

EVALUATION OF MECHANICAL LOSSES IN A LINEAR MOTOR PRESSURE WAVE GENERATOR

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ABSTRACT

A moving magnet linear motor compressor or pressure wave generator (PWG) of 2 cc swept volume with dual opposed piston configuration has been developed to operate miniature pulse tube coolers. Preliminary experiments yielded only a no-load cold end temperature of 180 K. Auxiliary tests and the interpretation of detailed modeling of a PWG suggest that much of the PV power has been lost in the form of blow-by at piston seals due to large and non-optimum clearance seal gap between piston and cylinder. The results of experimental parameters simulated using Sage provide the optimum seal gap value for maximizing the delivered PV power.

KEYWORDS: Pressure wave generator, clearance seal losses, pulse tube cooler, Sage.

INTRODUCTION

Attributes like light weight, compact, high efficiency and high MTTF have made high frequency pulse tube cryocoolers natural candidates for both space and tactical applications. Regenerative cryocoolers like Stirling type pulse tube coolers mainly consist of two elements: the PWG and the cooler. The PWG is the most critical component which supplies the necessary PV power to the cold head of the cooler. Efficiency of such a PWG is conventionally defined as the ratio of PV power measured at piston face to the electrical input power. Overall efficiency of the cryocooler system critically depends on the efficiency of the driving compressor which in turn is dictated by mechanical losses occurring in it. These mechanical losses occurring in the compressor consist of pressure losses that arise largely from the blow-by through the clearance seal between piston and cylinder and the flow losses resulting from the pressure drops through the flow channels

within the compressor. Therefore actual efficiency of PWG will be the ratio of delivered PV power to the input electrical power.

As a part of our research program, a compact moving magnet pressure wave generator of 2 cc swept volume, of dual opposed piston configuration, has been built in our Institute. Flexure bearing stacks are used at both front and rear end to suspend the piston. They also provide a controlled uniform radial clearance gap between the piston and cylinder, in the ideal case.

A radial clearance of 10-15 μm is required for achieving a satisfactory dynamic seal [1]. However due to fabrication deficiencies and assembly inaccuracies such an optimum and uniform clearance seal could not be achieved in the first prototype linear motor compressor that we developed. Experimental program was taken up to quantify piston-cylinder dynamic seal losses and overall linear motor compressor efficiency. In the experimental program a pulse tube cooler, though not acoustically matching to the compressor, was used for initial testing and a no-load cold end temperature of 180 K was obtained [2].

EXPERIMENTAL PROGRAM AND MEASUREMENTS

In order to characterize losses involved in the PWG, measurements of pressure amplitude, mass flow rate and phase angle between pressure and mass flow are carried out at the face of piston. The pressure amplitude is measured using Endevco 8510B-200 dynamic pressure sensor. Mass flow is estimated by measuring the piston displacement using LVDT (Micro-Epsilon, Germany). Phase angle between volume flow rate and pressure is measured using Gwinsteak oscilloscope model GDS-2104.

The ideal PV power obtainable at the face of piston is given by the equation (1).

$$\dot{W} = \frac{1}{2} P_1 \dot{V}_1 \cos\phi \quad (1)$$

Here P_1 is pressure amplitude, \dot{V}_1 is amplitude of swept volume flow rate and Φ is the phase angle between pressure and mass flow rate in the compressor. However due to losses in the PWG much less PV power is delivered to the acoustic load. These losses mainly consist of pressure losses and flow losses. Radebaugh et. al. have carried out extensive measurements to characterize pressure and flow losses in a linear compressor of 4.3 cc swept volume [4]. Same measurement technique has been adopted in our work. Pressure loss measurements are carried out by blanking off compressor thus ensuring no flow conditions and thereby resulting in no flow losses during these measurements. Pressure losses are calculated by subtracting the copper, iron losses and measured PV power at the piston face from the total electrical input power. Flow losses are evaluated in a separate experiment by connecting a large reservoir to the compressor such that there is only small pressure amplitude in the compressor. Flow losses are correlated to the mass flow rate measured in the reservoir and are calculated in a similar fashion to that of pressure losses. These measurement techniques exclusively quantify the inherent losses in the compressor.

