

A self contained ^3He refrigerator suitable for long duration balloon experiments

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Abstract

We describe the design, development and test of a self contained ^3He fridge, aimed to cool at $\lesssim 0.3$ K a bolometric detection system with a hold time of two weeks. The system is robust and suitable for operation on a long duration stratospheric balloon payload. A physical model of the fridge has been developed, which describes the measured hold time, limiting temperature and load curve of the fridge. The system has been flight tested successfully during two balloon flights of the BOOMERanG payload in summer 1997.

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1. Introduction

BOOMERanG^{1,2} is a balloon-borne millimeter-wave telescope devoted to map the microwave sky with high angular resolution (~ 12 arcmin) and high sensitivity ($\sim 20 \mu\text{K}$ per pixel). The aim of the experiment is to map the Microwave Background Radiation and to measure the power spectrum of the Microwave Background Anisotropies. The detectors of choice for this kind of experiments are composite semiconductor bolometers cooled at cryogenic temperatures^{3,4,5}.

Our group has developed and successfully tested a custom cryogenic system, able to cool the detectors and the reimaging optics at 0.3 K. The system is designed to operate automatically on a stratospheric payload for two weeks, and is suitable for long duration ballooning. In a separate paper⁶ we describe the main ^4He cryostat developed for the project. Here we concentrate on the ^3He fridge. This is basically an upgrade from the smaller one developed for the ARGO experiment⁷: a self contained, single shot, high pressure ^3He fridge of the kind described in references 8, 9, 10). However, a number of different, peculiar specifications impacted considerably the design of the new system. In particular:

A) The mechanical strength specifications for long duration balloon flights carried out by NASA-NSBF around Antarctica are more severe than the one which drove the design of the ARGO fridge. All the mechanical parts must be able to withstand accelerations as large as $10g$ on the vertical and $5g$ at 45 degrees from the vertical (any azimuth). The longer hold time needed (two weeks) forces us to increase considerably the quantity of ^3He contained in the fridge, and, as a consequence, the total volume of the fridge itself to maintain a reasonable internal pressure at room temperature. Thus the cryopump and the evaporator are heavier, and need special supporting structures to withstand the accelerations given above.

B) BOOMERanG features high efficiency total power bolometric receivers, and the thermal stability of the temperature of the evaporator must be extremely good to operate successfully the detectors in this observation mode. In fact, the sky observation strategy of BOOMERanG consists of azimuth scans, ~ 40 s long, at constant speed (1 deg/s). The beamsize is $\gtrsim 10$ arcmin. This means that the useful information in the detectors signals is contained in the band from about 0.02 Hz to about 10 Hz. Temperature fluctuations of the cold plate in this frequency range can affect significantly the noise figure of the detection system. We use Neutron Transmutation Doped Ge bolometers¹¹ with a resistance of a few $\text{M}\Omega$ at 0.3 K, and a temperature coefficient $\alpha = \frac{1}{R} \frac{dR}{dT} \sim 20 \text{ K}^{-1}$. In the operating point we have a ratio between temperature induced signal and fridge temperature variation $\frac{\Delta V}{\Delta T} \sim 0.1 \text{ V/K}$, which must be compared to an expected CMB anisotropy signal of $\sim 2 \text{ nV}$. This means that the temperature stability of the fridge has to be better than 20 nK on timescales between 0.1s and 100s. Similarly if we require that the temperature induced noise is lower than the intrinsic noise of the bolometer (spectral density $W_{VI}^{1/2} \sim 10 \text{ nV}/\sqrt{\text{Hz}}$) we get a limit on the spectral density of the temperature fluctuations of the cold flange of the fridge: $W_T^{1/2} \lesssim 0.1 \mu\text{K}/\sqrt{\text{Hz}}$ in the frequency range $0.02 \text{ Hz} - 10 \text{ Hz}$.

C) The mechanical stability of the cold flange and of all the other parts of the fridge is very important to avoid microphonics of the detectors. Mechanical resonance

frequencies above 90 Hz are required to push the microphonics induced by fridge oscillations well outside the useful electrical bandwidth of the detectors, in a region where the spurious signal is strongly attenuated by electronic filters in the amplification chain.

D) The BOOMERanG telescope has an alt-azimuth mount, and the cryostat is rigidly connected to the telescope. So must be able to operate with constant performance in a range of inclinations ($\pm 30^\circ$ from the vertical).

In the following we describe how the fridge design was implemented, with particular attention to the long hold time and the peculiar problems A, B, C and D. After that we describe the hardware construction and optimization, and the results of the final calibration tests of the system.

