Experimental Study and Application Prospect Analysis of Truss Heat Pipes for Space Optical Cameras

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Abstract. To meet the application requirements of space optical cameras, the concept of a truss heat pipe is proposed in this paper. This truss heat pipe is composed of multiple (two or more) heat pipe shells with inner capillary wicks connected to each other by special joints with additional inner capillary wicks. It can be assembled into a variety of truss structures according to the layout requirements of the structure. Truss heat pipes can be used as the supporting structure of space optical cameras and can simultaneously efficiently transfer heat and maintain an even temperature, conserving the thermal control resources of space optical cameras. The function and performance of truss heat pipes were verified through tests of a single T-joint prototype and a multi T-joint prototype. Two truss heat pipes were then applied to the thermal management of a space optical camera. The thermal vacuum test results show that as the supporting structure of the front lens barrel heat shields, the two truss heat pipes can transfer heat from the space optical camera heat source to the front lens barrel heat shield both efficiently and evenly. This indicates that truss heat pipes can meet the requirements of the structure-thermal integrated design and thermal management of space optical cameras. As a new thermal control technology and product, truss heat pipes have the potential for broad applications in many aerospace fields in the future.

1. Introduction
A satellite in orbit is alternately heated and cooled by orbital heat flow (including solar radiation heat flow, Earth albedo heat flow, and Earth infrared heat flow) and the 4 K background of cold black space. The space environment in which satellites operate is thus very harsh[1-2]. As a main satellite payload, space optical cameras are also affected by the space environment. Meanwhile, space optical cameras are also affected by thermal disturbances due to their internal heat sources, such as electronic equipments, CCD components, cryocoolers, etc. To ensure imaging performance in orbit, the precision of temperature control for the optical lens and its supporting structure must be high. The temperature control accuracy of a typical high-resolution camera is up to ±0.3°C[3]. This makes thermal design one of the decisive factors for the performance of space optical cameras and also makes thermal design a key problem to be solved by thermal designers at home and abroad[3-14].

There are several traditional thermal control methods for the optical lens and corresponding supporting structure in space optical cameras. Components that are exposed to space are covered with multilayer insulation (MLI) to reduce the adverse impact of orbital heat flow and the 4 K background cold black space. Meanwhile, active temperature control heating circuits are arranged around the space
optical camera to achieve accurate temperature control of the optical lens. Traditionally, internal heat sources are heat-insulated from the optical structures of the camera. The heat produced by the heat sources is transferred to an external radiator through an axially grooved heat pipe and is discharged to space. Simultaneously, additional active temperature control heating circuits are arranged on the internal heat sources to ensure that the temperatures of these components do not fall below the required lower limit when they are not working.

The traditional thermal control methods for space optical cameras are resource-intensive, requiring power, weight, layout considerations, installation space for the heat pipe and radiator, and so on. As long as the thermal control resources are sufficient, traditional thermal design ideas can keep the temperature of space optical camera components within the requirements. However, with the development of space optical camera technology, space optical camera thermal control is facing more and more challenges. The most prominent issue is that the number of internal heat sources with large power consumption and long working time is large. For example, the heat sources in one particular space spectrometer include three video processors and two pulse tube cryocoolers, whose total peak consumption can be up to 362 W[15]. In this case, using traditional thermal control measures would consume a large number of thermal resources, such as needing a large radiator and a large number of heat pipes, as well as having high power consumption.

Satellites cannot meet unlimited thermal control resource demands. Thermal engineers need new design ideas or methods to effectively solve thermal control challenges with limited resources. For instance, in the thermal design of the space spectrometer mentioned above[15], the thermal engineers used the power consumed by the electronic devices for thermal control by using multiple axially grooved heat pipes to connect the electronic devices with the bottom plate and cover shell of the space spectrometer, forming a thermal control network. This allowed them to simultaneously conserve both the thermal control power consumption and the radiator for the electronic devices. However, in the thermal control network, there are overlapping thermal resistances between multiple heat pipes, which leads to lower heat transfer efficiency. This is therefore not the best solution.

In this paper, truss heat pipe (THP) technology was proposed to better solve the above problems. Tests of the prototype were carried out. Additionally, two THPs were applied to a space optical camera and tested. Finally, the application prospects of the truss heat pipe were analyzed.

