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Design of the Large Dimension Cold Shield Cooled by G-M Cryocooler

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***Abstract*—**The design of methods of the 20 K large dimension cold shield used for infrared radiation demarcating in space environment simulation test were introduced in this paper. The cold shield were cooled by five G-M cryocoolers , and the dimension of the cold shield is the largest in our country.Cold shield installation and distribution and compensator for contraction on cooling were introduced detailedly. The temperature distribution and cool-down time of cold shield surface were also calculated and analysed in this paper. The design of cold shield resolves the difficulty of compensator for contraction on cooling successfully. Test results show that the actual technical performance indicators of cold shield met and exceeded the design requirements.

***Keywords*—**cold shield, G-M cryocooler , infrared radiometer demarcating, satellite, space environment simulation equipments

I. INTRODUCTION

A REMOTE sensing satellite is an artificial satellite to the Earth used to observe various features and phenomena of the Earth and atmospheric layer using remote sensing technology. Before the remote sensing satellite being launched, its remote sensor should undergo performance check/test and radiometer demarcating test in some special equipment which can simulate the relevant space environment, terrestrial radiation and reflected solar radiation to quantitatively measure the solar radiation scattered and reflected by the Earth and the terrestrial infrared radiation. Generally speaking, the test equipment used for the demarcation of the satellite radiation should be oil-free and in an ultra-high vacuum. Its cold shield temperature should be lower than 25K and its heat sink temperature should be under 100K[1].

Up until now, several infrared radiometer demarcating devices have been developed domestically, but most of whose cold shields still apply the helium refrigeration system which is complicated and unstable. To cool the cold shield with G-M cryocooler by connecting the cold shield with the cold head is a less complicated and more stable way. The cold shield using G-M cryocooler as its cold source has been developed successfully in China, however, this kind of cold shield is cooled by a single G-M cryocooler and its dimension is relevantly small. The cold shield in certain space environment simulation equipment described in this paper has a dimension up to 1m×1m and needs to be cooled by 5 G-M cryocoolers. Because it is the largest one among the cool shields of the same kind, people may have to cope with various problems concerning its installation, vacuum sealing and production.

II. STRUCTURAL DESIGN OF THE COLD SHIED

A. Cold Head

The cold shield uses the G-M cryocoolers for refrigeration, which is transferred to the shield through the cold head. The model of the chosen cold head, which is a dual-stage cold head, is 10MD made by Leyhold Company. The refrigeration capacity provided by a single cold head is 18W@20K for a second stage cold head and 110W@80K for a first stage cold head. 5 cryocoolers are used because according to the design requirement, the required refrigeration capacity is 80W@20K. The shape of the dual-stage cold head 10MD is illustrated in Fig. 1.

Fig. 1 Schematic Diagram of the Shape of the Cold Head

B. Configuration of the Cold Shield

Fig. 2 shows the configuration of the whole cold shield. The upper surface of the cold shield is connected with the flange surfaces of the second stage cold heads through screws. The lower surface is of the honeycomb texture and can absorb the heat discharged by the radiation cryocooler. The heat will then be transmitted to the flange surfaces of the second stage cold heads and taken away by the cryocooler. There are holes in the cold shield baffle through which the second stage cold heads cross and are connected with the flange surfaces of the first stage cold heads with screws. The funtion of the cold shield baffle is to separate the cold shield from the container to avoid direct radiant heat exchage between them so as to decrease the thermal load of the cold shield. The cold shield can absorb the radiant heat of the container and the heat sink and then transmit the heat to the flange surfaces of the first stage cold heads. The heat will eventually be taken away by the cryocooler. The cold shield baffle bends downward and forms a close cavity with the cold shield. This design can reduce the radiant heat received by

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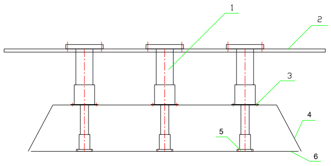
the cold shield as much as possile because the temperature of the cold shild baffle is lower than that of the heat sink and container. At the same time, the downward bending of the cold shield baffle can also reduce the radiated area of the cold shield, so the received radiant heat will also be reduced.

