Cryogenics xxx (2016) xxx–xxx

Contents lists available at ScienceDirect

Cryogenics

journal homepage: www.elsevier.com/locate/cryogenics

Research paper

Development of a compact cryocooler system for high temperature superconductor filter application

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ARTICLE INFO

Article history:
Received 20 January 2016
Received in revised form 12 May 2016
Accepted 2 June 2016
Available online xxxx

Keywords:
Thermoacoustic theory
Co-axial pulse tube cryocooler
Power density
High frequency

ABSTRACT

Seeking a higher specific power of the pulse tube cryocooler is an important trend in recent studies. High frequency operation (100 Hz and higher), combined with co-axial configuration, serve as a good option to meet this requirement. This paper introduces a high efficiency co-axial pulse tube cryocooler operating at around 100 Hz. The whole system weighs 4.3 kg (not including the radiator) with a nominal input power of 320 W, namely, power density of the system is around 74 W/kg. The envelop dimensions of the cold finger itself is about 84 mm in length and 23 mm in outer diameter. Firstly, numerical model for designing the system and some simulation results are briefly introduced. Distributions of pressure wave, the phase difference between the pressure wave and the volume flow rate and different energy flow are presented for a better understanding of the system. After this, some of the characterizing experimental results are presented. At an optimum working point, the cooling power at 80 K reaches 16 W with an input electric power of 300 W, which leads to an efficiency of 15.5% of Carnot.

1. Introduction

Superconductivity has, over the century since its discovery by Kamerlingh Onnes in 1911, found important applications such as generating high magnetic field for scientific research and high sensitivity SQUID device [1]. In 1980s, the discovery of high temperature superconductors (HTS) led to a new boom for superconductivity research. As people are trying to extending the application fields for HTS [2,3], HTS filter is one of the most attractive applications. For this, small-scale and easy-to-be-packaged cryocooler plays an important supporting role. Among several options, Stirling or Stirling type pulse tube cryocooler are good candidates [4,5]. Especially, the pulse tube cryocooler (PTC) is famous for its advantages such as low vibration, high reliability, robustness and low cost due to absence of the moving parts at the cold end. However, existence of a hollow pulse tube and reservoir brings down the system specific power if compared with Stirling cryocooler [6]. A natural choice to make up for this disadvantage is to increase the working frequency [7]. Lots of studies on this have been reported since 2005. In 2007, NIST in America reported a 120 Hz in-line pulse tube cryocooler [8], which could obtain 3.55 W cooling power at 80 K and the relative efficiency of Carnot reached 19.7% based on input acoustic power. NGST developed a high frequency co-axial pulse tube cryocooler in 2009 [9]. The cooling power at 77 K was 1.3 W with 46 W electric power input under 124 Hz operating frequency. The estimated mass of the mechanical cooler in a flight configuration is only 900 g, including the drive electronics. Gan et al. developed a 120 Hz pulse tube cryocooler in 2009 [10]. A cooling power of 8.0 W at 78.5 K as well as a rapid cooling down to 79.8 K in 5 min was achieved.

The research of 100 Hz PTCs at our laboratory has been carried out since 2007. The paper [11] designed an in-line PTC driven by a single piston linear compressor, the no-load temperature reached 59.6 K. In 2009, an in-line PTC combined with a single-piston linear compressor was designed and experimented. A no-load temperature of 31.8 K and a cooling power of 12.5 W at 77 K were obtained with 185.2 W electric power input, the corresponding relative Carnot efficiency of the whole system reached 18.9%, which was the highest efficiency reported at that time [12].

Although the efficiency of the in-line PTC in paper [12] is high, the whole system weighs about 8.9 kg, which is far more than that of a Stirling cryocooler with a similar cooling performance. Meanwhile, in in-line configuration, the cold end heat exchanger is located in the middle of the cryocooler, which is not suitable for practical applications. Therefore, further development is

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http://dx.doi.org/10.1016/j.cryogenics.2016.06.004
0011-2275/

needed. This paper introduces an important upgrade of our previous system, which is of co-axial configuration and weighs only 4.6 kg. The compactness and specific power of the system is much improved. In the following sections, the numerical model is firstly presented. Secondly, some simulation results are discussed. Thirdly, the system configuration is briefly introduced and characterizing experimental results are given. Finally, some conclusions are drawn.

2. Numeric method

2.1. Numeric model

The simulation model employed to design the cold head in this paper is based on thermoacoustic theory. According to classic thermoacoustic theory, the momentum, continuity and energy equations are as follows [13,14].

$$dU_1/dx = -\frac{i\omega A}{\gamma p_m}[\xi_c + (\gamma - 1)\xi_k f_k] \cdot p_1 + Z \cdot U_1 \tag{1}$$

$$dp_1/dx = -(i\omega\xi_l l + \xi_v r_v)U_1 \tag{2}$$

$$dH/dx = -Q \tag{3}$$

where p_1, U_1 are the pressure and volume flow rate, respectively. ω is the angular frequency. p_m is the mean pressure. f_k is related to the configuration and dimensions of the flow channel. Z is acoustic impendence which is a function of the temperature gradient. r_v, l represent the viscous resistance and inertance, respectively. H is the total energy flow. Q is the heat transfer between each component and the surroundings. Considering the turbulent flow in practical pulse tube cryocooler, four correction factors $\xi_c, \xi_k, \xi_l, \xi_v$ are introduced in Eqs. (1) and (2). Details of these formulations can be found in [13–15]. The calculation algorithm can be found in Ref. [16].

