Research paper

Systematic approach to determine the transient cooling power and heat leak of a commercial pulse tube cryocooler

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ABSTRACT

In a typical pulse tube cryocooler application, modeling the cool-down dynamics of a load attached to the cold tip require knowledge of the transient cooling power. Although this data may be calculated for in-house developed cryocoolers, the only data provided in commercial off-the-shelf cryocoolers is the steady state cooling power. In this paper, we show a systematic approach to derive transient cooling power data by performing heating and cooling experiments. The data obtained is verified with cooling data of an attached mass, which showed a good agreement. We also present a method to determine the parasitic heat load in a steady and transient operation of a commercial pulse tube cryocooler.

1. Introduction

The usage of cryocoolers once a workhorse for detector technology is branching into new application areas. To name a few, a device cooled by a pulse tube cryocooler to snap freeze human tissues for cancer diagnosis is developed by our team [1]. Recently, conduction cooling of superconducting radio frequency cavities for particle accelerators using a cryocooler was successfully demonstrated [2,3]. Owing to low vibrations, there is an increasing interest in using a pulse tube cryocooler in high definition microscopy [4]. In addition to these new applications areas, commercial-off-the-shelf cryocooler is also assessed in the detector community [5].

The data often provided by a cryocooler manufacturer are the load curve and the no-load cool-down of the cryocooler. The load curve is the net available steady-state cooling power as a function of cold tip temperature. The cool-down characteristic is a plot of temperature measured at the cold heat exchanger as a function of time. The details such as the cold heat exchanger mass, transient cooling power, other internal geometry, and material parameters are often not provided as it is proprietary information of the manufacturer. However, in order to calculate the cooling dynamics in an application where a load is attached to the cryocooler, the data provided by the manufacturer is insufficient. For example, in a typical application using the steady-state cooling power values to determine the cool-down time of a mass attached to the cold-end will lead to a significant error.

A study by Grossmann et al. [6] reports a significant difference in the net steady-state and net transient cooling power values for a cryocooler. The reason for this difference is not clearly mentioned but is suspected due to different regenerator temperature profiles that lead to different parasitic heat loads at the cold-end. This is an interesting outcome, as to date, transient effects inside the regenerator are not considered for the parasitic heat load calculations. For the steady-state operation, parasitic heat load values are usually approximated from transient measurements using a methodology reported by Vanapalli [7]. In this approach, first, the cooler is cooled to its minimum no-load value, followed by turning off the compressor and allowing it to warm up while applying different amounts of heat input at the cold-end. The warm-up rate of the cold heat exchanger is then calculated using the temperature-time measurements. For any particular cold end temperature value, the parasitic heat load is obtained by extrapolating the warm-up rate to zero.

Earlier attempts to determine the intrinsic cool-down time of a cryocooler were to attach several cold masses on the cold heat exchanger and measure the cool-down time. From these temporal temperature measurements and the procedure explained by Grossman et al. [6] and Vanapalli [8], the intrinsic cool-down time is derived. This method also allows the calculation of cooling transients with any load attached to the cold heat exchanger. However, it is to be noted that in the previous reported work, the inner dimensions of the cryocooler and the details of the components such as the heat capacity of the cold heat exchanger, porosity of the regenerator, effective cross-sectional area of the cold finger are known.

The objective of this paper is to show an experimental approach to determine the transient cooling power and parasitic heat leaks of a...
commercial off-the-shelf pulse tube cryocooler. First, we perform a pulse heating experiment to determine the effective cold mass of the cold finger, followed by cool-down with several continuous heating values at the cold heat exchanger. The parasitic heat leaks are determined from heating experiments after the cryocooler has attained its lowest temperature. The tissue snap freezer developed by us consisted of a 1.25 kg copper mass attached to the cold heat exchanger of a 3.8 W at 80 K pulse tube cryocooler. The cooling of this mass is compared with the model that uses the transient cooling power determined in the above diagnostic experiments.

