ABSTRACT
Since 2005 Thales Cryogenics has been producing coaxial pulse-tube coolers under CEA license for applications that are very sensitive for mechanical vibrations and require a long lifetime.

In order to optimize the existing baseline design of the coaxial pulse tube to its customers needs, Thales Cryogenics has been working on several of the critical elements inside the pulse tube. This optimization should lead to a wider application of these pulse-tube coolers into high-end civil applications.

This paper describes the work carried out on the optimization of the heat exchangers at the cold tip, the warm end and the buffer including irreversible heat losses caused by disruptions of the gas flow. Moreover, the heat exchange of warm end gas to the surroundings has been investigated. Also, the sensitivity to internal contamination has been tested.

Results will enable a design optimization of the whole range of coaxial pulse-tube coolers, varying from 1 and 4 W at 80 K to pulse-tube coolers of more than 12 W cooling power at 80 K. In this paper, test result, trade-offs and benefits of the new design will be discussed and evaluated.

INTRODUCTION
Thales Cryogenics BV has been designing and building pulse-tube refrigerators (PTRs). U-shape PTRs have been developed in-house; coaxial PTRs have been developed in cooperation with CEA/SBT (Commissariat à l’energie atomique, Service des basses températures). This has led to two production models of pulse-tube refrigerators, the LPT 9510 [1] and LPT 9310 [2], shown in FIGURE 1, with cooling powers of 1 W and 4 W at 80 K respectively.

Both pulse tubes use flexure-bearing compressors with long Mean Time To Failure (MTTF). The LPT 9510, the 1 W pulse tube, uses a flexure-bearing compressor of the second generation [3]. Pulse-tube refrigerators find their applications where low induced vibration to the application is extremely important or if large masses are connected directly to the tip of the cold finger. The absence of moving parts in the cold finger and the dual opposed piston compressor yields low vibrations and long lifetime. Typical users are high-end civil applications, such as electronic equipment and sensors for analysis and diagnostics.

The current versions of the coaxial pulse tubes are industrialized versions of the original prototypes developed by CEA. They use state-of-the-art components and manufacturing techniques. Typical manufacturing capabilities required for the production of the cooler are wire-EDM (electrical discharge machining), high-vacuum brazing, and electron-beam welding. To ensure wider application into high-end civil application, continuous improvement of performance and a consistent trade off between cost, performance, and specifications is required. Therefore, a research program was started at Thales Cryogenics to improve manufacturability while maintaining the high cooling power and efficiency of the current product, and at the same time create design possibilities for pulse-tube coolers with a higher overall performance. Several aspects addressed in the research program were the optimization of heat exchangers, improvement of heat sinking, reduction of material costs and optimization of manufacturability.

The result of the research program is a complete form-fit redesign of the LPT 9310 coaxial pulse-tube refrigerator with an improved efficiency that can be produced at a lower price, and a newly designed large pulse tube, with a cooling power of approximately 12 W at 80 K. The latter will be driven by a flexure
bearing compressor with 300 W of PV-power, using the well-known and proven Thales flexure technology.

**DESIGN OPTIMIZATION**

**Heat Exchangers**

Heat exchanger design focuses on optimal heat exchange with minimal pressure drop. Optimal heat exchange means a large surface area. In practice this means that large numbers of channels with small cross sections are required.

![FIGURE 1. LPT9510 (foreground) and LPT9310 (background) coaxial pulse-tube coolers.](image)

![FIGURE 2. Cooling power at 80 K as a function of electrical input power, with the reference heat exchanger and the simplified heat exchanger.](image)

Such channels are usually manufactured using the wire-EDM method, which is a time-consuming and thus expensive process. Furthermore, material with high thermal conductivity such as Oxygen Free High Conductivity (OFHC) copper is used for minimum temperature gradients in the heat exchanger body itself. When OFHC is used for the heat exchanger body itself, a stainless steel body is used for structural support. We have investigated the use of aluminum as a body material, thus improving the heat exchange to the surroundings.
The improvement of heat exchange with the surroundings allows us to change the design of other components, without a loss in performance. This means that simpler components and production methods can be used, thus improving manufacturability, yield, and cost.