2. Fridge design

A sketch of the fridge is shown in fig.1. CP, HE, E label the cryopump, the heat exchanger and the evaporator, respectively. We used a vertical geometry with hot components above cold components during normal operation, in order to minimize the heat convection in both the cryopump to heat exchanger tube and the heat exchanger to evaporator tube. Convective heat transfer has been found to be effective in different fridge configurations, heavily affecting the ^3He condensation efficiency³. ^3He condensation is obtained inside the high purity copper heat exchanger (HE), which is connected to a pumped ^4He bath through a reed bolted joint (labeled BJ in fig.1).

The total geometrical volume inside our fridge is 1050 cm^3 . The volume available for the gas is about 20% less, due to the volume occupied by the charcoal grains. We assumed to have a typical total heat input on the evaporator of $\sim 27\ \mu\text{W}$, a condensation efficiency of 80% (see ref.9), and that 25% of the condensed liquid ^3He evaporates to cool down to 0.3 K the remaining. This means that for 15 days of operation we need 60 liters STP of ^3He in the fridge, with a room temperature pressure ~ 67 bar. This number sets the thickness of all the mechanical components of the fridge.

The stainless steel spherical cryopump (120 mm diameter) thickness has been decided based on the formula¹²:

$$T = \frac{PDX}{5R\psi} \quad (1)$$

where T is the thickness, P is the pressure, D is the inside diameter, R is the breaking strength ($\sim 5 \times 10^8\text{ N/m}^2$ for stainless steel), ψ is a form factor (23/22 for a sphere) and X is a safety factor, which turns out to be ~ 6 for $T = 2\text{ mm}$. The cryopump is enclosed in a cylindrical aluminum radiation shield, to prevent radiation from the cryopump to affect the performance of the bolometers. A copper tube, connected to the cryopump heater and to the cryopump heat switch, crosses the cryopump body on two antipodes. The tube is bent in several coils inside the cryopump, thus increasing the conductivity to the charcoal grains. In fact these are not glued, and are simply inserted to fill the sphere. A cylindrical stainless steel mesh (labeled M in fig.1) continues after pumping tube inside the cryopump, avoiding migration of grains in the system, without reducing too much the surface available to the incoming gas for cryosorption. The volume of the cryopump is 905 cm^3 . We used 346 g of charcoal. So the volume available for the gas

is 785 cm^3 . The cryopump is quite heavy ($\sim 0.8 \text{ kg}$) and must be held by suitable low conductivity supports. In our case we have the stainless steel pumping tube, taking care of vertical accelerations, and two lightweighted fiberglass dishes (2 mm thick) connecting the cryopump to the cylindrical shield, taking care of horizontal accelerations.

The evaporator is a stainless steel cylinder (56 mm inner diameter, 55 mm inner height, 2 mm thick), with a thick stainless steel upper cover (8 mm thick) and a thick copper base. The thickness of the stainless steel cylinder has been computed from the formula¹²:

$$T = \frac{PDX}{2R\alpha} \quad (2)$$

where α is a coefficient depending on the quality of weldings $\alpha \sim 0.8$ in our case. We get a safety factor $X \sim 5$ for $T = 2 \text{ mm}$. The thickness of the copper baseplate has been computed from the formula¹²:

$$T = 0.866 \frac{D}{2} \sqrt{\frac{PX}{R}} \quad (3)$$

For high purity copper, after oven welding, we have quite a low breaking strength ($R \sim 1.5 \times 10^8 \text{ N/m}^2$), and a full 10 mm thickness is necessary to get a safety factor ~ 4 . The inner side of the copper base flange has been slotted to reduce the thermal impedance between the liquid ^3He and the load, and to reduce the temperature change when the cryostat is tilted from the vertical. We obtain a contact surface of 50 cm^2 . The evaporator weight is $\sim 0.5 \text{ kg}$. The stainless steel pumping tube easily withstands vertical accelerations, but is vulnerable to modest horizontal accelerations. So a very low heat conduction supporting structure has been designed, consisting in a set of three vespel¹³ tubes ($A/\ell \sim 0.06 \text{ cm}$) supporting an aluminium ring concentric to the evaporator cylinder. The ring holds six horizontal vespel spikes with set-screws. These avoid horizontal shifts and tilts of the evaporator with a very small additional heat input ($\ll 1 \mu\text{W}$). We have verified that this support system pushes the resonant oscillation modes of the evaporator at frequencies significantly higher than the highest interesting frequency in the signal ($\sim 10 \text{ Hz}$). For oscillations along the vertical direction, the resonant frequency is