2. Materials and procedure

A pulse tube cryocooler of the type LPT9310 manufactured by Thales Cryogenics is used in our experiments. The compressor and the warm end of the pulse tube are forced air cooled by a set of two fans, one for the compressor and another for the inertance buffer. The power electronics supplied with the cryocooler, which is proprietary, allows setting the desired cold tip temperature or the input voltage to the compressor. In all our experiments the input voltage to the compressor is set to a constant value. The temperature of the cold heat exchanger is measured using a silicon diode sensor, which has a maximum error of ±0.5 K for the temperature range of 54 K to 300 K. An electrical resistance of 22 Ω is used to input heat at the cold heat exchanger. The regenerator and the cold heat exchanger are made up of stainless steel - 304 (SS304) and copper material, respectively, for which temperature-dependent thermal properties are shown in Fig. 1.

The experiments performed are summarized in Table 1, and the results are discussed in the next section.

3. Results and discussion

3.1. Effective cold mass at the cold heat exchanger

The heater attached to the cold heat exchanger is triggered with a heat pulse of duration 15 s and an electrical power of 35 W. The heat diffusion time into the regenerator \( t_{\text{diff}} \approx (L^2/\alpha) \) is \( 10^3 \) s, which is two orders of magnitude larger than the heat pulse duration. Therefore, to a good approximation, the heat diffusion to the regenerator may be neglected. The temperature increase of the temperature sensor attached to the cold heat exchanger is shown in Fig. 2. The data is fairly constant in the duration of the pulse except at the initial and final parts of the pulse, which is due to the limitation of the power supply, namely, the slew rate of the control electronics. The effective cold heat exchanger mass determined using this method is in the range of 130–135 g.

3.2. Steady and transient cooling power

The cooling power available at the cold heat exchanger is discussed in this section for two cases, namely ‘steady’ and ‘transient’. In the steady situation, the cooler is cooled to its lowest attainable

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**Nomenclature**

**Greek symbols**

- \( \alpha \): thermal diffusivity
- \( \Delta \): difference
- \( \lambda \): thermal conductivity
- \( \rho \): density

**Subscripts**

- \( \infty \): warm-end
- \( cc \): cold-end while cool-down
- \( c \): cold-end
- \( cw \): cold-end while warm-up
- \( Cu \): copper
- \( d \): diffusion
- \( eff \): effective
- \( h \): heater
- \( p \): parasitics

**Superscripts**

- \( s \): steady-state
- \( t \): transient

**Symbols**

- \( A \): area
- \( c \): specific heat capacity
- \( D \): diameter
- \( L \): length
- \( m \): mass
- \( Q \): heat load
- \( T \): temperature
- \( t \): time

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temperature, after which a controlled heat load is applied using an electric heater mounted on the cold heat exchanger. The cooler warms up to a steady temperature, which is recorded. The measured, steady cooling power and cold heat exchanger temperature are shown in Fig. 4.

On the other hand, the transient cooling power is determined in two scenarios, during cool-down and warm-up of the cooler. During cool-down, the compressor input voltage is maintained at a constant value, and during warm-up, the compressor is turned off. The measured cold heat exchanger temperature values during cool-down with a steady heating power of 0 W, 2 W, and 4 W with a constant compressor power are shown in Fig. 5. In the same figure, the temperature-time derivative is also shown. The derivative is obtained from a curve fitted to the temperature-time data. The cooling power \( Q \) is determined from Eq. 1; in this case, the cold mass \( m_c \) of 130 g determined above is used. Fig. 4 (a) shows the transient cooling power as a function of temperature, which is fairly linear. The dip seen in the data points at room temperature is similar to the observation by Grossman et al. [6], which could be due to the start-up of the cooler. However, due to a lack of instrumentation in our commercial cooler, we cannot ascertain the reason for this behavior. A fit for all the data points for the transient cooling power values shows a linear trend (see Fig. 4 (a)). Extrapolating the transient cooling power fit to room temperature we could obtain the expected cooling power for the transient case.