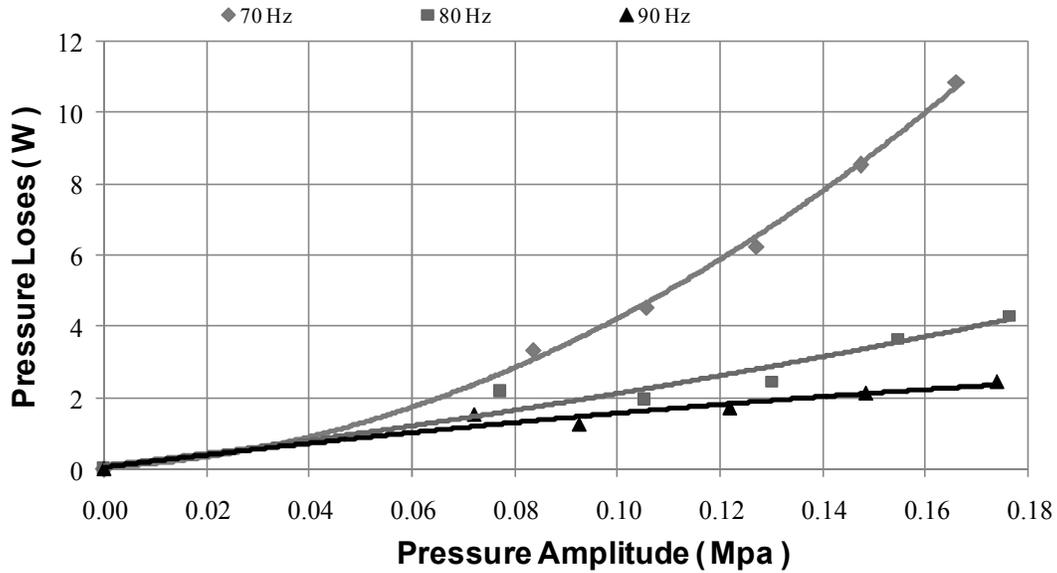


FIGURE 1. Pressure losses at different operating frequencies, filling pressure 1.5 MPa.

FIGURE 1 and FIGURE 2 show the variation of pressure and flow losses respectively at different operating frequencies for a filling pressure of 1.5 MPa. Similarly, pressure and flow losses at higher filling pressure of 2.5 MPa are shown in FIGURE 3 and FIGURE 4 respectively. It is seen from FIGURE 1 and FIGURE 3 that pressure losses are higher at lower frequencies, though it is in contradiction to the results given by Radebaugh et.al. [3]. It is postulated that at higher operating frequency clearance seal gas flow time duration is shorter leading to lower pressure losses. The flow losses evidenced from FIGURE 2 and FIGURE 4 show significantly higher losses at higher frequencies. However, the flow losses are realistically small due to the lower mass flow rate achievable when an acoustic load is connected to the PWG.

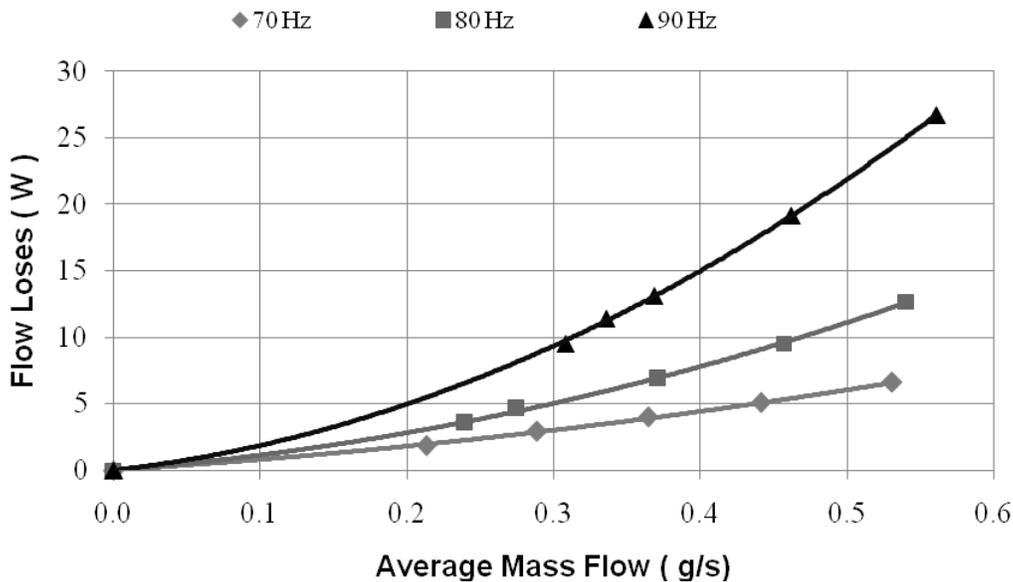


FIGURE 2. Flow losses at different operating frequencies, filling pressure 1.5 MPa

