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ABSTRACT

In order to provide cryogenic cooling for applications that are extremely sensitive to vibrations, a Pulse Tube Cooler and associated cooler drive electronics are developed at Thales Cryogenics.

Initially, the development focussed on the double inlet design because of its potential high efficiency. The DC flow arising in this design can decrease the performance significantly. Although this DC flow is successfully suppressed in prototype double inlet pulse-tubes, the solution proves to be too complex to be acceptable for large production quantities. It is therefore concluded that due to the DC flow, the double inlet design is not suitable for mass production, and the research further focussed on the development of an inertance type pulse tube.

Optimisation of the U-shape inertance-type pulse tube results in a very reproducible cooling system that is easy to produce in large quantities. The cooling performance of 500 mW at 80 K, for 60 W of electrical input, is comparable to that of a double-inlet system without DC flow.

Based on previous experience with the vibration reduction of Stirling coolers, a DSP-based cooler drive unit is designed that reduces the vibrations of the dual-opposed piston flexure bearing compressor.

The paper describes the results of a reduction method for DC flow, gives the design trade-offs for the inertance pulse-tube, and describes the vibration control algorithm, -hardware and results.

INTRODUCTION

The objective of the development is to design and build fit-for-manufacture cooling systems with extreme low vibrations, fitting in a pre-described dewar envelope and meeting various challenging demands required by the customer. The most important technical requirements are a cooling power of 300 mW @ 80 K at 45 °C skin temperature, at a maximum electrical input power of 60 W. Furthermore the level of cooler-induced vibrations on the system should be minimised by using vibration electronics.

PULSE TUBE DEVELOPMENT

Double inlet pulse tube

Because of the limited diameter of the dewar, the development has initially focussed on adaptation of an existing Thales double inlet pulse tube design, of which 5 prototypes were successfully tested to the specified performance in 1998 in a development program for the French MOD. The secondary orifices of these prototypes were manually optimized, in order to overcome the often observed non-reproducibility in a cooler batch caused by DC flow.

The present development has aimed at finding a solution to the DC-flow problem which would be suitable for mass-production.

Contrary to the often used assumption that DC flow is caused by asymmetric flow impedance of the secondary orifice, it has been found that DC flow takes place also when the secondary orifice is perfectly symmetrical. Referring to Figure 1, in the secondary orifice gas flows from the warm end of the regenerator (I) to the warm end of the pulse tube (II) during the high pressure part of the cycle. At this part of the cycle, the gas has a density $\rho_{high} \propto \dfrac{p_{high}}{RT_h}$. The gas flows back from (II) to (I) during the low pressure part of the cycle, when the density is $\rho_{low} \propto \dfrac{p_{low}}{RT_h}$. This implies that for a symmetrical orifice, where the net volume flow over one cycle is zero, there is a net mass flow and thus an enthalpy flow.

A solution has been found in creating an asymmetry in the secondary orifice, thereby creating a net volume flow in the direction of the warm end of the regenerator, i.e. from (II) to (I). Calculations have been performed with Thales' pulse tube simulation program[2] in order to find the values of the pressure waves in the spaces on either side of the secondary orifice, and it is found that an asymmetry of 9 % would be sufficient to neutralize the DC flow (see Table 1).

Table 1: Net volume- and massflow over 1 cycle

filling pressure 24 bar	**0 % asymmetry**	**9 % asymmetry**
pressure wave amplitude (I)	2.10	2.10
pressure wave amplitude (II)	1.19	1.19
phase difference (I) and (II)	18°	18°
net volume flow over 1 cycle	0	$2.7 \cdot 10^{-8}$ m^3
net mass flow over 1 cycle	$1.0 \cdot 10^{-7}$ kg	0

Experimental pulse tubes have been built in which the secondary orifice is realized as a replaceable plug which contains the actual orifice, and which is sealed with two small O-rings to prevent gas leak along the orifice (see Figure 1, detail B).