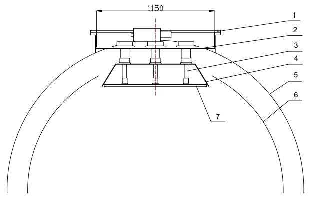


1-Cold Head; 2-Flange; 3-First Stage Cold Head Flange; 4-Cold Shield Baffle; 5-Second Stage Cold Head Flange; 6-Cold Shield
Fig. 2 Schematic Diagram of the Configuration of the Cold Shield

C. Installation and Layout

The installation positions of the cold shield and the container are shown in Fig. 3. Open a hole in the container and sink the flange assembly into the container through the hole. The upper flange surface and the container's flange surface are connected in the way of vacuum sealing. The cold head and the installation flange surface are connected in the way of vacuum sealing.

The layout of the cold heads is shown in Fig. 4. In order to save space, the hole should be as small as possible. But at the same time, enough space should be guaranteed for the installation of the helium pipe (The whole length of the straight pipe head and the bended pipe is 150mm, so 170mm needs to be preserved). The cold head needs to be rotated by 50°horizontally. The ultimate decided diameter of the hole is φ1150.



1-Upper Flange Surface; 2-Intallation Flange Surface; 3-Head Cold; 4-Cold Shield Baffle; 5-Container; 6-Head Sink; 7-Cold Shield
Fig. 3 Schematic Diagram of the Installtion Positions of the Cold Shiled and the Container

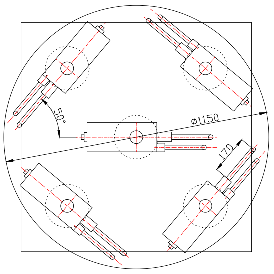


Fig. 4 Schematic Diagram of the Layout of the Cold Heads

D. Compensation Structure When the Cold Shield Suffers from Cold Contraction

The cold shield (with honeycomb made of aluminum foil) contracts by 4.8‰ when its temperature drops from the normal temperature to 20K. The cold shield baffle (made of aluminum panel) contracts by 4‰ when its temperature drops from the normal temperature to 40K. If installed in the normal way, i.e., fixed on the flange through the installation flange surface of the cold heads, the 5 cold heads will certainly endure some sheer forces induced by the contraction of the cold shield and baffle when the temperature drops. But the allowable lateral force of the cold head decided by its own characteristic is no more than 6kg. In order to compensate the sheering force to the cold head brought by the cold contraction of the cold shield, the cold shield described in this paper is made into a whole and the connection between the cold head and the container is designed to be laterally movable. Weld one end of the bellows with the fixed flange and screw the other end of the flange with the cold head flange to form a vacuum sealing. The supporting seat holds the bellows to keep it from contracting axially under the atmospheric pressure. Balls are also installed between the bellows and the supporting seat to reduce the friction between them by changing it into rolling friction. At the same time, the cold shield baffle is divided into 5 pieces and installed on the cold head respectively to abate the force induced by the change of temperature so that it is affordable to the cold heads.

According to the layout of the cold heads and the dimension of the cold shield, it can be calculated that the displacement of each cold head does not exceed 2mm when the temperature drops from the normal temperature to 20K. In order to testify the feasibility and reliability of our design, a special test device is designed in this paper (shown in Fig.5) to measure the force required to pull the cold head for 2mm laterally with the test device. The test result shows that with the design structure in this paper, the force needed for a 2mm lateral displacement does not exceed 5kg, which is less than the allowable sheer force of the cold head. In a word, the compensation problem of the cold shield's cold contraction is successfully figured out.