2. Overview of THP
A THP is a special new type of heat pipe that is composed of multiple (two or more) heat pipe tube shells with inner capillary wicks connected to each other by special joints with additional inner capillary wicks. The basic elements of a THP include the joint, heat pipe tube shell, sealed end cover, filling end cover, and filling tube. Different types of joints can be used, including L-type, T-type, cross-type, and others. According to the layout requirements, a THP can be assembled into a variety of truss structures. As a heat transfer element, a THP also has a mechanical bearing capacity and can be used as a supporting structure. The capillary wicks and vapor passages of the THP’s different heat pipe tube shells can be connected to each other through joints. The working medium of the vapor-liquid phase change heat transfer can then circulate in the THP, just as in a single common heat pipe.

Figure 1 shows diagrams of plane single T-joint and multi T-joint truss heat pipe structures. The THP described in this paper is composed of an aluminum alloy, and the working medium for the vapor-liquid phase change heat transfer is high purity ammonia. Other compatible materials and working fluids can also be used according to requirements.

3. Functional verification of THPs
In this paper, prototypes of a single T-joint truss heat pipe and a multi T-joint truss heat pipe, shown in Figure 1, were manufactured and tested.
Filling tube
Filling end
cover
longitudinal
tube shell
joint
sealed
end cover
transverse
tube shell

Figure 1. Diagrams of plane single T-joint and multi T-joint THP structures

3.1. Tests and results of the single T-joint prototype

The diagram and photo of the single T-joint prototype and its test system are shown in Figure 2. The size of the single T-joint prototype can be seen in Figure 2. During tests, 14 thermocouples connected to temperature measuring equipment were arranged on the single T-joint prototype for real-time temperature recording. A heating circuit was affixed on the evaporation section of the single T-joint prototype. The condensation section of the prototype was installed on the cold plate of the refrigerator. The prototype was covered with insulating cotton. The test cases considered are shown in Table 1.

![Figure 2. Diagram and photo of the single T-joint prototype and its test system](image)

| Evaporation section and condensation section | Thermal load on evaporation section (W) | Temperature of refrigerator cold plate (°C) |
|---------------------------------------------|----------------------------------------|------------------------------------------|
| A is evaporation, B and C are condensation   | 60 90 120 150                           | 10                                       |
| B is evaporation, A and C are condensation  | 60 90 120 150                           | 10                                       |

Curves of the single T-joint prototype temperature over time under different test conditions are given in Figure 3. The maximum difference and standard deviation of all temperature measuring points on the single T-joint prototype are shown in Table 2.

For test cases in which A is the evaporation section and B and C are condensation sections (case A) and the thermal load on A is within 150 W, the maximum difference between all of the temperature measuring points on the single T-joint prototype varied between 1.3°C and 3.7°C with a standard deviation of about 0.4°C to 1°C, showing that the single T-joint prototype has good isothermal properties.

For cases in which B is the evaporation section and A and C are condensation sections (case B) and
the thermal load on B is within 120 W, the maximum difference between all of the temperature measuring points varied between 1.1°C and 3.8°C with a standard deviation varying from about 0.3°C to 1°C, showing that the single T-joint prototype has good isothermal properties in these conditions as well. When the thermal load on B is increased to 150 W, however, the maximum difference and standard deviation are 8.8°C and 2.2°C, respectively. Both the maximum temperature difference and standard deviation are larger than those for the same heat load in case

![Figure 3. Curves of the single T-joint prototype temperature over time under different conditions](image)

**Table 2.** Maximum difference and standard deviation of all temperature measuring points on the single T-joint prototype

| Thermal load on evaporation (W) | Maximum difference(°C) | Standard deviation(°C) |
|---------------------------------|------------------------|------------------------|
| A is evaporation, B and C are condensations | 1.3 2.0 3.5 3.7 | 0.4 0.5 0.8 1.0 |
| B is evaporation, A and C are condensation | 1.1 1.9 3.8 8.8 | 0.3 0.5 1.0 2.2 |

This is because, compared with case A, both the backflow resistance of liquid in the capillary core and the flow resistance of vapor in the vapor channel at the joint are larger in case B. Thus, theoretically, the maximum heat transport capability at capillary limit is larger in case A than in case B. With the same 150 W load, the maximum temperature difference and temperature standard deviation of all temperature measuring points are larger in case B than in case A, indicating that 150 W is close to the maximum heat transport capability at capillary limit in case B. In contrast, the gap between 150 W and the maximum heat transport capability at capillary limit in case A is relatively large.

