# Performance Improvement of the 15K Pulse Tube Cooler

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## ABSTRACT

A 15 K pulse tube cooler engineering model has been developed by Air Liquide, Thales Cryogenics BV and CEA through an ESA program. It has now reached TRL6 and provides more than 400 mW of cooling power at 15 K with more than 3 W at 90 K on the first stage at Air Liquide. It has been integrated as part of a 300 K-50 mK cryochain demonstrator for ATHENA/X-IFU and is also baseline on the ATHENA/X-IFU dewar where the cryogenic constraints are important. An improvement of the 15 K pulse tube performance would drastically improve the robustness of the cryochain and possibly reduces its complexity.

A new ESA program, awarded to Air Liquide jointly with CEA-SBT and Thales Cryogenics is focusing on increasing the maturity of this 15 K pulse tube cooler and an additional objective is increasing the cooling power of the cold finger. This paper will focus on the work done at CEA-SBT for this latter objective on the 15 K cold finger.

## INTRODUCTION

The need for very low temperatures (50-100 mK) in future astrophysical space missions are becoming predominant for some of the most sensitive astrophysics missions. The use of a liquid cryogen like on Herschel is not favored due the long lifetime required for the new mission (typically >3 years). To reach such a low temperature level, a cryogenic chain architecture shall be used including a sub-K cooler, one or several Joule-Thompson coolers and pre-coolers. In order to have the best helium Joule-Thompson coolers operating parameters, precooling is needed around 15 K or lower. To reach such a low temperature with Stirling or Pulse Tube coolers, a multi-stage configuration is needed. A significant effort is being made on the development of those pre-coolers over the several years. Hence, a 2 stage Stirling cooler, made by SHI, have already flown successfully onboard Astro-H [1]. Lockheed-Martin [2] and NGST [3][4] have both developed 3-stage pulse tubes working at 10K. On the future JWST-MIRI instrument, a 3-stage 18 K pulse tube is being integrated on the spacecraft [5].

In CEA-SBT, we are developing low temperature pulse tube for more than 10 years under CNES and ESA contracts as well as self-funding. Our work is focused on the development of novel regenerator materials for low temperature [6 -8] and on the architecture of multi-stage pulse tube coolers [9][10]. Based on CEA-SBT results and on common developments, Air Liquide Advanced Technologies (ALAT),

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jointly with us and Thales Cryogenics B.V., developed an engineering model of a heat intercepted 15 K pulse tube cooler. This cooler is now baseline on ATHENA/X-IFU instrument cooling chain [11] and could be used to cool thermal shields, harnesses and to precool a 2K and 4K JT coolers from JAXA. The architecture of the cooler is a single 15K pulse tube cooler precooled at 80 -120 K by a Large Pulse Tube Cooler (LPTC) from Air Liquide [12]. A new EM 15K pulse tube with a Demonstration model (DM) cooler drive electronics is being developed through an ESA contract and ALAT leadership jointly with SITAEL, Thales Cryogenics B.V. and CEA-SBT.

This paper presents the work performed on the first EM 15K pulse tube cooler and on the optimization we made to improve its performances. Then a focus is made on the last test campaign on a cold regenerator to consolidate and possibly improve the performance of the future cooler.

## STATUS ON THE 15 K PT

The 15K pulse tube cooler consists of a single compressor that drives through a single transfer line 2 pulse tube cold fingers and a 15K cold finger precooled by a LPTC cold finger (Figure 1). Part of the heat of the 15 K cold finger is removed by the LPTC at the intercept which separates the hot regenerator and the cold regenerator. The 1st-stage of the pulse tube (LPTC) has an inertance/buffer dephasing system while the 15 K cold finger has an active phase shift setup to actively control the phase between pressure and mass flow.

