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# A compact remote heat transfer device for space cryocoolers

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**Abstract**

In this paper a compact remote heat transfer device (CRHD) for cryocoolers is proposed. This device is especially attractive in cases where cryocoolers are not easy to set near the heat source, generally the infrared sensor. The CRHD is designed on basis of the concept of loop heat pipes, while the primary evaporator is located near the cryocooler cold head and a simple tube-in-tube secondary evaporator is remotely located and thermally connected with the heat source for cooling. With such a device a cooling power of 1 W is achieved across a heat transfer distance of about 2 m. The major problem of this device is the low heat transfer efficiency (1 W of net cooling power at the cost of about 7 W of cooling power from the cryocooler), and in the future a secondary wicked evaporator will be used instead of the tube-in-tube evaporator in order to improve the efficiency.



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*Keywords:* remote heat transfer, heat pipe

## 1. Introduction

Space infrared detecting systems became more and more popular in recent years in many countries. Infrared detectors are key elements in an infrared detecting system. Infrared detectors generally work under cryogenic temperatures, e.g., below 80 K and are cooled by cryocoolers. As a most widely used configuration, an infrared detector is generally directly connected to the cold tip of a cryocooler. Since a cryocooler is relatively large in volume, such a configuration is not available in some cases where there is not enough space behind the detector. Therefore the cryocooler has to be located apart and a remote cryogenic heat transfer device is required as the thermal link between the cryocooler cold tip and the detector.

Cryogenic loop heat pipes (CLHPs) are promising cryogenic heat transfer devices. A normal CLHP is a two-loop

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structure with a second loop for the starting-up. Details of CLHP technology are referred to in Ref. [1-4]. As for space infrared detector cooling, a normal CLHP is not applicable, because the structure is complicated and the evaporator is relatively large. In this paper a CLHP-based compact remote heat transfer device (CRHD) is proposed which is very compact with a small cooling end (evaporator not as the cooling end but as a capillary pump), at the cost of cooling efficiency. The structure and the working principles of the CRHD are introduced, and some experimental results on a prototype are shown and analyzed.

## 2. Structure and working principle of the CRHD

Fig. 1 shows the schematic diagram of a CRHD, which consists of an evaporator, a primary condenser, a secondary condenser and a closed loop hydraulically connecting all above. A certain amount of fluid is filled in the loop which exists in a liquid-gas phase equilibrium status. At the beginning the fluid is condensed in the secondary condenser and then flows into the evaporator. Then a heat load $Q_{eva}$ is applied onto the evaporator and the liquid inside the evaporator evaporates. The generated gas accumulates and flows to the primary condenser. The gas condenses in the primary condenser, giving out a heat $Q_{cond1}$, and the condensed liquid flows to the secondary condenser. On the way the liquid collects the parasitic heat load $Q_{para}$ and a heat load $Q_{load}$ specially applied on the cooling end which is a copper tube sleeved on the pipe, simulating the infrared detector heat generation and gradually becomes gas. The gas flows into the secondary condenser and condenses, giving out the collected heat $Q_{cond2}$. The condensed liquid flows back into the evaporator and then a cycle completes. The driving force of the fluid flow is the capillary force on the meniscus of the outer surface of the wick, therefore the CRHD is a completely passive heat transfer device without moving part.

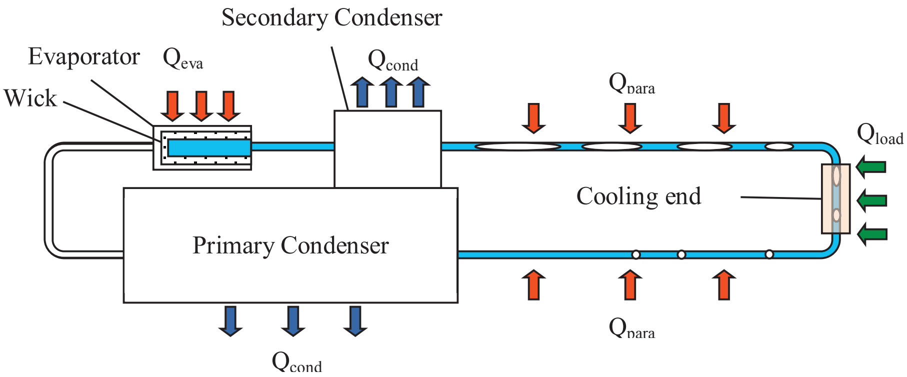



Fig. 1. Schematic diagram of a CRHD

From the descriptions above it is easy to determine the energy balance equation of the CRHD:

