The Operational Characteristics of a Novel Miniature Cryogenic Loop Heat Pipe

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ABSTRACT

A novel miniature cryogenic loop heat pipe (MCLHP) has been developed for future aerospace applications at CAST (China Academy of Space Technology). This article presents the general operational characteristics of a miniature cryogenic loop heat pipe related to the operational principle of the secondary evaporator and its effect on the start-up process and the heat transfer capability of the total system. The MCLHP studied in this work has demonstrated to be able to start up successfully from supercritical condition placed horizontally and can operate in liquid-nitrogen temperature range using nitrogen as the working fluid. The experimental results show that the temperature of the MCLHP can be decreased to 80K from room temperature within 20 minutes without gravity assistance, and the primary evaporator can transfer 6W over a long distance of 500 mm with a power of 1W applied on the secondary evaporator.

Keywords: Miniature cryogenic loop heat pipe, Liquid-nitrogen temperature, operational characteristics

1. INTRODUCTION

Loop heat pipes (LHPs) are effective two-phase heat transfer devices that have gained increasing acceptance for space applications and are emerging as the baseline design of thermal control systems since they are invented in 1980’s [1-6]. Several space missions performed by America have tested loop heat pipe successfully, which demonstrate the feasibility and reliability of LHPs in space application. However, with the development of cryocooler-based applications such as space exploration, it is necessary to develop the cryogenic LHP which can operate at liquid-nitrogen temperature range and even lower cryogenic temperature ranges.

Due to the limit of cryogenic condition and space application background, the investigation of cryogenic LHP has two difficulties: 1) Start up from supercritical condition without gravity assistance; 2) In order to be applicable to the requirement of small power (in magnitude of 1W), the miniaturization of cryogenic LHP is becoming inevitable. To solve the first problem, a secondary evaporator can be located near the heat sink to get sufficient driving force in the beginning [7-9]. The references in recent years are mainly focused on how to implement the startup and normal operation, the size of the prototypes is too large to be satisfactory [10-13], thus there is a long distance far from the realization of engineering application. The miniature cryogenic LHP studied in this article can solve both of the problems mentioned above at the same time. The configuration of the condenser makes it possible to couple directly with the cooling finger of the cryocooler. Using nitrogen as the working fluid, the MCLHP can be cooled down and start up from supercritical condition by heating the secondary evaporator without gravity assistance.

Secondary evaporator, a key component of the cryogenic loop heat pipe, can be used to accelerate the cooling process of the whole loop from room temperature to operating temperature. The presence of the secondary evaporator divides the loop into two parts: the primary loop and the secondary loop. In general, the secondary loop is needed during the start-up process by heating the secondary evaporator; after that in the normal operation, the primary loop can transfer heat by heating the primary evaporator only without the secondary loop assistance. But, although the secondary loop doesn’t work, its presence will affect obviously the heat flux distribution and flow resistance of the whole system, resulting in some peculiar operational characteristics. Thus, this article has tried to analyze the operational principle of the secondary evaporator, and also investigate its function during the cooling process, its operational characteristics all alone and its effect on the heat transfer capability of the primary evaporator based on the experimental data.

2. CONFIGURATION AND PROTOTYPE OF A CRYOGENIC LHP

2.1. The Typical Configuration of a Cryogenic LHP

The typical configuration of a cryogenic LHP is
shown in Fig. 1. Obviously, the primary loop is actually a room-temperature LHP which consists of a primary evaporator and its compensation chamber (CC), a primary vapor line, a primary condenser and a primary liquid line. In order to be applicable to the cryogenic condition, the cryogenic LHP needs to include two other components: a hot reservoir connecting to the primary vapor line and a secondary evaporator (thermally and hydraulically linked with a secondary CC). It should be noted that the secondary CC is necessary for a cryogenic LHP to ensure the function of the secondary evaporator during the cooling process whereas the primary CC can be canceled in some cases to get a more compact configuration.

Figure 1. The typical configuration of a cryogenic LHP

The presence of the hot reservoir is to concern the safety of the cryogenic LHP under room temperature. The function of the secondary evaporator is to shorten the cooling process of the primary evaporator without gravity assisted: sufficient capillary force can be produced by heating the secondary evaporator due to the connection of the secondary CC to the heat sink directly, and the resulting vapor will flow toward the primary condenser through the secondary vapor line, thus the condensate in the primary condenser will be pushed into the liquid core of the primary evaporator. Moreover, to keep the continuous operation of the secondary evaporator, there must be the same quantity of liquid returning back into the secondary CC as that of the vapor. Therefore, to form another loop called as the secondary loop (relatively to the primary loop), the secondary CC is connecting with the primary evaporator liquid core through the secondary liquid line, the part of which near the secondary CC is thermally joined with the heat sink directly, as shown in Fig. 1. This part of line is called as secondary condenser which can make sure of the liquid with sufficient subcooling going back into the secondary CC during the cooling process.