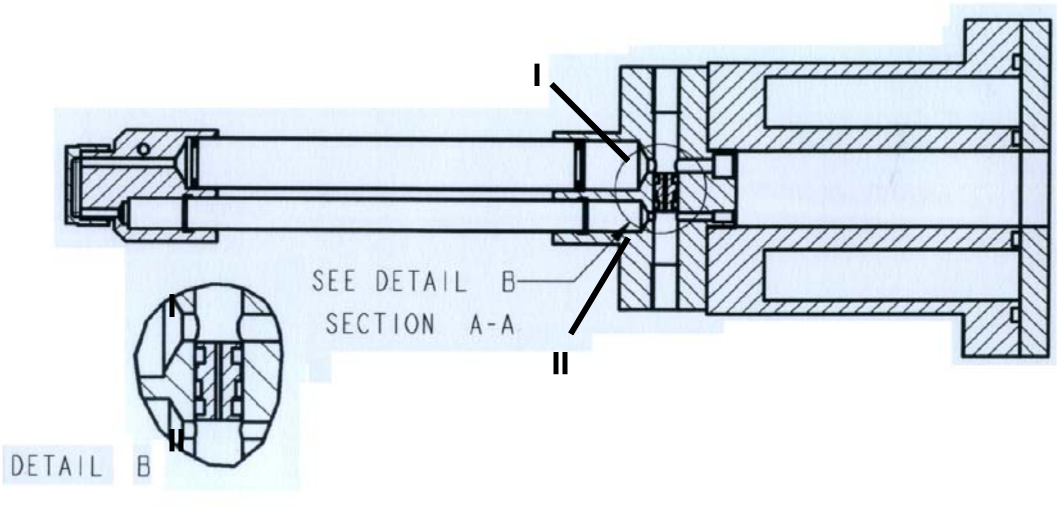

Figure 1: Double inlet pulse tube

The flow resistance of the orifice is tuned with the diameter of the hole (optimum ø0.4 mm), and it has been shown that reproducible asymmetries between 4 and 20% can be realized by chamfering the outlet of the orifice with a drill angle off 20 degrees[3], over a depth between 3 and 6 mm.

The reproducibility of the cooling performance with an asymmetrical secondary orifice has been tested using 2 identical prototypes. Even though all measurable properties of these prototypes are identical (including their flow resistance for steady flow), it turned out that there is a significant difference in the asymmetry required for their optimal performance (Table 2). It is therefore concluded that manual tuning of each pulse tube would be required in production, making the double inlet concept unsuitable for mass production.

Table 2 : Performance for two identical double inlet pulse-tubes

performance @ 25 W_{pdV}	**PT 1**	**PT 2**
symmetrical orifice	398mW @ 140K	425mW @ 140K
5% asymmetry	0mW @ 110K	110mW @ 80K
10% asymmetry	unstable	0mW @ 95K

Inertance type pulse tube

After dismissing the double inlet pulse tube, the development has focussed on optimizing a pulse tube that uses an inertance to provide the necessary phase shift. The limited diameter of the dewar reduces the degrees of freedom in optimizing regenerator and pulse tube diameters. The space available for the tubes is further decreased by the space needed for the vacuum-brazed connection between tubes and warm- and cold end (see Figure 1).

An optimization run performed with Thales' simulation program has revealed that the optimum configuration within these constraints is a regenerator diameter of ø9 mm and a pulse tube of ø5.5 mm. In other words, the regenerator diameter is maximized, while maintaining enough space to accommodate a pulse tube with a volume that is approximately one third of the total volume of the regenerator.

For the first prototypes, it was intended to use stainless steel regenerator gauzes that have the same properties as the gauzes used in the regenerators of Thales' free displacer stirling coolers. Calculations performed for the geometry of these prototypes have revealed that the heat capacity of these regenerators is insufficient for efficient operation of the pulse tube at 80 K. The calculations indicate that the optimum filling factor of a pulse tube regenerator is higher than the optimum filling factor of a stirling regenerator. It was also found that a higher filling factor in the entire pulse tube regenerator would significantly increase the pressure drop over the regenerator, thereby reducing the performance of the pulse tube cooler. An optimization run where the filling factor and wire diameter were varied along the regenerator has indicated that it is optimal to fill the coldest part of the regenerator with gauzes with a high filling factor, and the warm part with gauzes with a lower filling factor. Combining the simulation results with practically available gauzes, an optimum is found with a high filling factor in the cold end, and a low filling factor in the warmer parts (Table 3). Measurements with three differently stacked regenerators have been performed to confirm the predictions of the simulation model.