Fig. 2. (above) Photos of the experimental set up. (below) Schematic representation of the cold finger. Here, \( L = 80 \) mm and \( D = 25.2 \) mm.

In an earlier work by Grossman et al. [6], the data presented in Fig. 8 in their publication, shows an opposite trend, where the transient cooler temperature, after which a controlled heat load is applied using an electric heater mounted on the cold heat exchanger. The cooler warms up to a steady temperature, which is recorded. The measured, steady cooling power and cold heat exchanger temperature are shown in Fig. 4.

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The transient cooling power during cool-down is lower than the steady-state values for a corresponding cold temperature. This trend is expected because during cool-down the temperature profiles develop in the regenerator from the cold end towards the warm end, resulting in a higher temperature gradient at the cold-end during transient cool-down and thus higher heat leak to the cold-end. A linear fit to the measured, steady-state cooling power data is also shown in Fig. 4 (b). At room temperature, the steady and transient curves should meet, which is not the case if a linear interpolation is assumed for the measured steady-state cooling power values. Using the measured cooling power values and the transient cooling power value at the room temperature, a second-order polynomial fit is obtained as the best fit for all data points. The reduced steady-state cooling power at lower temperatures is expected due to the non-linear effects in the regenerator material.

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**Table 1**

| Exp. No. | Initial state of the cooler | Heater input | Compressor | Objective |
|----------|-----------------------------|--------------|------------|-----------|
| 1        | Cooled to no-load temperature | pulse, 23 W, 15 s | OFF | Determine intrinsic cold mass |
| 2        | Room temperature | 0, 2, 4 W | ON | Transient cooling power during cooldown |
| 3        | Set cold temperature | several values | ON | Steady state cooling power |
| 4        | Cooled to no-load temperature | 2, 4, 8, 10.8 W | ON | Transient cooling power during warm-up |
| 5        | Cooled to no-load temperature | 0, 2 W | OFF | Effective cold finger cross-sectional area |
| 6        | Room temperature | – | ON | Cooling of a 1.25 kg copper mass attached to the cold heat exchanger |

**Fig. 3.** Cold heat exchanger temperature \( T_c \) variation, its temporal derivative \( \frac{dT_c}{dt} \) and estimated values of the effective copper mass \( m_c \) using heat pulse experiments with the cryocooler.
power is larger than the steady-state cooling power. In their work, a temperature averaged specific heat capacity of copper is used to compute the heat capacity of the cold heat exchanger. However, specific heat capacity decreases rapidly at low temperature (see Fig. 2). Their data is further analyzed here to determine the affect of including a temperature-dependent heat capacity on the outcome of their work. The temperature-time data points are extracted using image analysis followed by the interpolation of extracted data points. A four-term Fourier series interpolation fits all the curves very well. The built-in mass (copper, stainless steel flanges, and screws) of the cold heat exchanger is 26.40 g, and the excess copper masses used for the experiments are 9.18 g, 42.10 g, 51.32 g, 83.73 g, and 92.91 g. The weights for the stainless steel flanges and screws in the cold heat exchanger are not mentioned, and therefore for simplification, the cold heat exchanger is assumed to be made up of only copper material. The calculated transient cooling power $Q_{tc}$ values during cool-down for all the six cases are now compared with the measured steady-state cooling power $Q_s$ values in Fig. 6. The shape of the transient cooling power curve for all the six cases differs from that reported by Grossman et al. [6] and is very similar to the shape of the $Q^t$ curve. In addition, similar to the results presented in Fig. 4 (a), the $Q_{tc}$ values for all six cases are slightly lower than the measured $Q^t$ values. The difference between the $Q_{tc}$ values among six cases is suspected due to the limited number of data points and also due to the inaccuracy associated with the data extraction using the image.

Fig. 4. Steady-state (a) and transient (b) cooling power values for different cold-end temperature values.

