Numerical Analysis of a Pulse Tube Cryocooler with Inertance Tube-Bounce Space as Phase Shifter

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Abstract: ***A performance comparison of a Pulse Tube Cryocooler (PTC) that uses inertance tube- bounce space and inertance tube-reservoir as a phase shifter is conducted using numerical simulations. The initial design cryocoolers were carried out using Sage software. A CFD model was developed using ANSYS Fluent to analyze the Cryocooler performance. The CFD model was used to simulate the effect of different volumes of reservoir and bounce space on Cryocooler performance. The thermal non-equilibrium mode was chosen to consider the effect of temperature difference between solid and fluid temperature difference in porous zones. The numerical model was validated with experiments from referred journal. The simulation results showed a phenomenal increasing trend in cooling capacity up to 400cm^3, and thereafter, a marginal increment in performance with increase in volume. The Stirling cryocooler with inertance tube-bounce space as phase shifter has over performed the cryocooler with inertance tube-reservoir. The COP of cryocooler with 400cm^3 bounce volume was found to be 0.042, is 1.38 times higher than that of 200 cm^3 and for higher volumes difference in COP was less significant.***

Keywords : Pulse Tube, Stirling, Cryocooler, Bounce space, CFD, Inertance tube.

I. INTRODUCTION

A Pulse Tube Cryocooler is a regenerative cryocooler, which works on the reversed Stirling gas cycle with working fluid Helium. It has the least number of moving parts as the oscillating gas inside a thin walled tube known as pulse tube is responsible for the temperature gradient along regenerator. This tube containing gas is equivalent to a displacer in a Stirling cryocooler. At low temperature regions, the pulse tube cryocoolers does not have a moving part, which makes them reliable, low cost and less vibrations. These features are attractive to many Cryogenic temperature applications like cooling of superconducting magnets for MRI application, Cryosurgery, cooling of infra-red sensors for surveillance[1], etc. The pulse tube refrigerators were first introduced and phenomena's were reported by Gifford and Longworth[2][3], this refrigerator called a Basic pulse tube refrigerator(BPTR) produced little temperature drop. Mikulin et al.[4] placed orifice and reservoir to provide cooling to 104K and expected to cool below 60K. Kanao et al.[5] used the inertance tube instead of an orifice to achieve cryogenic temperature.

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Boer et al.[6] reported that use of inertance tube and reservoir has better performance compared to that of an orifice connected cryocooler. The inertia of the gas provides phase shift [7] for the inertance tube- reservoir phase shifter. There are many types of pulse tube cryocoolers and are named based on the phase shifters they use, like double inlet pulse tube coolers[8], tandem[9], active phase control[10], adjustable inertance[11], looped pulse tube cooler[12], etc.

An Inertance Pulse Tube Cryocooler (IPTC) uses an inertance tube-reservoir as a phase controller. IPTC has a linear compressor, Aftercooler, regenerator, Acceptor, pulse tube, warm heat exchanger (WHX), inertance tube and reservoir. Compressor used for this type of system is a valve less linear compressor, which creates an oscillating pressure typically about 5 to 15% of the charge pressure. A pressure oscillation by itself hardly produces any refrigeration[13]. The phase-shifting mechanism controls the motion of gas in an inertance pulse tube cryocooler, i.e. inertance tube-reservoir combination along with pulse tube.

The design of pulse tube is complex and it is not mathematically established. According to the available literature, the pulse tube acts as an insulation between the two process at hot end and cold end. For this the pulse tube should be large enough so that the gas flowing from the hot and travels only a part of the way through the tube before the flow reverses. Therefore, the flow from the hot and cold end never reaches the hot end. The flow of gas depends on the on the phase shifter used i.e. in this case a inertance tube- reservoir. Phase sifter is responsible to maintain an oscillatory flow and compensate for any additional fluctuations. The phase shift is primarily dependent on geometric parameters of the inertance tube, namely its length, diameter as well as the size of the adjoining reservoir for a given acoustic power and frequency[14].