The cooler was first fabricated and tested at Air Liquide where environmental testing (mechanical and thermal) has been performed without any change to the overall performances [12]. The target performance of the 15K pulse tube cooler was to generate the maximum cooling power at 15 K, for the precooling of a JT cooler. The measured performances were more than 400 mW at 15 K with an electrical input power of 300 W [12]. Due to this dual stage configuration, an extra cooling power of 2 W at 80 K is available on the intermediate stage. The lowest achieved temperature was about 9 K with the intermediate temperature stage around 65 K. Our measurements showed that for low temperature pulse tube, the lower the operating frequency, the better the performances. However, it is difficult to build compact low weight space compressor based on flexure bearding technology with large swept volume, operating at low frequencies. Hence, the operating frequency of the system has been set to 41 Hz as a compromise between compressor and cold finger efficiency.

At Air Liquide and at CEA/SBT, we have shown that the impact of the 1st stage temperature on the performances of the 2nd stage is significant. An increase from 65 K to 100 K on the first stage leads to a 100 mW loss at 15 K.

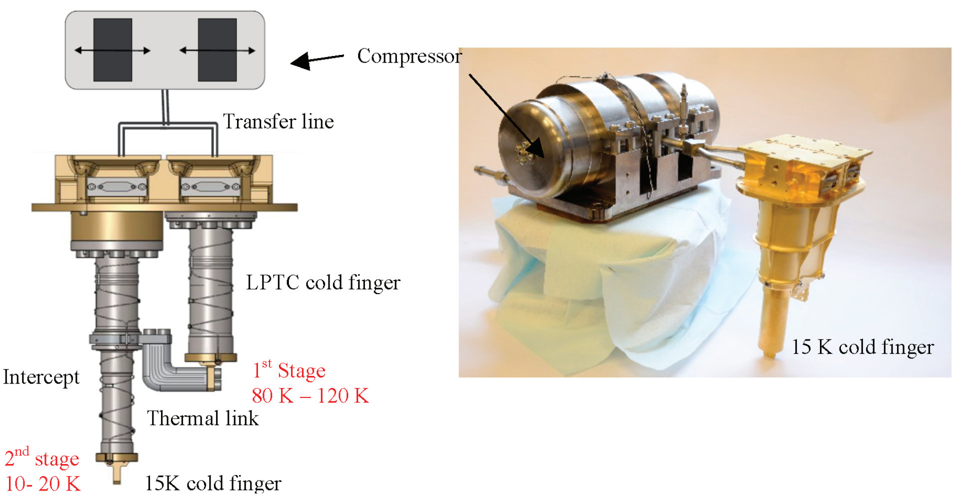


**Figure 1.** Schematic and picture of the 15 K pulse tube cooler.

## FURTHER WORK ON THE EM 15 K PT

### Improved Cryogenic Performance

After the test campaign at ALAT, the 15 K pulse tube cooler has been transferred to CEA-SBT in order to be integrated with a Joule Thompson cooler as part of an overall space cryogenic chain [13]. An extended test campaign has been carried out in CEA-SBT in order to further optimize the performances of the 15 K pulse tube cooler:

- The $1^{st}$ stage inertance has been modified to increase the efficiency of the $1^{st}$ stage. As a result, the optimum frequency of the $1^{st}$ stage has been increased up to 43 Hz with a limited effect on the $2^{nd}$ stage (not shown here).
- For future integration into a demonstration model of the ATHENA/X-IFU instrument, the effect of a small inclination (about 10°) compared to vertical did not show a significant effect on the performances (not shown here).

The 15 K pulse tube cooler has been transferred into a dedicated cryostat (Figure 2a) for last test before coupling with the 1K-class Joule Thompson cooler from JAXA. During the integration, the MLI around the $1^{st}$ stage and the $2^{nd}$ stage have been modified and the wiring of the thermal sensors too. Those two modifications increased the $2^{nd}$ stage performances with a gain of 100 mW at 15 K. The effect on the $1^{st}$ stage is limited.

Finally, the optimization process gives a gain of 5.6 K on the $1^{st}$ stage, a gain of 2.4 K on the $2^{nd}$ stage and an optimum operating frequency 2 Hz higher, so a better compressor efficiency (Figure 2b).