$$Q_{eva} + Q_{load} + Q_{para} = Q_{cond1} + Q_{cond2} \tag{1}$$

It is noted that the parasitic heat load on parts of the CRHD other than the pipe connecting the primary condenser and the secondary condenser (referred to as the heat transfer pipe hereinafter) is neglected.

In Equation (1) it is seen that only $Q_{load}$ is useful in the CRHD, $Q_{eva}$ is used for pumping the condensed fluid from the heat sink to the remote heat source, and $Q_{para}$ is the inevitable consumption of the cooling power. Therefore the heat transfer efficiency of the CRHD can be defined as:

$$\eta = \frac{Q_{load}}{Q_{eva} + Q_{load} + Q_{para}} = \frac{Q_{load}}{Q_{cond1} + Q_{cond2}} \tag{2}$$

In order to improve the heat transfer efficiency, $Q_{eva}$ and $Q_{para}$ shall be as small as possible. As $Q_{eva}$ is the driving force of remote cooling power pumping, it is basically a constant when other parameters are fixed, including the CRHD dimension and the cooling power requirement. Therefore the key parameter for improving the heat transfer efficiency is $Q_{para}$.

## 3. Design and fabricating of the CRHD

In designing the CRHD, liquid nitrogen is used as the heat sink. The primary and secondary condenser are serpent pipes directly integrated to a cold plate cooled by liquid nitrogen. The heat transfer pipe is 4 m long, and therefore the heat transfer distance is 2 m. In order to minimize the heat transfer pipe and the cooling end, which is essential for remote cooling power transfer within a narrow space, a pipe-in-pipe structure is adopted for the heat transfer pipe. One end of the inner pipe is hydraulically connected to the primary condenser and the other end open to the outer pipe. One end of the co-axially outer pipe is hydraulically connected to the secondary condenser and the other end sealed with a copper cover as the cooling end. Actually the outer pipe has an annular cross section. Considering possible ground applications of the CRHD, such as HTS or MRI, the vacuum shell surrounding the CRHD is also designed very compact, with a 2 m long pipe only a little larger in diameter than the CRHD outer pipe (see also Fig. 2). The advantages for the pipe-in-pipe structure are the compactness of the whole cooling system and blocking of parasitic load from the inner pipe.

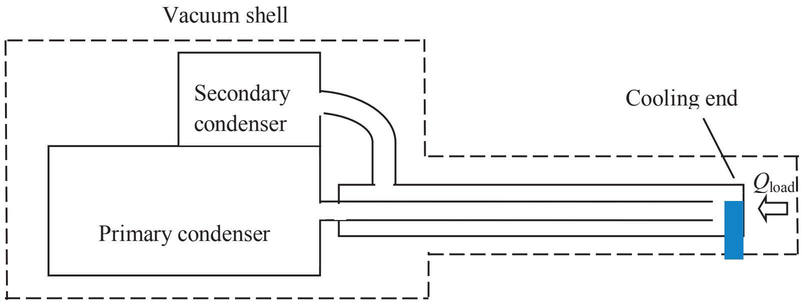



Fig. 2 Pipe-in-pipe structure of the heat transfer pipe

A CRHD prototype is fabricated utilizing the above design. Fig. 3 is a photo of the CRHD prototype. The prototype is welded with the vacuum pipe and vacuum shell as a whole, so it is impossible to show only the CRHD itself. From this photo we can see the liquid nitrogen dewar as the cooling source, the vacuum shell containing the two condensers, the 2 m long vacuum pipe surrounding the 2 m long heat transfer pipe and the relatively big end of the vacuum pipe containing the CRHD cooling end and a heater for applying $Q_{load}$. The condensers and the cooling end are made of copper, the evaporator shell is made of stainless steel, the wick is made of stainless steel powder with a mean pore size of about 15 μm, and the heat transfer pipes are made of stainless steel. The main dimensions of the CRHD are listed in Table 1.

