The Study on High Efficiency and Low Vibration Flexure Bearing Stirling Cryocooler

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ABSTRACT

In this paper, a high efficiency and low vibration Stirling cryocooler has been designed and manufactured. The high efficiency compressor implementing the technology of dual opposed moving magnet motor and flexure bearing has been optimized to drive pneumatically a Stirling cold finger also implementing flexure bearing technology. Through theoretical study and experimental study, the cryocooler can achieve performance of 3W/80K under 60 WAC of electrical power. The vibration of compressor is suppressed by reducing the weights of moving-masses and controlling the assembly process. The vibration suppression of the cold finger is implemented in terms of a mass-spring passive balancer. The vibrations of the compressor and the cold finger could be reduced to below 5.6 mg and 1.9 mg respectively using the above-mentioned methodology.

INTRODUCTION

The application of mechanical cryocoolers such as Stirling cryocooler and pulse tube cryocooler are increasing rapidly in the aerospace field for the cooling of infrared detectors and optic devices[1]. The general requirements are high reliability and efficiency, low vibration export, ability to survive launch vibration extremes and long-term exposure to space radiation[2]. Compared with the pulse tube cryocooler, the Stirling cryocooler could offer a very high efficiency in a limited volume and low mass because the moving displacer can store sufficiently expansion work. However, a disadvantage of the Stirling cryocooler in comparison to the pulse tube cryocooler is the higher induced vibration. Normally, the major portion of this vibration export occurs at the driving frequency. So far, there are two approaches to active control and passive control for suppression of this vibration[3, 4].

In the paper, a high efficiency and low vibration Stirling cryocooler has been demonstrated for cooling down sensitive IR devices. A high efficiency compressor implementing the technology of dual opposed moving magnet motor and flexure bearing has been optimized to drive pneumatically a Stirling cold finger also implementing flexure bearing technology. Through theoretical study and experimental study on the compressor and the cold finger, the cryocooler can reach a performance of 3W/80K under 60 WAC of electrical power.

It is also particular important to reduce the vibrations coming from the compressor and the cold finger. The vibration of the compressor is caused by the unbalance forces between dual op-

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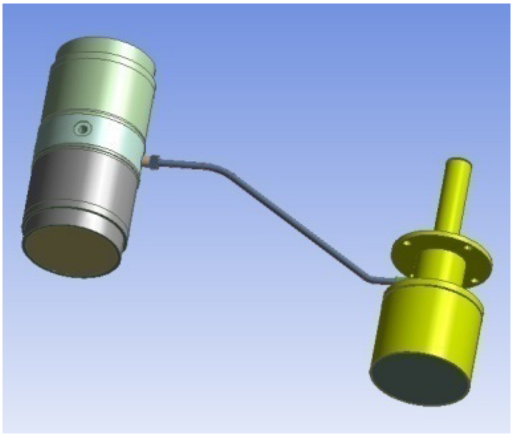

Figure 1. The schematic presentation of the stirling crycooler.

posed structures. It is suppressed by reducing the weights of moving-masses and controlling the assembly process. The vibration suppression of the Stirling cold finger is implemented in terms of a mass-spring passive balancer. The vibrations of compressor and the cold finger can be decreased to below 5.6 mg and 1.9 mg respectively under the above-mentioned methodology.

THE DESIGN AND EXPERIMENTAL STUDY OF HIGH EFFICIENCY STIRLING CRYOCOOLER

The Structure of the Stirling Cryocooler

A schematic representation of the Stirling cryocooler is provided in Fig. 1. It is a split Stirling cryocooler and has the characteristics of higher reliability and smaller vibration in comparison to the integral Stirling cryocooler. A linear motor structure with moving magnet in the compressor and a pneumatically driven displacer in the cold finger are proposed and designed. In the moving magnet linear motor, the magnets are connected directly to the pistons and the coil holders are part of the compressor housing. The displacer is supported by flexure bearings with sufficient radial stiffness and appropriate axial stiffness.