Fig. 5. Cold heat exchanger temperature $T_c$ variation and its temporal derivative $dT_c/dt$ during cool-down experiments performed with different heat load $Q_h$ values at the cold heat exchanger.

Fig. 6. Transient cooling power values calculated from the temperature-time data reported by Grossmann et al. [6] are compared with the corresponding steady-state cooling power values.
analysis, resulting in slightly in-accurate interpolation for the temperature-time curve, especially at temperature values close to the room temperature.

The transient cooling curves derived in this paper show a linear dependence, whereas that derived from Grossmann et al. [6] are not linear, which could be due to how the experiments are performed. In our case, the compressor power is kept constant, whereas, in the other work, the pressure ratio is kept constant. Nevertheless, using a simple experimental technique of a pulse heating experiment, followed by a cool-down experiment, we demonstrated a method to determine the transient cooling power of a commercial cooler where the dimensions of the cold heat exchanger are unknown.

To calculate the transient cooling power during warm-up $Q_{cw}$, experiments are performed for different values of the external heat load $Q_b$ at the cold end and with the compressor turned on. The net transient cooling power during the warm-up is calculated using Eq. 1. The temperature-time measurements from the experiments are shown in the Fig. 7a and the calculated values transient cooling power are shown in the Fig. 7b. It can be seen that the transient cooling power values during the warm-up experiments are different from the cool-down experiments and also are slightly higher compared to the net steady-state cooling power values. The higher $Q_{cw}$ values are suspected due to lower parasitic heat load values compared to the steady-state conditions. For $Q_b = 2$ W, the warm-up rate is lowest. This would give more time for the heating front from the warm end to diffuse towards the cold end of the regenerator and, therefore, would result in the temperature profile, which is closest to the steady-state conditions. Now, for higher $Q_b$ values, the warm-up rate of the cold heat exchanger is higher, and therefore, the diffusion time for the heating front to move towards the cold-end of the regenerator will be lower. This would result in slightly lower temperature values close to the regenerator’s warm end than the steady-state case and, therefore, lower parasitic heat load values. In conclusion, the temperature profiles inside the regenerator during the transient cool-down/warm-up experiments are different from the steady-state case, resulting in different parasitic heat load values.

3.3. Background heat losses

In this section, a procedure to determine the background losses or parasitic heat leak is explained. It must be noted that in this discussion, only heat conduction through the regenerator is accounted for, which is a major source of heat leak. First, heat leak for steady-state conditions $Q_{lc}^s$ is derived, followed by heat leak during the transient cool-down $Q_{lc}^t$ of the cryocooler.

a) Steady state heat leak

The heat flux through the regenerator $Q_c^s(T_c)$ spanning a temperature of $T_c$ and $T^*$ is given by,

$$Q_c^s(T_c) = \frac{1}{T^*} \int_{T_c}^{T^*} \lambda(T) dT \tag{2}$$

The length of the regenerator in our case is 80 mm resulting in a heat flux value of $3.25 \times 10^4$ W m$^{-2}$ for $T_c = 80$ K and $T^* = 295$ K. To determine the heat leak $Q_{lc}^s$, the effective cross-sectional area $A_{eff}$ should be known. The procedure adopted to determine the effective area is as follows: First, the cooler is allowed to attain a steady-state cold temperature. The electric power to the compressor is turned off, and the cold heat exchanger temperature is measured during warm-up. In the second set of experiments, also a constant heat is supplied by a heater equal to 2 W. Fig. 8 shows the temperature rise in both cases. In the first minute during the warm-up, the temperature rise is less than 25 K from the steady-state value. We assume that the regenerator temperature profiles during this short time scale are not much different from the steady-state situation. Following this, we can write,

$$Q_{lc}^s(T_c) = Q_c^s(T_c) = m_e, (T_c) \frac{dT_c}{dt}; Q_b \tag{3}$$

where $Q_b = 0$ and 2 W for the two cases. The effective cross-sectional area is calculated from this equation, which is also shown in the snippet of Fig. 8. The effective area of 133 mm$^2$ is 21% of the cold end cross-sectional area. This value is reasonable, given the reduced area due to the pulse tube. The steady-state parasitic heat leak $Q_{lc}^s$ as a function of cold heat exchanger temperature is shown in Fig. 9. The gross cooling power obtained as a sum of the parasitic heat leak and the net cooling power is also shown in the Fig. 9.