$$f_v = \frac{1}{2\pi} \sqrt{\frac{\pi E(D^2 - d^2)}{4ML}}$$

where E is the Young modulus, D and d are outer and inner diameter of the tube, L is the length of the tube and M is the suspended mass (evaporator). In our case we have frequencies around 2000 Hz, assuming that the vespel supports are not effective, and that the evaporator is fully suspended from the stainless steel tube. More dangerous are radial oscillations. The resonant frequency is

$$f_r = \frac{1}{2\pi} \sqrt{\frac{3\pi}{64M} \left[\frac{E_v(D_v^4 - d_v^4)}{L_v^3} + \frac{E_s(D_s^4 - d_s^4)}{L_s^3} \right]}$$

where the subscripts i and s refer to the stainless steel tube and to the vespel columns respectively. In our system we get $f_r \sim 100 \text{ Hz}$, still satisfying our requirements.

The support system for the evaporator is strong enough to withstand the $10g$ loads vertical and $5g$ radial. To verify this, we consider the vespel column as a deformable system, constrained at the upper flange, and loaded by a known force. The equilibrium can be tested by comparing the vespel breaking strength $\sigma_A \sim 8.6 \times 10^7 Pa$ and the actual stress. In our case we can use the diagonal component of the stress tensor σ_z . The two components of σ_z are maximum in the section of the column nearest to the constraint. We have

$$\sigma_{z,\perp} = \frac{10Mg}{A}$$

and

$$\sigma_{z,\parallel} = \frac{5MgLD}{2I_x}$$

with $I_x = \frac{\pi}{64}(D^4 - d^4)$. We get $\sigma_{z,\perp} + \sigma_{z,\parallel} < \sigma_A$ with a safety factor ~ 25 .

The pumping line is divided in three sections: one (vertical) from the evaporator to the heat exchanger, one (horizontal) inside the heat exchanger, and one (vertical) from the heat exchanger to the cryopump. The offset between the two vertical sections of the pump line allows us to reduce the radiative load from the cryopump to the fridge. The inner diameter of the pumping line heavily affects the performance of the fridge, in particular the ultimate temperature and the load curve. In fact the impedance of the pumping line and the heat input on the still determinate the pressure drop between the evaporator and the cryopump. Since the pressure inside a properly designed cryopump is very low, the pressure of the vapours and the temperature of the liquid are mainly determined by the pumping line diameter. Previous experience with the ARGO fridge⁷ and with the MITO fridge¹⁶ has shown that a diameter $\gtrsim 8mm$ is needed to obtain a fridge temperature below 0.3 K with a heat input of $30 \mu W$ on the still. On the other hand, increasing the diameter of the tubes requires to increase their thickness to withstand the high pressure at room temperature, thus increasing the conduction heat load on the fridge. We find that a good trade-off for the evaporator section of the pumping line is a $250 \mu m$ thick stainless steel tube, 72 mm long, with 9.5 mm inner diameter. This gives us a safety factor ~ 3 . Further support to this choice comes from a physical model of the fridge (see section 5). Also for the cryopump section we used the same tube. The length of the cryopump pumping line is determined by the requirement that, during condensation, the power required to warm up the cryopump is not too high, so that the condensation point (heat exchanger) does not heat up. In practice we require that the conduction heat input is negligible with respect to the convection through the desorbed gas in the pumping line.

The bolometers, filters and radiation concentrators are mounted on a separate copper flange, thermally connected to the evaporator cold flange by means of a flexible copper sheet. The bolometers flange is supported by four Vespel columns with the same dimensions of the three columns supporting the evaporator.

3. Fridge construction

The parts for the fridge were prepared using standard machine shop techniques. The hemispheres for the cryopump were lathe flosspinned from 2 mm thick stainless steel flats. We used a high purity copper (99.999%) rod for the heat exchanger and the heat strap to the ^4He reservoir. We used thin wall stainless steel tubes for the pumping lines. Assembling was carried out as follows. First, all the welds were made (labels W in fig.1). Second, the parts were oven-brazed (labels B in fig.1) under an inert atmosphere to avoid oxidization of the internal surfaces.

At this point the system was pressure checked with ^4He at a pressure of 100 bar. After leak test, the charcoal grains were inserted.