Design of cryocoolers is carried out with numerical tools and application software. Abolghasemi et al.[15], Abraham et al.[16] and Wang et al.[17] used Sage software for the design of Pulse tube cryocooler. Iwase et al. [18] and Wang et al.[19] used an electric circuit analogy for simulation of IPTC. CFD based on the thermal equilibrium model of Pulse tube cryocooler was carried out by cha et al.[20], Ashwin et al.[21] and Abraham et al. [22]. Dang et al.[23] and Zhao et al.[24] developed thermal non-equilibrium CFD model for coaxial Stirling type pulse tube cryocooler.

The reservoir along with inertance tube is necessary to obtain higher phase-shifting ability, which makes the pulse tube cryocooler bulkier compared to the Stirling cryocoolers with displacer.

A reservoir occupies most of the volume for an Inertance Pulse Tube Cryocooler, which could be replaced by the use of bounce space to reduce the total space occupied. The volume behind the compression space of a linear compressor is known as a bounce space. The bounce space of the linear compressor has a phase shift of approximately 180º compared to that of compression space. This phase difference is utilized as a method of enhancement in performance of Pulse Tube Cryocooler and reduce the size. Numerical methods were conducted with Cryocoolers, which uses inertance tube-reservoir and inertance tube-bounce space as a phase shifter to establish the performance enhancement. A cryocooler experimental set up is proposed which can use inertance tube-bounce space or inertance tube-reservoir as phase shifter at a time. To analyze the effect of the different reservoir and bounce space volumes on the performance of the cryocooler a numerical model of the cryocooler is developed using ANSYS Fluent. The model was validated with the experiments (from journal), and study on different bounce space, and reservoir volumes were conducted.

II. DEVELOPMENT OF CRYOCOOLER

A. Cryocooler with Inertance tube –bounce space

The cryocooler experimental setup with instrumentation is shown in figure 1. The PTC consists of a linear compressor, a cold head including aftercooler, regenerator, acceptor, pulse tube and warm heat exchanger, inertance tube, reservoir, and bounce space as in figure 1. The cold head contains acceptor, pulse tube, regenerator, and the two hot heat exchangers, namely aftercooler and warm heat exchanger. The cold head is coaxial type, regenerator is the annular region and Pulse tube as the inner tube. The working fluid of cryocooler is helium, charge pressure is 3.2MPa, and working frequency is 55Hz. Table I and II gives the input values, specification, and dimensions of the proposed setup. The compressor is a moving magnet linear single piston compressor, with a maximum compression space volume of 3.70cm^3, and offer a power of 100W.

Heat exchangers acceptor, warm heat exchanger, and aftercooler were made of stacked copper mesh with mesh size 100. The regenerator, a key component for the Stirling system is made of micro-porous metallic structure, where the oscillator flow and regenerative heat transfer of working fluid happens. The regenerator consists of stacked ss304 wire mesh with mesh number 400. The pulse tube and outer casing of cold head were made of SS304 with a wall thickness of 0.2mm to reduce conduction heat loss between the hot and cold ends. Inertance tubes and the intermittent connections were made of copper tubing. The choice of the phase shifter, inertance tube-bounce space or inertance tube reservoir depends on the opening and closing of valves 1 and 2. When experiments are conducted with inertance tube-bounce space as phase shifter valve-1 closed and valve-2 will be open. As in case of inertance tube-reservoir as phase shifter, valve-2 will be closed and valve-1 is opened. Simulations were conducted for 400cm^3 reservoir and bounce space volume. The temperature variation at acceptor and pressure variations inside reservoir and bounce space are acquired using data acquisition systems. The conduction losses from the acceptor at the cryogenic temperature to the aftercooler and warm heat exchanger through the walls of regenerator and pulse tube cannot be eliminated. The cold head is enclosed in a vacuum jacket and is connected to a turbo molecular pump. The vacuum created inside the jacket reduce the parasitic heat loss due to the molecular conduction of water vapor present in the air.

