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Research paper

Opposed piston linear compressor driven two-stage Stirling Cryocooler for cooling of IR sensors in space application

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ABSTRACT

A two-stage Stirling Cryocooler has been developed and tested for cooling IR sensors in space application. The concept uses an opposed piston linear compressor to drive the two-stage Stirling expander. The configuration used a moving coil linear motor for the compressor as well as for the expander unit. Electrical phase difference of 80 degrees was maintained between the voltage waveforms supplied to the compressor motor and expander motor. The piston and displacer surface were coated with Rulon an anti-friction material to ensure oil less operation of the unit. The present article discusses analysis results, features of the cryocooler and experimental tests conducted on the developed unit. The two-stages of Cryo-cylinder and the expander units were manufactured from a single piece to ensure precise alignment between the two-stages. Flexure bearings were used to suspend the piston and displacer about its mean position. The objective of the work was to develop a two-stage Stirling cryocooler with 2 W at 120 K and 0.5 W at 60 K cooling capacity for the two-stages and input power of less than 120 W. The Cryocooler achieved a minimum temperature of 40.7 K at stage 2.

1. Introduction

The most attractive feature of Stirling Cycle cryocoolers is its high COP. Stirling machines with linear motor drive are compact and require no valves for operation. Stirling coolers can be operated at high speeds and high pressures (cooling capacity being proportional to speed and pressure). This leads to a reduction in the specific mass (mass of cooler per unit cooling capacity). The use of very high effectiveness regenerative heat exchanger (with very large area density) also contributes to the compactness. The system operates on Closed Thermodynamic cycle using Helium or Hydrogen as working fluid [1]. Cyclic Analysis proposed by Atrey et al. [2] was used to simulate the performance of the proposed.

Thermoaccoustic theory to optimize the system operating and structure parameters. The Cryocooler achieved a minimum temperature of 27.6 K and cooling power of 78 W at 40 K with an electric input of 3.2 kW. Alan Caughley et al. [4] discuss a novel concept for free piston Stirling Cryocooler using pair of metal diaphragms to seal and suspend the displacer. The diaphragm allows the displacer to move without rubbing resulting in reduced friction and improved life. The Cryocooler achieved a minimum temperature of 56 K and 29 W of cooling at 77 K. de Jonge et al. [5] discussed analysis and optimization of linear motor for linear compressor, the said analysis has been used to design the linear motor for the present application.

Cryocooler in the present work. Cyclic analysis divides the Stirling Cycle in equal number of intervals pressure and volume of the compressor and expander vary with each interval. The ideal compressor power per cycle is estimated by integrating Compressor PV area for one cycle. The ideal cooling power is estimated by integrating PV area for the expansion process. Various losses (Pumping loss, Temperature swing loss, Shuttle conduction loss, regenerator loss, Pressure drop loss) are considered to predict actual Compressor power and cooling power. Xiaotao Wang et al. [3] discusses development of a two-stage high capacity free piston Stirling Cryocooler driven by linear compressor for HTS application. The paper The thermal analysis was carried out using Cyclic presents results from a numerical model based on

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2. Configuration of the Cryocooler

analysis proposed by Atrey et al. [2]. This analysis first estimates the ideal compressor power and ideal cooling power. The losses due to pressure drop in Regenerator, Connecting tube, dead volumes for the compressor power are added to the ideal power to predict the actual compressor power. The losses on the cooling side viz. Regenerator ineffectiveness, Shuttle conduction loss, Temperature Swing loss, Pumping loss are subtracted from the ideal cooling power to predict the actual cooling power. The input parameters used for achieving the objective of 2 W at 120 K at stage 1 and 0.5 W at 60 K at stage 2 and output parameters are shown below in the Table 2.1.

3. Hardware development

The optimized parameters shown in Table 2.1 were used to develop the Opposed piston driven Two-stage Stirling Cryocooler unit for testing. Moving coil linear motor was developed to drive the two pistons and displacer. The phase shift between the two supply voltages to compressor motors and expander motor was maintained using a two-channel Ac supply source so as to maintain the mechanical phase shift between the motion of piston and displacer. Piston was coated with Rulon which is an antifriction material for achieving sealing and oil less operation.