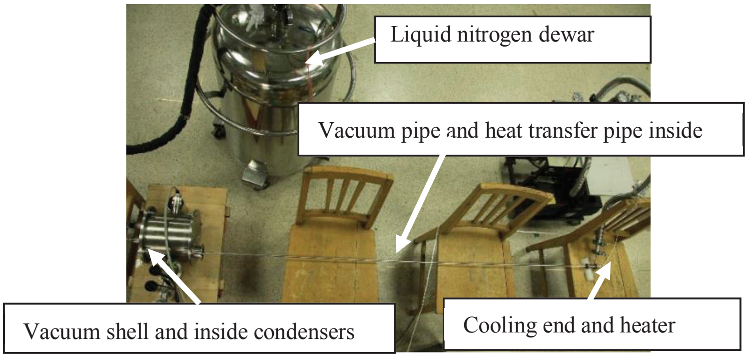



Fig. 3 Photo of the CRHD prototype

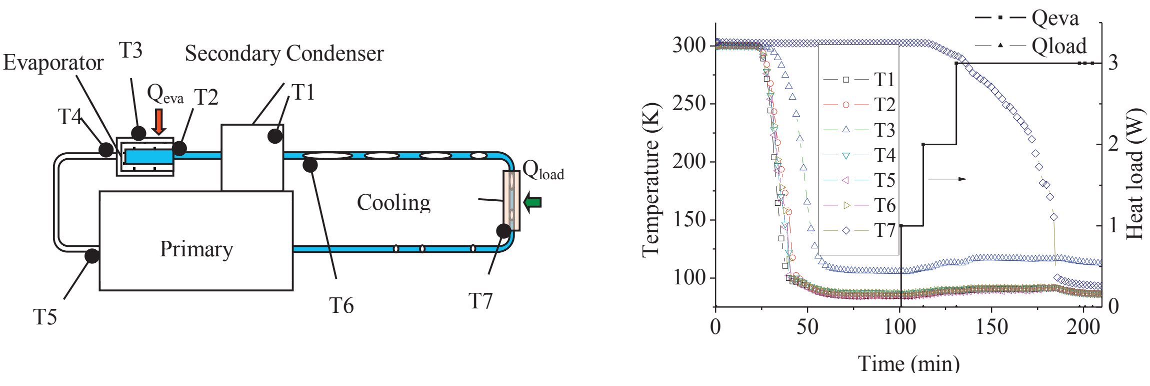
Table 1 Main dimensions of the CRHD (Unit: mm)

| Parts | Size | OD | ID | Length |
|---|---|---|---|---|
| Primary condenser | 50X70X15 | - | - | - |
| Secondary condenser | 50X70X10 | - | - | - |
| Evaporator | | 16 | - | 27 |
| Inner pipe | - | 2 | 1.6 | 2000 |
| Outer pipe | - | 4 | 3 | 2000 |
| Vacuum pipe | - | 6 | 5 | 2000 |
| Cooling end | - | 5 | 3 | 15 |

## 4. Experimental results and discussion

Experiments are carried out on the CRHD prototype. The experimental procedure is as follow: i. Charge a certain amount of nitrogen into the CRHD to ensure that there will be enough liquid in the cooling end in normal operation; ii. Make sure all sensors and heaters work well; iii. Fill in liquid nitrogen to cool the cold plate and the condensers; iv. After the temperature of the evaporator reaches the saturation temperature of nitrogen, which means that liquid reaches the evaporator, apply a load $Q_{eva}$ to the evaporator to push liquid from the primary condenser to the cooling end; v. After the temperature of the cooling end nearly reaches the saturation temperature of nitrogen and gets stable, which means that gas-liquid mixture reaches the cooling end, apply a load $Q_{load}$ to the cooling end, simulating a remote heat load from the cooling source; vi. Adjust and study the operation characteristics of the CRHD as scheduled.

