Steady-State Performance Analysis of Lithium Heat Pipes Based on Improved Lumped Parameter Method

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Abstract

High-temperature heat pipes hold broad application prospects in space nuclear power and small mobile nuclear power sources. Due to their complex internal mechanisms, developing computational methods with relatively complete physical models yet simple solutions has remained a research focus. This paper proposes an improved lumped-parameter numerical heat pipe model that accounts for the effects of different flow regimes and Mach numbers on vapor flow, as well as variations in wick working fluid, and analyzes the flow and heat transfer characteristics of ultra-long lithium heat pipes for the HP-STMC space reactor. For both solid and fluid regions, a system of lumped-parameter differential equations for heat pipes based on annular mesh wick structures is established, solved using a Python program, and validated through comparison with other codes. This model is employed to simulate variations in lithium heat pipe operating parameters under constant heat sink and constant operating temperature conditions. The results demonstrate that: 1) the program exhibits good accuracy; 2) under constant heat sink conditions, thermal resistance, vapor velocity, and wick dryness decrease with increasing heating power; 3) under constant operating temperature conditions, both total thermal resistance and vapor thermal resistance change minimally when vapor does not reach turbulence, whereas thermal resistance increases rapidly when vapor enters turbulence, accompanied by increases in vapor velocity and wick dryness, while the liquid working fluid does not become turbulent and its velocity remains very low, with maximum Reynolds number and velocity of approximately 260 and 0.12 m/s, respectively; 4) for ultra-long lithium heat pipes operating at 1800 K and below, vapor thermal resistance constitutes approximately 3.9% of total thermal resistance. The findings of this study provide a theoretical foundation for the application of alkali metal heat pipes, with lithium heat pipes as a representative example.

Full Text

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Abstract

High-temperature heat pipes offer broad application prospects in space nuclear power and small mobile nuclear power sources. Due to their complex internal mechanisms, developing computational methods with complete physical models yet simple solutions has remained a research focus. This paper proposes an improved lumped-parameter numerical heat pipe model that accounts for the effects of different flow regimes and Mach numbers on vapor flow, as well as variations in wick working fluid. The flow and heat transfer characteristics of ultra-long lithium heat pipes in the HP-STMC space reactor are analyzed using this model. For both solid and fluid regions, a system of lumped-parameter differential equations is established for heat pipes with annular mesh wicks. The equations are solved using a Python program and validated against other codes. The model is then employed to simulate the operating parameter variations of lithium heat pipes under fixed heat sink and fixed working temperature conditions. The results indicate that: (1) the program exhibits good accuracy; (2) under fixed heat sink conditions, thermal resistance, vapor velocity, and wick dryness decrease with increasing heating power; (3) under fixed working temperature conditions, both total thermal resistance and vapor thermal resistance change minimally when vapor flow remains laminar, but increase rapidly when vapor transitions to turbulence, accompanied by rising vapor velocity and wick dryness, while liquid working fluid never reaches turbulent flow and maintains very low velocity with maximum Reynolds number and velocity of approximately 260 and 0.12 m/s, respectively; (4) for ultra-long lithium heat pipes operating at 1800 K and below, vapor thermal resistance accounts for approximately 3.9% of total thermal resistance. These findings provide a theoretical foundation for the application of alkali metal heat pipes, with lithium heat pipes as representative examples.

Keywords: lumped parameter method; lithium heat pipe; Python; flow and heat transfer; ultra-high temperature; numerical model

Introduction

Heat pipe reactors, which employ alkali metal heat pipes as heat transfer components, offer advantages including passive operation, long lifespan, compact size, and high reliability, making them promising for deep-sea nuclear power, deep space exploration, and defense applications [1-4]. As the core heat transfer equipment in heat pipe reactors, alkali metal heat pipe performance is critical for safe and effective reactor operation [5-7]. Since experimental research on alkali metal heat pipes faces difficulties in internal parameter measurement, high costs, and long testing cycles, constructing effective theoretical models for simulation analysis is essential [8].

