IOP Conf. Series: Materials Science and Engineering **1327** (2025) 012153 doi:10.1088/1757-899X/1327/1/012153

Design and Performances of a Dry Table-Top Optical Cryostat at 2K

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Abstract. We have developed a versatile cryogenic platform for research in optics. The optical cryostat offers an experimental volume of 100 x 100 x 100 mm with up to five 2-inches optical accesses. The cold plate is cooled to a temperature below 2 K with more than 50 mW power available. The cryogenic system is based on a two-stages 4 Kelvin Pulse Tube (PT410RM) and a 2 K helium Joule-Thomson refrigerator. The Pulse-Tube cold head is located in a remote 'cold box' connected to the cryostat by a cryogenic line in order to minimize the occupation on the optical table. The thermal coupling is made by two cooling loops, coupling the two stages of the PT with a radiative shield (50 K) and the cold plate (2 K). This solution allows a distance of a few meters between the cryostat and the 'coldbox' and limits the vibrations induced by the cryocooler on the experimental plate, which are generally very detrimental in optics.

1. Introduction

The achievement of temperature between 1 – 2 K is classically made with liquid helium pumping through a flow impedance, i.e. a Joule Thomson refrigerator (JT). Before the advanced of commercial Gifford Mc Mahon (GM) or Pulse-Tube at 4 K, the coupling with a JT cooler was the only solution to build a small scale cryocooler at liquid helium temperature [1].

Today, coupling a JT cooler with a 2-stages 4K PT or GM is an attractive solution to build a closed-cycle system that provides more than 100 mW below 2 K [2][3]. The advantage of the PT is the robustness (no moving parts at low temperature) with more than 80000 h MTBF (only the PT compressor requires a periodic maintenance at 20000 h).

The theoretical basis of the JT cryocooler is well established [4] and is still a key component in large scale helium liquefaction. The drawbacks of those systems are 1) the high sensitivity to helium impurity, which is the main cause of blockage in long duration exploitation (more than a few months) 2) the addition of a pump and/or a compressor for the helium circulation. A clear advantage is the possibility to realize a solution that avoid conductive links between the 'application cryostat' and the active cryocooler part. The advantage of a 'cooling loop' is clear if the distance exceeds one meter [4]. Moreover, the vibrations induced by the cryocooler, are expected to be reduced by using flexible pipes for the various flows.

The cryogenic plateform developed here integrates those concepts to build a low vibration optical table-top cryostat.

2. System architecture and design

2.1 Mechanical architecture

The 'coldbox' contains the PT head mounted with a welded bellow and a cryogenic FAN for the high pressure cooling loop. The box is evacuated and mounted on soft absorbers on a mechanical structure independent of the optical table, allowing a few mm horizontal displacement with less than 50 N force. The cryogenic line is mounted into a reinforced bellow in order to avoid a large displacement during evacuation (Fig.1).

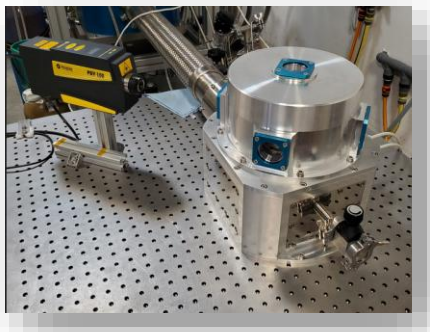

Figure 1. Left: The ACE-CUBE system with the cold box and a cryogenic line (1 meter). Right: the optical cryostat with a Laser velocimeter installed on the optical table for vibration measurement.

2.2 Cryogenic system design

A pressurized helium cooling loop (~1.0 – 1.5 MPa) is used to couple the 1st stage PT (~50 K) with the radiative shield of the optical cryostat (Fig.2). The return line is coupled to a cooled shield into the cryogenic line in order to limit the heat load on the cold inner lines. The helium circulation is realized with a small cryogenic FAN that allows a flow of ~300-400 mg/s at the nominal conditions (T < 50 K), making a thermal conductance of 1.6 – 2.1 W/K.