Flexure bearing were used to maintain the clearance between the piston and Cylinder to avoid leakage of the compressed gas. Compressor cylinder was fitted with a Cast Iron Liner to avoid wearing of the cylinder walls. A 15 μm radial clearance between the piston and the cylinder walls was maintained to avoid leakage of working fluid. Vacuum of the order of 10^{-6} torr in the ambient space of the cold tip was maintained using diffusion vacuum pump. Following figures shows important components of the Cryocooler.

Fig. 3.1 Shows Instrumentation used for the two-stage Cryocooler. A Two-channel power source was used to maintain the phase difference between the two power supply. LVDT was used

Table 2.1
Optimized input and output parameters of the two-stage Stirling Cryocooler.

Item	Value	Remarks
Input Parameters		
Frequency, Hz	40	
Operating pressure, bar	16	
Working fluid	Helium	
Compressor		Piston coated with antifriction material for oil less operation and sealing
Piston diameter, mm	22.5	
Total Stroke of the piston, mm	10	
Transfer tube or Connecting tube		
Diameter, mm	2	
Length, mm	200	
Regenerator		
Diameter		
Stage 1, mm	16	
Stage 2, mm	8	
Length		
Stage 1, mm	50	
Stage 2, mm	25	
Mesh Size	250–48	
Material	Phosphor Bronze	
Wire diameter, mm	41 × 10	
Cryo-Cylinder		Uses motorized displacer to maintain the required phase difference between the piston and displacer motion (80 deg)
Diameter		
Stage 1, mm	17	
Stage 2, mm	9	
Total Stroke of the expander, mm	3	
Total Cryocooler Dead Volumes, mm^3	5 × 10	
Output results from Cyclic Analysis		
Ideal RE (stage – I), W	8.9	Ideal predicted performance
Ideal RE (stage – II), W	3.5	
Ideal Power, W	46	
Net Losses (stage - I), W	5.1	Actual Predicted Performance
Net Refrigerating effect (Stage-I), W	**3.8**	
Net Losses (stage – II), W	1.8	
Net Refrigerating effect (Stage-II), W	**1.7**	
Net Losses Compressor side, W	29	
Actual power, W	**75**	
Motor parameters	**Compressor motor**	**Expander motor**
Magnet material	Nd-Fe-B	Nd-Fe-B
Length of the air gap, mm	6	4.5
Height of air gap, mm	15	5
Coil height, mm	20	6
Diameter of the copper wire, D_w mm	0.45	0.45
Total resistance of coil, $R_{T,}$ Ohm	16.5	1.4
Flexure Design	**Compressor**	**Expander**
Total stiffness required, N/m	24,000	15,400
Flexure thickness, mm	0.3	0.3
Material for flexures	Beryllium Copper	Beryllium Copper
Flexure diameter, mm	69	45

to monitor the instantaneous position of the compressor piston, expander displacer and there phase difference. Piezoelectric Pressure transducer was installed at the exit of the compressor unit in transfer tube to measure dynamic pressure discharged by the compressor.

Fig. 3.2 Shows Compressor Piston coated with antifriction material used for sealing and oil less operation of the compressor.

Fig. 3.3 shows the developed linear motor (Fig. 3.3a Magnet and Iron assembly, Fig. 3.3b Coil former with Copper winding). Fig. 3.4a and b shows integral Cryo-Cylinder and displacer made out of one piece two avoid rubbing friction arising by misalignment between the two-stages. Fig. 3.5 shows the developed flexure bearing of 0.3 mm thickness and 69 mm diameter for compressor and 45 mm diameter for expander. Figs. 3.6 and 3.7 show the cut section of the assembled compressor and expander unit.

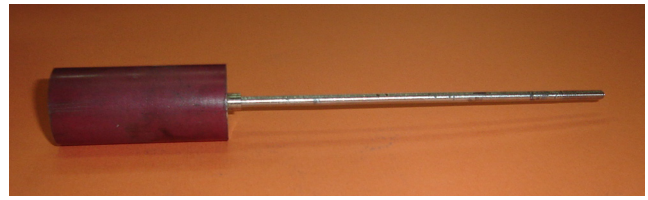


Fig. 3.2. Piston coated with anti-friction material.

