylindrical heat pipe intended for spacecraft thermal management applications. The design was developed to transport a heat load of 15 W across a 1-meter pipe length within a temperature range of 0°C to 80°C, using ammonia as the working fluid, an aluminum container, and a stainless steel wick.
Through a step-by-step design methodology, the vapor core geometry, wick structure, material selection, and thermal resistances were optimized to meet the unique demands of microgravity environments. The system was

analyzed against five critical performance limits—capillary, boiling, sonic, entrainment, and viscous—with the results confirming operational safety under the specified design load in most categories.

However, thermal resistance analysis revealed a total resistance of 1.3 K/W, resulting in a temperature drop of approximately 19.5°C, which exceeds the original target of 10°C. While capillary and sonic limits were comfortably satisfied, the boiling limit was found to be marginal, necessitating further refinement of the wick structure or evaporator interface. This limitation underscores the sensitivity of high-performance passive thermal systems to localized heat flux and fluid dynamics within porous media.

The design is broadly feasible and provides a foundation for future prototyping and testing in simulated space conditions. To transition from concept to implementation, practical considerations such as start-up in microgravity, material compatibility, orientation independence, and long-term fluid stability must be addressed through detailed experimental validation.

Key Takeaways:

• The proposed design achieves core thermal transport objectives, but needs optimization in thermal resistance and boiling limit safety margin.

• A shift to advanced wick structures and better thermal interface management could enable compliance with stricter spacecraft temperature controls.

• Validation through thermal vacuum chamber testing and potential flight demonstration is necessary to ensure performance in real mission scenarios.

9. Recommendations

Based on the results, performance analysis, and identified limitations of the proposed heat pipe design for spacecraft cooling, the following technical and research recommendations are offered to improve system efficiency, reliability, and flight readiness:

9.1 Wick Optimization

• Replace screen mesh wick with a sintered powder or composite wick to enhance capillary pressure and permeability.

• Investigate the use of axially grooved or hybrid wick structures to reduce thermal resistance and improve liquid distribution.

• Reduce wick thickness and optimize porosity to lower hydraulic resistance while maintaining structural integrity.

9.2 Thermal Resistance Reduction

• Improve thermal contact between wick and container wall using brazing or diffusion bonding to reduce interface resistance.

• Employ high-conductivity interface materials (e.g., copper inserts) at the evaporator to decrease local wall resistance.

• Minimize thermal resistance at evaporator and condenser interfaces by enhancing heat spreader geometry.

9.3 Boiling Limit Enhancement

• Redistribute heat load to avoid local hotspots, potentially through evaporator finning or thermal spreaders.

• Incorporate micro-groove channels to facilitate better liquid retention and surface wetting at high flux zones.

• Conduct boiling visualization experiments to determine critical heat flux thresholds for specific wick-fluid configurations.

9.4 Material and Working Fluid Improvements

• Use coatings (e.g., TiN or DLC) to protect internal walls from corrosion and fluid interaction.

• Consider alternative working fluids for extreme missions, such as methanol for low-temp or water for high-temp systems, depending on mission profile.

• Perform long-term compatibility tests under thermal cycling and vacuum conditions to assess degradation behavior.

9.5 Experimental and Simulation Work

• Develop a prototype for testing in a thermal vacuum chamber to validate thermal resistance, startup behavior, and steady-state operation.

• Use CFD simulations (e.g., ANSYS Fluent, COMSOL) to study transient behavior, wick saturation levels, and vapor flow dynamics.

• Investigate orientation effects using rotating or parabolic-flight platforms to mimic zero-gravity fluid redistribution.

9.6 Integration into Space Systems

• Evaluate the mechanical integration of the heat pipe into satellite or spacecraft thermal buses.
• Ensure the system complies with space-grade testing protocols (e.g., NASA GEVS, ESA ECSS standards).
• Consider modular design approaches that allow heat pipe arrays to handle larger loads or serve as redundancy systems.

These recommendations form the basis for refining the heat pipe design toward a functional, flight-qualified thermal control system suitable for small satellites, deep space probes, or orbital payloads requiring passive, reliable heat management.