Table- I: Input parameters to the pulse tube cryocooler

Operating Parameter	Value
Frequency	55Hz
Charge pressure	3.2MPa
Acceptor temperature	80K
Ambient temperature	300
Working fluid	Helium

Table- II: Dimensions and material used for Cryocooler

Part name	Dimensions	Material
Compressor	19.80mm o.d, Amplitude 6mm	--
Aftercooler	26.38mm o.d, length 5 mm	Copper mesh #100
Regenerator	26mm o.d, length 74.2 mm	SS304 #400
Cold heat exchanger (Acceptor)	26.38mm o.d, length 2 mm	Copper
Pulse tube	14mm o.d, length 76.2 mm	SS304 tube
Warm heat exchanger(WHX)	14 mm o.d, length 3 mm	Copper mesh #100
Inertance tube	3.5mm i.d, length 3.147m	Copper tube
Bounce space/ Reservoir (400 cm^3)	80mm o.d, length 80 mm	Copper

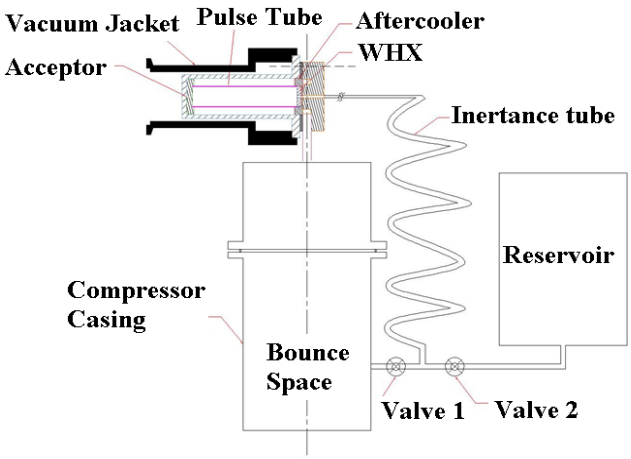

Fig. 1.Schematic of proposed cryocooler setup that can use inertance tube-bounce space or inertance tube-reservoir as phase shifter

B. Cryocooler simulation and results

Design of cryocooler was done with the help of Sage-v11 software. This software is created by Gedeon associates, Ohio, USA. Which is a proprietary software for the design of cryocoolers, Stirling coolers and engines. Using this software a cryocooler as in figure. 1 is designed. The dimensions and operating conditions of the cryocooler are shown in Table I and II. The simulation was conducted according to the proposed experimental conditions from previous sections.

Sage model was developed based on the proposed experimented system. It represented the actual cryocooler system including DC gas flow which happens between the compression space and bounce space.

As the system is coaxial heat transfer effects between regenerator and pulse tube is incorporated. Figure 2 show the simulation using sage, Pressure variation inside bounce space and reservoir. In bounce space, the amplitude of pressure is 0.035MPa, whereas in reservoir it is 0.025MPa. The phase difference between the bounce space and reservoir pressure waves is 36°.

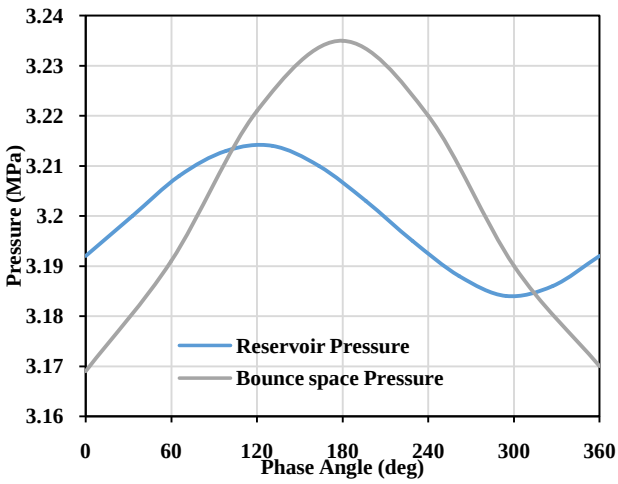

Fig. 2.Pressure varaition in reservoir and bounce space(400 cm^3).