Fig. 4a illustrates the temperature sensor arrangement of the CRHD and shows the cooling down process in a CRHD operation test. In this test the CRHD is kept horizontal, i.e. the axis of the evaporator, the condensers and the cooling end are at a same level. It is seen from Fig. 4b that the temperature of T1 ~ T6, except T3, drops quickly after liquid nitrogen filling-in at about 30 min, reaching about 85 K and getting stable at about 60 min. The cooling down process at T3 is a little behind, reaching about 106 K and getting stable at about 70 min. The temperature gap between T3 and others (except T7) is due to the poor thermal conductivity of the evaporator shell (made by stainless steel). The gap always exists during the CRHD operation. At about 100 min when T1 ~ T6 are all stable, a 1 W heat load $Q_{eva}$ is applied on the evaporator, and at about 135 min $Q_{eva}$ is gradually increased to 3 W. The temperature T7 starts to drop down at about 125 min when $Q_{eva}$ is 2 W, and finally reaches about 93 K at about 210 min, when $Q_{eva}$ is 3 W. From the above data it is clearly seen that the condensed liquid in the primary condenser is pumped into the cooling end 2 m away with a 3 W capillary pump (the evaporator), as always happens in a normal CLHP.




(a) Temperature sensors arrangement (b) Cooling down process

Fig. 4 Cooling down process in a CRHD operation test

Fig. 5 shows the variation of temperature difference (T7 – T1) as $Q_{load}$ varies while $Q_{eva}$ is kept constant (3 W). It is seen from this figure that the temperature difference increases together with $Q_{load}$, also similar with normal CLHP, and when $Q_{load}$ equals 1 W the temperature difference increases significantly, indicating that $Q_{load}$ is approaching its upper limit. For infrared sensor remote cooling, a cooling power of 1 W at 80 K is enough for most cases.

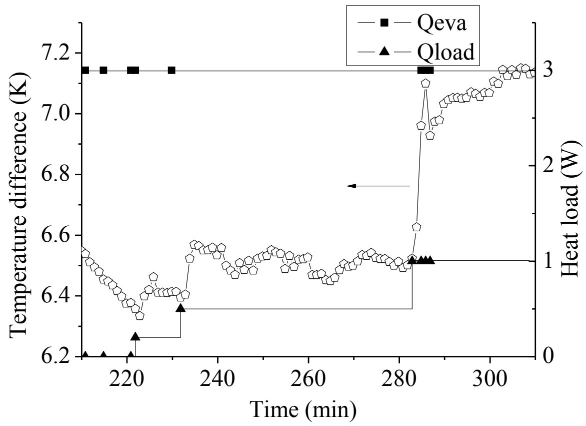



Fig. 5. Temperature difference vs. $Q_{load}$ in a CRHD operation test

Now we will consider the heat transfer efficiency as defined in Eq. (2). At about 310 min in the above test the whole system is deemed thermally steady. At this time $Q_{eva}$ equals to 3 W, $Q_{load}$ equals to 1 W, and $Q_{para}$ is the net radiation heat from the inner surface of the vacuum pipe to the outer surface of the outer pipe of the CRHD, roughly estimated with the radiation heat transfer equation:

$$Q_{para} = \left( \frac{1}{\frac{1}{\varepsilon_{vp}} + \frac{1}{\varepsilon_{op}} - 1} \right) A_{op} \left( T_{vp}^{\ 4} - T_{op}^{\ 4} \right) \qquad \text{(Equation holds for } A_{op} \approx A_{vp}\text{)} \tag{3}$$

where the subscript "vp" stands for vacuum pipe, "op" stands for outer pipe of the CRHD, $\varepsilon$ for emissivity, $A$ for surface area and $T$ for radiation surface temperature, roughly equal to pipe material temperature. The view factor of the outer pipe radiation surface to the inner surface of the vacuum pipe is 1. In this case $A_{op}$ is 0.025 m$^2$, $T_{vp}$ is 300 K, $T_{op}$ is 85 K, $\varepsilon_{vp}$ is set to 0.6 (rough and oxidized steel surface), and $\varepsilon_{op}$ is set to 0.3 (polished steel surface). Therefore $Q_{para}$ is 2.97 W, roughly 3 W. Since there is only a 0.5 mm gap between the vacuum pipe and the CRHD outer pipe, point or line contact therebetween is inevitable, which will lead to some conduction heat flow. This part of heat load is neglected in this paper.