2.2. The Miniature Cryogenic LHP Prototype

The MCLHP (miniature cryogenic loop heat pipe) used in this work has the configuration as Fig.2. The MCLHP consists of two evaporators (each with a compensation chamber), two condensers, transportation lines and a pressure reduction reservoir (about 500ml). The condensers are two stainless steel thin-walled pipes welded in the spiral groove on the outside surface of a copper cylinder. The top surface of the cylinder is contact with the secondary compensation chamber. Such a kind of structure can implement the miniaturization of the condenser, so that the condenser can be mounted with the cryocooler cooling finger directly through its bottom flat surface or its inside cylindrical surface. The structure of the primary evaporator is similar to that of the secondary evaporator. The outer diameter of both evaporators is only 13mm. The heat transfer distance between the primary evaporator and the heat sink is about 500mm.

Figure 2. The miniature cryogenic LHP prototype

In the experiment, liquid nitrogen is directly used to cool the copper cylinder so that it can serve as a heat sink of the spiral condenser. The whole loop (including the hot reservoir) is placed inside a vacuum vessel horizontally. To weaken the negative effect of the parasitic heat gains from ambient, the MCLHP is wrapped in multi-layer insulation consisting of more than 10 layers. The temperatures along the MCLHP are measured with T type thermocouples and the measuring locations have been given in Fig. 2: T1 is the temperature of heat sink, T2 is the temperature of primary evaporator, T3 is the temperature of secondary evaporator, T4 is the temperature of primary CC, T5 and T6 are temperatures of the inlet and middle of primary liquid line, T7 and T8 are temperatures of the inlet and middle of secondary liquid line.
3. THE OPERATING PRINCIPLE OF THE SECONDARY EVAPORATOR

The secondary evaporator is one of the key components of cryogenic LHP, the function of which is to implement and ensure the cool down process of the whole loop from room temperature to operating temperature. At the starting up, the side of primary evaporator is at room temperature, but the capillary structure in the secondary evaporator is wetted by the liquid due to the direct connection between the secondary CC and the heat sink. Therefore, sufficient capillary force will be produced by heating the secondary evaporator and force the liquid in the primary condenser to flow toward the primary evaporator, so that the temperature of the entire loop can be decreasing.

Figure 3. The mass flow direction during the cooling process

The second As shown in Fig. 3, the operating principle of the secondary evaporator is as follows: 1) When a power (W2) is applied on the secondary evaporator, a certain pressure difference (i.e. the capillary pressure) will be presented between point A and point E; 2) In order to make the secondary evaporator operate continuously, there should be the same mass flow rate of working fluid returning back into the secondary CC through the secondary liquid line while a mass flow rate (m2) of vapor is produced by W2. Therefore, after the successful start-up of the secondary evaporator, point A is the place with the highest pressure in the whole loop, and the pressure of point E (inside the secondary CC) is the lowest. 3) The function of the secondary evaporator is to compel the primary evaporator to be cooled down so that the primary CC will be gradually filled with liquid, which obviously is accumulated by the working fluid with a mass flow rate of m1 coming out gradually from the hot reservoir.

When the hot reservoir is connected to the primary vapor line, as shown in Fig. 3, two parallel paths to the primary CC exist for the working fluid with mass flow rate of m1: 1) A part of working fluid with mass flow rate of m1 will go from point A into the primary CC through the primary condenser and primary liquid line; 2) Whereas another part of the working fluid with mass flow rate of m2 will reach the primary CC by penetrating the primary wick directly through the primary vapor line.

However, the flow resistance of the second path is several orders of magnitudes greater than the first path due to the presence of the primary wick. Hence, the mass flow rate of m1 for the first path is the majority of m1 while m2 can be neglected. Moreover, from viewpoint of the function, the small part of working fluid with mass flow rate of m2 can hardly contribute to the temperature decrease of the primary evaporator because its entering the primary CC without going through the condenser. Based on the above discussion, it is important how to design the resistance of the primary wick: the resistance of the primary wick should be large enough to force the most part of the working fluid to go through the first path.

The heat dissipation of the working fluid in the first path is as follows: the working fluid with mass flow rate of (m11+m2) dissipates a part of heat (Q1) in the primary condenser to get certain subcooling before reaching the primary CC through the primary liquid line. Then the primary evaporator is cooled down by cooling capacity of W1, and the corresponding working fluid with mass flow rate of m11 is left within the primary evaporator to compensate the necessary amount of working fluid resulting from temperature falling. The residual working fluid with mass flow rate of m2 is kept going through the secondary liquid line and reaches the secondary condenser, where it dissipates another part of heat (Q2) and gains a suitable subcooling before returning back into the secondary CC. Hence, W1+W2=Q1+Q2 (neglecting the parasitic heat to the loop).