The governing equations for linear compressor are

$$\hat{E} - BL\hat{u} = \hat{I}(R_e + i\omega L_e) \tag{4}$$

$$\hat{I}BL = \hat{p}A + k\frac{\hat{u}}{i\omega} + R_m\hat{u} + mi\omega\hat{u} \tag{5}$$

where $\hat{E}$ is applied electric voltage, BL is force factor, $\hat{u}$ is the piston velocity, R_e and L_e are resistance and inductance of the motor coil, respectively. $\hat{p}$ is the pressure wave at the piston front side. k is spring constant. R_m is the damping coefficient. Details of the calculation algorithm of matching compressor and cold head can be found in Ref. [17].

2.2. General procedures in design

Firstly, systematical calculations have been done to optimize the dimensions of the 100 Hz co-axial cold head with the thermal efficiency as the primary target. Considering the possible discrepancy between simulation and experiments from previous experiences, some margins on both efficiency and cooling power are kept. Final dimensions of the cold head, listed in Table 1, are determined with some practical considerations due to the co-axial configuration.

Table 1
Main parameters of the pulse tube cryocooler.

Components	Diameter (mm)	Length (mm)
Main ambient exchanger	i.d. 20	30
Regenerator	Outside diameter, 22.5 Inside diameter, 10.4	35
Cold end heat exchanger	i.d. 20	8.4
Pulse tube	10	54
Secondary ambient heat exchanger	10	5
Inertance tube	3	1500
Reservoir	130 cm^3	

Once the cold head design is determined, based on the cold head acoustic impedance and the compressor mechanical governing equations, i.e. Eqs. (4) and (5), parameters of the linear compressor are calculated and determined as shown in Table 2.

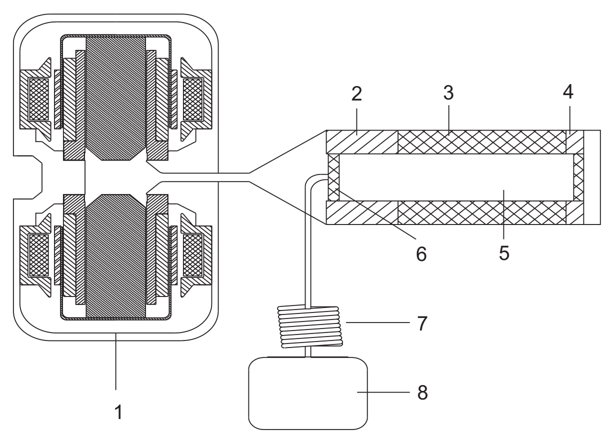
Table 2
Details of the compressor parameters.

BL (N/A)	R_e (Ω)	R_m (N s/m)	M (kg)	D (mm)	L_e (mH)	K (kN/m)	Void volume of the front chamber (cm^3)
16	0.43	14	0.32	22	0.2	62.7	7

3. Simulation results

The geometric model is presented in Fig. 1. Temperatures of the ambient heat exchanger and the cold end heat exchanger are set at 308 K and 80 K, respectively. The mean pressure is set to be 3.5 MPa with helium and the operating frequency is set at 100 Hz. For the regenerator, 600-mesh stainless steel screens with a wire diameter of 18 μm are used.

According to the numerical model introduced in last section, the optimized performance is obtained with 300 W input electric power, and the optimized relative Carnot efficiency of the whole system is about 17.4%. In order to further understand the



1.compressor 2.main ambient heat exchanger 3.regenerator 4.cold end heat exchanger 5.pulse tube 6.secondary ambient heat exchanger 7.inertance tube 8.reservoir

Fig. 1. Schematic of a 100 Hz co-axial pulse tube cryocooler.

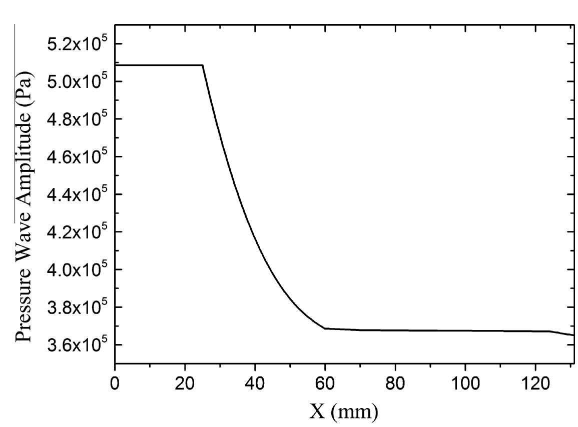


Fig. 2. Axial distribution of pressure wave amplitude.