### Integration with the 2 K Joule Thompson Cooler

A demonstration model of a cryogenic chain made of space cryocoolers has been designed and made jointly with CEA-SBT and JAXA [13]. For this purpose, a new 2 K Joule Thompson cooler has been assembled where the pre-cooler is now the 15 K pulse tube cooler. Figure 3 shows the final integration where the counter flow heat exchanger of the 2 K JT cooler is wounded around the 15 K cold finger.

Extra cooling power was available than the 2 K JT required, because the pre-cooler was operated at a nominal electrical input power of 300 W, of more than 400 mW at 15 K on the $2^{nd}$ stage and more than 3 W at 90 K on the $1^{st}$ stage has been measured while the 2 K JT cooler was running at nominal conditions (10 mW at 1.7 K). The coupling was successful with a fully operational 2 K JT cooler. The results of the JT cooler will not be discussed in this paper.

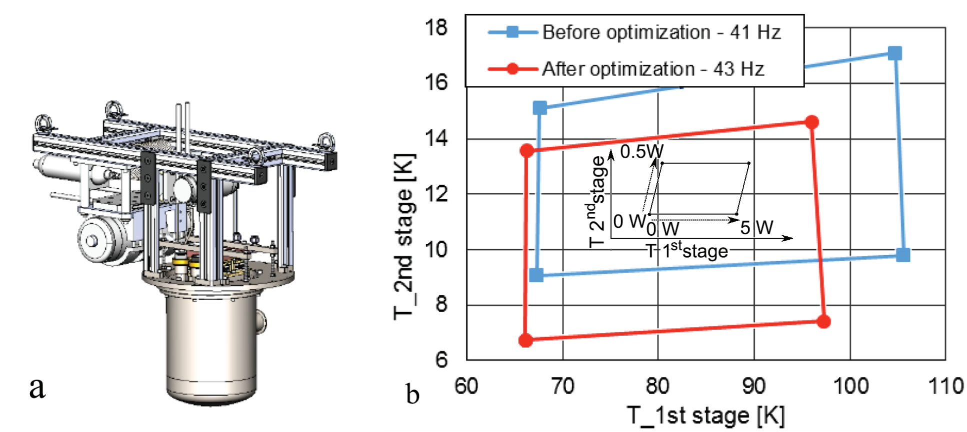


**Figure 2.** a. Schematic of the test cryostat b. Cryogenic performances (300 W electrical power) before optimization (41 Hz) and after optimization (43 Hz).

