ABSTRACT

The demand for more cooling power for infrared imagers, which may require up to 3 W of cooling power at 77 K, is nowadays surpassed as other industries are getting interested in cryogenic cooling as well. These potential markets require robust, efficient and affordable coolers with cooling capacities in excess of 6 W. As announced at the previous SPIE conference in 2000, Thales Cryogenics has been working on the development of a cryocooler based on the LSF 918x series consisting of a flexure bearing compressor in combination with a 20 mm Stirling cold finger in order to meet the demands of this emerging markets. Based on the proven principles of Thales LSF 91xx flexure bearing compressors, a moving magnet compressor was designed that delivers the required pressure wave for this larger cold finger. The compressor has been successfully tested in combination with the 20 mm cold finger resulting in the LSF 93xx cooler. For the second half of 2002, tests are planned for the combination of a version of this compressor with a 5 W pulse tube cold finger.

At present, the European Space Agency is funding the space qualification of a modification the LSF 93xx cooler, in order to use it to provide the cryogenic cooling required for future manned missions. A test program for the specific requirements for the CRYOSYSTEM program is under progress.

This paper describes the trade-offs that have been considered in the design phase, and gives a detailed overview of the test results and the resulting specification of the LSF 93xx coolers.

BASIC PRINCIPLES

In conventional linear Stirling cryocoolers, the lifetime of the cooler is limited by the compressor. Inside a linear compressor, usually two pistons, driven by moving coil linear motors, are translating in opposite phase generating a pressure wave in the compression space between the pistons, which is connected to the warm end of the coldfinger. The magnitude of the generated pressure wave directly determines the cooling performance obtained with a certain Stirling coldfinger or pulse tube. To prevent gas leakage along the pistons inside the compressor, which reduces the generated pressure wave and thus the cooling performance, close tolerance seals are often applied. The principle of close tolerance seals is a very small annular gap between the piston and the cylinder. Typically, this gap has a length of a few centimetres, which prevents gas to flow from the compression space to a larger (buffer) space behind the pistons. To limit the gas flow along the pistons to an acceptable level, the gap between the piston and the cylinder should be as small as possible. However, it should still allow piston movement at different ambient temperatures between –52 °C and +71 °C. In practice the initial gap between the piston and cylinder will be around 10 microns. Increase of this gap due to wear of the coating applied on the piston increases the flow along the piston. In fact, the gas flow through the annular gap between the piston and cylinder is dependent on the gap height to the third power. The impact of an increase of the gap height between piston and cylinder on the efficiency of the cooler in practice, has been reported at the SPIE conference in 2000.
For the generation of the required cooling power of 6 W a 20 mm Coldfinger is developed and tested extensively. The use is of a 20 mm cold finger is made because the efficiency of a cold finger with a larger diameter is higher as the efficiency of a smaller cold finger. The effect of the cold finger diameter in the efficiency of the overall system is shown in Figure 2.

Figure 2. Impact of the coldfinger diameter on the efficiency of the cooler (measurement results @23°C)

COMPRRESSOR DEFINITION

A schematic representation of the compressor is provided in Figure 1. THALES Cryogenics developed in the past years several different sized flexures bearing compressor based on flexure bearing and moving magnet technology. With a moving magnet linear motor, the magnets are connected directly to the pistons
and the coilholders are part of the compressor housing. This offers a number of advantages that increase reliability of the cooler and allow the compressor design to be more compact.

First, the fact that the coils are no longer moving means that flying leads or flexible leads are no longer necessary to supply current to the coils. Absence of these components simplifies the compressor design and assembly and reduces the length. In the final flexure-bearing compressor design, the coil-holder on which the coils are wound is part of the housing of the compressor. This means that the coils are outside this housing and outside the working gas. As the coil insulation consist of synthetic materials which can absorb moisture, bake out of components and curing of the cooler under high vacuum at elevated temperatures is probably the most critical process in the cooler production. With the removal of the coils outside the working gas, no more outgassing components are present inside the cooler. This reduces the risk of gas contamination during the life of the cooler.

Finally, the fact that the coils are located outside the hermetically sealed compressor means that also glass feed-throughs are no longer required. Under extreme temperature shocks and severe mechanical stresses on the cooler glass feed-throughs are known critical components which can crack resulting in gas leakage.

All these advantages have lead to the conclusion that for a high reliability cryocooler moving magnet technology in combination with flexure bearing suspension is the best solution to guarantee long lifetimes.
At present THALES Cryogenics has two sizes of compressors developed on the above stated technologies. One compressor limited to a maximum input power of 100 Watt at 15 Vac with a diameter of 60 mm and a length of 165 mm and one compressor with a maximum input power of 150 Wac at 30 Vac with a diameter of 90 mm and a length of 204 mm. This later compressor design can also be used up to 200 W input power.