The system was filled with ^4He at 70 bar and cycled to check for possible cold leaks and general performance. After this test, the fridge was filled with ^3He . Since the volume of the ^3He bottle was comparable to the volume of the fridge, we had to use a small cryopump immersed in a liquid helium storage dewar to adsorb all the gas from the bottle. Then the cryopump was connected to the fridge and heated up. Meanwhile the fridge was maintained at 77K, imbedded in a liquid nitrogen vessel. At equilibrium the pressure was read and the copper filling tube was crimped, thus sealing the fridge. We find that the ratio between the pressure readout at equilibrium in these conditions and the pressure at 300 K is 3.7, in rough agreement with the ideal gases law. Our best estimate of the ^3He charge is 52 liters STP. We plan to increase the charge to 60 liters STP in a second unit of the fridge.

The cryopump heater was prepared using an EvanOhm wire, wound and glued with Stycast FT2850 epoxy to the copper support on the cryopump. The total resistance of the heater was 350 Ω , a value suitable to warm up the cryopump at 40-50 K with a reasonable current (40 mA). Three calibrated thermometers were mounted on the fridge. We used a Germanium Resistance Thermometer (GRT model LakeShore GR-200) for the evaporator, and silicon diodes (LakeShore DT-470-SD13) for both the heat exchanger and the cryopump. All the thermometers were wired for four wires readout. The electronics for fridge operation has been mounted in a shielded, RF tight box on the top of the main cryostat. The box includes precision current sources to supply the thermometers (10 μA for the Si diodes and 125 nA for the GRT) and instrumentation differential amplifiers for readout of the thermometer voltages. A variable voltage supply for the cryopump heater and two variable voltage supplies for operation of the motor are also included. All these circuits are commandable through the relays present in the box, which are operated either by means of push bottoms on the box front panel for laboratory operation or by means of the payload telemetry for in flight operations.

The fridge was mounted in a test cryostat for debug and qualification, and moved in the BOOMERanG cryostat after the first successful test.

4. Measured Performace

The most important result of the first tests is that this fridge does not need a heat switch for the evaporator. The large quantity of ^3He present in the circuit and the large size of the pumping line make convective cool down quite effective. In practice, the evaporator thermalizes at 2 K simultaneously with the heat exchanger. Even with a

large load, like the ~ 1 kg of copper of the focal plane of the BOOMERanG experiment, the evaporator follows the condensation point with a delay smaller than 1 hour.

On the other hand, the cryopump has an important heat capacity, and the time required to heat it up to desorb the ^3He and to cool it down starting cryopumping is quite long (several hours).

We made a test cycle starting from room temperature and avoiding to cool down the cryopump (heat switch open), so that the gas was not adsorbed and condensed completely in the evaporator. Since we know the quantity of gas we put in the fridge, this peculiar cycle allows us to measure the total heat input on the fridge from the hold time. Since the temperature of the condensation point was 2K during condensation, we assume a 80% condensation efficiency in this case (see also ref. 9). We measured a hold time of 336 hours at 0.3K, limited by the hold time of the ^4He in the main cryostat, which was maintained at ~ 1.7 K for the whole hold time test. From this a total heat input of $\lesssim 28\mu W$ is estimated. This is consistent with our estimate of $\sim 15\mu W$ due to conduction through the evaporator pumping line, $\sim 7\mu W$ due to 4 vespel supports for the bolometers flange and $\sim 5\mu W$ from the bolometers array wiring.

A normal cycle starts with the cryopump cold at 2 K, instead. The cryopump is heated up to at least 35 K. Inversion of the temperature rise under constant applied power is a signature of desorption of the ^3He . We find that to have a good condensation efficiency we need to heat the cryopump to temperatures higher than 35 K while maintaining the condensation point (heat exchanger) below ~ 2.5 K. This requires to dissipate about 0.35 W in the heater (for a total of about 5 kJ). To achieve this, the thermal conductivity of the heat strap connecting the fridge and the ^4He tank has to be larger than ~ 1.8 W/K. This is achieved using very high purity copper and a reed joint clamped by a M8 bolt. We have measured the heat conductivity of the joint to be ~ 2.6 W/K. A typical path in the $T_{CP} - T_{cond}$ plane during the condensation phase is shown in fig.2. When the heat switch is closed, the cryopump cools down with a time constant of about 2 h. The evaporator reaches 0.3 K in about 5 hours. With the normal heat load of the BOOMERanG experiment (total $27\mu W$ on the fridge), the low temperature is maintained for 14 days. The ultimate temperature obtained with such a heat input is 0.297 K. With lower loads (no Vespel, less wiring) we have measured temperatures as low as 0.28 K. The temperature stability is remarkably good. Without any stabilization system we get a typical temperature drift of 0.1 mK/h (see fig.3), mostly due to variations of the temperature of the ^4He cryostat (which is pumped at $\sim 1.7K$). In order to get rid of these long term drifts a temperature control system can be used. Also, the presence of a temperature controller is required when the system is mechanically shocked. Operation of some balloon-borne bolometric experiments, for example, requires fast (few degrees per second) scans obtained by azimuth rotation of the entire payload. During the turnarounds, the system experiences significant accelerations, and ^3He temperature changes of the order of $0.1\mu K$ are developed. A temperature controller can easily and effectively fight this problem, while nulling the long term drifts. Due to the intrinsic stability of the temperature, it is possible to precision regulate the temperature of this fridge using a minimal power dissipation ($\sim 2\mu W$), so that the hold time is basically unaffected.

