The Development of a New Generation of Miniature Long-Life Linear Coolers

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ABSTRACT

Thales Cryogenics (TCBV) has an extensive background in developing and delivering long life cryogenic coolers for military, civil and space programs. This cooler range is based on two main compressor concepts: close tolerance contact seals (UP) and flexure bearing (LSF/LPT) compressors. With both concepts Thales Cryogenics has achieved excellent lifetime results.

New market developments require more compact linear cryocoolers with long lifetimes. In this paper the development of two new compact linear cryocoolers will be outlined. Both coolers are initially designed for a ¼" IDCA cold finger and operation between 77 K and 120 K.

The first cooler type, which is the UP8497, uses a long-life fully miniature close tolerance contact seal compressor with an expected lifetime of 15,000 hours minimum. The second type, which is the LSF9997, uses an extremely reliable compressor, which is based on the proven flexure bearing technology with an expected lifetime of well over 25,000 hours.

The differences in cooler concepts will be described, including design and application trade-offs. Furthermore, experimental results will be shown and discussed. Also, the expected lifetimes of both concepts and the roadmap towards even more improvements regarding the reliability of both concepts of Stirling coolers will be discussed.

Finally, the development of miniature drive electronics suitable for this new generation of compact coolers will be presented.

INTRODUCTION

Many of today's applications require cryogenic temperatures of 80 K – 120 K with heatloads of typically less than 400 mW. These applications are often limited in terms of volume and mass available for the cryocooler. Therefore rotary mono bloc Stirling coolers are often used as cryocoolers. The main reason for this is that rotary coolers offer a very high efficiency in a very limited volume and low mass. The Carnot efficiency of coolers is expressed as

$$\eta_{Carnot} = \frac{T_{\mathbf{low}}}{T_{\mathbf{High}} - T_{\mathbf{low}}}, \tag{1}$$

being the theoretically absolute maximum efficiency. Cooler efficiency is often expressed as a percentage of this maximum.

The relative efficiency of rotary coolers between 77 K and 296 K can be as high as 17% of Carnot, whereas higher power tactical linear coolers can reach 10%-15% of Carnot efficiency with lower efficiencies for linear coolers with a cooling power comparable with rotary coolers. However, disadvantages of high efficiency rotary coolers in comparison to linear Stirling coolers are the lower lifetime expectancy, higher acoustic noise, and higher induced vibration. Therefore a program was initiated at Thales Cryogenics to develop a miniature linear Stirling cooler that can offer an alternative for those applications where lifetime, vibration levels and acoustic noise are the main drivers, but still a compact lightweight cooler is required.

COMPRESSOR DEVELOPMENTS

The starting point for any linear compressor development is the targeted mechanical power the compressor should be able to deliver. The mechanical power is the output power (work) from the compressor available for the cooling cycle. In this case, a target nominal cooling power of around 400-500 mW at 80K had been defined. It was therefore chosen to set the compressor's nominal mechanical power at around 16 W. Secondly, it was decided to choose the commonly used dual piston split Stirling cooler approach because of the low induced vibration that is inherent to this concept. The split configuration offers high integration flexibility. The lowest amount of induced vibration was not only considered for optimum line of sight stability in for instance thermal imagers, but also for low acoustic noise and to avoid possible negative impact on lifetime of the complete system resulting from an unbalanced single piston compressor.

Both coolers have been designed to operate in full resonance, meaning that the motor force only has to overcome the internal damping and provide the PV work needed. This maximizes cooler efficiency by minimizing the phase shift between drive voltage and current.

Using the resonance conditions as a starting point, the first design parameters were established such as piston diameter, piston amplitude, axial stiffness, system damping, and required motor force.

Two different types of linear compressors were then developed: a close-tolerance contact-seal compressor (UP8497) and a full flexure-bearing-supported non-contact compressor (LSF9997). The outline dimensions of the resulting compressors are given in Fig. 1. Both compressors have initially been designed to match a ¼" IDCA cold finger. In Fig. 2, a photograph is shown of the different types of compressors. The top compressor at the back is the standard LSF95xx compressor with a diameter of 60 mm and a length of 122 mm. In the middle, the LSF99xx compressor is shown. The UP84xx compressor is shown at the bottom front. The masses of the coolers have been measured at 855 grams for the LSF9997 and 580 grams for the UP8497.