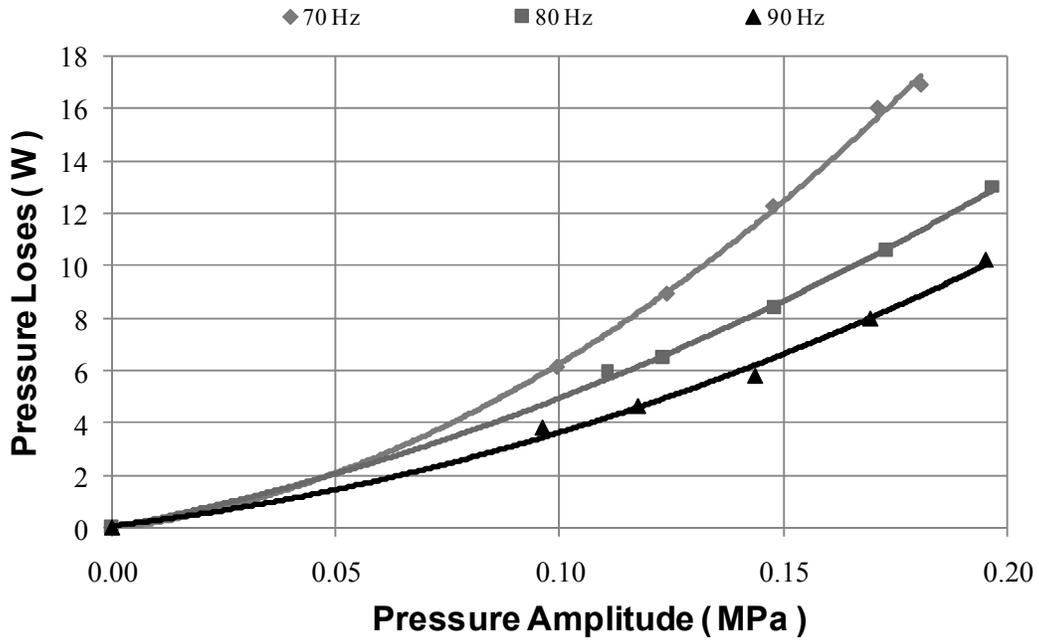


FIGURE 3. Pressure losses at different operating frequencies, filling pressure 2.5 MPa