A. Faghri et al. treated the wick as pure conduction and ignored vapor thermal resistance, proposing a thermal resistance network method for off-design simulation of sodium heat pipes [9]. Chai Baohua et al. developed a coupled method that considered two-dimensional compressible laminar flow in the vapor region while treating the wall and wick regions as pure conduction for potassium heat pipe simulation [10]. Wang Chenglong et al. divided the vapor region into three stages during startup and coupled it with wick and wall equations to simulate the startup process of sodium and potassium heat pipes [11-12]. A. Faghri proposed a computational method that treated the entire heat pipe as an integrated system rather than analyzing vapor flow separately, but this approach assumed uniform working fluid distribution and could not accurately predict the effects of evaporation and condensation on fluid distribution and capillary forces [13]. C. Ferrandi et al. proposed a lumped-parameter method that considered the overall cycle of both vapor flow and wick fluid flow, but only accounted for laminar, incompressible vapor flow [14]. Li Huaqi et al. developed the SNPS-HPD program using a one-dimensional vapor flow model that considered compressibility and Reynolds number effects to simulate heat transfer performance of lithium heat pipes in HP-STMCs (Heat Pipe-Segmented Thermoelectric Module Converters) reactors, but neglected internal wick flow, thus preventing complete investigation of the vapor-liquid circulation [15]. Current theoretical model research on alkali metal heat pipes mostly considers only vapor flow, rarely addressing liquid flow within the wick or the impact of evaporation and condensation on the overall pressure cycle, particularly for ultra-high temperature lithium heat pipes where relevant studies are scarce.

Building upon the work of C. Ferrandi et al. [14], this paper proposes an improved lumped-parameter method that considers vapor flow under different flow regimes and Mach numbers, as well as wick flow and its fluid variations. The effects of heating power on flow and heat transfer performance of ultra-long lithium heat pipes in HP-STMCs space reactors under different operating conditions are investigated.

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1 Lumped-Parameter Mathematical Model

The lumped-parameter numerical model divides the heat pipe into several control volumes represented by nodes, where thermophysical properties and average temperatures are assigned to the nodes. These nodes are interconnected through thermal resistance, thermal capacitance, and thermal inductance elements, with each node described by a set of one-dimensional linear differential equations that are solved numerically. The model consists of solid and fluid network models, with each component represented by nodes. The structural principle of the heat pipe and solid network model is shown in Figure 1 [Figure 1: see original paper].

The solid model primarily comprises the wall region and wick region, each divided into evaporator, adiabatic, and condenser sections. Radial and axial thermal resistances are given by Equations 1 and 2:

[Mathematical expressions for radial and axial thermal resistances would appear here]

The effective thermal conductivity and effective heat capacity of the mesh wick are calculated using Equations 3-4, while the effective thermal conductivity and effective heat capacity of the entire annular mesh wick are calculated using Equations 5-7 [16]. Other heat capacities are calculated using Equation 8.

Since the working fluid thickness in the evaporator and condenser sections changes during operation (decreasing in the evaporator due to evaporation and increasing in the condenser due to condensation), the annular wick must be updated after each time step. The update method varies the wick control volume thickness, with updated parameters including m_C, ε, k_{eff}, C_{eff}, V_w, and V.

Differential equations are established for the six nodes of the solid model:

[Mathematical expressions for solid model differential equations would appear here]

1.2 Fluid Model Figure 2 [Figure 2: see original paper] illustrates the fluid network model described in circuit diagram form, while Figure 3 [Figure 3: see original paper] shows the fluid dynamic cycle model. To simplify the control equations to ordinary differential equations, the following assumptions are made: (1) fluid flow is one-dimensional; (2) vapor follows the ideal gas law; (3) vapor expansion and compression are adiabatic processes; (4) evaporation and condensation occur only in the evaporator and condenser, respectively; (5) liquid mass flow rate is constant in the adiabatic section; (6) temperatures of the wick mesh and working fluid are identical.

1.2.1 Vapor Reservoir Based on the ideal gas assumption, $\rho = p/(R_g T)$. Using the continuity equation $\dot{m} = \sum \dot{m}_i$, differential equations for the evaporator and condenser vapor reservoirs are obtained:

Evaporator vapor reservoir: [Mathematical expression]

Condenser vapor reservoir: [Mathematical expression]

where $\dot{m}_{evap}$, $\dot{m}_{cond}$, and $\dot{m}_v$ represent evaporation mass rate, condensation mass rate, and vapor channel mass flow rate, respectively.

1.2.2 Vapor Flow Channel Based on the adiabatic compression assumption, the vapor reservoir temperature change is: [Mathematical expression]

Accounting for local resistance from bends in the vapor flow channel through an effective length L_{eff}, the one-dimensional differential equation under one-dimensional flow assumption is: [Mathematical expression]

Neglecting vapor property variations along the flow channel: [Mathematical expression]

where K represents the vapor local resistance coefficient. The viscous resistance formula considering turbulence and compressibility effects is [12]: [Mathematical expression]

where μ, γ, and M represent dynamic viscosity, adiabatic index, and Mach number, respectively. After rearrangement: [Mathematical expression]

where R_{va} and L_{va} represent vapor flow resistance and flow inductance, respectively.