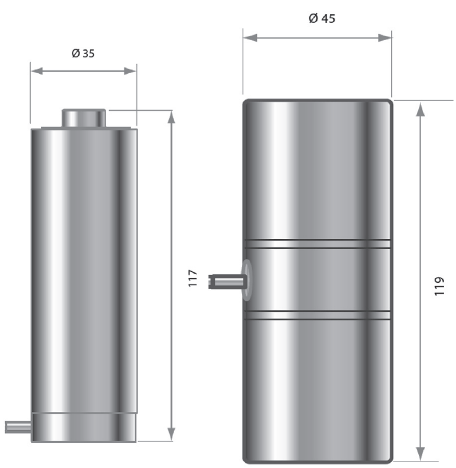

Figure 1. Outline dimensions of the UP8497 close tolerance contact seal compressor (left) and full flexure-bearing compressor (right).

Figure 2. Photograph of the LSF95xx, LSF99xx and UP84xx compressors including drive electronics.

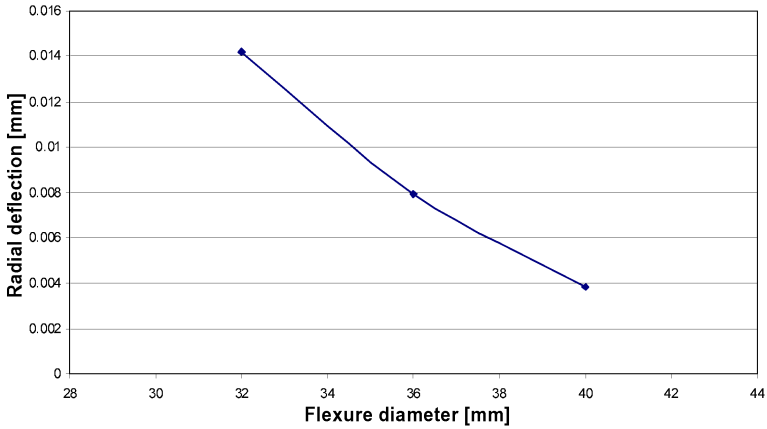

Figure 3. Flexure bearing diameter versus radial deflection simulations.

Development of the Miniature Flexure-Bearing Compressor

Before starting the design of the flexure bearing compressor, we investigated to what extent the flexure bearings could be downscaled. The goal was to minimize the compressor diameter while maintaining the required axial and radial stiffnesses of the flexure bearings. Simulations were performed to investigate the impact of the flexure diameter on the radial deflection under a certain radial load. The radial deflection was only allowed to be several microns to ensure a contact free piston in all compressor positions. The minimum flexure diameter determined was around 40 mm as is shown in Fig. 3. This value establishes the limit to which the diameter of such a compressor can be downscaled if correct operation of the flexure bearings is to be guaranteed.

Next, different motor concepts were evaluated taking into account their reliability aspects, dimensions, efficiency, and costs. For the LSF9997, a moving magnet motor concept was selected. Well-known advantages of this motor type are that the coils are positioned outside the helium gas, thus eliminating a potential outgassing source while also eliminating the need for flying leads and glass feedthroughs. The main disadvantage of this motor concept is that, with an imperfect axi-symmetrical magnetic field of the linear motor, a residual radial force will occur. However, in a correctly designed flexure bearing compressor this radial force can be absorbed by the flexure bearings, provided that a correct assembly and alignment method is used.

For the chosen flexure-bearing motor all the major losses were addressed and calculated. These included the Joule losses in the motor coils, the eddy current losses due to changing magnetic fields in conductive materials, the hysteresis losses, and the flow losses along the pistons.

The largest losses are the Joule losses, which for the entire compressor can be calculated with

$$P_{\mathbf{joule}} = 2I^2R \quad (2)$$

This represents the Joule losses of the two in-line linear motors where I is the current through one coil and R is the resistance of one coil.