The shell wall temperature of the evaporation section is determined by the temperature of the backflow supercooled liquid, local steam temperature, and ambient temperature. With a fixed ambient temperature, when the actual heat transfer load is close to the maximum heat transport capability at capillary limit, the reflux rate of the supercooled liquid under capillary suction is slightly smaller than its evaporation rate. This will cause the end of the evaporation section to be partially dried or have little supercooled liquid. As a result, the cooling effect of the cold liquid on the shell wall of the evaporation section is weakened. On the other hand, when the heat load on the evaporation section is large, more steam will be produced and the steam flow speed will be large. The flow resistance of vapor in the vapor channel will be large. The difference in steam pressure between the evaporation section and the condensation section is then large, which means that the temperature difference between the steam in the evaporation and condensation sections is large. The above two phenomena lead to the large temperature difference between the evaporation section and the condensation section.
3.2. Tests and results of the multi T-joint prototype

In this study, the multi T-joint prototype shown in Figure 1(b) was designed and manufactured. Its transverse and longitudinal dimensions were 2463 mm and 1670 mm, respectively. The multi T-joint prototype was tested in a low-temperature vacuum environment simulation room. The test system diagram is shown in Figure 4. A test support and test fixture were also designed. The test support was used to support the multi T-joint prototype and its test fixture. The test fixture includes a radiator and a cabin. The multi T-joint prototype was installed on the radiator thermally conductively. The radiator was then installed on the test support thermally non-conductively, and the side of the radiator facing the low-temperature heat sink was painted black to radiate heat. The cabin was installed on the radiator thermally non-conductively. The inner surface of the cabin was covered with multilayer insulation to ensure that the radiation heat exchange between the multi T-joint prototype and the cabin was as small as possible. Figure 5 shows the assembly diagram of the multi T-joint prototype, test fixture, and test support.

The heating circuit was located at the evaporation section of the multi T-joint prototype. Thermocouples were arranged on the shell to measure the temperature distribution on the prototype. The temperature control heating circuit and temperature control thermocouples were arranged on the radiator and cabin. During tests, the temperature of the environment simulation room was 100 K and the pressure was below $1.3 \times 10^{-3}$ Pa. The working temperature distribution of the multi T-joint prototype was tested under the condition that when the evaporation section was exposed to a certain heat load, the radiator and cabin were controlled at a certain temperature. Specific test cases are given.
in Table 3.

**Table 3.** Test cases of the multi T-joint prototype

| Thermal load on evaporation section (W) | Temperature of radiator and cabin (°C) | Temperature and pressure of the low-temperature vacuum environment simulation room |
|-----------------------------------------|----------------------------------------|----------------------------------------------------------------------------------|
| 90                                      | -29                                    | Temperature: 100 K, Pressure: ≤1.3 × 10^-3 Pa                                   |
| 110                                     | -28                                    |                                                                                  |
| 130                                     | -29                                    |                                                                                  |
| 150                                     | -31                                    |                                                                                  |

The temperature curves of the multi T-joint prototype over time under different test conditions are given in Figure 6. The maximum difference and standard deviation of all temperature measuring points are shown in Table 4. From the test results, we observe the following. When the thermal load on the evaporation section was between 90 W and 150 W, the maximum temperature difference between all of the temperature measuring points on the multi T-joint prototype varied between 4.0°C and 8.7°C and the standard deviation varied between about 1.1°C and 2.1°C, indicating that the multi T-joint prototype can work normally as a heat pipe. The temperature standard deviation is smaller than the maximum temperature difference, reflecting the fact that only a few data points have a slightly larger temperature difference, as can be seen in the temperature curves shown in Figure 6.

**Figure 6.** Temperature curves of the multi T-joint prototype over time under different test conditions

**Table 4.** Maximum difference and standard deviation of all temperature measuring points

| Thermal load on evaporation section (W) | Maximum difference (°C) | Standard deviation (°C) |
|-----------------------------------------|--------------------------|-------------------------|
| 90                                      | 4.0                      | 1.1                     |
| 110                                     | 6.1                      | 1.3                     |
| 130                                     | 7.0                      | 1.7                     |
| 150                                     | 8.7                      | 2.1                     |
4. Application and verification of the THP on a space optical camera