Table 3: Pulse-tube performance for different regenerator fillings

performance @ 25 W_{pdV} *warm part / cold part*	**calculated regenerator loss**	**calculated pressure drop over regenerator**	**measured pressure drop over regenerator**	**measured lowest tip temperature**
LOW / MED filling factor	2472 mW	0.85 bar	0.60 bar	80 K
MED / MED filling factor	2114 mW	1.10 bar	0.95 bar	75 K
LOW / HIGH filling factor	1896 mW	0.97 bar	0.82 bar	72 K

Finally, an inertance optimization has been performed to match the optimum frequency of the pulse-tube with the resonance constraints of the flexure bearing compressor, and secondly to optimize the phase shift in the optimal pulse-tube.

The resonance of a linear compressor depends on the filling pressure, cold finger volume and cold finger damping. It is very convenient to have a mechanism of fine tuning the optimum frequency of the cold finger, as this makes it possible to drive a large range of cold fingers with a limited number of compressor designs[4]. For a given regenerator and pulse tube volume, the impedance of the inertance determines the optimum frequency of the pulse-tube[5]. It is found that the optimum working frequency of the pulse tube can be shifted with approximately 5 Hz without a significant influence on the performance of the pulse-tube.

The impedance of the inertance consists of a resistive part R and an inductive part L. As the inductive part is responsible for the beneficial phase change over the inertance[6,5], one would expect that it is always beneficial to increase the ratio L/R for the inertance, as this would increase the inertia effect for a given flow resistance. However, an increase in L for the same R means that the overall impedance Z of the inertance increases, thereby decreasing the optimum frequency of the pulse-tube. Furthermore, as L is proportional to 1/A and R is proportional to $1/A^2$, the ratio is proportional to the surface area A of the inertance. For a large ratio a relatively large surface area A is therefore required, in which case the length Δx should be large to obtain the proper flow resistance. The resulting volume of the inertance causes a storage effect (or capacity C), which gives a phase lead that counterbalances the desired phase lag. Figure 2 shows the measured phase difference between the pressure wave in the pulse tube, and the (small) pressure wave in the buffer. The figure reveals that as the inertia effect in the inertance increases (larger diameter and / or larger length), the phase lead of the pressure in the tube increases, which is the desired effect of the inertance[5].