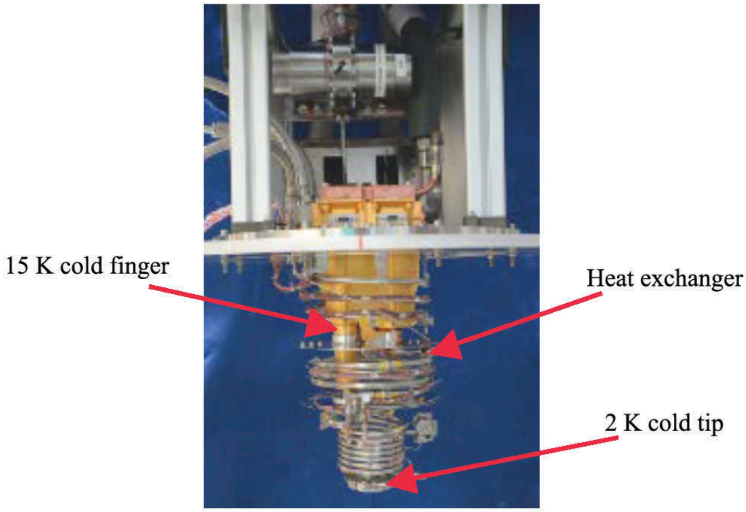


**Figure 3.** Picture of the 15 K pulse tube cooler integrated into the 2 K JT cooler as a pre-cooler.

### Impact of Transfer Line Geometry

A dedicated study on the impact of the transfer line length has been carried out. The possibility to use a longer transfer line would give more freedom for integration purposes in ATHENA/X-IFU instrument. The initial transfer line has a dead volume of 13 cc. The transfer line is a tube of 54 mm long with an inner diameter of 10 mm that then splits into 2 tubes 155 mm long with an inner diameter of 6 mm. Two added lengths on the transfer line have been studied, one 156 mm longer (added dead volume of 12.3 cc) and another 456 mm longer (added dead volume of 35.8 cc) both with an inner diameter of 10 mm. The study was done at a constant electrical input power of 250 W at compressor level and a constant frequency of 43 Hz. The effect of increasing volume of the transfer line has a very limited impact on the 2$^{nd}$ stage and is favorable for the 1$^{st}$ stage with a gain of 290 mW at 80 K (Figure 4a). This extra cooling power is explained by a better matching of the compressor with the cold finger. In practice, increasing the transfer line volume increases the available PV power at constant electrical input power (Figure 4b). Here the PV power is estimated by $P_{elec} - I^2R$, where R **is t**he resistance of compressor motors.

The main drawback is that the piston stroke of the compressor is significantly increased and goes above the nominal stroke given by the manufacturer Thales Cryogenics B.V. Between the initial length and the final length, there is a 33 % increase of the piston stroke.

In order to further lengthen the transfer line at constant volume, the diameter can be decreased too. The pressure losses in the transfer line will then increase and so decrease the final PV power in the cold finger. From previous tests, we measured the pressure swing which was of about 5 bar

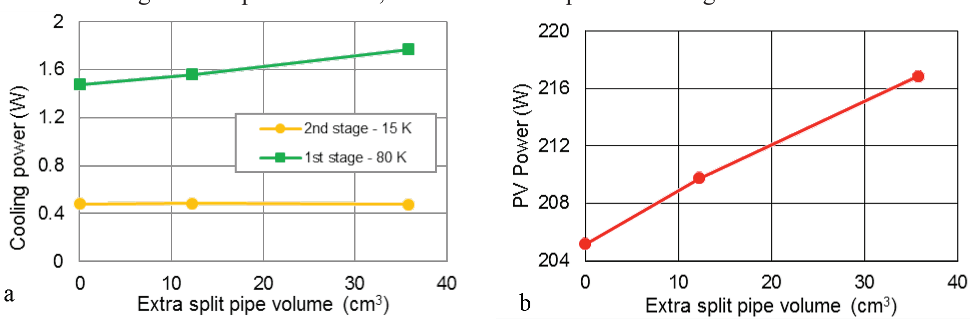


**Figure 4.** Effect of the transfer line on the cooling power at 1$^{st}$ and 2$^{nd}$ stage of the 15 K pulse tube (**a**) and on the PV power at 250 W electrical input power (**b**).

(slightly dependent of the operating condition like the active phase shifter setting) for a PV power of about 250 W (at 300 W electrical input power. So, a maximum decrease of PV power of 1% leads to a maximum acceptable pressure losses of 50 mbar. Pressure loss calculations are based on Darcy-Weisbach law. The flow rate is calculated, from the stroke measurements on the compressor at a 300 W electrical input power, to be about 1.3 g/s rms. Keeping the extra dead volume of 12.3 cc constant, a tube with an inner diameter of 4.6 mm and a length of 950 mm will generate a pressure drop of 50 mbar. Experimental validations are necessary to conclude on the topic.

To summarize, the transfer line can be increased significantly compared to current design, either at constant volume with a slight increase in pressure loss or with an impact on the swept volume.

## OPTIMIZATION OF THE COLD REGENERATOR

At CEA-SBT, a test campaign has been performed on the cold regenerator (below the intercept), to find the best ratio between standard regenerator material (stainless steel) and exotic material in order to improve the cooling capacity at 15 K. We focused the study only on the 15 K cold finger without dependency on the LPTC pre-cooler. Two different components have been tested. They will be named A and B. Component A was the one used in the previous 15 K cold finger.

### Test Philosophy

In order to find the optimum ratio, a systematic study has been conducted on the two materials. Five configurations have been built and tested, one is all of stainless steel, the two other ones are filled with half stainless steel and half of component A or B and the two last ones are filled with 100 % of component A or B (Figure 5a). According to the operating parameters and the kind of component, we should find an optimum ratio that will give the maximum cooling power at 15 K for each component (Figure 5b).