The short term (0.02 Hz - 10 Hz) temperature stability has been measured using an AC biased GRT, in the absence of temperature control. We did not see any evidence of fluctuations above the intrinsic noise of the GRT. From this we can place an upper limit to the power spectrum of the temperature fluctuations of the cold plate of the order of $W_T^{1/2} \lesssim 0.1 \mu K / \sqrt{Hz}$ in the band 0.02 Hz - 10 Hz.

We have measured the load curve of the fridge for loads ranging from 25 μW to 500 μW . The data are shown in fig.4. It is evident that the fridge can easily withstand extra heat loads (up to 20 μW) coming from temperature changes in the ^4He cryostat generated by changes in the float altitude, or from the temperature control system.

5. A physical model for the fridge

Here we set up a physical model which allows us to predict the ultimate temperature and the cooling power of a ^3He fridge of the kind we have described in this paper.

The pressure in the evaporator can be derived from standard pressure drop formulas¹⁴. The infinitesimal pressure drop dP in a tube length element $d\ell$ is

$$-dP = \frac{G^2 d\nu}{\alpha} + \frac{2fG^2 \nu d\ell}{D} \quad (4)$$

where $G = \frac{4\dot{m}}{\pi D^2}$ is the mass flow per unit area, ν is the inverse of gas density, η is the gas viscosity, D is the inner diameter of the tube, $\alpha \sim 0.5$ in the case of laminar flow, and $f = \frac{16\eta}{DG}$ is the friction factor in the case of laminar flow. Equation 4 has been integrated in the evaporator and cryopump pumping tubes, using the average temperature as an approximation¹⁴ for $T(\ell)$, obtaining

$$P_1^2 - P_2^2 \simeq 2R\langle T \rangle \left[\frac{G^2}{\alpha} \log\left(\frac{\rho_1}{\rho_2}\right) + \frac{32G}{D^2} \int_0^L \eta(T) d\ell \right] \quad (5)$$

while in the heat exchanger section we can neglect the pressure drop. If we assume that ^3He is an ideal gas, and that it is in each point at the same temperature as the pumping line, we can compute the total pressure drop using eq.5.

If we are able to estimate the pressure inside the cryopump, from the total pressure drop computed above we can find the pressure in the evaporator. From this, by means of the standard vapor pressure curve of ^3He , we can get in turn the temperature of the liquid in the evaporator.

We estimate the pressure inside the cryopump P_{cp} (in Torr) by means of the formula¹⁵

$$\log_{10}(P_{cp}) = [153 \log_{10}(q + 112) - 450] \left(\frac{1}{T_g} - 0.016 \right) + 1.9 + \log_{10}(q) \quad (6)$$

where T_g is temperature of the grains (in K), q is the quantity of gas adsorbed per unit mass of charcoal (in $\text{cm}^3\text{STP/g}$). From our measurements of the mass of charcoal and of the ^3He charge we have $50 < q(\text{cm}^3\text{STP/g}) < 150$. Despite of the presence of the heat switch, the cryopump heats up significantly during cryosorption. During normal

operation our cryopump is $\sim 2K$ warmer than the heat exchanger. The larger the heat input on the fridge is, the larger is the mass of ^3He evaporated per unit time (\dot{m}), the larger is the adsorption work per unit time ($W = L\dot{m}$ with $L \sim 50J$), the larger is the temperature rise of the cryopump. In our system there is an additional difficulty, i.e. the fact that, due to modest thermal contact, the grains are not at the same temperature as the cryopump container. We measure the temperature of the container T_{cp} and we assume that there is an effective conductivity G_e between the grains and the container: so we have in the end

$$T_g - T_{cp} = L\dot{m}/G_e \quad (7).$$

Combining eq.5,6,7 allows us to find the evaporator temperature and the charcoal grains temperature as a function of the heat input. These computations can be compared to actual measurements of T_e and T_{cp} . We define a

$$\chi^2 = \sum_i \left(\frac{T_{e,meas}^i - T_e^i}{\Delta T_e^i} \right)^2 + \sum_i \left(\frac{T_{cp,meas}^i - T_{cp}^i}{\Delta T_{cp}^i} \right)^2$$