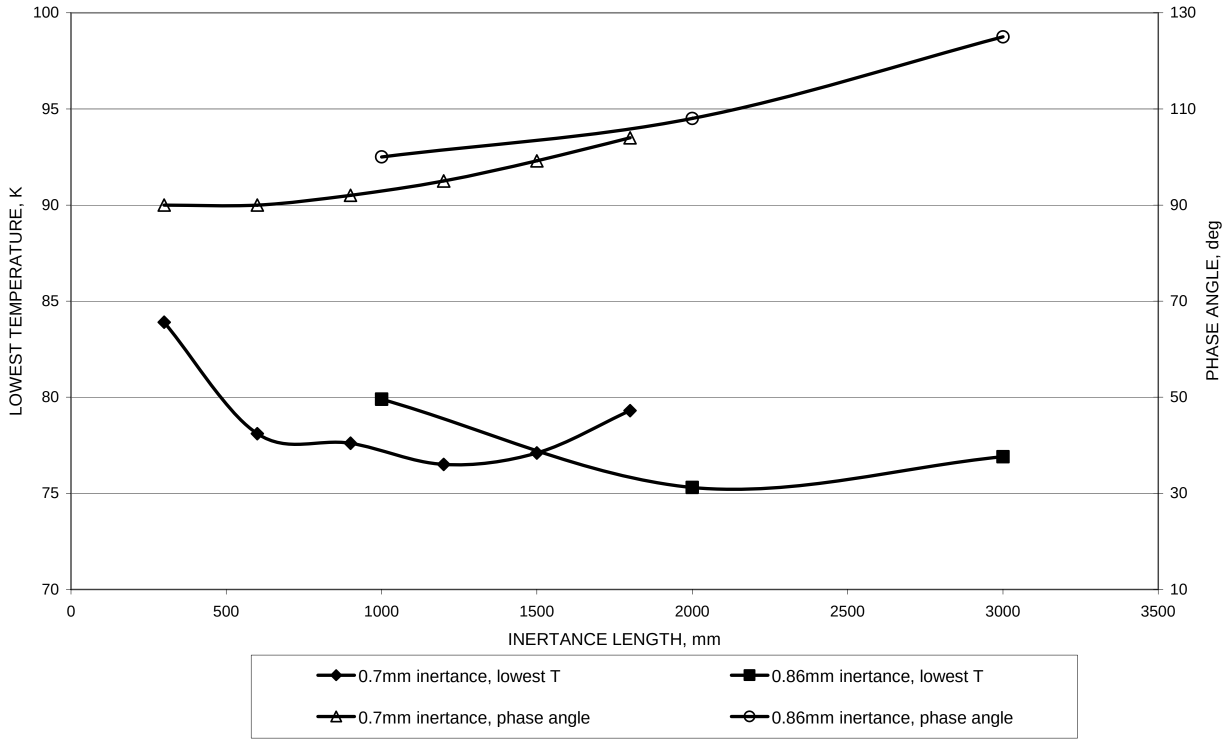

Figure 2 : Phase change over the inertance for different inertance properties, and associated pulse tube performance (MED/MED regenerator filling)

VIBRATION REDUCTION

Compressor vibration

The dual opposed piston compressor has an axial vibration level that is specified at 2.8 N_{rms}, which is acceptable for most applications. The axial vibrations originate from an unbalance in the movement of the two opposing pistons. Earlier work[7] has indicated that it is possible to reduce the vibrations by a factor of 50 by adapting the current through the coil of one piston.

In collaboration with CILTEC, the Centre for Interfacing Low Temperature Electronics and Coolers of the University of Twente, it was found that with present day electronics, a digital feed-forward system is an affordable, reproducible and reliable means of reducing the vibrations of the compressor. A DSP-based vibration control system has been realized in commercially available hardware.

The heart of the vibration reduction system consists of a DSP, an accelerometer and two analogue amplifiers. During startup of the cooler, the DSP measures the transfer functions of the system via a step response. In steady state operation, the transfer functions are used in a feed-forward loop to minimize the vibrations by adapting the phase and amplitude of the current through one of the compressor coils. Meanwhile, a software programmed PID loop maintains a constant cold tip temperature by changing the input power to the compressor. The DSP is capable of minimizing the vibration of the compressor at the drive frequency and the first two harmonics see Figure 4.

The vibrations of the compressor are measured by means of a commercial-of-the-shelf accelerometer. In order to get a large acceleration signal for small force levels, the moving mass should be kept to a minimum. For this reason, the acceleration transducer is attached directly to the compressor, and the compressor is suspended flexibly in order to be able to measure the acceleration. This implies that the moving mass is only the mass of the compressor, and not that of the entire application. Figure 4 shows the results of the vibration reduction.

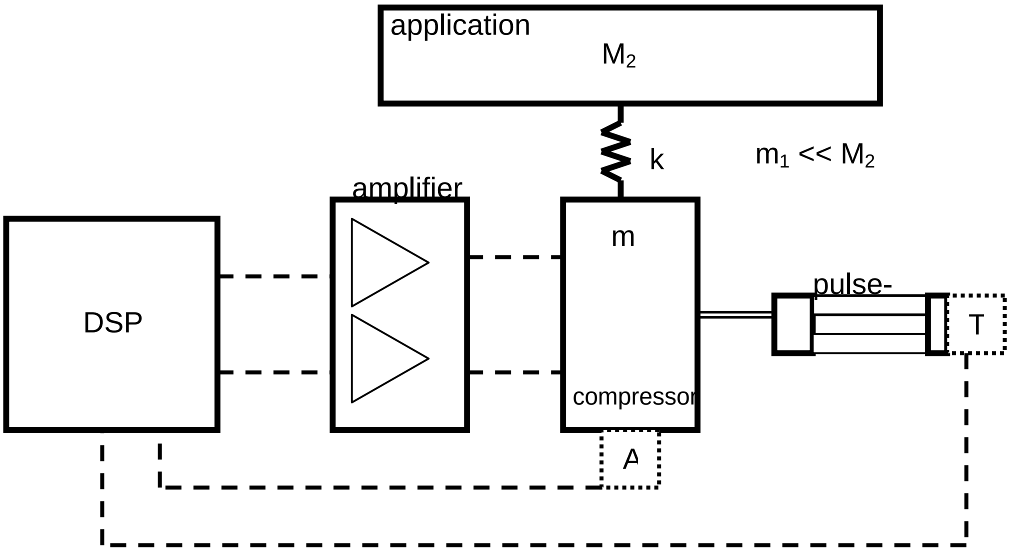

Figure 3: Block diagram and mechanical implementation of vibration- and temperature feedback loop