During the study, a large number of parameters have been tested in order to map the performance of the 15 K cold finger with different operation/boundary conditions that can be used to

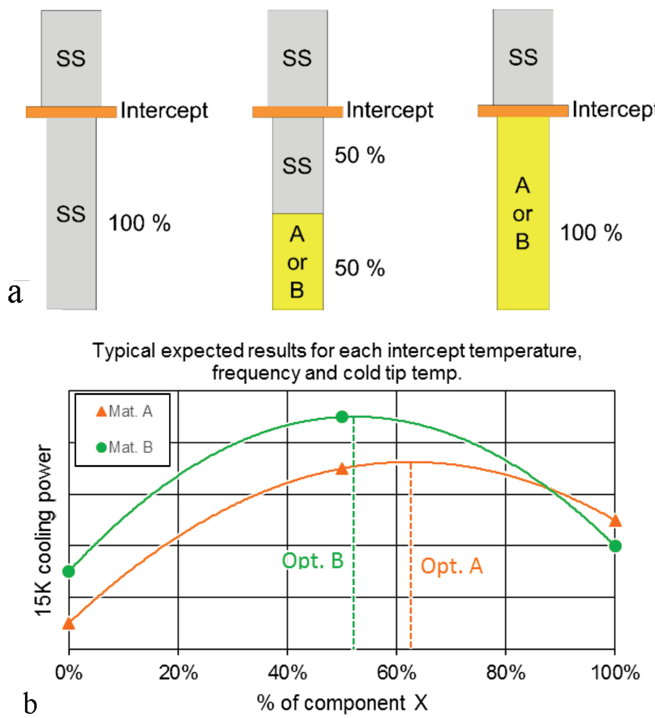


**Figure 5.** a. Schematic of the tested configuration b. Typical expected results for the two different components.

further optimized the cold finger for future applications with different requirements. The parameters that have been studied are the following:

- Operating frequency (36 – 46 Hz)
- Temperature of the intercept (50 – 120 K)
- Temperature of the cold tip ($T_{min}$ / 15 K / $T_{@1W}$)

We also measured the pressure losses due to each component. This was performed at ambient temperature under continuous flow instead of alternating flow as in a pulse tube.

### Test Setup

Figure 6 shows the test setup used during the optimization of the 15 K cold finger. We are not limited by the performance of the LPTC and we can vary the intercept temperature over a wide range with a Gifford Mc-Mahon cryocooler that can reach 20 K. The modification of the cold regenerator is also easily feasible thanks to a dismountable cold regenerator. A thermal shield is mounted around the cold tip in order to avoid radiative losses due to the hot surface (300 K) of the cryostat.

The warm flange interface was maintained to 15°C thanks to water cooling. In the previous 15 K pulse tube cooler, the PV power was estimated to be around 110 W, here we chose a PV power of 100 W. The compressor was a lab compressor (9710 from TCBV). So, we can expect that the cryogenic performance presented here results are slightly less than estimated compared to what can be achieved in a future 15 K EM cold finger that would implement the regenerator changes. The PV power is close enough that the trends measured can be extrapolated to the full 15 K PT cold finger.

### Discussion on the 2 Different Components

For each component, we measured the pressures losses of the pulse tube cold finger filled with regenerator material. The pressure drop also includes the heat exchangers, hot regenerator, cold regenerator, and pulse tube. Figure 7 shows that the component A has a pressure drop lower than component B. So, the hydraulic losses in component A are smaller than in component B.

However, the estimation of the specific heat of component B is about 20 % higher than component A. So, the thermal performances of component B should be higher.

The porosity of the 2 components are similar (Table 1). Finally, the performances of the two components will be dependent on the compromises between geometries and specific heat effect.

### Cryogenic Performance

For all the configurations, the active phase shift is optimized for each point to obtain the best performances. Figure 8 presents the cooling power curves obtain for the 100 % ratio of stainless steel and component A and B. The results are similar with 50 % SS – 50 % of component A or B. As expected, there is a strong gain to use exotic material instead of stainless steel only.