and we find the temperatures T_e and T_{cp} which minimize the χ^2 satisfying the previous equations set. The fit of the load curve data with the model outlined above are shown in fig.4. A remarkable agreement is evident for loads up to $100 \mu W$. The temperature of the grains is around 10 K in the same range. The parameter which best fits the measured data is $G_e \sim 2mW/K$. This is the effective conductivity of all the grains in the pump. At high ^3He evaporation rates the gas convection inside the cryopump becomes comparable to the conduction, and the value of G_e in eq.7 increases significantly. This explains the deviation of measured data from the simple model outlined in eq.7. The model is valid, however, in the full interesting range of heat loads $\dot{Q} \lesssim 100\mu W$ expected for the normal operation of the fridge.

6. Conclusions

We have developed a two weeks hold time, single shot ^3He fridge, suitable for stratospheric balloon payloads. The system has been optimized to be reliable, heavy duty, and simple to operate. The ultimate temperature is 0.28K (no additional loads, main bath at 1.7 K) and 0.30K with $30\mu W$ additional load (main bath at 1.7 K). The stability of the ultimate temperature is better than 0.1 mK/hour, and $0.1 \mu K/\sqrt{Hz}$ short term (0.02 to 10 Hz). The system has been flown on the BOOMERanG payload in two short flights, on Aug.12, 1997 and on Aug.29, 1997, from the National Scientific Balloon Facility of Palestine (TX, USA). The measured performance in flight is consistent with the specifications and the laboratory tests.

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8. Figure Captions

Figure 1: Section of the ^3He fridge. The drawing is to scale, the reference quote is in mm. CP is the cryopump, HE the heat exchanger (where ^3He is condensated), E is the evaporator. B marks a brazing, W marks a weld. The detail on the right shows the top view of the reed bolt joint, the thermal connection between the fridge and the ^4He cryostat. The joint is clamped by a brass M8 bolt.

Figure 2: A typical condensation process. The cycle starts with the cryopump cold (bottom left in the figure) and the condensation point at 1.6 K. The cryopump heater is switched on and starts to dissipate 0.35 W. In about 2 hours the cryopump reaches 35 K (top right). We maintain the temperature between roughly 34K and 36K for one hour by switching on and off the heater. After two hours, the heater is switched off, and the cryopump starts to drift towards lower temperatures. The heat switch is closed when the cryopump reaches 30K. The subsequent pump down is four hours long, and the final temperature of the cryopump is $\sim 2\text{K}$, with the ^3He temperature in the evaporator at 280 mK.

Figure 3: Typical temperature stability of the refrigerator cold flange in steady conditions. The drift is of the order of $100 \mu\text{K}/\text{h}$.

Figure 4: Experimental load curve of the fridge (panel A, data points). The data are compared to the best fit physical model predictions (continuous lines, see text) for loads lower than $100 \mu\text{W}$.