b) Transient heat leak

The gross cooling power of a pulse tube cryocooler is the product of the dynamic pressure amplitude, volume flow, and the phase difference between them. We assume that for a particular cold temperature, the gross cooling is not influenced by the temperature profiles in the regenerator. Therefore, the gross cooling calculated for the steady case will be the same for the transient case. The net transient cooling power during the cool-down $Q_{lc}^t$ is obtained earlier. The parasitic heat leak is the difference between the gross cooling power and the net transient cooling power shown in Fig. 9. The parasitic heat leak is higher in the transient operation compared to the steady-state as expected due to larger temperature gradients in the regenerator, which is derived next.
Temperature profiles in the regenerator

Assuming a one-dimensional temperature distribution in the regenerator, the temperature at any position along the regenerator’s length can be obtained from the parasitic heat leak calculated above. The temperature at the warm end of the cooler is \( T_w \), and at the cold side, the temperature is \( T_c \). Using these temperature boundary conditions and the parasitic heat leak for steady-state and transient conditions, the temperature at any location along the length of the regenerator is derived using,

\[
-A_{\text{eff}} \int_{T_c}^{T_w} \frac{dT}{dx} = Q_p(T_c) \int_0^L dx \tag{4}
\]

The calculated steady-state temperature profiles are shown in the Fig. 10, which appear non-linear due to the temperature-dependent thermal conductivity of the regenerator material. To determine the temperature profile during the transient cool-down, a cubic temperature profile inside the regenerator is considered, \( T(x) = a + bx + cx^2 + dx^3 \). A cubic polynomial is chosen due to the limited boundary conditions, which are shown below.

\[
T(x) = T_w \tag{5}
\]

\[
(T(x) = T_w) \left| \frac{dT}{dx} \right|_{x=0} = \frac{dT}{dx} \bigg|_{x=0} = 0 \tag{6}
\]

\[
T(L) = T_c \tag{7}
\]

\[
-A_{\text{eff}} \frac{dT}{dx} = Q_{\text{cc}}(T_c) \tag{8}
\]

For each setting of the cold temperature \( T_c \), the solution to the temperature profile is shown in Fig. 10. Comparing the steady-state and transient temperature profiles, the gradient at the cold-end in case of the transient case is larger than the steady case as one would expect.

It must be noted here that the warm-end of the cryocooler in the above discussion of background losses is taken as a constant value, \( T_w = 295 \) K. In reality, during the cool down, the warm-end rises in temperature and in our particular case to a maximum value of 310 K. To determine the impact of this temperature rise, we estimated the background losses using the procedure discussed above for \( T_w = 295 \) K and 310 K. The difference in the heat leak value is less than 0.25 mW. This data however, does not influence the estimation of transient or steady cooling power as the cold end temperature is used to determine these parameters.

### 3.4. Cool-down of an added mass to the cold-end

A copper mass of 1.25 kg is attached to the cold heat exchanger of the cryocooler with an aim to cool vials for snap-freezing tissues. The details of the system are given in our earlier publication [1]. Our aim is to predict the cool-down time of the cold mass as this is an important parameter to determine the preparation time of the system for subsequent snap-freezing procedures. Having determined the steady \( Q_c \) and transient \( Q_{\text{cc}} \) cooling power in previous sections, we will now check which of these powers better predict the cooling of the cold mass. The duration of the cooling to a particular cold temperature is,

