Numerical Simulations and Analyses of Temperature Control Loop Heat Pipe for Space CCD Camera Postprint

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Abstract

This paper addresses the improvement of output for heating and cooling cycles through the use of a work-fluid containing phase change material. The experimental investigation is conducted via heat exchange between the work-fluid and a heat transfer surface. The work-fluid is delivered to either a high-temperature or low-temperature heat transfer surface through a narrow passage. To enhance heat transfer, a trace amount of diethyl ether (boiling point 34.8°C), serving as a phase change material (PCM), is added to the work-fluid. The experimental parameters include the PCM additive amount, the rotational speed of the displacer piston, and the temperature of the heat transfer surface. The results demonstrate that the addition of PCM leads to an increase in engine cycle output. The effect of PCM addition is quantified using an output ratio defined based on experimental cycle output data. The conditions necessary to achieve output enhancement through PCM addition are identified.

Full Text

Preamble

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As a key unit of space CCD cameras, the temperature range and stability of CCD components directly affect critical image quality metrics. Reasonable thermal design and robust thermal control devices are therefore essential. A temperature control loop heat pipe (TCLHP) has been designed that highly satisfies the thermal control requirements of CCD components. To investigate the dynamic heat

and mass transfer behaviors of TCLHP, particularly under orbital flight conditions, a transient numerical model has been developed using well-established empirical correlations for flow models within a three-dimensional thermal modeling framework. This paper presents the temperature control principles and mathematical model details, then employs the model to study operating states, flow characteristics, and heat transfer performance through analyses of temperature, pressure, and quality variations under different operating modes and external heat flux conditions. The results demonstrate that TCLHP can effectively satisfy the thermal control requirements of CCD components while ensuring excellent temperature stability and uniformity. Comparison between flight data and simulated results shows the model accuracy is within 1°C, enabling reliable prediction and understanding of TCLHP transient performance.

Keywords: Loop heat pipe, Temperature control, Numerical simulation and analyses, Heat and mass transfer, Space CCD camera

Introduction

Space CCD cameras are widely used in military reconnaissance, resource exploration, surveying and mapping, and other fields [?]. As one of the key units, the temperature range and stability of CCD components directly or indirectly affect image quality metrics such as signal-to-noise ratio and geometric accuracy [?]. For high-performance cameras, traditional thermal control products are unsuitable for CCD components due to limitations in heat transfer capacity, mounting dimensions, and temperature stability [?], necessitating more advanced thermal design and robust control devices.

Loop heat pipe (LHP) is an advanced heat transfer device that uses capillary pressure generated in a porous structure to circulate working fluid from heat source to heat sink [?]. For precise thermal control of CCD components, a specialized TCLHP has been designed. By decoupling the evaporator from the heat source, capillary force is generated by applying heat directly to the evaporator, which drives fluid to absorb heat produced by CCD components and release it through condensers. Compared with traditional LHP applications, TCLHP offers superior start-up characteristics, excellent stability, and flexible piping layout, making it highly promising for CCD component thermal control.

Extensive ground and flight tests have demonstrated the excellent heat transfer performance of LHP [?]. However, theoretical analyses and mathematical modeling studies of LHP remain limited compared with experimental investigations. Accurate LHP models are needed to study thermal performance, particularly during orbital transients. The LHP system involves complex heat and mass transfer processes, making mathematical modeling highly challenging. Numerous steady-state models with varying complexity have been proposed that prove useful for predicting steady-state performance characteristics. Kaya et al. [?] developed a one-dimensional (1D) mathematical model of LHP that could reflect variable conductance characteristics, but model oversimplification led to large discrepancies between predictions and experimental results. Chuang [?] developed a 1D steady-state model based on energy balance equations, thermodynamic relationships, and heat transfer and pressure drop calculations, with improvements including pressure drop induced by bends, convective heat transfer in vapor grooves, and heat fluxes in the wick. Adoni et al. [?] established a model to study LHP thermal and hydraulic performance using mass and energy equations, with pressure drop across the capillary wick calculated by Darcy' s law. Bai et al. [?] developed a model to study the effect of a two-layer compound wick based on energy conservation equations and empirical correlations. Rivière et al. [?] modeled a LHP using a classic nodal network to study fluid distribution in a loop.