9. References

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FIG.1

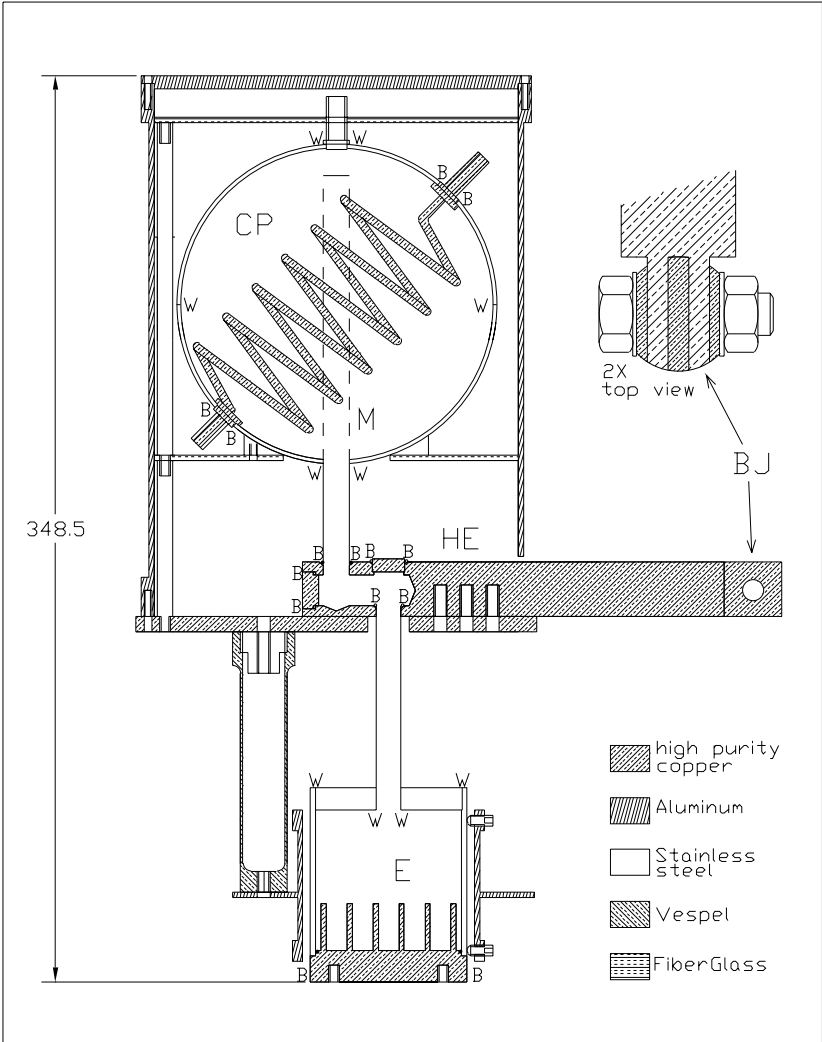
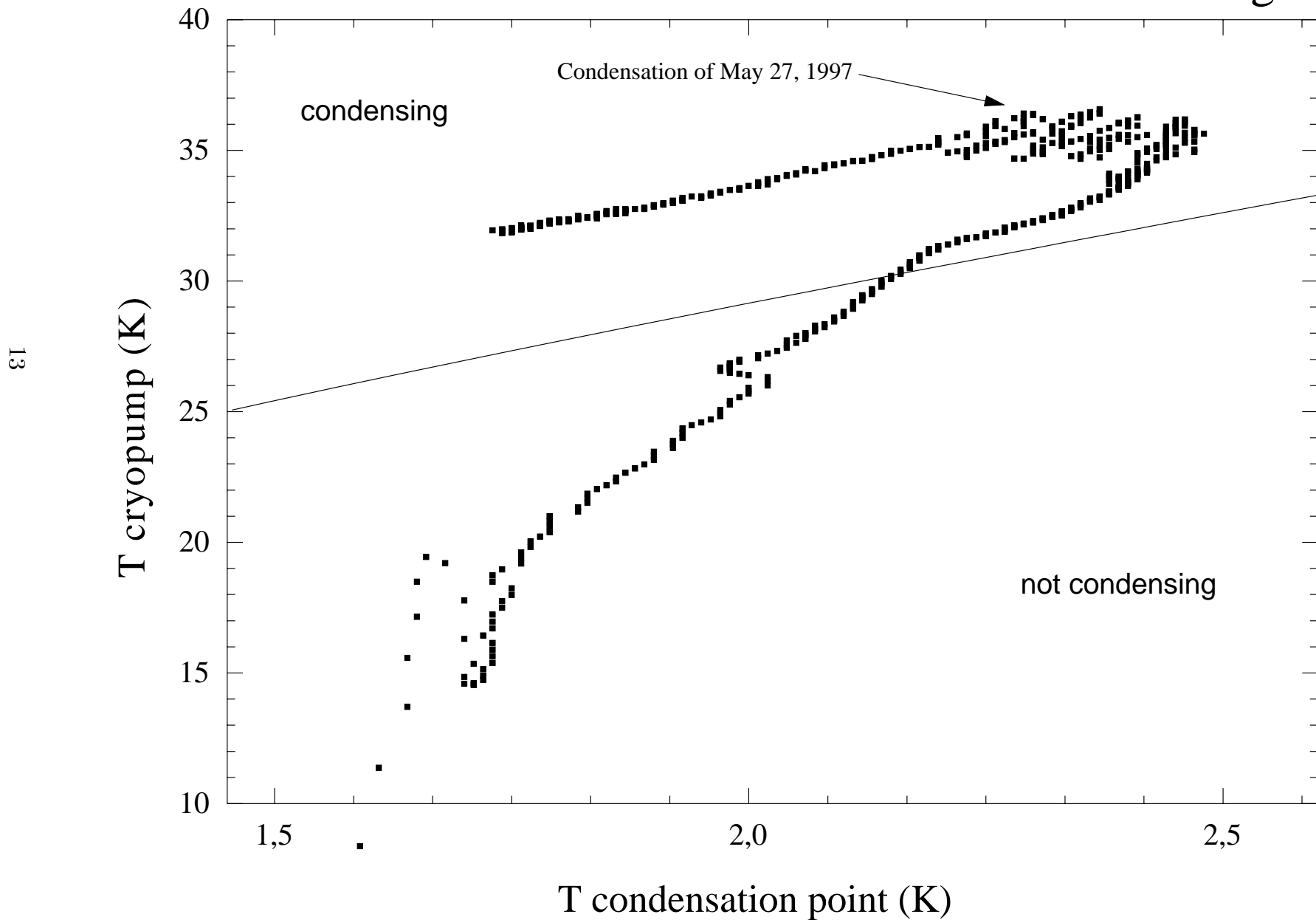