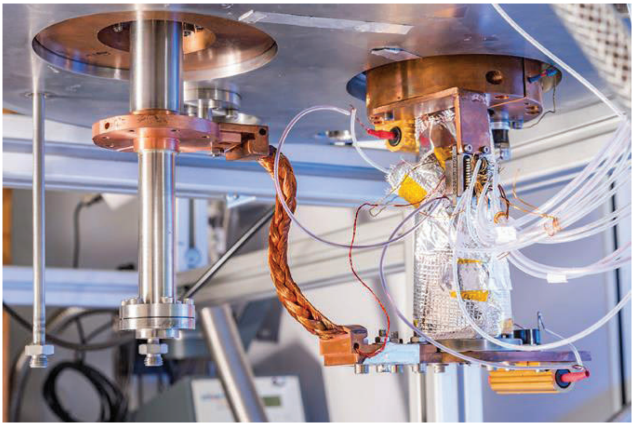

**Figure 6.** Picture of the test setup used for the optimization of the 15 K cold finger.

**Table 1.** Porosity of the stainless steel and of component A and B

| SS | Comp. A | Comp. B |
|---|---|---|
| 0.62 | 0.465 | 0.455 |

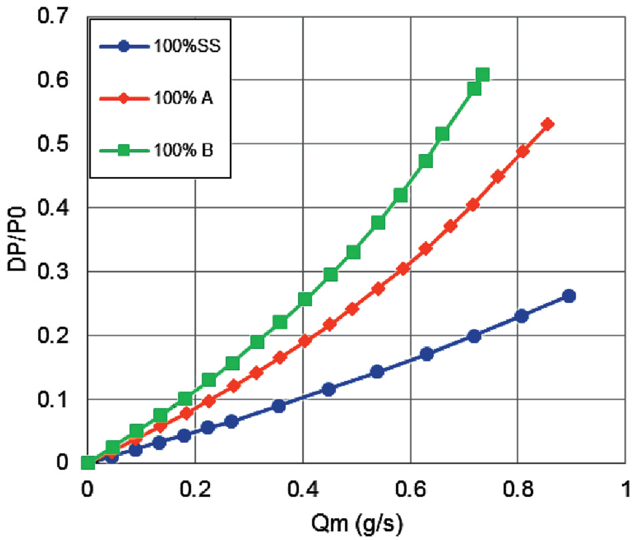


**Figure 7.** Pressure losses measurements under continuous flow

There is a non-negligible difference in cooling performances between component A and component B. As explained in the previous paragraph, despite a lower specific heat, the component A is better than component B. As seen before, the component B has higher pressure drop, so the expansion work inside the pulse tube is not enough to produce a sufficient cooling power.

On the cooling power curve, the minimum temperature for A and B is almost the same and there is significant difference at higher temperatures. This is linked to the PV power that is also associated to the pressure drop. The PV power has a limited impact on the minimum temperature but a high impact on the cooling power. So the lower the pressure drop, the higher the cooling power.