The eddy current losses are a function of the magnetic field, the velocity of the magnets and the reciprocal value of the electrical resistivity of the static iron material ($\sigma = 1/r$ [$\Omega^{-1}m^{-1}$]). It is evident that the lower the electrical conductivity σ is, the lower the eddy current losses will be. The eddy currents have been simulated and where found to be quite low in the motor design chosen: well below 1 W.

The flow losses along the pistons can be determined with:

$$P_{Flow} = 2\frac{\pi D_{\mathbf{piston}} h^3}{12\eta L_s}(p_c - p_m)^2 \quad (3)$$

where h is the gap between the piston and cylinder, and L_s is the sealing length of the piston. It is obvious that the main parameter with the highest impact is the piston gap. Because the LSF9997 compressor has a piston gap of only several microns, the piston flow losses are negligible.

The overall theoretical efficiency of the flexure bearing compressor was calculated using:

$$\eta = \frac{P_{\mathbf{mech}}}{P_{\mathbf{mech}} + P_{\mathbf{joule}} + P_{\mathbf{eddy}} + P_{\mathbf{flow}} + P_{\mathbf{hys}}} \tag{4}$$

The theoretical efficiency was calculated at 70%.

Development of the Close Tolerance Contact Seal Compressor

As a starting point it was decided to use a moving coil concept as the baseline for the close tolerance contact seal compressor (UP8497). As indicated before, a moving magnet concept is not very suitable for a concept without flexure bearings because of the potential residual radial motor forces. For the UP8497 the advantage for the design was that the required minimum diameter for flexure bearings did not need to be taken into account. Without this diameter constraint, and using the required swept volume, the internal damping and subsequently the required motor force to overcome this damping were calculated. After that, with again taking full resonance operation as a boundary condition, the bandwidth for the required moving mass was calculated after which the actual design was made.

Also in this design, the motor losses were evaluated. In a moving coil concept the main motor losses are also the Joule losses. Eddy current losses only occur in the support of the coils, which move within the magnetic field. However, because of the low volume of this support and the careful selection of a material with a high electrical resistance, the eddy currents were simulated and proven to be very low and practically negligible. The same applies for the hysteresis losses, which had been measured to be nearly zero on other moving coil compressors at Thales Cryogenics. The calculated efficiency using Equation (4) for the UP8497 compressor was 75%, which is slightly higher than the moving magnet compressor.

COOLER PERFORMANCE AND EFFICIENCY

Measured Cooler Performance

After building and characterizing the first compressors, both systems were optimized in combination with a ¼" IDCA cold finger for maximum coefficient of performance (COP) and input power.

In order to increase the maximum input power of the cooler, several options are available such as: 1) To increase the internal system damping by changes in the cold finger design, 2) increase the compressor piston diameter, 3) increase the charge pressure and consequently operating frequency. The first two options can decrease the COP of the cooler and were already optimized during the design phase. Experimental work has led to a drive frequency of 60 Hz, leading to an optimum in both cooler efficiency and maximum input power. The measured cooling performance is given in Figs. 4 and 5.

From these figures several observations can be made. First of all, the difference in compressor

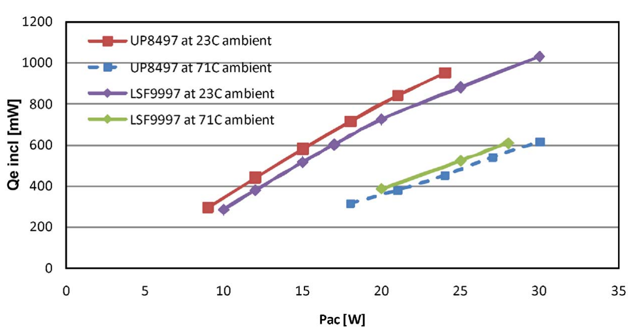

Figure 4. Measured cooling performance at 80K for 23°C and 71°C ambient temperatures.