During the cooldown, a helium flow is injected through a specific line that bypass the JT recuperative heat exchanger and the JT (blue line in Fig.2). The cooling flow uses a specific tube mounted on the regenerative PT heat exchanger in order to maximized the cooling power available at 4K [3]. That line can also be used to control the cold plate between < 5 – 300 K in the 'cooling loop operational mode' which is discussed in §2.3.

The JT cooler is operated when the cold plate has reached a temperature below 5 K by simply closing the cooling flow inlet valve. In order to limit the thermal loads due to the JT flow, three recuperative counter flow (CF) heat exchangers have been implemented between 300 K / PT 1st stage / PT 2nd stage and the JT evaporator (cold plate). They are made with a stainless steel tube-in-tube classical technique with efficiency > 90%. The flow is thermalized on the PT stages with ~1 m tubes welded on the PT interfaces (ε > 80% for the 5 mg/s flow). While there is no critical requirements on the JT cooler inlet pressure loss, the inner diameters have been chosen > 3 mm in order to limit the risk of clogging.

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2.2.1 The JT2K cooler
The JT cryocooler design is driven by the requirements on the cold plate. Supposing that the condensation of helium is fully achieved into the PT 2nd stage heat exchanger, the cooling power is approximatively given by the latent heat of helium at the low temperature (~90 J/mol @ 2 K).

Considering an available cooling power of 50 mW at 2 K, we have to take into account the thermal loads from the cryostat, i.e. mainly the conductive load from the struds made in titanium alloy (TA6V). The design considers a thermal load P_0 of ~20 mW from the struts supporting the cold plate (Table 1). For the design, we have considered a conservative cooling power needed of 100 mW. For that cooling power, we need ~100 mW/90 J/mol = 1.1 mmol/s (i.e. 4.5 mg/s). The design temperature drives the pressure on the evaporator, i.e. the pumping inlet pressure. The saturation temperature at 1.80 K is ~2000 Pa. Considering 1) a thermal conductance between the saturated liquid into the evaporator of ~0.1 K 2) a pressure loss into the pumping line (50%), the pumping speed needed at room temperature to get 1.9 K on the evaporator is ~10 m³/h. This is fully compatible with commercial dry scroll pumps.

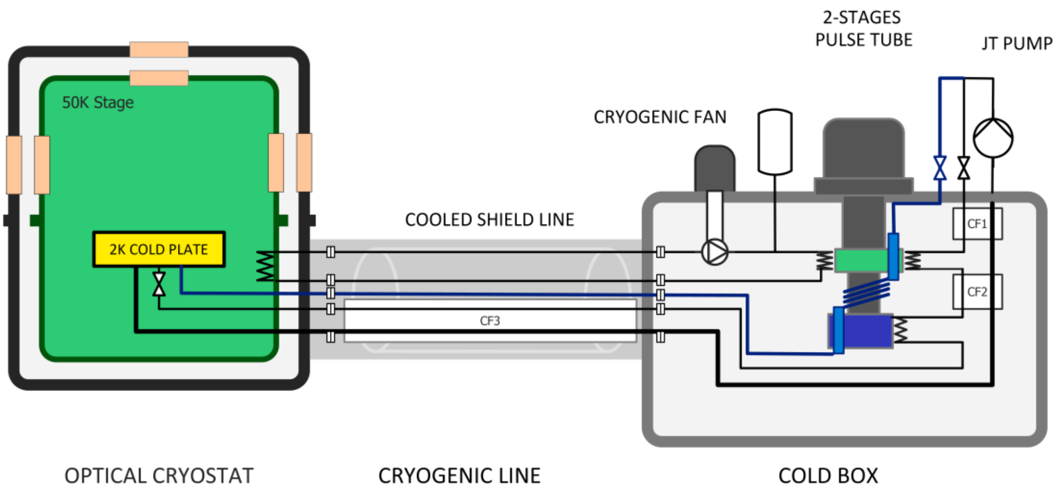

Figure 2. Cryogenic system architecture. The pressurized helium cooling loop is based on a small cryogenic FAN and designed to have a flow of 300-400 mg/s under cold conditions. The JT pump outlet is to two separate lines. The 'blue one' injects a flow into the evaporator during the cooldown phase. It may be used to have a variable temperature mode to control the cold plate between ~5 K to 300 K. The inlet valve is closed to operate the JT cooler at low temperature (2 K).