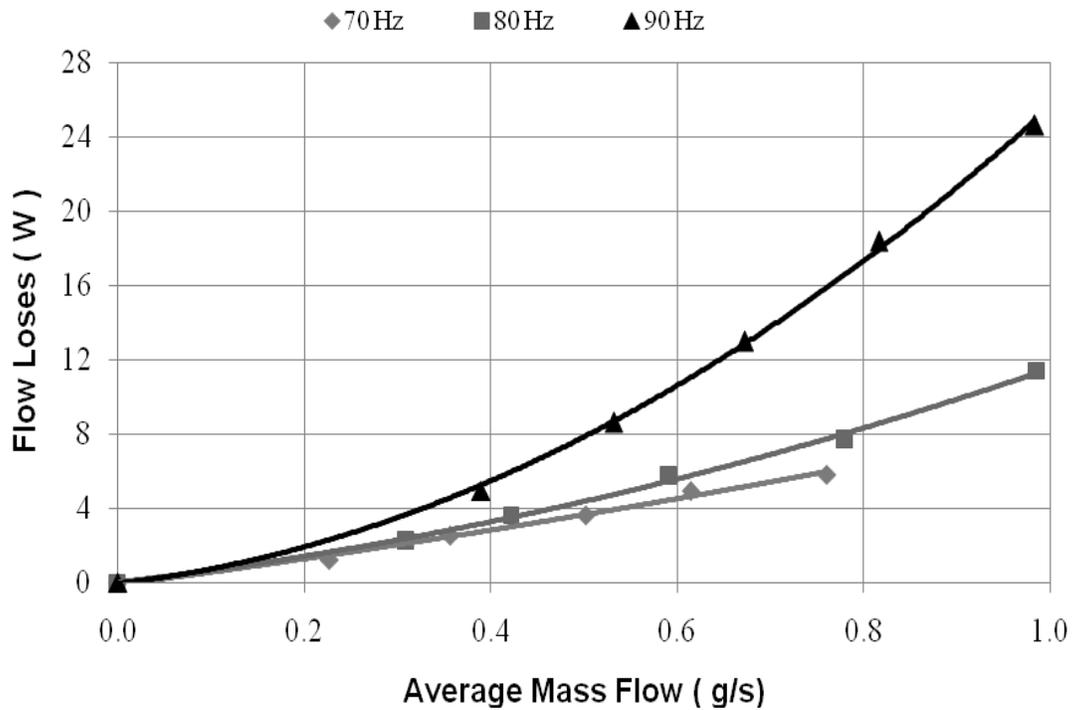


FIGURE 4. Flow losses at different operating frequencies, filling pressure 2.5 MPa.

TABLE 1. PV power and the losses in the PWG connected to the acoustic load

Measured Parameters	Power (W)
Total input power	35.0
Copper losses	8.0
Iron losses	0.2
Shaft power	26.8
PV power at piston face	17.8
Pressure losses	4.4
Flow losses	2.0
Net PV power delivered to PTC	11.4

Deliverable PV power to cold head

In this experiment PWG was connected to an acoustic load and operated at a resonance frequency of 69 Hz and mean filling pressure of 1.5 MPa. The pressure ratio in the PWG was 1.14 corresponding to pressure amplitude of 0.10 MPa. Measured phase angle, mass flow leading the pressure, was 39° . The mass flow through the acoustic load was estimated in a separate experiment, by the differential pressure method [4], at the cold end of the regenerator, carried out at ambient condition. The measurements carried out represent the conditions at the start of the system though mass flow rate may be a bit different at actual operating conditions of the acoustic load. The losses and deliverable PV power are given in the TABLE 1. Thus, only a net PV power of 11.4 W was delivered resulting in poor performance of the connected acoustic load.

SIMULATIONS USING Sage

The performance of the pulse tube cryocooler that was chosen as the acoustic load was analyzed using Sage software [5]. The values of input parameters like filling pressure, frequency and stroke correspond to the experimental data. The compressor is modeled as two pistons oscillating physically 180° out of phase with each other. An “annulus” component was used to model the leakage past the piston. Sage model for the compressor and the acoustic load is shown in FIGURE 5. Simulations were carried out by varying seal clearance from 5 to 45 μm . This allows the comparison of the Sage predictions with the experimental results. Results of Sage analysis are given below.