**Calculation of the motor dimensions**

The motor dimensions have been optimized with the FEM software PC Opera. The motor geometry has been changed compared to the geometry of the LSF91xx compressor. Instead of using 2 radial magnetized magnets on each motor half, one axial magnetized magnet is used with an inner stator on both sides of each magnet (see figure 2.1). The advantages of this magnet configuration are simplicity, each of manufacturing and minimization of failure mechanisms.

The only disadvantage of the axial magnet configuration is the slightly higher EMI level because there is more stray flux generated by an axial magnet as compared to a radial magnet. In figure 2.1 one half (this part is axis-symmetric around the vertical axis) of the geometry of half the compressor is plotted. The potential (flux) lines are shown as well in this figure. It can be seen that the flux is closed in this part of the motor.

![Figure 4. Geometry and potential lines of a moving magnet motor](image)

The efficiency of the optimal motor, when only the $I^2R$ losses are considered is calculated and measured to be 77%.

**Calculation of the hysteresis losses**

Next to the joule losses in the coil the electromotor will have other losses. These losses can be divided into hysteresis and eddy current losses. The hysteresis losses are very difficult to be calculated with the aid of the used software. However, no hysteresis losses are expected because the two coils on each motor half are oppositely wound. So the generated magnetic field in the inner and outer stator by one coil is mostly cancelled by the other coil.
Calculation of the eddy current losses

To calculate the eddy current losses in the coil support and the outer stator, some velocity analysis’s were done with PC Opera. The eddy current losses are a function of the height of the magnetic field, the velocity of the magnets and the $\sigma$ ($\sigma = 1/R [1/\Omega]$ is the reciprocal value of the electrical resistance of the material). If $\sigma$ is low, the eddy current losses will be low. The eddy current losses were calculated with PC Opera by doing a non-linear velocity analysis. The resistance force due to the eddy current at a velocity of 1432 mm/s calculated by PC-Opera is 6.28 N for one motor half. In case of the sinusoidal movement of the piston the eddy current losses of one motor half will be 4.5 W. The maximum needed force based on a the required pdV work will be 66.18 N. Because of this extra resistance force due to the Eddy current losses the system damping will increase and thus the efficiency of the motor will decrease to 68% at an input power of approximately 150 Wac

Radial clearance

In the linear motor concept used a non constant gap between magnets and iron will lead to additional radial forces on the magnet piston assembly. These additional radial forces will be have to be balanced by the radial stiffness of the flexure bearing support of the same assembly. To calculate the height of these forces, some calculations with PC Opera have been performed. The radial force will increase exponentially when the outer stator is placed closer to the magnet. But because the displacements in this situation are very small, it can be considered as linear. So the linear force coefficient is estimated to be approximately 0.285 N/mm$^2$. In figure 5 a sketch of the situation is shown in which the magnet is $\varepsilon$ out of the exact centre.

Using geometry relations it can be deducted that $F_\varepsilon = \varepsilon \cdot c_f \cdot r \cdot \pi$. When the magnet is for example 0.2 mm out of the centre of the outer stator the radial force will be: $F_\varepsilon = 0.285 \cdot \pi \cdot 0.2 \cdot 34.5 = 6.2 N$. The radial displacement of the piston with the radial flexure-bearing stiffness of 1600 N/mm will be $f_{rad} = \frac{F_\varepsilon}{k_{ax}} = \frac{6.2}{1600} = 0.0039 \text{ mm} = 3.9 \mu\text{m}$

![Figure 5. Situation sketch](image)
This possible presence of the radial force has to be dealt with during the assembly procedure of the compressors. The possible radial force due to miss alignment needs to be considered in the choice of motor design and gaps used.

**FLEXURE DESIGN AND VERIFICATION**

THALES Cryogenics uses the FEM simulation package Algor® for the design of the flexure bearing suspensions used in their compressors. The main important criteria in the design of the flexures are that the maximum stress level is far below the fatigue limit stress, the flexure package has sufficient radial stiffness over the complete stroke of the piston, the flexure has a constant axial stiffness and the flexure own resonance frequencies are sufficient high.

All of the above criteria have been simulated and tested, an example of the stress forces at a certain extension are provided in Figure 6. Figure 7 represents the verification of the radial stiffness of a mounted flexure pack. The displacement in radial direction due to a force upto 3 N is measured at different axial displacements of the flexure and represented in Figure 7.

![Figure 6. Stress levels in a flexure](image)
COLD FINGER DEFINITION

The cold finger is used to generate the cold on the tip of the cold finger with the aid of a Stirling cycle. The cold is generated with a displacer with integrated regenerator which is moving (45 to 90 degrees) out of phase with the pressure fluctuation. With this out of phase movement of the displacer cold is produced in the tip of the cold finger.

Two kinds of coolers with two kind of cold fingers have been designed and tested. The LSF 9320 consists of a “standard” cold finger in which the support of the displacer is performed with the aid of contact seals (figure 8a). The LSF 9330 consists of a flexure bearing cold finger in which the support of the displacer is made with flexures as well as clearance seal (figure 8b) to prevent the flow of gas along the displacer. Due to the large gas volume present at the position of the flexures it has been decided to use a pneumatic driven displacer concept to generate the movement of the displacer.