4. Experimental results

Tests were conducted on the developed two-stage Stirling Cryocooler. Fig. 4.1 shows the results at charge pressure of 16 bar and operating frequency of 40 Hz. The minimum cool down temperature achieved at stage 2 was 40.7 K and 90 K at stage 1 with

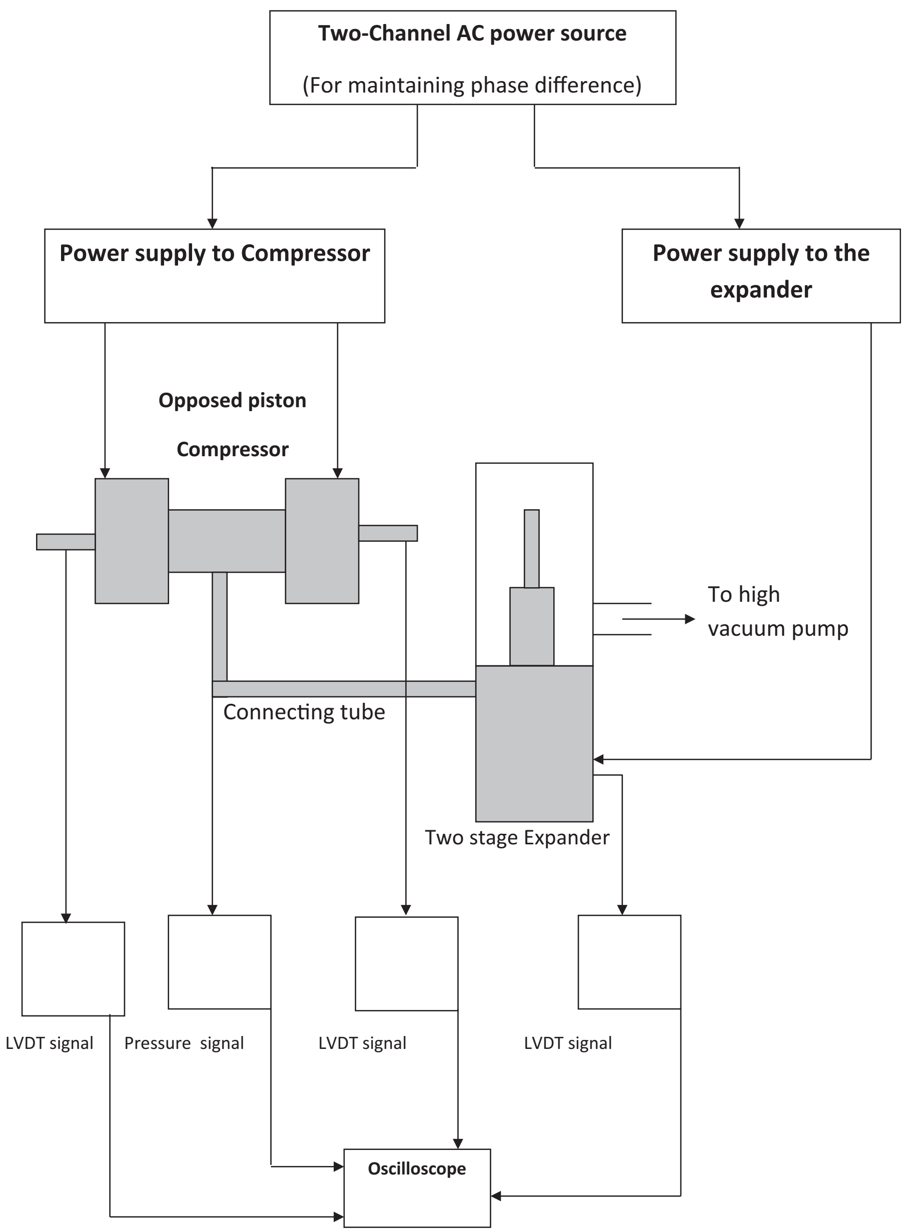
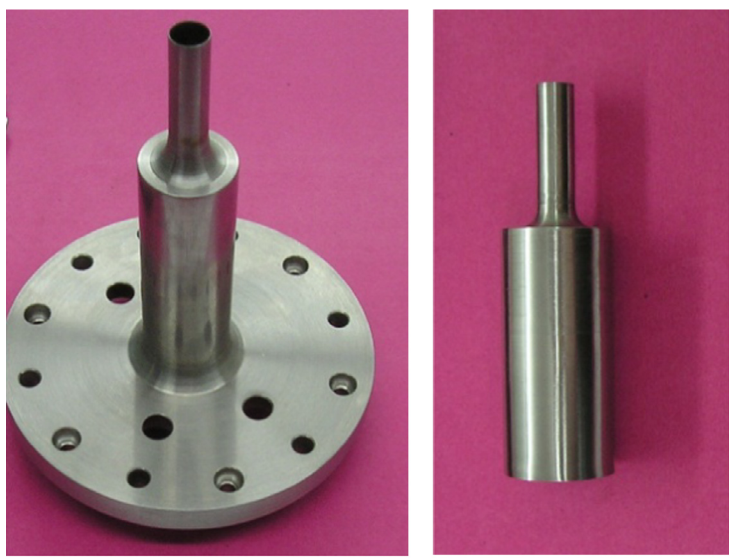