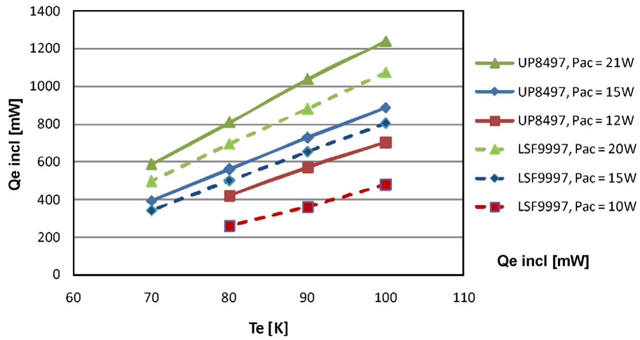

Figure 5. Cooling power versus cold end temperature for miniature coolers UP8497 and LSF997 at different input powers and 23°C ambient temperature.

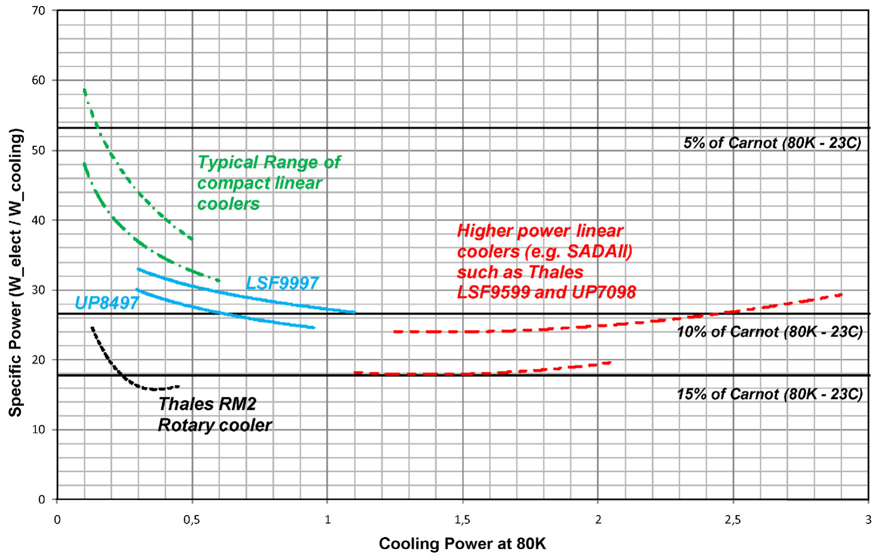

Figure 6. Comparison of cooler efficiencies.

efficiency is visible in the measured COP of the coolers. As both compressors are measured against a similar cold finger, the difference in cooling power is caused by the slight difference in efficiency.

Cooler Efficiency

The cooler efficiency can be directly calculated from the measured performance. In Fig. 6 the cooler efficiency is represented as a specific power (required input power in watts per watt of generated cooling power). A lower specific power means a higher efficiency. As a comparison, some average efficiencies of tactical coolers available today are also given. For reference, lines of constant relative Carnot efficiency are also shown. From this figure it can be seen that both miniature linear coolers offer a good COP that, in the higher power cooling range, can match the COP of a rotary cooler in the lower cooling power range. Moreover, it is demonstrated that these new coolers also match linear Stirling cooler efficiencies that previously were limited to the higher power Stirling coolers such as SADAII coolers.

This COP will result in steady state input powers of around 7-8 W_{AC} in a typical ¼" IDCA dewar of around 200 mW.

Table 1. Lifetime test results summary for TCBV cryocoolers.