4.1. Application mode and THP structure

A diagram of the structure of the space optical camera is given in Figure 7. The main structure includes the baffle, front lens barrel, front optical lens assembly (main mirror assembly and secondary mirror assembly), rear optical system, focal plane assemblies, and two cryocoolers for the focal plane assemblies. According to traditional thermal design, the front lens barrel temperature should be controlled with temperature control heating circuits to provide a good temperature environment for the front optical lens assembly. Heat consumed by the two cryocoolers should be transferred to a space radiator to dump the heat to cold black space. Alternatively, to use the thermal control resources more efficiently, heat consumed by the two cryocoolers can be transferred to the front lens barrel. This can not only reduce the power consumption of the temperature control in the front lens barrel but can also eliminate the radiator of the cryocoolers. The heat transfer path from the cryocoolers to the front lens barrel using an ordinary axially grooved heat pipe is shown in Figure 8. In this heat transfer path, there are multiple overlapping thermal resistances. Therefore, the total thermal resistance of the heat transfer path is large. To reduce the thermal resistance of the heat transfer path as much as possible, the bottom circumferential heat pipes and the optical axis axial heat pipes were combined into one THP to greatly improve the efficiency with which the heat consumed by the cryocoolers was utilized.

![Diagram of the structure of the space optical camera](image)

**Figure 7.** Diagram of the structure of the space optical camera

The Pictures of the THPs are given in Figure 9. Two sets of semicircular truss heat pipes, which can be seen in Figure 9(a) and 9(b), respectively, make up a cylindrical frame. The front lens heat shields are installed on the cylindrical frame of the THP, as shown in Figure 9(c). The inner diameter and height of the cylindrical frame of the truss heat pipe are 1602 mm and 1685 mm, respectively. The THP tube shell is an axially grooved heat pipe shell made of an aluminum alloy. The T-joint material is an aluminum alloy. The working medium is high purity ammonia.

Heat consumed by the cryocooler is first transferred to the circumferential tube shell evaporation section. The heat at the evaporation section is then transferred to other parts of the THP through the vapor-liquid phase transformation of the working medium in the THP. Finally, the THP transmits the heat to the front lens barrel heat shields through heat conduction. The evaporation section of THP A receives the heat consumed by one cryocooler. The evaporation section of truss heat pipe B receives the heat consumed by the other cryocooler.

![Heat transfer path from the cryocoolers to the front lens barrel using ordinary heat pipe](image)

**Figure 8.** Heat transfer path from the cryocoolers to the front lens barrel using ordinary heat pipe

4.2. Vacuum thermal test results

During the vacuum thermal test, the two truss heat pipes were installed on the space optical camera as
the supporting structure of the front lens barrel heat shields. The front lens barrel heat shields were installed on the two truss heat pipes thermally conductively. The space optical camera was vertically placed in the simulation room with the baffle vertically up. In this arrangement, the circumferential pipe shells of the truss heat pipe were at the bottom and the axial pipe shells of the truss heat pipe were on top.

![THP A and THP B](image)

**Figure 9.** Pictures of two THPs and their assembly diagram with front lens barrel heat shield

During the tests, thermocouples connected to temperature measuring equipment were arranged on the truss heat pipe for real-time temperature recording. The temperature of the simulation room was 100 K and the pressure was below 1.3×10⁻³ Pa. The two THPs and the front lens barrel heat shields faced the 100 K low-temperature heat sink directly through the light inlet. The working heat consumption of the cryocooler thermally connected with truss heat pipe A was 140 W, while the working heat consumption of the cryocooler thermally connected with truss heat pipe B was 100 W.

The temperature curves of the two THPs over time are given in Figure 10. The maximum difference and standard deviation of all of the temperature measuring points on the two THPs are shown in Table 5. From the test results, we can see the following. The maximum difference among all of the temperature measuring points on THP A is 8.2°C with a standard deviation of 2.3°C, while the maximum difference of all temperature measuring points on THP B is 6.9°C with a standard deviation of 2.2°C. These results indicate that the two THPs can transfer the heat consumed by the two cryocoolers to the front lens barrel heat shield efficiently and evenly. The power consumption of the front lens barrel temperature control is saved. The radiator of the cryocoolers is eliminated. The uniform temperature applied to the front lens barrel heat shield by the THP also provides a good temperature environment for high precision temperature control of the front optical lens.