The first component that has been investigated is the hot heat exchanger of the pulse tube. Currently this is a wire-EDM machined OFHC heat exchanger. We replaced it with a cylindrical heat exchanger. The results of this test are given in FIGURE 2. The performance of this system only slightly deteriorates at high input powers. The comparable performance for the new configuration is partly caused by the inertance tube. As this inertance tube is attached to the aluminum buffer, it also acts as a heat exchanger.

![FIGURE 2.](image)

**FIGURE 2.** Performance of the system at 80 K versus electrical input power for different hot heat exchanger configurations.

The second aspect that was investigated was the after cooler. This is the heat exchanger situated before the regenerator. This is currently a large slit-type heat exchanger constructed of OFHC. We have tested two new configurations, one with a simplified heat exchanger body in the stainless steel warm end and one with the same simplified geometry, diffusion-bonded in an aluminum body. The results are given in FIGURE 3.

![FIGURE 3.](image)

**FIGURE 3.** Cooling power at 80 K versus electrical input power for different after cooler configurations.

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![FIGURE 4.](image)

**FIGURE 4.** Cooling power at 80 K versus electrical input power for different cold heat exchanger configurations.
The simplified heat exchangers cause a cooling-power loss of approximately 0.5 W at 180 W input power. However, when an aluminum housing is used, the heat exchange with the surroundings is improved. As a result, the new configuration performs equal or better than the reference design.

A similar investigation has been done on the cold heat exchanger. In a coaxial pulse-tube refrigerator, it is particularly important as the flow reverses in this location. This means that special care has to be taken with respect to straightening of the flow into the pulse tube. The wire-EDM cold heat exchanger was replaced with a milled version, resulting in a wider slit width. Several configurations were tested with different void volumes \( V \) and surface areas \( A \). The results are shown in FIGURE 4.

Based on the results with several combinations of void volumes and surface areas, it cannot be concluded that any of those parameters alone is responsible for the reduction in performance despite additional flow straightening near the heat exchanger.

It can therefore only be concluded that the design of the cold heat exchanger is very important, and its function extends from heat exchange to flow conditioning as well, and that the loss in performance is caused by irreversible flow losses in the cold end. The configuration of this heat exchanger is still subject for further studies.

**Regenerator Performance**

The regenerator is the main source of losses in the pulse-tube cooler. Heat exchange, thermal conductivity, and pressure drop are the loss mechanisms that require optimization, but are conflicting with each other when it comes to geometrical requirements. De Waele et al. [4] have derived an expression for the specific entropy production rate per unit of volume of the regenerator. It reads

\[
\sigma_i = \frac{\beta (T_r - T_g)}{T_r T_g} + \frac{\kappa_g}{T_g} \left( \frac{\partial T_g}{\partial l} \right)^2 + \frac{\kappa_r}{T_r} \left( \frac{\partial T_r}{\partial l} \right)^2 + \frac{\eta z v^2}{T_g},
\]

with \( \beta \) the volumetric heat transfer coefficient, \( T_g \) and \( T_r \) the temperatures of the gas and matrix respectively, \( \kappa_g \) and \( \kappa_r \) the thermal conductivity of the gas and matrix respectively, \( l \) the axial coordinate, \( \eta \) the viscosity, \( z \) the specific flow impedance of the regenerator, and \( v \) the velocity of the gas in the regenerator. Equation (1) shows the four irreversible processes that lead to losses in the pulse tube: heat exchange, heat conduction through the gas, heat conduction through the matrix, and flow resistance.

Because the individual mechanisms depend on temperature, temperature gradient, and velocity, they are not distributed evenly over the regenerator. An estimation of the specific entropy production as a function of temperature is plotted in FIGURE 5. Since the temperature profile in the regenerator is approximately
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linear, this figure could also be seen as a representation of the entropy production rate as a function of position. It can clearly be seen that on the warm end the flow resistance is the dominant mechanism while imperfect heat exchange is dominant on the cold end. The velocity in the numerator of the flow resistance term and the temperature in the denominator of the heat exchange term are responsible for these effects. Heat conduction is negligible, but it should be noted that thermal conductivity through the regenerator wall, both axial and radial towards the pulse tube, are not taken into account.

These kinds of estimations are used to optimize the composition of the regenerator. The lower velocity on the cold end allows the use of finer gauze there, thus reducing the heat transfer losses. The high temperature on the warm end allows the use of coarser gauze, thus reducing the flow losses.