Cooler type	Description cooler	Running hours	Status
Flexure Bearing coolers			
LSF9088	Moving coil flexure	69.620	Still running
LSF9188	10 mm Stirling cold finger	77.780	Still running. Worn out displacer replaced at 69.000 hours.
LSF9188		48.741	Stopped. Displacer worn out around 30.000 hours: restarted with new cold finger
LSF9188		30.150	Stopped, displacer wear
LSF9188		47.328	Stopped, displacer wear
LSF9180	5 mm Stirling	44.325	Stopped, displacer wear
LPT9110	500 mW Pulse	73.140	Still running
LSF9320	20 mm Stirling cold finger	60.310	Still running. New displacer coating material after 20.000 hours
LSF9320		54.907	Still running. New displacer coating material after 15.000 hours
LSF9330	20 mm Stirling cold finger with flexure bearings	56.830	Still running
LSF9330		58.020	Still running
LSF9330		43.758	Still running
LSF9330		43.972	Still running
LSF9597	¼” IDCA cold finger	1.256	Still running, lifetime systems with new expander coating
		1.256	
		8.537	Still running
		8.541	
		12.267	
Close tolerance contact seals			
UP 7080	Moving coil 5 mm Stirling cold finger	12.955	Stopped, worn out
UP 7080		40.322	Stopped, worn out
UP 7080		35.229	Stopped, worn out
UP 7080		32.738	Stopped, worn out
UP 7080		14.346	Stopped for research project, still well within spec.
UP 7080		28.353	Stopped, worn out
UP 7080		32.131	Stopped for research project, cooler was still within spec

LIFETIME PREDICTION MINIATURE COOLERS

To investigate the reliability of the two miniature linear coolers, several aspects need to be taken into consideration. First, is the lifetime test data acquired on similar coolers. The design similarities enable the comparison with the newly developed miniature Stirling coolers. The second consideration is the advances in coating materials used, both for the compressor pistons and Stirling expander. The final aspect is the difference in designs that can have an impact on lifetime.

Table 1 provides a summary of life data for several of the coolers life tested by TCBV. The first, upper part of the table shows data for flexure bearing systems, while the second, lower part gives data for several close tolerance contact seal coolers.

The LSF9997 has a high similarity with the LSF91xx, LSF93xx and LSF95xx cooler families. These flexure bearing compressors have demonstrated lifetimes well over 70,000 hours of continuous operation. From the flexure-bearing cooler data, it is concluded that the Stirling cold finger has become the limiting factor concerning lifetime, as the moving expanders, except for a few exceptions[3], normally rely on contact bearings. The main difference between the expander contact bearings and the contact bearings of a linear compressor is the bearing pressure. Typically the mass of a Stirling expander is a factor 10-20 less than that of the moving mass of a linear compressor, resulting in a much lower contact pressure and consequently less wear.

During the last few years Thales Cryogenics has been working on the implementation of improved bearing materials in both compressors and Stirling expanders. In addition to this new bearing material, improvements have been defined to reduce the wear rate of this new material even more. Pin on disk tribological tests have been performed to quantify the improvement in wear rate possible and differences in coating hardness. The wear rate k (in m^2/N) has been measured with the existing coating material including the scratch depth caused by the pin on the moving metal disk. After that, the improved coating material was measured under exactly the same conditions. It was found that the improved coating material has a 30% lower wear rate than the baseline material. Moreover the scratch depth caused by the coating material on the counter plate was found to be 20% deeper than the baseline test, clearly demonstrating an increased hardness of the material resulting in significantly lower wear.

In order to assess the potential for lifetime of the UP8497 cooler the piston wear can be reviewed in more detail. Piston wear of a contact compressor is determined by the relation

$$h = k_i p_n s \quad , \tag{5}$$

with h representing the coating wear, k_i the specific wear rate coefficient, p_n the normal contact pressure, and s being the sliding distance of the piston. The contact pressure is mainly caused by the gravity force acting on the moving mass and the piston surface area. In an attempt to translate the lifetime data of the existing UP7080 cooler to the new UP8497, both moving masses and contact surface areas can be compared. For this the pressure ratio and, consequently the coating wear ratio, can be determined. This results in

$$h_{\text{ratio}} = \frac{F_1}{A_1}\frac{F_2}{A_2} \approx 2 \quad . \tag{6}$$

This ratio indicates that with the newly developed miniature compressor in the expected worst case position, which is compressor mounted horizontally, wear is expected to be a factor of 2 lower than the existing UP7080 compressor. The lifetime and wear characteristics of the UP8497 will of course be finally verified with lifetime tests. These tests will be initiated in the middle of 2010, with the first test data becoming available at the end of 2010.