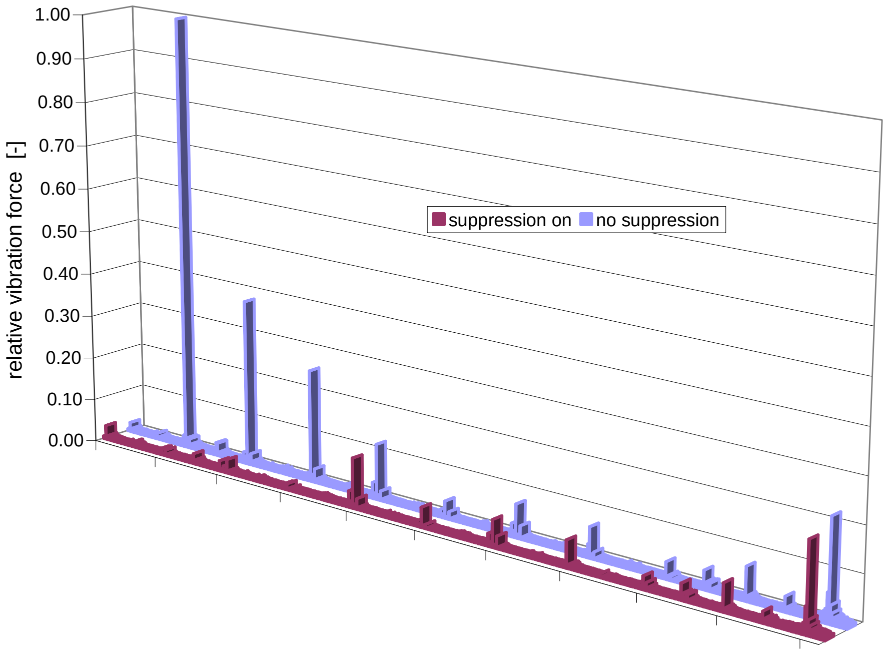

Figure 4 : Vibration reduction results

Pulse-tube vibration

The absence of moving parts in the cold finger generally makes a pulse-tube cooler very suitable for vibration-sensitive applications. The present research has revealed, however, that a pulse-tube cold finger is not completely vibration free. After de-coupling the compressor vibration there is still some small movement of the pulse-tube cold finger. Significant reduction in this level of movement has been achieved by studying the pulse-tube design and improving it with respect to vibration characteristics.

LIFETIME

The design of Thales' moving magnet flexure bearing compressors[4] removes all failure mechanisms commonly identified[8] in pulse-tube cryocoolers. Compressor wear is eliminated by the flexure bearings which fully support the piston mass at the front and back side with high radial stiffness. Because of the moving magnet concept, moving current leads and helium-tight current feedthroughs are not needed. The moving magnet concept also moves the synthetic material of the coil insulation outside the working gas, thereby strongly reducing the risk of gas contamination. All parts that are in contact with the working gas are metallic parts, joined together with laser- and electron beam welding techniques.

The lifetime expectancy is supported by lifetime tests that are being performed on 10 coolers with the described compressor design. The lifetime coolers have presently gathered a total of 118.000 running hours, with no failures and no performance degradation. The lifetime tests include tests where the cooler is running under high side loads (up to 10g), temperature cycles between the specified extremes (-54°C to 71°C), and cooler on / cooler off cycles.

CONCLUSION

Based on an inertance type pulse tube, a U-shape pulse-tube has been designed with a cooling performance of 500 mW at 80 K. The pulse tube is driven by a long-life moving magnet flexure bearing compressor, giving a virtually failure-mode free cooling system.

A separate DSP-based vibration control system reduces the vibrations originating from unbalanced motion of the compressor pistons with a sufficient reduction factor. Detailed design work on the pulse tube was needed to reduce the level of vibration induced by the pulse tube cold finger.

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