Design and Experimental Study of Compressor

As shown in Fig. 1, the main parts of the compressor include the pistons, the cylinder, the magnets, the flexure bearings, and the coils. This structure can offer a number of advantages such as coil outside the working gas without gas pollution, absence of flying leads and glass feed-throughs[5]. These increase reliability of the compressor and allow the compressor design to be more compact. The calculations simulating the complete dynamic behavior of the compressor have been done to find the optimal dimensions. The motor dimensions and flexure bearing structures have also been optimized with the Finite Element Analysis (FEA) software, Opera Simulation Software.

Moreover, the several main factors referring to electrical resistance of the coil, the axial stiffness of the flexure bearing and diameter of the piston that influencing efficiency of the motor have been experimentally tested. It should be noted that the experiments have been done with the compressor matching a cold finger in which the displacer is supported by cylinder spring. Experimental results of the above factors are shown in Tables 1-3. The results show that the coil resistance of 1.43 Ω reaches the best cooling capacity in Table 1. The optimal axial stiffness of flexures is 2.7 N/mm in Table 2, when further reducing the stiffness of flexure bearings in the experiment, the moving mass would not be supported by flexure bearings. In Table 3, the diameter of 14 mm could get a better cooling capacity because of a pressure ratio of 1.307 at the outlet of compressor. Nevertheless it is only 1.289 at the outlet of compressor with the piston's diameter of 15 mm. The main reason for the pressure ratio difference is that there is bigger void volume in the compressor with a piston

Table 1. The results of different coils experiments.

NO.	Diameter（mm）	Resistance（Ω）	Performance	Remarks
1	0.63	1.5	1.9W/80K@48.4WAC	Filling pressure: 2.2MPa
2	0.63	1.8	1.9W/80K@48.9WAC	Operating frequency: 50Hz
3	0.69	1.43	2.12W/80K@48.5WAC	Piston diameter: 14mm

Table 2. The results of different stiffness of the flexure bearings experiments.

NO.	Stiffness（N/mm）	Performance	Remarks
1	2.7	2.24W/80K@48.5WAC	Filling pressure: 2.2MPa Operating frequency: 50Hz Piston diameter: 14mm
2	3.4	1.9W/80K@48.4WAC	

Table 3. The results of different diameters of the pistons experiments.

NO.	Diameter（mm）	Performance	Remarks
1	14	2.24W/80K@48.5WAC	Filling pressure: 2.2MPa
2	15	1.9W/80K@48.4WAC	Operating frequency: 50Hz

diameter of 15 mm. The redundant void volume causes the pressure ratio and the performance of compressor to decrease.

Design and Experimental Study of the Cold Finger

The cold finger has also been designed and manufactured in which the support of the displacer is made with flexure bearing instead of cylinder spring. It was decided to use a pneumatically driven displacer to generate the movement of the displacer because of the large gas volume at the position of the flexures. The flexure bearings with the different radial stiffness and axial stiffness have been designed by the FEM simulation package. The design of the flexure follows the four main important criteria below: the flexure has sufficient radial stiffness over the complete stroke of the displacer to prevent the contact wear between the displacer and the expansive cylinder, the flexure has a constant axial stiffness, the flexure own resonance frequencies are sufficient high and the maximum stress level is far below the fatigue limit stress.

According to the dynamic behavior of the displacer, the axial stiffness of flexure bearing influences the performance of cold finger. The theoretical and experimental studies have been done to find the optimal axial stiffness of flexure bearing. It is shown that the theoretical result of axial stiffness is 4.6 N/mm by the dynamic analysis. The experimental results are shown in Fig. 2. In the experiment, the cryocooler maintains the temperature of 80 K at the cold finger's cold end with the electrical power of 60 WAC. The results show the cryocooler get the maximal cooling power of 3W/80K at the actual optimal axial stiffness of 4.8 N/mm. There is only 4.35% deviation between the theoretical result and experimental result.