Therefore, the larger the total condensing capacity of the primary and secondary condenser (Q1+Q2) is, the shorter the cooling process of the primary evaporator when W2 is given. Moreover, in general the cooling power of the cryocooler is limited for engineering applications, so that the designer should choose W2 based on the cooling capacity of the cryocooler and minimize the parasitic heat to weaken its bad effect on the cool down process.
4. THE EXPERIMENTAL RESULTS AND ANALYSIS

4.1. The Cooling Effect of the Secondary Evaporator

The miniature CLHP, which is placed horizontally in this research, can be cooled down successfully from room temperature to operating temperature. The cooling process of the prototype is shown in Fig. 4 under working fluid inventory of 6.3g corresponding to an initial charge pressure of 1.0MPa. As shown in Fig. 4, the heat sink temperature (T1) was decreasing from 300K to 77K in 5 minutes after being cooled by liquid nitrogen. With the decreasing of the heat sink temperature, the temperatures of the condenser and secondary CC were also decreasing gradually, as well as the pressure in the loop.

At about 17 min, the temperature of the secondary evaporator started to be below the critical temperature which indicated the capillary wick in it was gradually wetted by the liquid, and this part of liquid would be heated and become vapor due to the heat capacity of the secondary evaporator. As shown in Fig. 4, during the period between 17 min to 19 min, the temperature of primary liquid line (T6) decreased from 300K to 77K and the temperatures at the side of primary evaporator began to decrease rapidly. This is because that the resulting vapor flowed into the primary condenser and simultaneously pushed the condensate in it towards the primary evaporator through the primary liquid line. Therefore, after the secondary wick was wetted completely, the temperature of the liquid line increased quickly, and the primary evaporator temperature also stopped decreasing, which is also the time sign for starting-up the loop by heating the secondary evaporator. At about 26 min, a power of 4W was applied on the secondary evaporator, and then the temperatures at the side of the primary evaporator decreased from 250K to below 80K within 20 minutes.

Figure 5. The cooling process under inventory of 10.1g (1.6MPa)

As shown in Fig. 5, after the working fluid inventory was increased to 10.1g, the cooling process became much more difficult than the case of 6.3g. When the power on the secondary evaporator was less than 4W, the temperatures at the side of the primary evaporator increased very slowly instead of decreasing. When the same power of 4W applied on the secondary evaporator, the temperature decreasing rate was obviously slower than that of the above case: After about 30 minutes, the primary evaporator temperature only reached 150K. At about 98 min, when the secondary evaporator power was increased to 8W, the temperature of the primary evaporator decreased very quickly. The possible reason for this phenomenon is as follows: for a given power, the operating temperature of the secondary evaporator would be affected by the inventory. The larger the inventory, the higher the operating temperature and the smaller the capillary driving force produced. As shown in Fig. 3, when the driving force is small, the mass flow rate of m11 will be also small, thus the cooling process becomes longer due to the less of cold capacity. Based on the above discussion, larger driving force and cold capacity would be induced with the power increasing on the secondary evaporator, so that the temperature of the primary evaporator could be decreased very quickly.

4.2. The Individually Operating Characteristics of the Secondary Evaporator

The operating performance of the secondary evaporator is the key factor for the cool down process of the primary evaporator and the successful start-up of the whole loop. The individually operating characteristics of the secondary evaporator with different inventory are shown in Fig. 6 and Fig. 7 respectively. As there is no heat load applied on the primary evaporator, its temperature (T2) was stable for different inventory and power. However, the secondary evaporator temperature was up to 120K in the case of small inventory of 4.4g and large power of 4W. Moreover, the temperatures of the liquid lines
were oscillating and the frequency and swing were different with different inventory and power. The following will try to analyze the reason for these two phenomena.

**Figure 6.** The operating characteristic of the secondary evaporator with inventory of 4.4g (0.7MPa)

(1) The heat leak is large with insufficient working fluid inventory.

It is well known that the temperature difference will be small if heat is transferred in a manner of phase change (i.e. vapor produced by heating liquid) and taken away by the vapor, this part of heat can be called as “effective heat”. On the other hand, when the heat is transferred to the liquid core inside the evaporator by conducting axially through the evaporator shell or radially through the capillary wick, the resulting temperature difference will be large. This part of heat can be called as “heat leak”, which leaks into the liquid core. When the working fluid inventory was small enough, the large superheating of the vapor would result in high wall temperature of the secondary evaporator due to the increasing heat leak, which can be proved by comparing Fig. 6 with Fig. 7 to some extent.

(2) Temperature oscillation due to the insufficient subcooling.