Fig. 5 Test Device of the Pulling Force Measurement

E. Facial Material laid on the Cold Shield – Honeycomb Texture

In order to improve the facial emissivity of the cold shield so that it can absorb the maximum radiant heat sent by the radiant cooler, a texture with a larger radiant heat-exchange area is applied here, i.e., the hexagonal honeycomb texture on the surface of the cold shield facing the radiant source. The honeycomb texture is just like a blackbody with each honeycomb being like an empty cellar according to the radiant theory. After the rays sent by the radiant cooler enter into the hexagonal honeycomb, they will be reflected for many times. Each reflection means that part of the energy will be absorbed by the honeycomb. Almost all the radiant heat will be absorbed after being reflected time and time again. Fig. 6 is the schematic diagram.

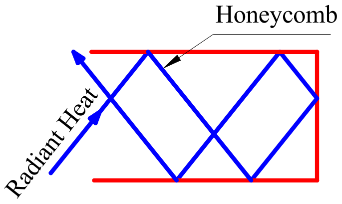


Fig. 6 Schematic Diagram of the Approximate Black Body Formed by the Honeycomb

On the other hand, to make the honeycomb texture better absorb the radiant energy, a special kind of black lacquer can be sprayed and painted on the surface of the honeycomb texture. The absorptivity of this kind of black lacquer is relevantly high and generally more than 0.9. Besides. The material of the honeycomb texture should have a good thermal conductivity and be thick enough so that its facial temperature is even. The material selected in this paper is the aluminum alloy with a thickness of $\delta = 0.1$mm. The length of each leg of the hexagonal honeycomb is 2mm and the height of the honeycomb is 8mm.

III. THERMAL DESIGN OF THE COLD SHIELD

A. Relationship between Cold Head Refrigeration Capacity and Temperature

The relationship between refrigeration capacity of 10MD cold head and temperature is shown in Fig. 7, from which it can be read that the 1st and 2nd stage cold head are of the same characteristics, that is, to attain a lower temperature requires a smaller registration capacity. The registration capacity of the 1st stage cold head is 18W@20K, and 110W@80K for the 2nd stage.

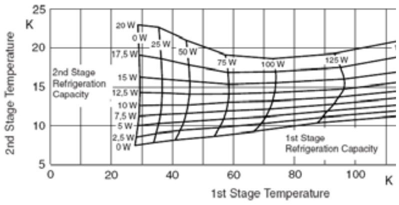


Fig. 7 Relationship between Cold Head Refrigeration Capacity and Temperature — Curve Graph

B. Selection of Cold Shield Material

It is required that the material for the cold shield is of sufficient strength and plasticity as well as good weldability, so the stainless steel, red copper and pure aluminum are usually selected as the material. The stainless steel material can bring not only good weldability and deformation resistance ability but also poor thermal conductivity, difficult processing and a great weight; the red copper material can give relatively good thermal conductivity, weldability and processing ease but unsatisfied deformation resistance ability and a relatively great weight; while the aluminum materials can help achieve satisfying thermal conductivity, processing ease, low weight, deformation resistance ability but also poor weldability, so the focus is fixed on how to weld aluminum and red copper honeycomb together. On the basis of intensive survey and research, the aluminum material having better thermal conductivity and lower weight is selected as the material for making the matrix of the cold shield. To enhance the emissivity and reduce the surface temperature of the cold shield, aluminum foil is adopted as the fundamental material for making the honeycomb. The thermal conductivity of copper and aluminum increases as the temperature decreases, and it has been looked up that at 40K, the coefficient of thermal conductivity of aluminum is 2300W/(m·K)[2].

C. Thermal Calculation of the Cold Shield Baffle

To reduce thermal load of the cold shield, the temperature of the cold shield baffle should be as low as possible. It can be read from Fig.7 that if the 2nd stage cold head needs to reach a relatively low temperature, there should be a relatively small refrigeration capacity, which requires the cold shield baffle receives heat radiance as slightly as possible. It is shown in Fig.3 that squarely facing the normal temperature vessel, the top surface of the cold shield baffle absorbs the radiant heat Q_1, which can be calculated [3] as per the formula (1) that follows:

$$Q_1 = \varepsilon_{1,1} A_1 \times 5.67 \left[\left(\frac{T_0}{100} \right)^4 - \left(\frac{T_1}{100} \right)^4 \right] \qquad (1)$$

in which,

Q_1 — heat absorbed by the cold shield baffle, W;