Fig. 3.1. Instrumentation for the two-stage cryo-cooler.

(a) (b)

Fig. 3.3. (a) Magnet and iron assembly. (b) Coil former and winding.



(a) (b)

Fig. 3.4. (a) Integral Cryo-Cylinder. (b) Integral Displacer.

Fig. 3.5. Expander and compressor flexure bearing.

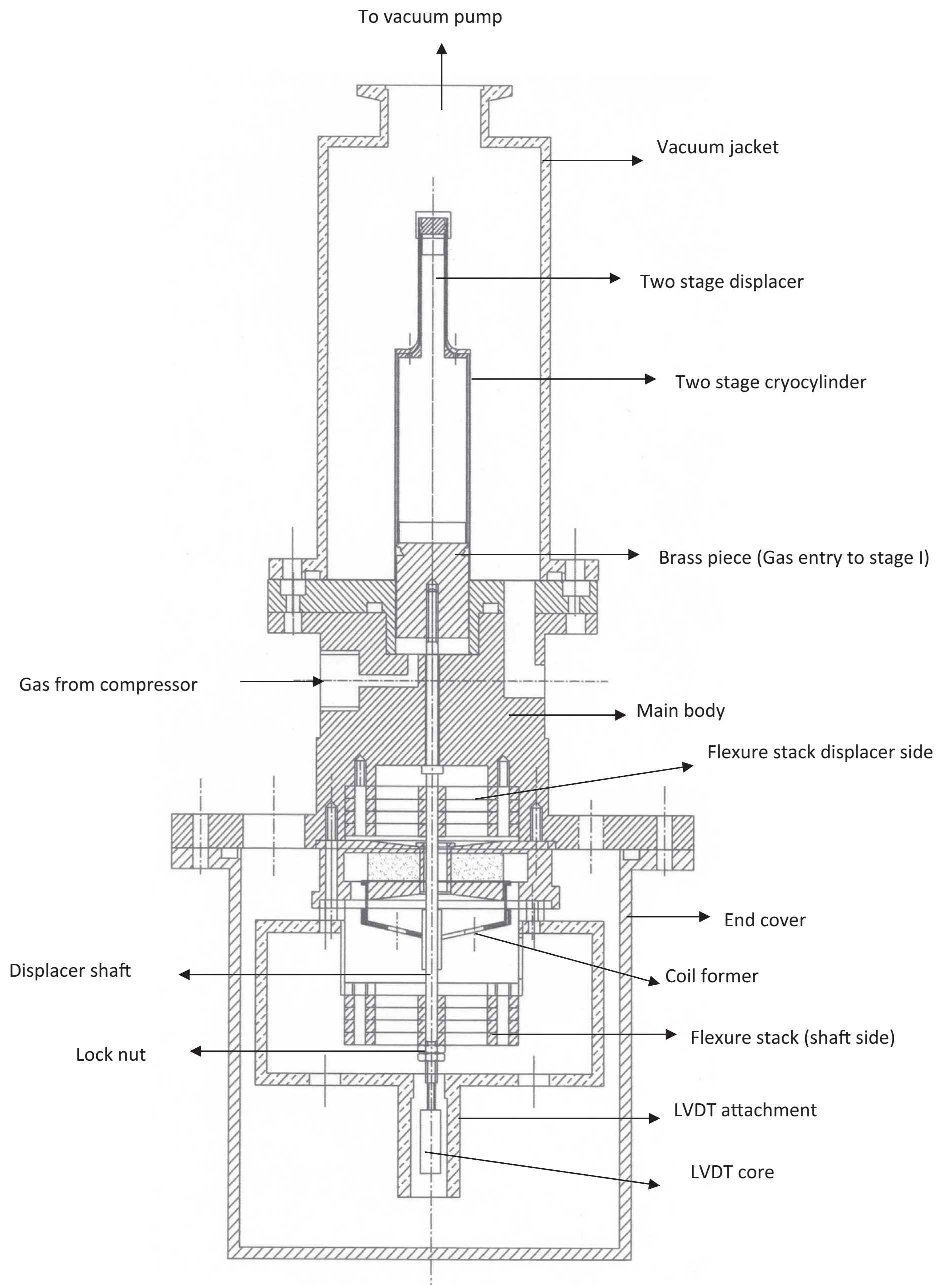


Fig. 3.6. Assembled expander unit (Expander I).

compressor power of 103 W. Table 4.1 shows as the load was applied to the two stages the viz. 0.5 W at stage 2 and 2 W at stage 1 the temperature at the stage 1 went to 128 K and at stage 2 went to 66.7 K.

5. Conclusion

A two-stage Cryocooler working on Stirling Cycle has been developed. It utilizes Moving coil linear motor and flexure bearing for suspension of the Piston and displacer**.** Rulon was coated on the piston and displacer unit to maintain the clearance seal and avoid friction. Two-stage Integral Cryo-cylinder and displacer were manufactured to avoid misalignment issue between the two stages and hence reduces the friction between the displacer and the Cryo-cylinder. The Cryocooler was successfully tested and achieved minimum temperature of 40.7 K at stage 2 and 90 K at stage 1 with a compressor power of 103 W. The Cryocooler achieved 2 W at 128 K at stage1 and

0. 5 W at 66.7 K at stage 2.