It can be seen from Fig. 6, the transportation line temperatures were oscillating very intensely with small inventory. However, the temperature oscillation on the line was only presented for small heat load of 1W with large inventory, as shown in Fig. 7. The possible reason for the above phenomenon might be as follows: when the working fluid inventory is small, the superheating of the vapor would be large due to the large heat leak, resulting in insufficient subcooling of the condensate left from the primary condenser into the primary liquid line. This is the reason why the temperature oscillation happened on the line. For the same reason, as the subcooling of the working fluid returning from the primary CC decreased further, the temperature of the secondary liquid line was oscillating more intensely. Moreover, the temperature oscillating for small power of 1W is mainly because the liquid flow would be affected more easily by the parasitic heat with small mass flow rate, resulting in the saturation of the working fluid (with bubbles) in the transportation lines.

Based on the above discussion, increasing the working fluid inventory is a method to minish the heat leak (i.e. to eliminate the temperature oscillation), but there would be other problems induced due to the higher operating temperature. Therefore, to solve this problem basically and to improve the working ability of the miniature CLHP for small heat loads, the configuration design of the evaporator and the material choice should be focused.

### 4.3. The Effect of the Secondary Evaporator Power on the Heat Transfer Capability

Fig. 8 and Fig. 9 show the heat transfer capability of the prototype with an inventory of 6.3g. When the power on the secondary evaporator is 1W and 4W, the maximal power 6W and 10W on the primary evaporator is obtained respectively. This experimental phenomenon shows that the power on the secondary evaporator can have an effect on the heat transfer capability of the primary evaporator.

**Figure 8.** The heat transfer capability with 1W on the secondary evaporator (6.3g)
Figure 9. The heat transfer capability with 4W on the secondary evaporator (6.3g)

It is well known that the capillary limitation is the heat transfer limitation restricting the heat transfer capability of the loop heat pipe, that is to say, the capillary pressure produced in the capillary structure can overcome the flow resistance of the working fluid in the whole loop. However, in fact the design value of the heat transfer capability calculating according to the above method is much larger than the experimental data. Taking this prototype for example, the design value over 30W on the heat transfer capability is 20W larger than measuring value in the test. The following will try to explain the possible reason for this difference: the power on the primary evaporator partially leaks into the liquid core and this part of heat will increase as power rising. When the working fluid rate produced by the secondary evaporator in the secondary loop can take this part of heat leak away, the primary evaporator can operate stably, whereas the capillary limitation will occur early due to the increasing of the operating temperature and decreasing of the capillary pressure. Thus the calculation of the heat transfer capability should consider synthetically the heat leak for a specific configuration of the primary evaporator instead of the heat transfer capability which is estimated just according to the maximum capillary pressure under ideal condition (without heat leak). Therefore, the more exact the estimation of the heat leak, the smaller the calculating error of the heat transfer capability. But how to determine the heat leak exactly is another challengeable topic, which is not discussed in detail in this article.

For the same reason, the heat leak will have negative effect on the primary evaporator for a low level of power. As shown in Fig. 10, the primary evaporator can still kept in a stable operation for its constant power 2W while the secondary evaporator power was changed from 2W to 0.5W. However, after removing the power on the secondary evaporator, the primary evaporator temperature (T2) started increasing quickly as well as the temperature of the primary liquid line inlet (T5), which indicated that working fluid circulation was broken off and there is no condensate entering into the primary liquid line any more.

Figure 10. The operating characteristic of the primary evaporator with low level of power (6.3g)

5. CONCLUSIONS

The miniature cryogenic loop heat pipe studied in this article has implemented the miniaturization of the condenser and evaporator. The experimental data on both cooling process and heat transfer characteristics show that the whole loop can be cooled down successfully from supercritical condition and can transfer heat over relatively long distance with small temperature difference using nitrogen as the working fluid. Based on the above mentioned experiments, the results of the testing can be summarized as follows:

1) During the cooling process, sufficient driving force can be provided only by heating the secondary evaporator to accelerate the temperature decreasing rate of the primary evaporator without gravity assistance. The temperature of the primary evaporator can decreased from room temperature to 80K within 20 minutes after a power of 4W applied on the secondary evaporator with a working fluid inventory of 6.3g.

2) For a given power, the cooling time needed for the whole loop is longer with larger working fluid inventory. The two ways to shorten the cooling process are rational design of the flow resistance and the increasing of the hot reservoir volume.

3) Keeping a power of 1W on the secondary evaporator, the primary evaporator can transfer 6W over a distance of 500mm. The heat transfer capability of the primary evaporator can be increased by increasing the secondary evaporator power.

4) The presence of the secondary evaporator and the secondary loop can have significant effect on the heat transfer characteristics of the primary
loop. A key factor for the miniature CLHP to operate stably with small heat load is to design rationally the flow resistance and heat transfer resistance of each component in the loop.

Therefore, the miniature cryogenic loop heat pipe with such a configuration is a potential device for future aerospace application. With further improving, it will have more excellent heat transfer capability.

REFERENCES