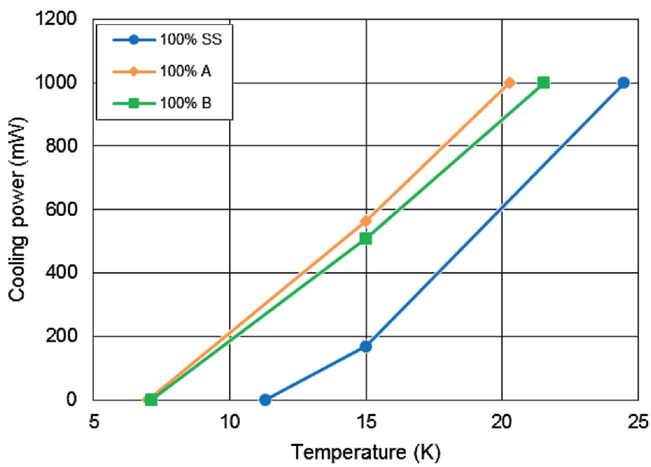


**Figure 8.** Cooling power plot at 100 W PV, 80K at the intercept and 41 Hz.

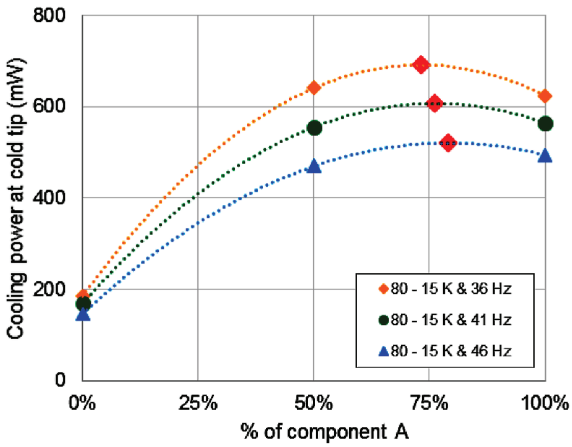


**Figure 9.** Typical curve of cooling power vs ratio of component A. Here the effect of frequency is presented.

### Optimum Ratio

As discuss above, a wide range of parameters have been studied for each configuration. The component A gives better cooling performance whatever the operating parameters, so the following discussion will deal only with the component A.

As shown on Figure 9, the optimum ratio is affected by the drive frequency of the pulse tube cooler. We observed that the drive frequency affects the optimum by only a few percent. The optimum ratio is mainly affected by the temperature gradient between the cold tip and the intercept (Figure 10). For a low temperature of the cold tip (< 15 K), the intercept temperature affects the optimum ratio of less than 10% between 50 K and 120 K. The lower the temperature of the intercept, the higher the need for exotic materials close to the intercept. For a high temperature of the cold tip (> 20 K), the effect of the intercept temperature is now stronger with more than 20% difference in the ratio and the highest for the 50 K intercept temperature.

This test campaign gives access to a wide range of parameters that has consolidated the performances of the cooler. It will also provide an optimization tool for future operating parameters of the cooler.

For the purpose of the future 15 K pulse tube cooler, the objective is to consolidate and improve the performances at 15 K. With this setup, at an intercept temperature of 80 K and a frequency of 41 Hz, we estimate the cooling power at the optimum ratio to be above 600 mW (round points on Figure 9) at a ratio which is similar to the previous cold finger.

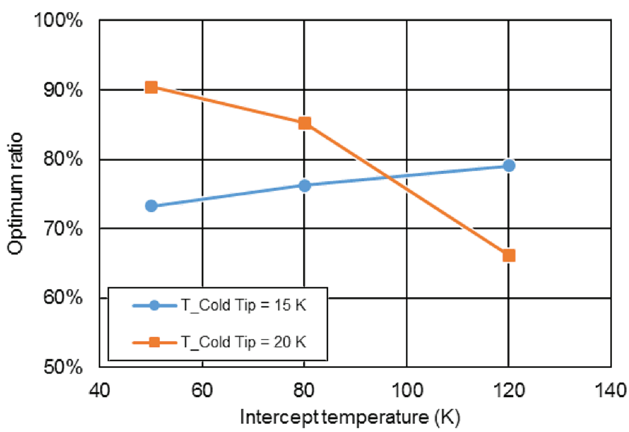


**Figure 10.** Effect of the 1st-stage temperature on the optimum ratio for 2 different 2nd-stage temperature (f = 41 Hz)

## CONCLUSION

After the end of the test campaign on the first EM 15 K pulse tube cooler at Air Liquide, further work has been carried out in CEA-SBT in order to further characterize the 15 K pulse tube cooler in the framework of the ATHENA/X-IFU mission. The performances of the overall system has been improved on both 1$^{st}$ and 2$^{nd}$ stage by changing the 1$^{st}$ stage inertance and 2nd stage radiative insulation.

Dedicated work on the cold regenerator has been performed to consolidate and possibly improve the cold finger performances. Thanks to the large number of studied parameters, we developed a data base for future optimization of the cooler in alternative operating conditions. However, the initial material (A) has better performances than the novel material (B). The expected gain due to higher specific heat has been decreased by a higher pressure drop. The ratio of the first EM model was already close to the optimum. For further optimization, other regenerator materials are under study.

## AKNOWLEDGMENT

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