2.3 Heat balance estimation
The thermal balance is estimated on the three cooling stages available (the PT 1st and 2nd stages, the JT evaporator). The heat loads are given from the three components:

- The cold box, with the radiative shield mounted on the PT 1st stage (MLI is used to reduce the radiative load from the room temperature walls)
- The cryogenic line: the radiative loads on the cooled-shield is intercepted on the active cooling loop, i.e. on the PT 1st stage; the mechanical supports used to center the inner lines (JT inlet, pre-cooling and LP pumping) is applied on the JT cold stage
- The cryostat: the cold shield loads (conduction from the supports, radiation from the ambient temperature walls and optical filters) are applied on the cooling loop, i.e; the PT 1st stage; the thermal loads on the cold plate are applied on the JT evaporator

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The following table (Table 1) gives a consistent view of the main thermal loads for the 'design point', i.e. with 50 mW applied on the cryostat cold plate and the temperatures achieved during the validation test (Table 2).

Table 1. Thermal balance estimation at the design point (5 mg/s) corresponding to 50 mW applied on the cryostat cold plate and the temperatures measured on the nominal point. The temperatures of the cryostat are 35 K on the radiative shield and 2.0 K on the evaporator. The static heat load on the evaporator is estimated to ~20 mW, dominated by the conduction from the 50K/2K struts made in TA6V.

Cooling stage	PT 1st stage	PT 2nd stage	JT evaporator
Cold box			
Radiation (MLI protection on the shield)	3.8 W	10 mW	-
Cryogenic FAN @ 300 mg/s	~4 W (3-5 W)	-	-
JT Cooler flow (5 mg/s with 90% efficiency HX)	0.70 W	160 mW	-
Cryogenic line (1 meter)	~2 W *	-	-
Application cryostat			
Radiation (MLI protection on the 50K shield)	1.1 W *	-	0.6 mW
Conduction from the struts	0.40 W *	-	18 mW
Application cryostat (user loads)			
Filters Radiation (5 x BK7 filters 2-inches diameter)	1.1 W *	-	0.9 mW
User load (electrical heater on the cold plate)	-	-	50 mW
TOTAL	**13 W**	**170 mW**	**70 mW**

(*) Those loads are applied on the cooling loop flow and contribute to the temperature difference between PT 1st stage and the cryostat radiative shield (~ 4.5 W)

3. Experimental results

3.1 Cooldown time

The cooldown is controlled by the pre-cooling flow achieved by the JT circulation pump. Because we do not use a compressor, the main limitation is the outlet pressure (p_{MAX} ~ 130 kPa). The maximum flow achieved during the cooldown is ~50-100 mg/s and should be reduced to ~10-15 mg/s to reach the conditions that allows the JT condensation, i.e. T < 5 K on the evaporator. (see §2.3). Without added mass, the cooldown time achieved on the prototype plateform is ~7 h.

3.2 Cooling capacity

In order to validate the cooling capacity, we used 1) an electrical heater mounted on the evaporator plate 2) a heater mounted on the flow impedance. The power is estimated from the electrical measurements from the voltage and the current measurements. The massflow and the temperature responses are reported in Fig.3 in both cases.