Only a few mathematical models have been established to study transient behaviors in response to power or sink cycles. Wrenn et al. [?] and Hoang and Ku [?] developed a transient model based on 1D time-dependent conservation equations, extending the transient model of capillary pumped loop (CPL), but large errors existed between simulation and experimental results. Pouzet et al. [?] presented a comprehensive mathematical model that successfully captured the whole dynamic behavior of a CPL due to power steps, which proved helpful for LHP transient simulation. Cullimore and Baumann [?] developed an advanced node-type LHP model written in FORTRAN code and embedded into Sinda/Fluint analyzer, which has been used by several researchers [?, ?]. Compared with test results, the predicted temperature showed small differences (less than 1°C). For orbital simulation, Ferrandi et al. [?] established a transient model to design and simulate LHP thermal control systems using Sinda/Fluint analyzer for cooling the cryomagnet avionics box on the Alpha Magnetic Spectrometer (AMS-02). Xin et al. [?] built a system-level LHP model for thermal control of cryocoolers on AMS-02 to analyze transient performance under orbital environment changes and understand LHP operation during mission time. In any case, LHP models based on Sinda/Fluint code can aid system design and predict on-orbit working performance.

In this study, a transient mathematical model of TCLHP is established using Sinda/Fluint code. The following sections first present the operating principle and mathematical model, followed by numerical results and discussions, and comparison between flight data and simulated results. Finally, conclusions are drawn.

Analysis and Modeling

Operating Principle

For heat dissipation and temperature control of multiple heat sources on the space CCD camera, the TCLHP flow schematic is shown in Fig. 1 [Figure 1: see original paper]. The operating principle is as follows: When heat is applied to the evaporator, liquid vaporizes and capillary force develops to push vapor out of the evaporator. Vapor condenses into liquid in condenser A, and capillary force continues to push liquid into the pre-heater, where the fluid changes into two-phase fluid. The saturated fluid is then forced into the cold plates, where heat produced by CCD components is absorbed by the latent heat of vaporization while temperature remains nearly constant. Two-phase fluid condenses into sub-cooled liquid again in condenser B, and capillary force pushes it into the accumulator, repeating the cycle.

Because the saturation states of fluid in cold plates and accumulator are related, the following condition must be satisfied [?]:

$$\frac{dP}{dT}_{sat}(T_{CCD} - T_{accu}) = P_{CCD} - P_{accu}$$

where T_{CCD} and T_{accu} are the saturation temperatures of fluid in cold plates and accumulator, $P_{CCD} - P_{accu}$ is the saturation pressure difference between cold plate and accumulator, and $(dP/dT)_{sat}$ is the slope of the pressure-temperature saturation line at T_{accu}. Eq. (1) states that for a given pressure difference between these two elements, a corresponding saturation temperature difference must exist. Generally, the magnitude of $(dP/dT)_{sat}$ is relatively large; for example, the value at 5°C is 3.78 kPa/°C. For TCLHP, the mass flux is small and tubes are smooth, so the pressure difference is small. If the magnitude of $P_{CCD} - P_{accu}$ is 1 kPa, the temperature difference is only 0.26°C according to Eq. (1). Hence, the working temperature of CCD components can be controlled by stabilizing the accumulator temperature.

Mathematical Model

The Evaporator and Accumulator The schematics and thermal network of evaporator and accumulator are shown in Fig. 2 [Figure 2: see original paper]. Since the accumulator is integrated with the evaporator, complex heat and mass transfer relationships exist between them.

The heat loaded on the evaporator wall divides into three parts. The first part transfers from evaporator wall to wet wick in evaporator by convective heat transfer, as shown by:

$$Q_{evap} = hA(T_{wall} - T_{wick})$$

with R an empirical constant that accounts for heat transfer in the wick. The resulting value for R is a combination of conduction, convection, boiling, and evaporation between the wet wick and evaporator wall [?]. This part also splits into evaporation power and back-conduction into the accumulator.