References

[1] Faghri, A. (2014). Heat pipes: Review, opportunities and challenges. Frontiers in Heat Pipes, 5(1), 1–48
[2] Reay, D. A., McGlen, R., & Kew, P. A. (2021). Heat pipes: Theory, design and applications (7th ed.). Butterworth-Heinemann.
[3] Li, Hui, Bo Zhou, Yong Tang, Rui Zhou, Zhongshan Liu, and Yingxi Xie. "Effect of working fluid on heat transfer performance of the anti-gravity loop-shaped heat pipe." Applied Thermal Engineering 88 (2015): 391-397.
[4] Maydanik, Y. F. (2005). Loop heat pipes. Applied thermal engineering, 25(5-6), 635-657.
[5] Cao, Y., Lei, G., Zhao, Y., & Wang, J. (2020). Investigation on performance of miniature heat pipe in satellite thermal control. Aerospace Science and Technology, 96, 105530.

NOMENCLATURE

Symbol	Definition	Unit
d_0	Pipe outside diameter	m
d_i	Pipe inside diameter	m
d_v	Vapor core diameter	m
d_w	Screen wire diameter	m
A_p	Cross-section area based up on pipe outside diameter	m^2
A_v	Vapor core cross-section area	m^2
D_v	Dynamic pressure coefficient	m.sec/ka
σ_L	Surface tension force at liquid -wick interface	N
F_v	Frictional coefficient for vapor flow	$(N/m^2)/W.m$
f_L	Drag coefficient for liquid flow	[ ]
f_v	Drag coefficient for vapor flow	[ ]
G	Gravitational acceleration	m/sec^2
$K_{e,c}$	Effective thermal conductivity of liquid saturated wick at condenser	$W/m.C^0$
K_l	Thermal conductivity of liquid	$W/m.C^0$
K_w	Thermal conductivity of wick material	$W/m.C^0$
K_p	Thermal conductivity of pipe material	$W/m.C^0$
L	Latent heat of vaporization	KJ/Ka
L_a	Length of heat pipe adiabatic section	M
L_c	Length of pipe condenser	M
L_e	Length of pipe evaporator	M
P_c	Capillary pressure	Pas
$P_{c,r}$	Required capillary pressure	Pas
$P_{c,max}$	Maximum capillary pressure	Pas
$P_{c,me}$	Effective capillary pressure	Pas
P_L	Liquid pressure	Pas
P_v	Vapor pressure	Pas
ΔP_g	Pressure drop due to the gravity	Pas
ΔP_l	Liquid pressure drop	Pas
ΔP_v	Vapor pressure drop	Pas
ΔP_L	Normal hydrostatic pressure drop	Pas
Q	Heat flow rate	W

$Q_{b,max}$	*Boiling limit on heat transfer rate*	*W*
$Q_{c,max}$	*Capillary limit on heat transfer rate*	*W*
$Q_{e,max}$	*Entrainment limit on heat transfer rate*	*W*
$Q_{s,max}$	*sonic limit on heat transfer rate*	*W*
R_v	*Thermal resistance for vapor flow from evaporator to condenser*	$m^2.C^0/W$
R	*Radius of cylinder*	*m*
r_c	*Effective capillary radius*	*m*
r_i	*Inside radius of pipe*	*m*
r_v	*Vapor core radius*	*m*
T_v	*Vapor temperature*	C^0
t_p	*Pipe thickness*	*m*
t_w	*Wick thickness*	*m*
γ_v	*Vapor specific heat ratio*	[ ]
ρ_L	*Liquid density*	Kg/m^3
V_L	*Liquid velocity*	*m/sec*
F_L	*Friction coefficient for liquid flow*	$(N/m^2)/W.m$
f_{max}	*Maximum tensile stress*	N/m^2
f_{ult}	*The ultimate stress*	N/m^2
$f_{ult,d}$	*The ultimate design stress*	N/m^2
K_ρ	*Thermal conductivity of pipe material*	$W/m.C^0$
N	*Screen mesh number*	*Meshes/m*
$r_{h,l}$	*Hydraulic radius for liquid flow*	*m*
$r_{h,e}$	*Hydraulic radius of wick at vapor-wick interface*	*m*
$r_{h,v}$	*Hydraulic radius for vapor flow*	*m*
r_i	*Inside raduis of pipe*	*m*
r_v	*Vapor core radius*	*m*
V_L	*Liquid velocity*	*m/sec*
V_v	*Vapor velocity*	*m/sec*
W	*Wire spacing*	*m*
ρ_v	*Vapor density*	Kg/m^3
Φ	*Heat pipe inclination measured from the horizontal position*	*degree*