$\varepsilon_{1,1}$— emissivity of the top surface of the cold shield baffle;

A_1— surface area of the cold shield baffle, through calculation, A1=1.5m2;

T_0— surface temperature of the vessel, as per T=300K;

T_1— surface temperature of the cold shield baffle, K;

As indicated in formula (1), to gain a smaller Q_1, make the $\varepsilon_{1,1}$ smaller, therefore select an aluminum plate with lower emissivity and make the emissivity as low as 0.1 by means of polishing treatment. T_1 is lower in relation to T_0 so the formula can be simplified into $Q_1 = \varepsilon_{1,1} A_1 \times 5.67 \times \left(\frac{T_0}{100} \right)^4$.

It is calculated that Q1=69W. Because there are holes in the cold shield baffle, the contact surface between the baffle and the flange face of the 1st stage cold head decreases to 82.7% in area. In view of contact thermal resistance and other factors, an individual 1st stage cold head needs to provide the refrigeration capacity $Q = \frac{69}{5 \times 0.827 \times 0.8} = 21W$. It can be read from Fig. 7 that the temperature of the flange face of the 2nd stage cold head is 35K.

The cold shield has 5 1st stage cold heads which equivalent to 5 ring fins. Take an individual ring fin as an example to establish a thermal physical model as given in Fig. 8. The cold shield baffle absorbs the radiant heat Q_1 which will then be transferred to the aluminum plate contacting the 1st stage flange face (in the direction of Q_2), the 1st stage flange face (in the direction of Q_3), and finally taken away by the cryocooler. Since the 1st cold head flange face is attached to the cold shield baffle with a sheet indium being sandwiched in the middle which is of excellent thermal conductivity and relatively good flexibility, a satisfying contact thermal conductivity is therefore ensured. In view of contact thermal resistance existing between the sheet indium and the cold head, it is considered that in the contact area the temperature of the aluminum plate is 35.5K, 0.5K higher than the flange face.

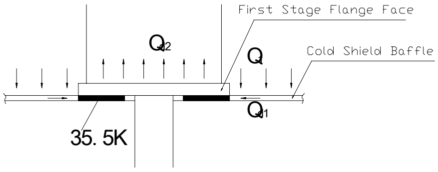


Fig. 8 Physical Model of a Ring Fin of the 1st Stage Cold Head

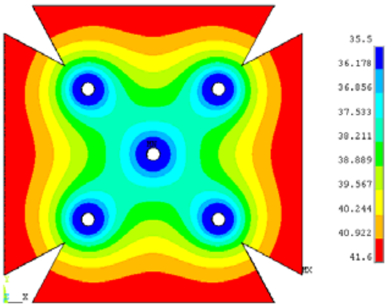


Fig. 9 Cold Shield Baffle Temperature Distribution Diagram

The aluminum plate used here is 2 mm thick. Through calculation and analysis mentioned above, temperature distribution of the cold shield baffle in stable conditions can be figured out by means of ansys calculation, as is shown in Fig. 9. From the figure, it can be read that the highest temperature of the cold shield baffle is 41.6K.

D. Cold Shield Heat Calculation

Since in finally stable conditions, the cold shield and the baffle are about 20K and 40K, even though both of them have high surface emissivity, the radiant heat exchanged between them is still low. However, in the initial phase, the baffle can be easily cooled down due to its relatively low thickness and relatively high refrigeration capacity of the 1st stage cold head, which makes it a cold background for the cold shield. As a result, to shorten the precooling time of the cold shield, apply black paint to the top surface of the cold shield and bottom surface of the baffle in order to increase its emissivity.