The evaporation power can be calculated by:

$$Q_{evap} = \dot{m}h_{fg}$$

The back-conduction power from wet wick to accumulator is given by [?]:

$$Q_{back} = k_{wick} A_{wick} \frac{T_{wick} - T_{accu}}{L_{wick}}$$

The other two parts of heat are radial heat leak from evaporator to accumulator and heat used for superheating vapor in grooves. Thus:

$$Q_{leak} = h_{groove} A_{groove} (T_{wall} - T_{vapor})$$

The thermal balance equation of accumulator satisfies the following relation:

$$Q_{accu} = Q_{back} + Q_{leak} + Q_{env} - Q_{sub}$$

where Q_{accu} is the power of temperature control loaded on the accumulator, and Q_{env} and Q_{sub} can be calculated by:

$$Q_{env} = h_{env} A_{env} (T_{accu} - T_{env})$$

$$Q_{sub} = \dot{m} c_p (T_{sat} - T_{sub})$$

Two-Phase Flow Model Two-phase flow phenomena exist in pre-heater, CCD cold plate, and condensers. The heat transfer coefficient of two-phase flow is related to heat flux density, stability, and flow pattern.

The Shah correlation is used to calculate Nu as follows [?]:

$$Nu = 0.023 Re^{0.8} Pr^{0.4} \left[\frac{(dP/dT)_{sat} \Delta T_{sat}}{h_{fg}} \right]^{0.04}$$

Transport Tubes The transport tubes of TCLHP are typically small-diameter smooth pipes with long transport lengths. By neglecting heat exchange between tubes and environment, the energy conservation equation of working fluid in lines can be written as [?]:

$$\frac{dT}{dx} = \frac{q'' P}{\dot{m} c_p}$$

The heat transfer coefficient at the inner wall can be calculated by:

$$h = \frac{Nuk}{D_h}$$

Since transport lines have large length-to-diameter ratios, developed flow is assumed. Nu can be determined by the following equations:

$$Nu = 64/Re \quad (Re < 2200)$$

$$Nu = 0.0791 Re^{0.8} \quad (2200 \leq Re \leq 10^5)$$

The qualitative temperature in Eqs. (16)-(19) adopts the saturation temperature of the two-phase section.

Capillary Wick and Vapor Grooves According to Darcy' s law [?], the pressure drop through the capillary wick is determined by:

$$\Delta P_{wick} = \frac{\mu \dot{m} L_{wick}}{\rho A_{wick} K}$$

Vapor is generated along the entire groove length, with its mass flowrate increasing approximately linearly along vapor grooves. Hence, the effective length of vapor groove is approximated as half of its total length.

The pressure drop of vapor grooves can be calculated by Eqs. (14)-(15). To determine the Reynolds number, the hydraulic diameter of each vapor groove is used:

$$D_h = \frac{2tw}{t + w}$$

where t and w are the height and width of the vapor groove, respectively.

Parameters and Conditions

Model Parameters

The working fluid is ammonia. The accumulator wall temperature is controlled at 5°C during simulations. Table 1 shows the main parameters of each component.

The External Heat Flux Variations and the Working Mode of CCD Components

Fig. 3 Figure 3: see original paper shows the simulated external heat flux variations of condenser A and B. Fig. 3(b) gives the working mode of CCD components. The orbital period is 5852 s, and the operation period is 900 s. During operation, the power dissipation of each CCD component is 9 W, and the total heat power is 36 W.

Results and Discussion

The temperature contour maps of condenser A and B for cold case (a) and (b), and for hot case (c) and (d), are shown in Fig. 4 [Figure 4: see original paper]. The cold case corresponds to the time instant of smallest external heat flux at camera shutdown, while the hot case corresponds to the time instant of largest external heat flux during operation. In this application, the vapor line is first coiled on the upper right section of condenser B, then enters condenser A where the fluid becomes sufficiently sub-cooled. The working fluid is heated to saturated condition in the pre-heater. Two-phase fluid absorbs heat produced by CCD components. The "hot" pipe is then coiled on the lower left of condenser B. After the working fluid becomes sub-cooled again, it flows into the accumulator. The bulk temperature of condensers A and B is low in the cold case but high in the hot case. Comparison of the four contour maps reveals that sections where vapor and two-phase lines are coiled exhibit higher temperatures.