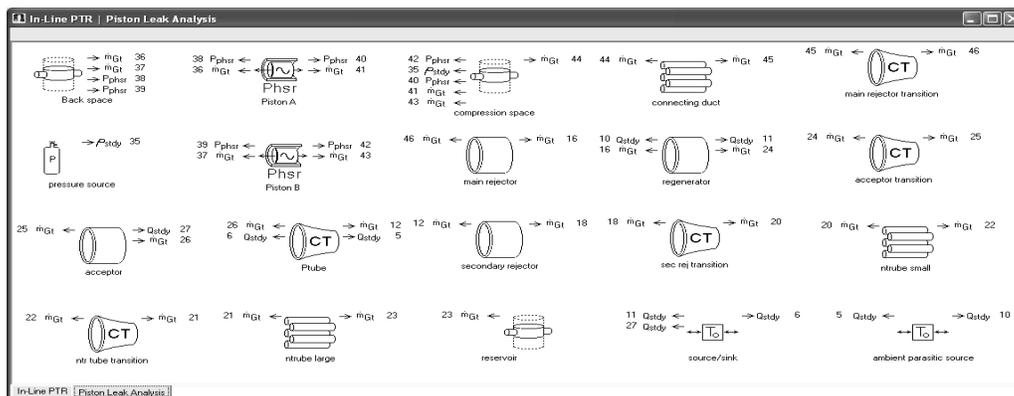


FIGURE 5. Window of Sage simulation scheme

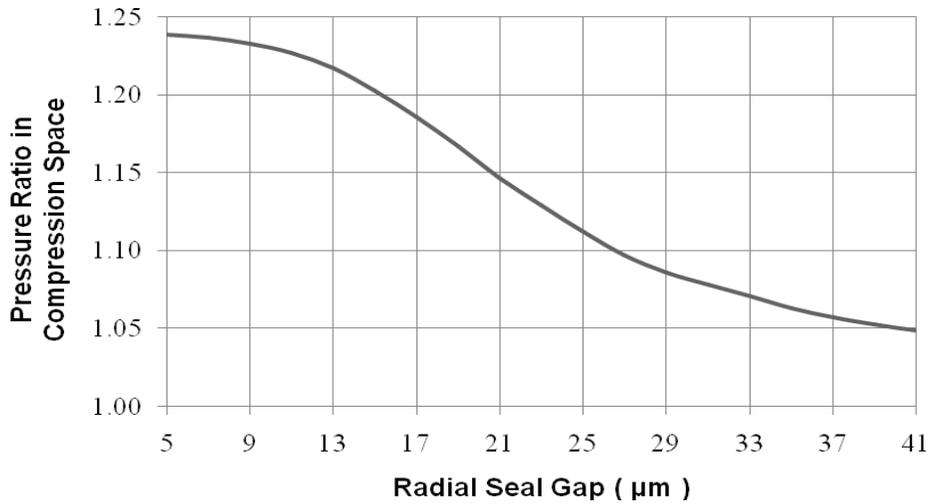


FIGURE 6. Effect of varying radial seal gap on pressure ratio.

Effect of radial seal gap on the pressure ratio

FIGURE 6 shows the effect of the radial seal gap on the pressure ratio in the compression space. As the radial seal gap increases there is a significant decrease in the pressure ratio. In the experimental program, a pressure ratio of 1.14 was obtained. The corresponding uniform radial seal clearance from Sage simulation is about 21 µm.

Effect of radial seal gap on the mass flow

Figure 7 shows the effect of the seal gap on the mass flow rate delivered to the cryocooler and the mass flow rate leaking to the back side of the pistons. As the radial seal gap increases there is a significant increase in the mass flow leaking to the back side. At a seal gap of more than 20 µm larger flow is leaking to the back space than being delivered to the cryocooler.

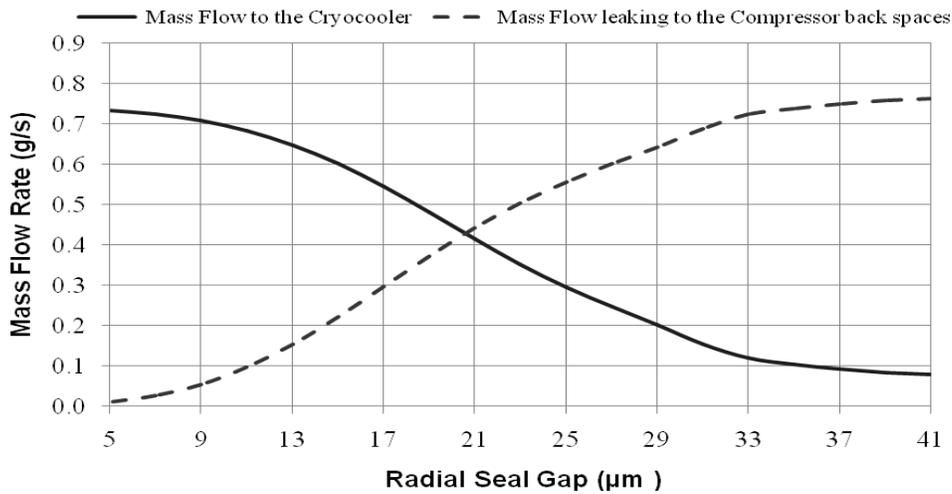


FIGURE 7. Effect of varying radial seal gap on mass flow rate.

Effect of radial seal gap on delivered PV power

The effect of the piston radial seal gap on the PV power delivered to the cryocooler simulated using Sage is shown in FIGURE 8. There is a significant reduction in the PV power delivered to the cryocooler as the radial gap increases. A radial seal gap less than 15 μm is needed to maximize the delivered PV power. For Sage simulated radial seal gap, which gives matching experimental conditions including no-load temperature of 180 K, the delivered PV power is about 17 W. This is higher than 11.4 W obtained by the experimental loss measurement program. This difference could possibly be due to the non-uniform seal gap in the actual case leading to higher leakage past the pistons.