VIBRATION CONTROL OF HIGH EFFICIENCY STIRLING CRYOCOOLER

The Stirling cryocooler is composed of a dual-piston linear compressor and a pneumatically driven cold finger. The reciprocating motion of dual-piston provides the required pressure wave and mass flow of the working gas for the cold finger. The vibration export comes from mainly the unbalance forces due to the obvious dissimilarity of the opposite components and the single-mass displacer in the cold finger. Normally, the major portion of this vibration export occurs at the driving frequency. In the paper, the vibration come from compressor is suppressed by reducing the weights

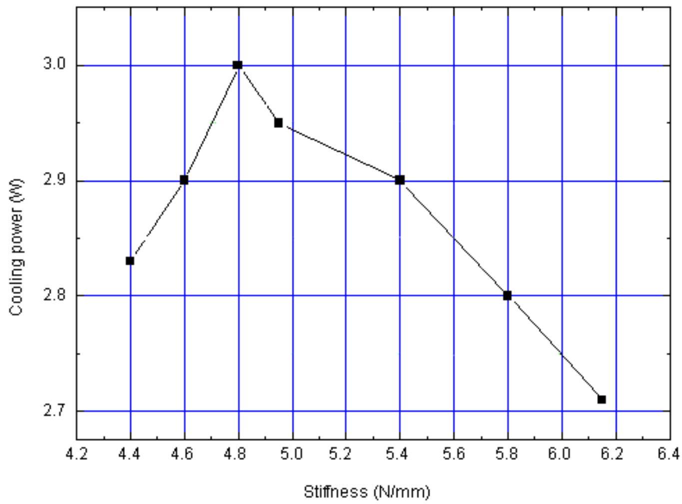

Figure 2. The experimental results of axial stiffness vs cooling power.

of moving-masses and controlling the process of assembling. The vibration suppression of the cold finger is implemented in terms of a mass-spring passive balancer.

Vibration Control of the Compressor

In theory, the linear compressor should not generate vibration because of a pair of back-to-back components. But there are some dissimilarity in the mass of the opposing components, such as the moving masses, the spring constants, the clearance sealings, and the friction factors and these cause a unbalance force. This force generates the vibration. That can be given as:

$$F = m_c a = m_{c1}\ddot{x}_1 - m_{c2}\ddot{x}_2 \tag{1}$$

where m_c and a are the mass and instantaneous acceleration (also named vibration export) of the whole compressor, m_{c1} and m_{c2} are the moving masses of opposite components, $\ddot{x}_1$ and $\ddot{x}_2$ are the second derivatives of the opposite components' strokes.

The formula shows that the vibration export is related to the moving masses. In the design phase of compressor, the moving masses have been decided to adopt a light weight concept. The moving mass of the compressor prototype is smaller by 100 g than the original design.

Vibration Control of the Cold Finger

In the cold finger, the displacer and the flexure bearings form a single-degree-of-freedom vibration system that resonates with the desired stroke and optimal phase lag relative to the pressure waves arriving from the compressor. The unbalanced force of the single-degree-of-freedom vibratory system causes the vibration export. The control of vibration export relies on the implementation of a passively driven counterbalance. Its dynamic model is shown in Fig. 3. In particular, m_1 and m_2 are the masses of the cold finger and the tuned dynamic absorber, c_1 and c_2 are the dampings of the cold finger and the tuned dynamic absorber, k_1 and k_2 are the stiffniss of the cold finger and the tuned dynamic absorber, x_1 and x_2 are the strokes of the cold finger and the tuned dynamic absorber. The dynamic analysis for this paper has already been completed [6]. According to the analysis, the tuned dynamic absorber is designed and manufactured with a moving mass of 218 g, a stiffness of 22 N/mm and a resonant frequency of 60 Hz.