\[
\int_0^t dt = \int_0^T \frac{m c_p}{Q_{\text{cc}}(T)} \frac{dT}{T} \tag{9}
\]

where, \( Q_{\text{cc}} = Q_c' \), and \( Q_{\text{cc}} = Q' \) (for values of \( Q_c' \), and \( Q' \) see Fig. 4). The temperature-time values for both cases are compared with the cool-down measurements in Fig. 11. The temperature variation using \( Q_c' \) values agrees well with the measurements, whereas, the calculated values using \( Q' \) values show a larger difference from experimental data.
In deriving the transient cooling power, we used Eq. 1, which require as input the cold mass $m_c$. Therefore, the question is why the agreement of our prediction with experimental data is rather good in spite of a large cold mass (1.25 kg) attached to the cooler. To throw light on this issue let us consider the diffusion time inside the regenerator material, which is, $t_c = (L^2 / \alpha) \times 10^3$ s. Now, let us estimate the cooper mass for which the temperature drop ($\Delta T$) in $10^3$ s is negligible. For calculations, we assumed $\Delta T = 1$ K, heat capacity 200 J·kg$^{-1}$·K$^{-1}$ at 80 K and a transient cooling power at 80 K equal to 2.86 W. The calculated value of the copper mass is about 14 kg. This value is an order in magnitude larger than the attached mass in our experiments. With such a large heat capacity, the regenerator temperature profiles will be similar for both steady-state and transient cases. This results in the same cooling power values for both cases. The maximum possible error value, which is the difference between the cool-down time calculated using steady-state and transient cooling power values compared to the transient case, is also shown in the Fig. 11. The maximum error in predicting the cool-down time increases with the decrease in the cold-end temperature and is roughly 14% for a temperature of 80 K.

4. Conclusion

We presented a systematic approach to determine key operational parameters of a commercial off-the-shelf pulse tube cryocooler.

- By releasing a short heat pulse on the cold heat exchanger, we make use of the fact that the diffusion time in the regenerator is much larger, and all the energy introduced is absorbed by the cold mass. From this experiment, the effective cold mass is determined.
- The temperature of the cold heat exchanger is measured during the cool-down of the cryocooler with and without constant heat input. The transient cooling power during cool-down is determined from the above derived cold mass, and the temperature-time derivative from these experiments. Note that the transient cooling power during cool-down is lower than the steady state cooling power.
- After attaining the lowest no-load temperature, steady heat input is applied to the cold heat exchanger causing the temperature at the cold heat exchanger to increase. The transient cooling during the warm-up is determined following the same procedure as above. Note that the transient cooling power during warm-up is larger than the steady state cooling power.
- Experiments with a copper mass of 1.25 kg attached to the cooler. The temperature-time predicted using transient cooling power agrees well with the experiments and not the steady cooling power data. This is due to the fact that the thermal diffusion time in the regenerator is longer than the cool-down speed of the cold mass. When a large mass is attached, to give a figure of merit 14/2.86 kg W$^{-1}$ mass to cooling power ratio, the cool-down of this heat capacity is of the same order of magnitude as the regenerator. In this case, steady state cooling power can be used to determine the cool-down dynamics.
- We showed that parasitic heat leaks in the transient and steady state can be determined from warm-up experiments. The heat leak is a useful parameter to benchmark the performance of a cryocooler.

CRediT authorship contribution statement

S. Jagga: Investigation, Software, Writing - original draft, Validation. H.J. Holland: Investigation. S. Vanapalli: Conceptualization, Methodology, Writing - review & editing, Supervision, Project administration, Funding acquisition.

Fig. 11. Measured and predicted temperature-time values during cool-down of a copper mass of 1.25 kg attached to the cold-end of the cryo-cooler. Error when using transient and steady cooling power $\left(\frac{\text{Temperature}}{\text{Temperature}}\right)$, which is the difference between the cool-down time calculated using steady-state and transient cooling power values compared to transient case is shown.

Declaration of Competing Interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

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