It is interesting to derive the lower limit of seal gap analytically by considering the flow through the seal gap to be like the laminar slit flow with a moving wall i.e. plane Couette flow [6]. Then the viscous stress (τ) acting on the gas in the seal gap is given by

$$\tau = \frac{\Delta p}{2L} \delta - \frac{\mu v}{\delta} \quad (2)$$

The mass flow rate through the seal gap is given by

$$\dot{m} = \frac{\Delta P \delta^3 W \rho}{12 \mu L} + \frac{\rho \delta W v}{2} \quad (3)$$

Where ΔP is the pressure drop across the seal gap, δ is the seal gap, L , W are the length and the width of the channel, v is the velocity of gas, ρ , μ are the density and the viscosity of the gas respectively.

It can be seen from the equations (2) and (3) that there is a specific seal gap for which the drag force equals the pressure drop (ΔP) driving force for the seal blow by. As the seal gap increases beyond this value the blow by increases as the pressure force is more than the drag force and the flow is predominantly Poiseuille type. Below this seal gap there is only a small reduction in the seal blow by and the flow is purely Couette type. Hence there is an optimal seal gap and its value for the experimental conditions (pressure ratio 1.14, Fill Pressure 1.5 MPa, Frequency 69 Hz, Stroke ± 2 mm) was calculated to be 0.95 μm .

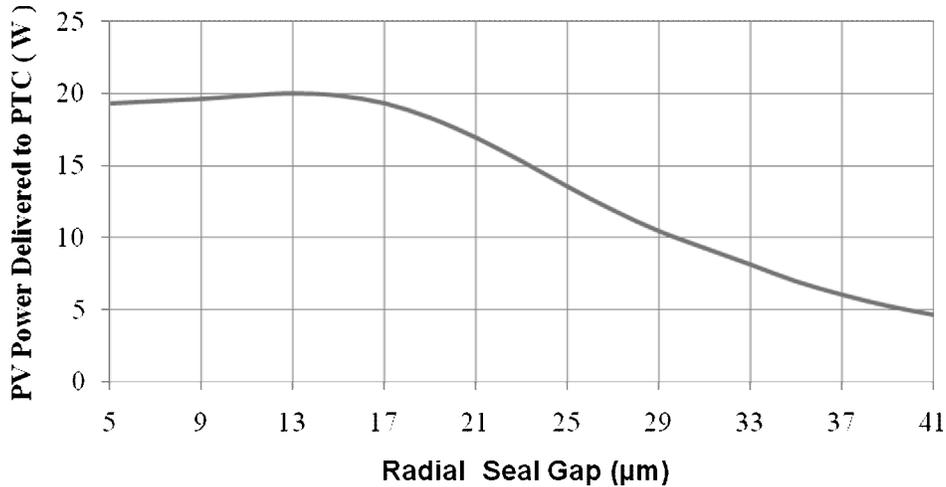


FIGURE 8. Effect of varying radial seal gap on PV power delivered to the PTC

However in practice it is difficult to achieve a concentric non-contacting gap of about 1 μm . Even if perfect coaxiality of the cylinder and piston and perfect cylindricity is assumed, a gap less than 10 μm will result in an close fit, leading to greater damping forces during oscillations of the piston. Further, the viscous drag in the Couette flow regime will reduce the stroke for a given input power. Therefore in practice a radial seal gap of about 10 μm is probably an acceptable value. This concurs with the typical value of the seal gap used in linear compressors for low power Stirling type pulse tube coolers [1].

CONCLUSIONS

The pressure loss and flow loss characteristics of the in-house developed prototype moving magnet pressure wave generator of 2 cc swept volume have been carried out. These losses have been also established when a pulse tube cryocooler acoustic load was operated with the PWG. The analysis indicated large blow by losses caused by non-optimum piston seal radial gap. This resulted in delivering low PV power of 11.4 W to the pulse tube cooler, that could only reach a no load temperature of 180 K. The experimental parameters of the PWG and the pulse tube cooler were simulated using Sage for varying piston seal radial gap values. From the analysis it is shown that the PWG is having an equivalent radial seal gap of 21 μm . Based on the analysis reported here, efforts are underway to achieve the ideal seal gap of about 10 μm by adopting new fabrication sequences and assembly procedures for the PWG.

ACKNOWLEDGEMENTS

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