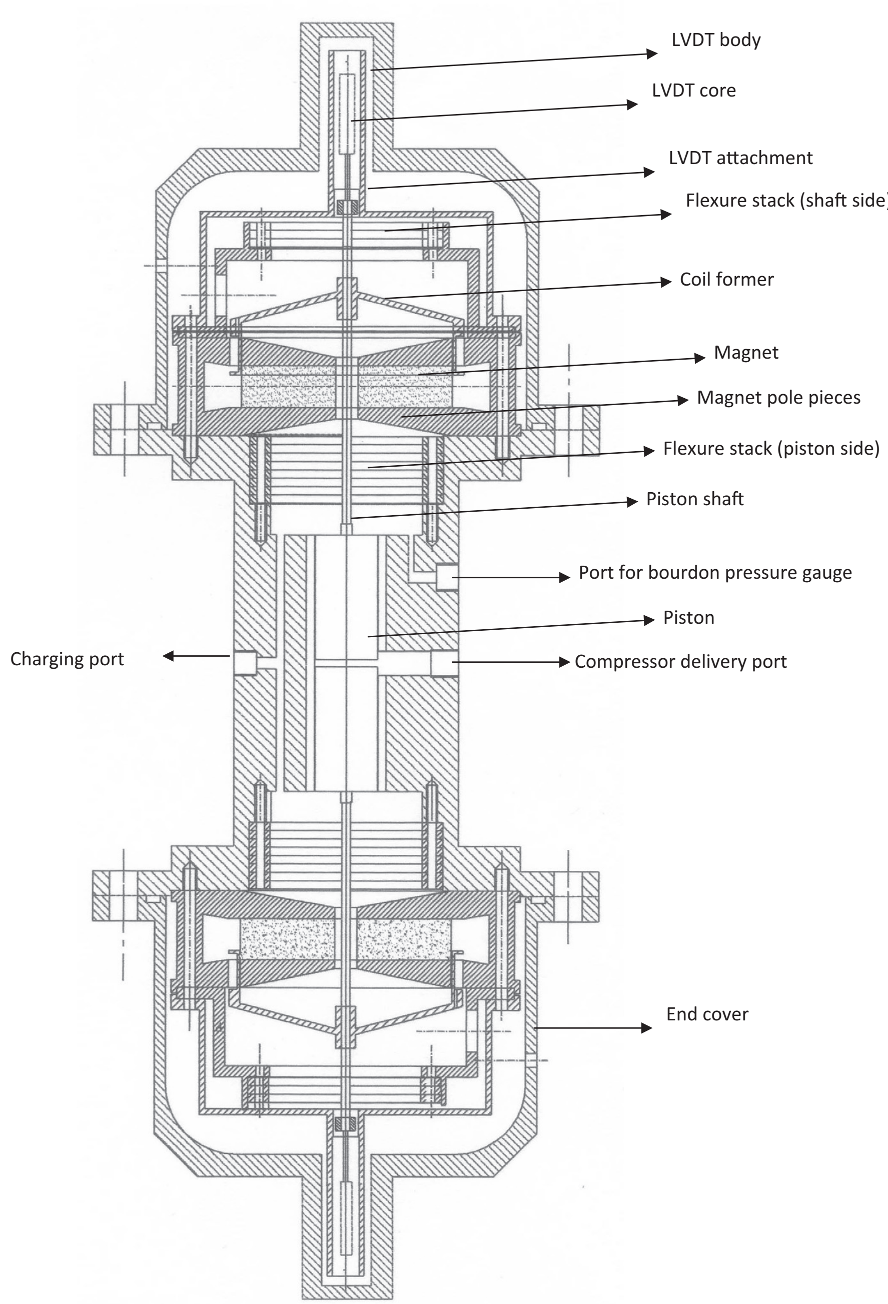


Fig. 3.7. Assembled opposed piston linear compressor.

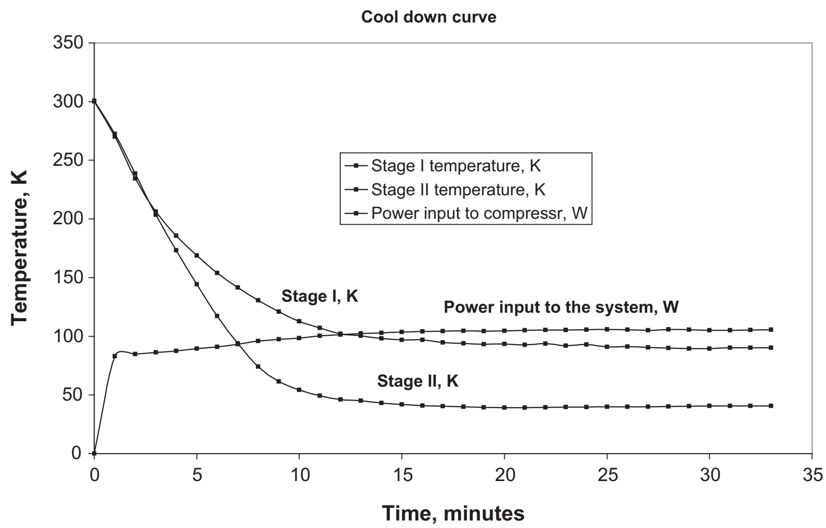


Fig. 4.1. Cool down curve for the two stages at no load condition.

Table 4.1
Load characteristics of the two-stage Cryocooler (load applied to both the stages).

Heat load applied on stage I, W	No load	2.0
Heat load applied on stage II, W	No load	0.5
Stage I temp., K	90	128.0
Stage II temp., K	40.7	66.7
Comp. power	103	100

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