1.2.3 Liquid Reservoir Continuity equations are established for the two liquid reservoirs: [Mathematical expressions]

1.2.4 Liquid Flow Channel For annular mesh wicks, fluid primarily returns through the annular gap. Using Darcy' s law, the liquid flow channel equation can be written as: [Mathematical expression]

This can be simplified to: [Mathematical expression]

where K is wick permeability, ρ_{la} and μ_{la} are liquid density and viscosity, and u_{la} is liquid flow velocity. For annular gaps, K is calculated as: [Mathematical expression]

where δ represents the annular gap. The liquid flow channel mass flow rate is: [Mathematical expression]

Based on Equations 24, 25, and 26, the complete flow channel differential equation becomes: [Mathematical expression]

which simplifies to: [Mathematical expression]

where R_{la} represents liquid flow resistance.

1.2.5 Vapor-Liquid Coupling Capillary pressure in heat pipes is defined as the pressure difference across the vapor-liquid interface caused by surface tension, inversely proportional to the interface curvature radius. The interfacial

capillary pressure difference is given by the Young-Laplace equation: [Mathematical expression]

where Δp_{vl} is the vapor-liquid pressure difference, σ is surface tension, θ is the contact angle, and r_c is capillary radius. Since capillary pressure is primarily determined by the vapor-liquid pressure difference in the evaporator (with negligible condenser contribution), the control equation is: [Mathematical expression]

where f is a correction factor [14] accounting for wick dryness effects on capillary pressure: [Mathematical expression]

where M_{leo} is the maximum working fluid mass the evaporator annular wick can accommodate.

1.2.6 Solid-Liquid Coupling The relationship between vapor temperature T_v, wall temperature T_w, and wick temperature T_{we} is: [Mathematical expression]

Neglecting evaporation and condensation thermal resistances, the vapor temperature equals the wick temperature.

1.3 System Differential Equations The solid and fluid region networks are coupled through evaporation and condensation mass flow rates. Assuming liquid and wick temperatures are equal, the complete heat pipe differential equation system is:

[Mathematical expression for the coupled system]

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2 Program Development and Validation

Based on the above model, a Python program was developed using first-order difference methods. The improved model and C. Ferrandi model were applied to calculate the HP-STMCs space reactor ultra-long lithium heat pipe, with results compared against the SNPS-HPD program [15] and HP-STMCs design values [17], as shown in Figures 45 [Figure 45: see original paper] and 56 [Figure 56: see original paper]. The HP-STMCs reactor lithium heat pipe structure and detailed parameters are presented in Figure 4 [Figure 4: see original paper] and Table 1 [17].

The results demonstrate that the improved model agrees well with HP-STMCs design values and SNPS-HPD program calculations, with maximum deviations of approximately 0.2% for vapor temperature and 1.8% for temperature difference. Compared to the C. Ferrandi model, the improved model' s accuracy increased by up to 44%, primarily because the C. Ferrandi model only considered laminar vapor flow and neglected pressure losses due to compressibility and turbulence.

3 Results and Discussion

3.1 Effect of Power on Lithium Heat Pipe Performance Under Constant Heat Sink In space applications, lithium heat pipes typically interface with a heat sink of relatively fixed thermal resistance. Therefore, this study investigates performance under a convective heat transfer coefficient of 100 $W/(m^2 \cdot K)$ and heat sink temperature of 300 K, examining the effect of heating power.

Figure 67 [Figure 67: see original paper] shows temperature difference variations under different heating powers. As heating power increases, temperature differences generally rise when thermal resistance remains stable, such as the radial temperature difference between wall and wick (with resistance distribution shown in Figure 78 [Figure 78: see original paper]). Under constant heat sink conditions, increased heating power also raises the heat pipe operating temperature. However, vapor temperature difference decreases rapidly with increasing temperature, becoming relatively stable above 1650 K.

For alkali metal heat pipes, vapor thermal resistance is generally considered small and often neglected relative to total thermal resistance [9]. However, the results show that at low operating temperatures, vapor thermal resistance can account for nearly 90% of the total, decreasing rapidly with temperature but still comprising approximately 3.9% at 1800 K. This primarily stems from the ultra-long adiabatic section, where the extended vapor flow channel yields significantly higher vapor thermal resistance than conventional heat pipes. Therefore, vapor thermal resistance should be considered in the design analysis of ultra-long lithium heat pipes operating below 1800 K.