The massflow indicates a strong relation with the power applied in both cases. Supposing an extrapolation of the massflow response to zero, we estimate a contribution from the system P_0 ~40-60 mW on the evaporator, which is higher than the expected value of ~20 mW (Table.1). Also, the base temperature is significantly higher than the expected one with the measured pump inlet pressure (700 Pa - T_{sat} = 1.6 K). That suggests an excessive pressure loss into the pumping line which has been identified in the JT regenerative HX mounted between the PT 1st stage and the ambient interface.

A temperature sensor mounted on the line after the PT 2nd stage indicates that the temperature is well correlated to the saturation temperature corresponding to the pump outlet pressure (Fig.4). It suggests that at least a partial condensation is achieved into the HX.

On the application cryostat, the radiative screen interface temperature is 35 K, closed to the PT 1st stage (33 K). This is well in accordance with a cooling flow of ~330 mg/s predicted into the cryogenic loop which is based on the FAN characteristics and the cooling loop pressure losses.

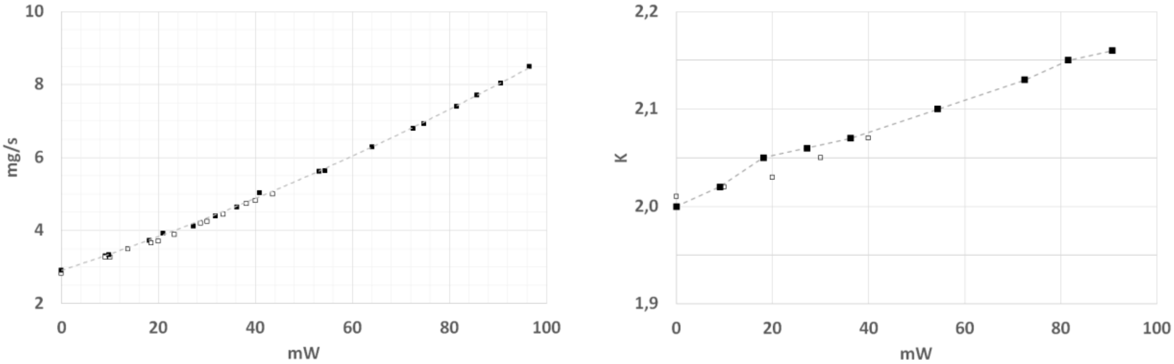
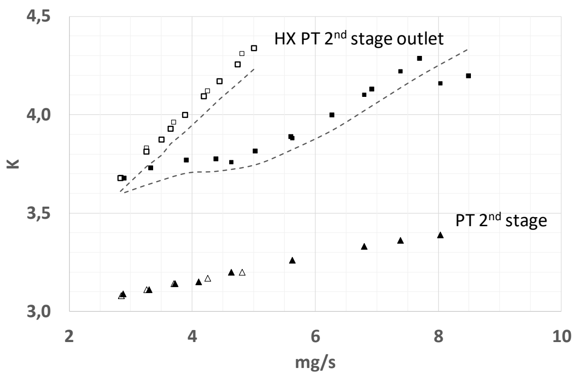


Figure 3. Power response of the massflow and the evaporator temperature. Dark squares = power applied on the evaporator, open squares = power applied on the JT impedance.



(dark symbols = power applied on the evaporator, open symbols = power applied on the JT impedance)
The squared symbols indicate the temperature measured on the JT line at the outlet of the PT 2nd stage.
Dotted lines are the saturation temperature corresponding to the pressure measured at the pump outlet (p < 130 kPa). The temperature drop after 8 mg/s suggests that the condensation is not complete below that value.
The PT 2nd stage temperature reflects the heat load induced by the JT flow.

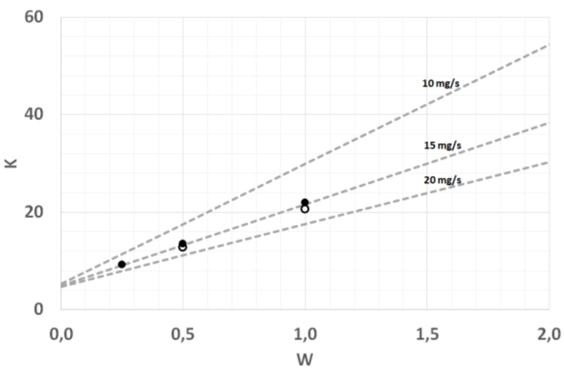
Figure 4. Temperature of the PT 2nd stage and the JT line in the cold box.