![Temperature curves of THP A and THP B](image)

**Figure 10.** Temperature curves of the two THPs over time

**Table 5.** Maximum difference and standard deviation of all of the temperature measuring points on the two THPs

| Truss heat pipe | Thermal load | Maximum difference(°C) | Standard deviation(°C) |
|-----------------|--------------|-------------------------|------------------------|
| Truss heat pipe a | 140          | 8.2                     | 2.3                    |
| Truss heat pipe b | 100          | 6.9                     | 2.2                    |

5. Application Prospect Analysis of THP
5.1. Structure-thermal integrated design and thermal management of space optical cameras

The THP proposed in this paper can be used as the supporting structure of a space optical camera and can also efficiently transfer heat and contribute to even temperature distributions. In addition, when bending an ordinary heat pipe, the bending radius generally must remain at least 5 times larger than the outer diameter in order to avoid damaging the internal axially grooved channel. In contrast, the joint design of the THP is not limited by this minimum bending radius, which can significantly conserve space in the heat pipe layout and reduce the difficulty of designing the heat pipe layout. Therefore, the THP has a wide application prospect in the structure-thermal integrated design and thermal management of space optical cameras.

5.2. Thermal control of on-board equipment with high power and high temperature stability requirements

With the development of large-capacity communication satellites, high-resolution remote sensing satellites, and other civil spacecraft in China, more and more devices with high power and high stability requirements are used. For example, there are many TWTs with high power dispersed across the decks of communication satellites. In another example, there are many SAR antenna T/R modules on remote sensing satellites. The arrangement of these T/R modules is compact and the heat consumption of the T/R module array is large, while the temperature uniformity requirements of the T/R modules are high. If the heat cannot be drawn away, the performance of the feeder unit will be affected. At present, the traditional method is to use the outer surface of the satellite deck as a radiator surface to disperse the working heat consumption of TWTs and TR modules into cold black space. However, because the satellite deck is a sandwich structure with a honeycomb core between the composite panels of both sides, the thermal conductivity in both the plane direction and the thickness direction of the satellite deck is limited. In order to reduce the thermal resistance between heat sources on the inner panel and the outer panel as much as possible, and to avoid the uneven temperature of the deck caused by the uneven heat consumption of heat sources and the uneven orbit heat flow incident on the outer surface of the deck, an orthogonal lap heat pipe system is arranged on the satellite deck to strengthen the heat conduction in the thickness direction of the cabin and to promote uniform temperature in the plane direction. At present, there are three types of orthogonal arrangements of heat pipes, i.e. the fully embedded type, the hybrid type, and the fully externally protruding type, as shown in Figure 11. These three arrangements all have the problem of overlapping thermal resistance. There are also other problems in the layout of the 2-layer overlap fully embedded type, such as the thick deck and the complex fabrication process of the deck. The externally protruding type, meanwhile, affects the layout of the equipment on the deck.

![Figure 11. Three types of orthogonal arrangements of heat pipes on board](image)

In view of the shortcomings of the traditional thermal control measures mentioned above, if the single-layer plane orthogonal integral truss heat pipe is embedded in the satellite deck, the overall heat transfer efficiency can be increased and the temperature equalization effect can be improved greatly. Meanwhile, both the difficulty of the manufacturing process and the thickness of the satellite deck can be greatly reduced. Therefore, truss heat pipes have a wide application prospect in the thermal control of equipment with high power and temperature stability requirements.

6. Conclusion

Based on the application requirements of space optical cameras, the concept of a THP composed of...
multiple heat pipe tube shells connected with each other by special joints was proposed in this paper. According to the layout requirements, the THP can be assembled into a variety of truss structures. THPs can be used as the supporting structure of a space optical camera and can also efficiently transfer heat and help maintain an even temperature, efficiently conserving the space optical camera’s thermal control resources. Prototypes of a single T-joint THP and a multi T-joint THP were manufactured and tested. The test results fully verified the function and performance of THPs as heat pipes. Two truss heat pipes were then applied to the thermal management of a space optical camera. The thermal vacuum test results show that as the supporting structure of the front lens barrel heat shields, the two THPs can transfer heat from the space optical camera heat source to the front lens barrel heat shield both efficiently and evenly. The function and performance of the THP were also verified.

THPs can be used as the supporting structure of space optical cameras while efficiently transferring heat and maintaining an even temperature. In addition, the joint design of truss heat pipes is not limited by a minimum bending radius, which can significantly reduce the space needed by the heat pipe layout and reduce the difficulty of the layout’s design. The plane orthogonal truss heat pipe can replace the traditional orthogonal lap heat pipe system and solve problems associated with current orthogonal lap heat pipe systems, such as large lap thermal resistance, thick satellite decks, and the complex fabrication process of the satellite deck. Therefore, as a new thermal control technology and product, THPs require further research and optimization but have the potential for broad application in many aerospace fields in the future.

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