Using the lifetime test data of Table 1 and the intermediate cooling performance data measured on these coolers, the average performance evolution over time of the different types of coolers can be plotted as has been done in Fig. 7. From this figure a very gradual degradation of cooling power can be observed. For the flexure bearing coolers, this degradation is of course much more gradual because of the absence of wear in the compressor. For the UP series of coolers, the potential reliability improvement is plotted by using the improved Stirling expander coating material. This will not only lead to less degradation, but also to a longer lifetime and most likely less variance in

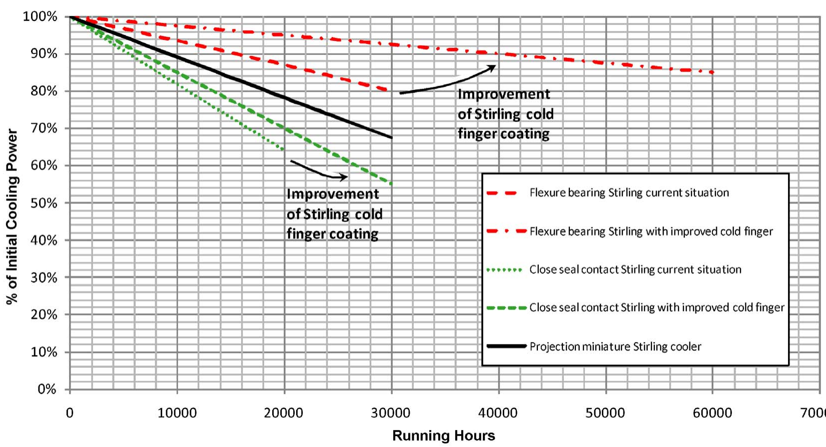

Figure 7. Average and projected cooler performance.

Figure 8. Integrated coolers in a Thales Claire MW IR camera dewar: left UP8497, right a RM4 rotary cooler.

wear behavior. The same applies to the Stirling flexure bearing systems. Here the projected lifetime is much longer because the failure chance for the compressor is much lower than that of a close tolerance contact seal compressor.

Finally, the projected curve for the miniature UP8497 cooler is given. This cooler will not only benefit from the coating improvements, but also from the lower coating pressure as outlined earlier. For the LSF9997 the same lifetime is expected as the existing flexure bearing Stirling coolers.

DEVELOPMENT STATUS AND FUTURE WORK

For both the UP8497 and LSF9997 types, the first coolers have been build and tested successfully. The final step of the development will be the environmental qualification. For this, both systems will be subjected to random vibration, sine sweep vibration, and shock testing.

A first pilot has already been started with the Claire Midwave infrared camera from Thales Optronics, the Netherlands.

The Claire is a multipurpose mid-wave infrared camera with a detection range up to 10.7 kilometers making it a suitable candidate for continuous observation applications. This camera is currently equipped with a Thales RM4 rotary cooler. In Fig. 8 the two different cooler types integrated in the Claire dewar are shown. As a first trial, the rotary cooler including the drive electronics has been replaced by a UP8497 and linear drive electronics. A photograph of this working demonstrator containing the small linear cooler is shown in Fig. 9. Such a camera upgrade enables its use in applications for surveillance where continuous use of the camera and consequently the cooler is required. Later in 2010 the possibility of integrating a LSF9997 into a Claire camera will be investigated.

Figure 9. Thales Claire midwave infrared camera equipped with a UP8497 cooler.

CONCLUSIONS

With the UP8497 and LSF9997 miniature coolers, a new range of high-efficiency long life coolers has been added to the product portfolio of Thales Cryogenics. With cooling powers of well over 800 mW at 80 K and efficiencies of around 10% of Carnot, these miniature coolers will enable the use of long life linear coolers in applications that could previously only be addressed by the use of rotary coolers. Especially those applications that require longer lifetimes and lower induced vibration, such as surveillance cameras, will benefit from these developments.

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