Fig.2



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Fig.3

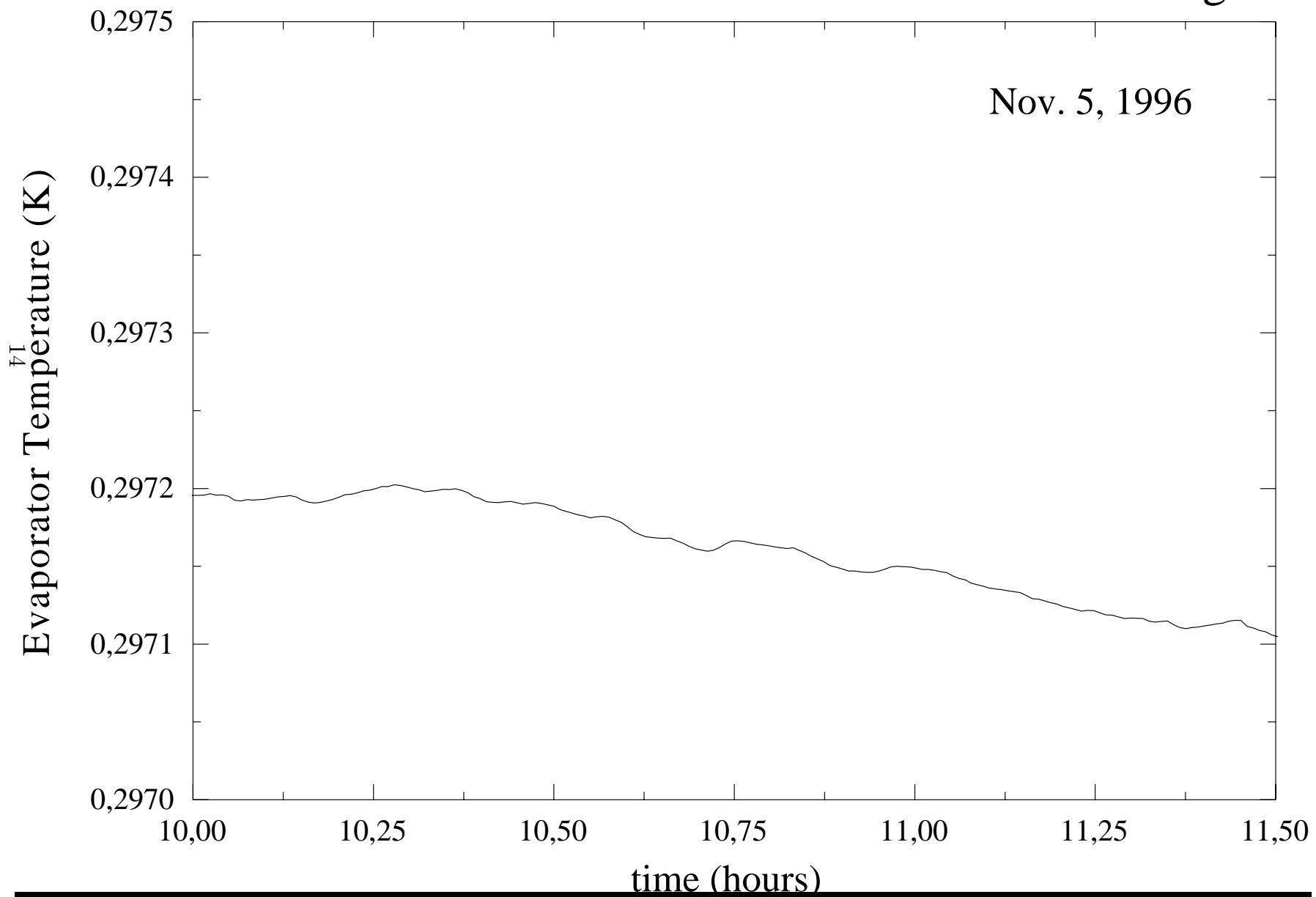


Fig.4