The radiant heat that the cold shield shall take away can be calculated as per formula (2)[3]:

$$Q_2 = \frac{A_2}{\frac{1}{\varepsilon_{2,1}} + \frac{A_2}{A_1}\left(\frac{1}{\varepsilon_{1,2}} - 1\right)} \times 5.67\left[\left(\frac{T_1}{100}\right)^4 - \left(\frac{T_2}{100}\right)^4\right] \tag{2}$$

in which,

Q_2 — heat absorbed by the cold shield, W;

$\varepsilon_{2,1}$ — emissivity of the cold shield top surface, as per 0.9;

$\varepsilon_{1,2}$ — emissivity of the cold shield baffle bottom surface, as per 0.9;

A_1 — cold shield baffle surface area, through calculation, $A_1 = 1.5m^2$;

A_2 — cold shield surface area, $A_1 = 1m^2$;

T_1 — cold shield baffle surface temperature , 42K;

T_2 — cold shield surface temperature , as per $T_0 = 20K$;

Through calculation, $Q_2 = 0.11W$, a very tiny value that can be ignored.

According to requirements on technical indexes, the heat the cold shield should take away from the radiant cryocooler is 80W, which mainly concentrates in a circle of $\oint$ 540 mm located in its center; while the cold shied not radiated by the radiant cryocooler has to absorb the heat from heat sinks, approximately 5W, so an individual 2nd stage cold head should provide a refrigeration capacity of 17W. It can be read from Fig. 7 that the flange face temperature of the 2nd stage cold head is 18.5K.

The cold shield has 5 2nd stage cold heads which equivalent to 5 ring fins. Take an individual ring fin as an example to establish a thermal physical model as given in Fig. 10. The cold shield baffle absorbs the radiant heat Q_1 which will then be transferred to the aluminum plate contacting the 2nd stage flange face (in the direction of Q_{d3}), then the 2nd stage flange face (in the direction of Q_{d4}), and finally taken away by the cryocooler. Since the 2nd cold head flange face is attached to the cold shield baffle with a sheet indium being sandwiched in the middle which is of excellent thermal conductivity and relatively good flexibility, a satisfying contact thermal conductivity is therefore ensured. In view of contact thermal resistance existing between the sheet indium and the cold head, it is considered that in the contact area the temperature of the aluminum plate is 19K, 0.5K higher than the flange face.

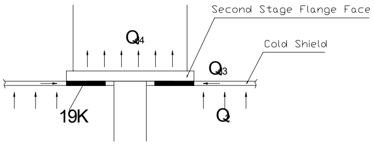


Fig. 10 Physical Model of a Ring Fin of the 1st Stage Cold Head

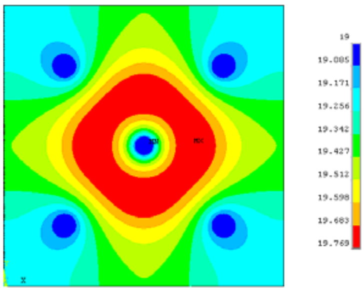


Fig. 11 Cold Shield Temperature Distribution Diagram

The cold shield temperature distribution in stable conditions, as shown in Fig. 11, is calculated through ansys. It can be read from the figure that the maximum variation in terms of the temperature distributing over the cold shield is 0.769K, which meets the indexed requirement that the variation must be lower than 20K. Here the thermal conductivity coefficient of the cold shield is calculated at the temperature of 40K. If it is about 20K, the thermal conductivity coefficient of the aluminum plate increases and the temperature over the cold shield will therefore be much more evenly distributed. It can also be read that the highest temperature is 19.769K, meeting the indexed requirement that the highest temperature must be lower than 20K.

E. Calculation of Precooling Time of the Cold Shield

Fig. 12 shows the heated cold shield and cold shield baffle during precooling process: precooling of the cold shield is achieved by taking the heat away which can be divided into three parts, that is, radiant heat Q_2 to the cold shield baffle, heat Q_{z2} that taken away by the 2nd stage cold head, part of the radiant heat Q_3 to the heat sinks.