With the above analysis the heat transfer efficiency of the CRHD prototype can be calculated as:

$$\eta = \frac{Q_{load}}{Q_{eva} + Q_{load} + Q_{para}} = \frac{1}{3+1+3} \approx 14\% \tag{4}$$

From a thermal view point the CRHD does not have a good heat transfer performance in comparison with CLHP and CCPL, which may have an efficiency of over 85% ($Q_{load}$ over 40 W with 2 W $Q_{eva}$ and roughly 3 W $Q_{para}$) [5]. Meanwhile, the CRHD has its own advantage of remote heat transfer and very compact cooling end.

In the future the authors will focus on improving the CRHD efficiency, mainly through lowering the surface emissivity of both the vacuum pipe and the CRHD outer pipe. It is estimated that if the surface emissivity can be lowered to below 0.1, the efficiency can be improved to over 20%.

Another possible way to improve the efficiency is to decrease $Q_{eva}$, therefore another test is carried out studying the effect of $Q_{eva}$ on thermal stability of the CRHD. Fig. 6 shows the test results. It is seen from the figure that when $Q_{eva}$ is 3 W, the system is thermally stable, whereas if $Q_{eva}$ is decreased to 1.5 W, a significant temperature fluctuation occurs at the cooling end, indicating that the two-phase flow inside the cooling end becomes unstable.

After $Q_{eva}$ is adjusted to 2 W and fluctuation disappears. It is hard to explain how the two-phase flow gets unstable at a relatively low $Q_{eva}$, but the test results tell us that $Q_{eva}$ is not allowed too small for a steady operation.

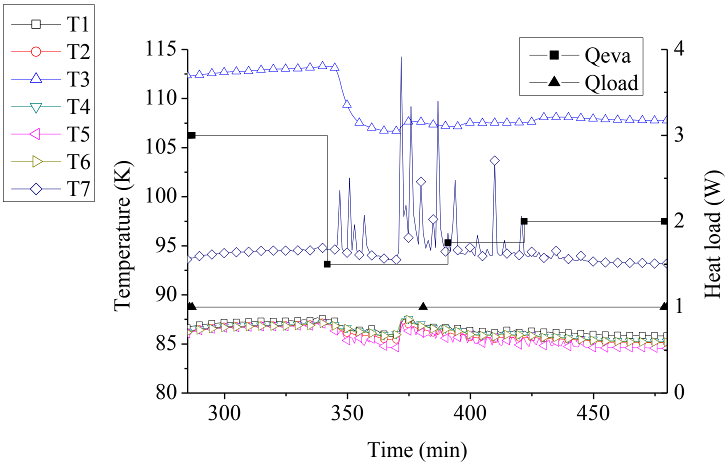



Fig. 6. System thermal stability at different Qeva in a CRHD operation test

## 5. Conclusion

A CRHD prototype is designed, fabricated and tested in Key Laboratory of Space Energy Conversion Technologies. The CRHD is designed on basis of a CLHP, and is aimed at solving the problem of space/ground remote cooling and narrow space cooling at about 80 K. The fabricated CRHD prototype is capable of cooling an object 2 m away from the cooling source with a 5 mm in diameter and 15 mm long cooling end. The performance of the CRHD is as follow:

i. The CRHD prototype has a heat transfer efficiency of about 14%, which is to say, it can transfer 1 W of cooling power at about 80 K over 2 m at a cost of another 6 W of cooling power;

ii. The cooling power can be decreased to 5 W for this prototype, and further improvement of the efficiency is expected through reducing the radiation heat load from the vacuum shell to the CRHD;

iii. For 1 W of transferred cooling power, a pumping power of at least 2 W is required to keep the whole system thermally stable.

## 6. Acknowledgement

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