![Image](image.png)

**FIGURE 6.** Estimated cooling curve for a mixed-gauze regenerator compared to the reference measurement.

Mixed-gauze regenerators also allow optimization of the regenerator for particular temperature ranges. In FIGURE 6, the simulated performance of a mixed-gauze regenerator optimized for higher temperatures is given. By replacing part of the warm-end gauze by coarser gauze, the cooling power at higher tip temperatures increases. As a drawback, the minimum temperature also increases. Similar optimizations can also be done for optimizing cooling power at lower temperatures. A mixed regenerator is not introduced yet, as it is an application-specific modification, subject to further investigation.

Curing and Drying

Tests were performed for determining the influence of the curing and drying procedures. A Stirling and a pulse-tube cooler with similar contamination levels were compared. These tests revealed that the main contamination failure in a PTR is blocking of a flow straightener causing additional flow losses, whereas a Stirling expander is much more sensitive to low contamination levels that cause the expander motion to be compromised.

Comparison to High-End Pulse-Tube Refrigerators

The performance of the pulse tube with the above-mentioned modifications, designed for the civil and industrial market where cost price is an issue, can be compared to the high-end PTRs available on the market. These are usually space-qualified PTRs that are optimized with only minor cost limitations. In FIGURE 7 the specific power is plotted versus the cooling power. The dashed line represents the range of several currently available high-end (space) PTRs known from literature. The performance of our PTR is about 50% lower in terms of specific power or efficiency. This can be attributed to the following components:

*Compressor efficiency:* In the current design, a commercial flexure-bearing compressor is used. Without cost requirements, compressor efficiency would increase from 60% to 75%, thus resulting in a specific power decrease of approximately 10 W/W.
Materials and machining: In a cost-no-object design, low thermal conductivity materials such as titanium would result in an increase in cooling power of approximately 0.5 W. Furthermore, the simplified heat exchanger reduces the cooling power with another 0.5 W. Without these choices, the specific power will decrease to approximately 26 W/W.

![Specific power versus cooling power for the current design versus state-of-the art, high-end space PTRs.](image)

**FIGURE 7.** Specific power versus cooling power for the current design versus state-of-the art, high-end space PTRs.

It can clearly be seen that the compromises that are taken, especially with respect to the motor design, lead to a decreased performance. However, it can also be seen that without these compromises the current design can be brought relatively easy to the performance level of the high-end PTRs. The compromises were necessary introduce the LPT 9310 and LPT 9510 coolers in the industrial market at acceptable prices.

**PULSE TUBE REDESIGNS**

Based on the findings of this research, a redesign of the LPT 9310 is foreseen. The proposed new design is shown in **FIGURE 8**. The redesign is form-fit compatible with the current version of the LPT 9310. The most apparent difference is the material of the warm end and buffer. This is made of aluminum in the new design and contains the redesigned heat exchangers.

Most of these features are also taken into account in a new product of Thales Cryogenics BV, a 12 W pulse tube refrigerator. An outline drawing of the 12 W pulse tube, together with the proposed specification is shown in **FIGURE 9**. The complete pulse-tube cold finger is designed taking into consideration the results of the tests conducted on the LPT 9310.
A new compressor has been designed for this 12 W pulse tube. It delivers 300 W of mechanical power. A photograph of the new compressor is shown in FIGURE 10. An LSF-91xx compressor is shown for size reference. The compressor uses existing flexure bearing technology, and uses a moving-magnet linear motor. The compressor efficiency is expected to be above 80%.

CONCLUSIONS

An extensive research program is ongoing at Thales Cryogenics BV, aiming at the improvement of performance and manufacturability of the entire range of coaxial pulse-tube refrigerators. The improvement of performance allows the redesign of components to improve manufacturability, without overall loss in performance, and offers design options for high-end and high-performance pulse-tube coolers, when this is required for specific applications. Significant improvements have been found in a redesign of the heat exchangers. Regenerator optimization by using mixed-gauze configurations is ongoing. An optimization of performance in terms of COP and cooling power is expected and should be utilizable for different temperature and cooling-power ranges.

The findings have been applied to a redesign of the existing LPT 9310 coaxial 4 W pulse-tube refrigerator. Next to this, the insight gathered by the performed studies will be applied to a completely new product, a 12 W pulse tube with newly designed 300 W mechanical power, flexure bearing compressor.

REFERENCES