Vibration Export Experiment on the Cryocooler

Figure 4 shows the Stirling crycooler mounted on the vibration export test platform. The cold finger is placed inside the envelope of the simulation dewar. In the experiment, the heat load of

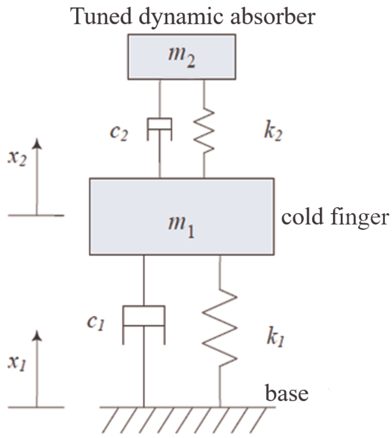

Figure 3. The dynamic model.

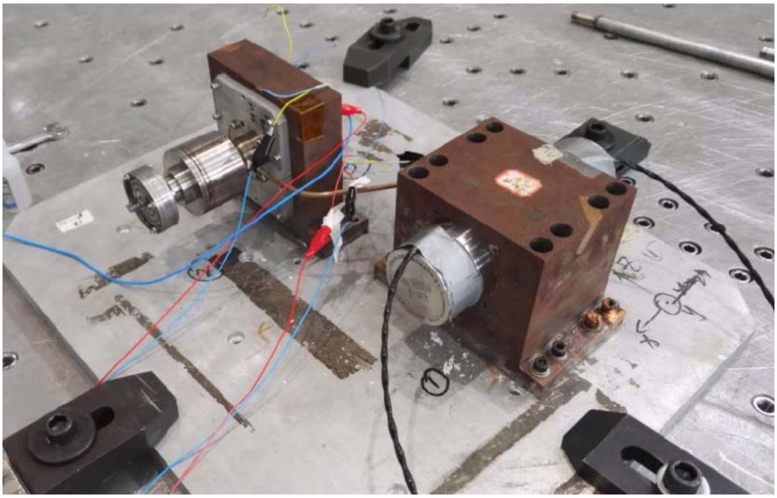

Figure 4. The vibration export test.

0.5 W at a reject temperature of 300 K is applied to the cold finger's cold end with the temperature of 80 K, which has a drive frequency of 60 Hz.

The Tables 4 and 5 show the test results of compressor and the cold finger. It can be seen the maximal vibration exports of the compressor and the cold finger occur at the driving frequency. The maximal vibration exports are 5.599 mg and 1.895 mg, respectively.

CONCLUSION

A Stirling cryocooler has been designed and manufactured, in which the technology of linear moving magnet motor and flexure bearing is implemented. Through theoretical study and experimental study, the cryocooler could reach performance of 3W/80K under the electrical power of 60 WAC. Through reducing the weights of moving-masses and applying a mass-spring passive balancer, The vibrations of compressor and the cold finger could be decreased to below 5.6 mg and 1.9 mg, respectively.

Table 4. Compressor vibration test results.

Frequency	Direction	Vibration export
60Hz	X	2.054mg
	Y	5.599 mg
	Z	5.109 mg
120Hz	X	0.184 mg
	Y	0.267 mg
	Z	0.313 mg
180Hz	X	0.041 mg
	Y	0.022 mg
	Z	0.014 mg
240Hz	X	0.021 mg
	Y	0.032 mg
	Z	0.015 mg
300Hz	X	0.021 mg
	Y	0.097 mg
	Z	0.094 mg

Table 5. Cold finger vibration test results.

Frequency	Direction	Vibration export
60Hz	X	0.049 mg
	Y	1.539 mg
	Z	1.895 mg
120Hz	X	0.013 mg
	Y	0.052 mg
	Z	0.105 mg
180Hz	X	0.022 mg
	Y	0.166 mg
	Z	0.250 mg
240Hz	X	0.051 mg
	Y	0.250 mg
	Z	0.213 mg
300Hz	X	0.381 mg
	Y	1.066 mg
	Z	0.287 mg

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