Figure 8. Presentation of a standard and a flexure supported displacer
The movement of displacer depicted in figure 8a is generated by the pressure drop over the displacer (in phase with \( \frac{dp}{dt} \)) while the movement of the displacer depicted in figure 8b is due to the pressure drop over the displacer (in phase with \( \frac{dp}{dt} \)) and the pressure difference over the rod diameter (180 deg out of phase with \( p \)).

To find the optimal rod diameter of the displacer with flexure support calculations have been done simulating the complete dynamic behavior of the system. To find the optimal rod diameter, two things have to be considered: the optimum resonance frequency of the assembly flexures and displacer is different for each rod diameter and the damping (generated cooling power as a function of the piston compressor stroke) of each displacer with a different rod diameter is different, so the piston stroke of the compressor needed for a certain mechanical input power is also different. The latest criteria has an impact on the motor efficiency.

Experimental results of different rod diameters with the same compressor are depicted in figure 9.

![Figure 9. Results of generated cooling power at different rod diameters as a function of the resonance frequency of the displacer flexure bearing assembly at identical pdV work of the compressor](image)

With respect to efficiency considerations it can be stated that the flexure cold finger has a 10% higher efficiency compared to the standard cold finger

**MEASURED COOLING PERFORMANCES**

In figures 10 and 11 the generated cooling power of the LSF 9330 are depicted. The generated heat load is measured with a heater and temperature sensing diode connected on the top of the cold finger.
Development of a 6W high reliability cryogenic cooler at Thales Cryogenics

Figure 10. Measured cooling performance of a LSF 9330 (20 mm flexure cold finger with corresponding flexure bearing compressor) with a skin temperature of 35°C as a function of the electrical input power.

Figure 11. Measured cooling performance of a LSF 9330 (20 mm flexure cold finger with corresponding flexure bearing compressor) with a skin temperature of 35°C as a function of the cold finger tip temperature.
SPACE QUALIFICATION

The LSF 9330 is selected by Air Liquide DTA as the cooler for the cooling of the dewar of the ESA Cryosystem program. This program at THALES Cryogenics is funded by the ESA and has as aim the qualification of a modified version of the LSF9330 for the specific requirements of this program. The qualification program started in the beginning of 2002 and is in full process. Main focus within this program will be further optimization of the efficiency, low parasitic heat losses in cooler off mode and all other aspects required for the qualification for the Cryosystem program.

ENVIRONMENTAL SPECIFICATIONS AND LIFETIME TESTS

The coolers LSF 918x presented above have been tested in a very demanding qualification test program prior to the lifetime testing. The test specifications submitted to the coolers are; Repetitive Shock (operating): IEC 68-2-29 Test Eb Shock (operating) : IEC 68-2-27 Test Ea, Random vibration (operating): IEC 68-2-36 Test Fdb, Sinusoidal vibration (operating): IEC 68-2-6 Test Fc. The coolers LSF 9320 and LSF 9330 are in process to undergo the same environmental tests. Lifetime tests on the LSF 91xx are still in progress and an operational time of 18000 hrs has already passed successfully. The lifetime test of the LSF 9320 and LSF9330 have already passed 6000 hrs for both types.

FUTURE DEVELOPMENTS

Besides the development of the new flexure bearing cryocooler family, THALES Cryogenics has several other development programs running. One of these new developments is the development of a miniature pulse tube cryocooler driven by a flexure bearing compressor delivering 500mW at 80K including vibration reduction electronics and a high capacity pulse tube cryocooler with flexure bearing compressor capable of delivering 5W at 80K. This later program is part of the French SUPRACOM program and performed in close co-operation with CEA-SBT and is funded by the French ministry.

CONCLUSIONS

From the presented work, it may be concluded that after five years of work we have been able to develop a range of affordable and compact flexure bearing cryocoolers. These coolers are currently available for prices that are in the same order of magnitude as conventional tactical coolers currently available in the market. The LSF cryocooler family can be split in two families. The LSF 918x family capable of delivering up to 3 W of cooling power with a flexure bearing compressor which has a diameter of only 60 mm and a length of 165 mm. And the LSF 93xx family which is capable of delivering 6 W of cooling power at 80 K with a cooler diameter of 90 mm and a length of 204 mm.

Within the LSF 93xx family a study has been performed for standard and pneumatic driven displacer supported by flexure bearings. The flexure supported coldfinger has a larger mass (additional 1 kg) and requires more volume but has a 10% higher efficiency. Also the design is thought to be more robust due to the flexure support of the displacer.

New developments currently running at THALES Cryogenics offer opportunities for future markets requiring ultra-low vibration levels using a small scale pulse tube cooler and active vibration reduction electronics and markets who want to use a pulse tube cooler with a cooling capacity of 5 W @80K combined with a flexure bearing compressor.