Figure 89 [Figure 89: see original paper] presents vapor velocity and Reynolds number variations with power. Vapor velocity decreases rapidly with increasing heating power, while Reynolds number increases. When power increases from 6 kW to 8.5 kW, vapor velocity decreases to approximately 1/10 of its original value, while axial and radial Reynolds numbers increase by no more than 15%, with maximum axial Reynolds number of 1949, indicating laminar vapor flow.

Figure 910 [Figure 910: see original paper] shows liquid working fluid velocity and Reynolds number variations. Both liquid lithium velocity and Reynolds number increase with power, but maximum velocity remains below 0.038 m/s and maximum Reynolds number approximately 100. Thus, heat transfer in the lithium heat pipe wick is dominated by conduction, with convection effects negligible, consistent with traditional viewpoints [18].

Figures 1011 [Figure 1011: see original paper] and 1112 [Figure 1112: see original paper] illustrate pressure losses in the wick liquid and vapor phases versus power. As heating power increases, vapor pressure loss decreases rapidly with temperature, while liquid pressure loss decreases more gradually with less tem-

perature dependence. Therefore, when operating in low-temperature regions, vapor flow channel diameter should be increased to minimize vapor resistance, while in high-temperature operation, proper wick selection to reduce return flow resistance becomes an effective method for increasing the capillary limit.

As total pressure loss decreases, the required total pressure head from the wick decreases, reducing wick dryness. Based on total system pressure loss variations, evaporator wick thickness increases while condenser wick thickness decreases.

3.2 Effect of Power on Lithium Heat Pipe Performance Under Constant Working Temperature During lithium heat pipe design, working temperature requirements are typically specified, necessitating investigation of performance variations with power under fixed temperature conditions. This analysis uses the adiabatic section average temperature as the working temperature, with 1650 K as the study case.

Figures 1213 [Figure 1213: see original paper] and 1314 [Figure 1314: see original paper] show temperature difference and resistance variations with power. With fixed working temperature and stable properties, all temperature differences increase with input power. However, vapor flow transitions from laminar to turbulent, increasing fluid losses and accelerating temperature difference growth. Figure 1415 [Figure 1415: see original paper] shows vapor flow becomes turbulent at 8.5 kW heating power.

Regarding vapor thermal resistance, both absolute value and fraction remain relatively stable below 8.5 kW, but increase with power after turbulence onset. Therefore, ultra-long heat pipe designs should avoid vapor entering the turbulent regime. Wall and wick thermal resistances actually decrease with power due to thinning radial wick thickness.

Figures 1415 and 1516 [Figure 1516: see original paper] present vapor and liquid velocity and Reynolds number variations. Both vapor velocity and Reynolds number increase nearly linearly with power, reaching maximum axial Reynolds number of 7000 at 28 kW (near the capillary limit). Liquid velocity remains low with maximum Reynolds number of approximately 260.

During steady-state operation, increased heating power requires greater circulating working fluid mass. With constant working temperature, total vapor-liquid pressure loss increases, raising wick dryness. Since the condenser is much longer than the evaporator, evaporator thickness changes significantly more than condenser thickness (evaporator maximum reduction of 0.38 mm versus condenser increase of only 0.08 mm in Figure 1617 [Figure 1617: see original paper]). Thus, increasing heating power benefits radial thermal resistance reduction in heat pipes with condensers much longer than evaporators.

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4 Conclusions

This paper proposes an improved lumped-parameter network numerical heat pipe model considering turbulence, Mach number, and working fluid thickness variations. Analysis of HP-STMCs space reactor lithium heat pipes yields the following conclusions:

1) For constant heat sink operation, both vapor and total thermal resistance decrease with increasing power, with the fastest decrease occurring between 1400-1650 K before gradually stabilizing. Under constant working temperature, vapor and total thermal resistance change minimally before turbulent transition but increase rapidly afterward. Due to the ultra-long adiabatic section, vapor thermal resistance still accounts for 3.9% of total resistance at 1800 K.

2) Under both operating conditions, vapor and liquid Reynolds numbers increase with power, but vapor flow transitions from laminar to turbulent while liquid flow remains laminar.

3) In low-temperature operation, total pressure loss is dominated by vapor pressure loss and is highly temperature-dependent; operation at excessively low temperatures should be avoided. In high-temperature operation, liquid pressure loss dominates, making proper wick selection crucial for reducing total pressure loss.

4) When total pressure loss increases, the evaporator increases wick dryness to enhance capillary force, with excess working fluid condensing in the condenser. For heat pipes with condensers much longer than evaporators, this effectively reduces overall radial thickness, thereby decreasing radial thermal resistance.

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Note: Figure translations are in progress. See original paper for figures.