Table 2. Nominal point measures (50 mW applied on the evaporator).

Parameter	Value	Comment
Temperature PT 1st stage	33 K	Est. load PT410RM~ 6-12 W
Temperature PT 2nd stage	3.3 K	Est. Load PT410RM ~ 0.2-0.3 W
Cryostat temperature - Radiation shield	35 K	Est. cooling loop flow ~330 mg/s
Cryostat temperature – Evaporator plate	2.1 K	Note: Psat @ 2.0K = 3100 Pa >> 700 Pa
JT cooler flow	5.5 mg/s	Est. load ~90-110 mW
Inlet pump pressure	700 Pa	
Outlet pump pressure	65 kPa	

3.3 Variable temperature mode

In the JT mode, the evaporator temperature can only be controlled by reducing the pumping speed in order to increase the saturation temperature (up to ~4 K). In order to control the temperature into a larger range, we use the 'cooling loop mode'. The temperature response to the power applied on the cold plate heater is driven by the cooling flow. The cooling power available on the regenerative PT heat exchanger limits the cooling flow to ~25 mg/s in order to keep a temperature < 5 K. The temperature response is indicated in Fig.5 for various flows. While not tested above 22 K (1 W), that system could easily be used to control the temperature up to 300 K.



The cooling loop flow is controlled by a valve installed within the inlet line. The helium pressure is below the condensation temperature of helium within the PT 2nd stage HX, i.e. the temperature increase within the evaporator is driven by the mass flow of the ideal gas enthalpy : $\dot{m} \cdot c_p \cdot (T_{evap} - T_{PT}) = P + P_0$

To reproduce the measurements made at 15 mg/s, we have to consider a thermal resistance (1 K/W) at the cold plate interface and an efficiency of the PT 2nd stage HX (~80%). The dotted lines indicate the expected temperature response for several flows below the limit of 25 mg/s.

Figure 5. Cold plate temperature control in the 'cooling mode'

4. Conclusions

The prototype system has demonstrated a cooling power of 50 mW at 2.1 K on the cold plate with 5 optical windows (BK7 2-inches diameter). The maximum power applied is ~90 mW, due to the pump outlet pressure limit (130 kPa) and the JT impedance choice.

The base temperature achieved (2.0 K) is limited by the pressure losses identified into the LP line. A modified design of the HX has already been implemented on a similar system and demonstrated 50 mW @ 1.6 K.

An excessive thermal load of 20-40 mW is estimated on the cold plate, suggesting that the system could be improved. Finally, the vibration level achieved is still under investigation and not reported here.

Acknowledgments

This work has been financed by Absolut System, CNRS-G2ELAB-Absolut System (research con-tract#116420) and the ANRT/CIFRE grant contract N°2021/0803 to support Marek Zeman's PhD the-sis

References

[1] Poncet J.-M., Claudet G., Lagnier R., Ravex A., Large cooling power Gifford Mc Mahon / Joule Thomson Refrigerator and liquefier. Cryogenics 1994 Vol 34 ICEC Supplement.
[2] Wang C. et al, A closed-cycle 1 K refrigerator cryostat. Cryogenics 64 (2014) 5-9.
[3] Wang C., Hanrahan T., Johnson M., High capacity closed-cycle 1 K cryocooler. Cryogenics 95 (2018) 64-68.
[4] de Waele A., Basics of Joule-Thomson Liquefaction and JT cooling. J Low Temperature Phys (2017) 186:385-403.
[5] Trollier T. et al, 30 K to SubK vibration free remote cooling systems. Conf Proc. CEC-ICMC (2019).