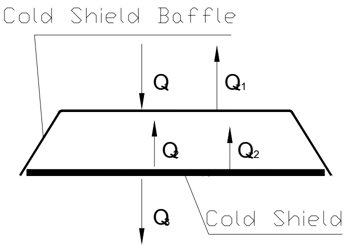


Fig. 12 Sketch Map of The Heated Cold Shield and Cold Shield Baffle

On the one hand, the baffle absorb the radiant heat Q_1 from the vessel, and radiant heat Q_2 from the cold shield, on the other hand, the 1st stage cold head takes away the heat Q_{z1}, so the baffle is pre-cooled under the combined operation of the baffle and the cold head.Due to the thickness of the baffle and a much greater refrigeration capacity of the 1st stage cold head than the 2nd stage, the cold shield baffle can be easily cooled down. It can be seen from Fig. 7 that in the beginning, the 1st stage cold head precools the baffle to a temperature of 116K at the maximum refrigeration capacity. Assume that during this process, the cold shield keeps the same temperature (actually the temperature falls, but it is not obvious), so the temperature variation of the baffle can be figured out. It is calculated that the temperature of the baffle varies with the precooling time, as is shown in Fig. 13. From the figure, it is read that if the precooling time is 3300s, the temperature of the baffle falls to 116K. At this point, the radiant heat Q_2 from the cold shield to the baffle becomes obvious, and the decreasing of the cold shield temperature can be calculated from this point. Through calculation, variation of the cold shield temperature with the precooling time is shown in Fig. 14.

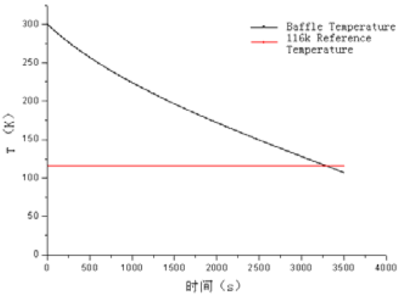


Fig. 13 Variation Curve of the Cold Shield Baffle Temperature

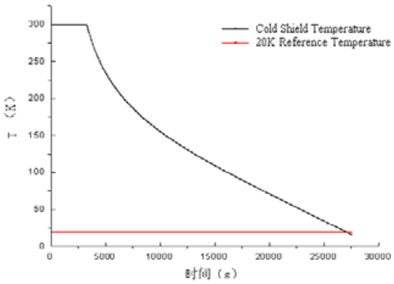


Fig. 14 Variation Curve of the Cold Shield Temperature

It can be read from Fig.14 that it takes 27000s, that is, 7.5 hours for the cold shield to get a temperature of 20K, which meets the indexed requirement that the precooling time must be less than 8 hours. During the initial 3300s, the temperatures are shown as a straight-line segment because of the above-mentioned assumption that the cold shield keeps the same temperature in this period, while actually the cold shield cools down at the beginning of the precooling process so that the actual precooling time of the cold shield is less than 7.5 hours.

IV. APPLICATIONS IN TESTS

The cold shield researched in this paper has been applied to infrared radiometer demarcation of satellite optical remote sensing devices many times upon its successful development.

During the test, 4 hours after the G-M cryocooler started working, the average temperature of main test points on the cold shield was 15K with a temperature homogeneity error of ±1K, which has achieved and surpassed the design demand and satisfied the test requirements. The variation curve of the cold shield temperature is given in Fig. 15.

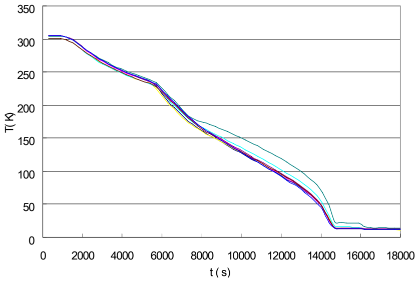


Fig. 15 Variation Curve of the Cold Shield Temperature in Test

V. CONCLUSION

The development of the cold shield in this paper has not only solved some regular problems in terms of cold shield design, but also cracked a hard nut in cold shield design — compensation for cold contraction of large-dimension cold shields, which has successfully made the maximum dimension of domestic cold shield of the same type enlarged to 1m × 1m. The design thought and method discussed in this paper will bring meaningful reference and useful experience for design of cold shields in larger dimension in future.

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