Fig. 5 Figure 5: see original paper and (b) show the temporal evolution of temperature and quality of working fluid in the pre-heater. The pre-heater heats sub-cooled liquid to saturated condition (i.e., set-point) before entering CCD cold plates. Numbers 1 through 15 represent fluid at different locations from inlet to outlet of the pre-heater. Fig. 5(a) indicates that external heat flux and CCD component working mode affect fluid temperature in the pre-heater, with outlet fluid temperature near 4.5°C after heating. Fig. 5(b) shows quality changing from 0 to 0.12–0.17, indicating the working fluid has reached saturated state. The pre-heater ensures fluid entering CCD cold plates remains in two-phase condition throughout the mission.

Fig. 6 Figure 6: see original paper and (b) give the temporal evolution of temperature and quality of working fluid in CCD components. After pre-heating, fluid entering the cold plate becomes two-phase saturated state. During camera shutdown periods, saturated flows through CCD cold plates at near-constant temperature. During operation periods, liquid in the two-phase fluid evaporates, causing quality to increase. Fluid temperature remains in the range of 4.37–4.53°C, providing a stabilized working boundary for CCD components. A decreasing temperature trend exists along the four cold plates due to two-phase pressure drop, but temperature uniformity is assured by two-phase fluid characteristics. Unlike temperature, quality changes significantly, increasing from 0.12 at the inlet of the first CCD cold plate to 0.90 at the outlet of the fourth CCD cold plate. These results indicate that influences of external heat flux and working mode on temperature can be adjusted by two-phase fluid quality.

The profiles of temperature, pressure, and quality of working fluid along flow distance for cold case (a) and (b) and hot case (c) and (d) are shown in Fig. 7 [Figure 7: see original paper]. During both cases, temperature and quality initially decrease as vapor from evaporator becomes sub-cooled liquid. To save power, heat exchange occurs between sub-cooled and return two-phase lines. Liquid then flows into the pre-heater and turns into two-phase fluid. Temperature increases to saturation point and quality increases from zero to a value greater than zero. In the cold case, temperature and quality of two-phase fluid remain unchanged along the four CCD cold plates. In the hot case, temperature stays constant while quality increases quickly along heating elements. Saturated two-phase fluid condenses into sub-cooled liquid and is pushed into the accumulator. Pressure drops along the flow direction.

Fig. 8 [Figure 8: see original paper] shows comparison of average temperature of four CCD cold plates between flight data and simulation results for non-operation period (a) and operation period (b). During shutdown periods, CCD cold plates maintain approximately 4.7°C in orbit, while simulated temperature is around 4.4°C—a difference of only 0.3°C. During operation, average temperature of four CCD cold plates increases from 4.9°C to 5.5°C for flight results, and from 4.8°C to 4.9°C for simulation results. Temperature difference between the two cases ranges from 0.1°C to 0.7°C. The model slightly under-predicts loop operating temperature, but error between model and curve fit is generally less than 1°C, and in many cases less than 0.5°C, verifying the validity and reasonableness of the TCLHP model. The model can be used to predict transient LHP characteristics and understand TCLHP performance.

Conclusions

Numerical simulations have been conducted to better understand heat and mass transfer phenomena in TCLHP for space CCD cameras. The main conclusions are: (1) Fluid flow states in tubes coiled on condensers affect temperature distribution of condensers A and B, with sections where vapor and two-phase lines are coiled exhibiting higher temperatures while other sections remain cold. (2) The pre-heater between condenser and CCD cold plates ensures fluid entering CCD cold plates remains in two-phase condition throughout mission time. (3) Fluid temperature in cold plates is stable and uniform, with external flux and working mode influences on temperature adjustable through two-phase fluid quality. (4) Comparison between flight data and simulated results indicates the model slightly under-predicts loop operating temperature, with error generally less than 1°C and often less than 0.5°C.

These results demonstrate that TCLHP can provide the temperature boundary required for CCD components to ensure normal operation. The model can be effectively used for predicting and understanding TCLHP transient performance.

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