III. NUMERICAL ANALYSIS OF CRYOCOOLER

Numerical modeling of the cryocooler was developed using ANSYS Fluent. The geometry parameters of the numerical model is given in table 2. Figure 3 shows the geometry of the 2D axis-symmetric CFD model, which consist of aftercooler, regenerator, acceptor, pulse tube, warm heat exchanger, inertance tube, bounce space and reservoir. The compressor effect is applied by pressure inlet-1 at the aftercooler wall. Similarly, the bounce space effect is introduced using pressure inlet-2 as shown in figure 3. To study the characteristics of the cryocooler with different reservoir and bounce space volume, simulations were carried out for cases given in table III. Cases 1, 3, and 5 uses a reservoir, i.e., single pressure source, Case 2, 4, and 6 uses bounce space, i.e., two pressure source are used simultaneously. The pressure fluctuation is created by User Defined Function (UDF), which is given by eq. (1) and (2):

Table- III: Cases considered for simulation.

Cases	Reservoir/ Bounce space volume
Case 1:	200cm^3 Reservoir
Case 2:	200cm^3 Bounce space
Case 3:	400cm^3 Reservoir
Case 4:	400cm^3 Bounce space
Case 5:	600cm^3 Reservoir
Case 6:	600cm^3 Bounce space

$$P_{inlet\ 1} = P_{charge} + \Delta P_{\ inlet\ 1} \sin(2\pi f t). \quad (1)$$

$$P_{inelt\ 2} = P_{charge} + \Delta P_{inlet\ 2}\, sin(2\pi f t + \pi) \quad (2)$$

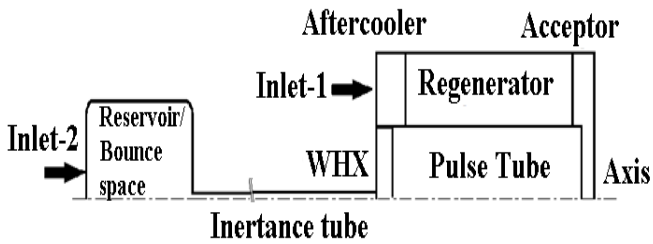

Fig. 3. 2D axis-symmetric model for inertance pulse tube cryocooler[22]

Where the frequency of operation is f, P_{charge} is the charge pressure, $\Delta P_{\ inlet\ 1}$ is the pressure amplitude inside the compressor, $\Delta P_{inlet\ 2}$ is the pressure amplitude inside the bounce space. The amplitude of the pressure is different at inlet-1 and inlet-2. Equations 1 and 2 are the function for the pressure generated at the inlet, i.e., the compressor and bounce space, respectively.

A. Governing equations and method of solution:

The equations[23] governing mass and momentum conservation for porous zones are eq. (3) and (4).

$$\frac{\partial}{\partial t}(\varepsilon\rho_f) + \nabla(\varepsilon\rho_f\vec{v}) = 0 \quad (3)$$

$$\frac{\partial}{\partial t}(\varepsilon\rho_f\vec{v}) + \nabla(\varepsilon\rho_f\vec{v}\vec{v}) = -\varepsilon\nabla P + \nabla.(\varepsilon\bar{\bar{\tau}}) + S_i \quad (4)$$

Here density, velocity, and pressure of working fluid is denoted by ρ_f, $\vec{v}$, and P. Stress tensor is indicated by"$\bar{\bar{\tau}}$". The values of porosity ε and momentum source term S_i depends on porous or non-porous zones. For porous-zones, porosity value depends on the filling of mesh and S_i is given by eq. (5).

$$S_i = -\left(\frac{\mu}{\alpha}\vec{v} + \frac{C_2}{2}\rho_f|\vec{v}|\vec{v}\right) \quad (5)$$

Where 'C_2' and 'α' is the inertial resistance and permeability factors respectively. To accommodate for the actual heat transfer effects, thermal non- equilibrium mode is introduced. Here the simulation will consider temperature difference between solid and fluid zones to calculate the overall heat transfer. The governing equation for the same is given in eq.(6).

$$\frac{\partial}{\partial t}(\rho_f E_f) + \nabla[\vec{v}(\rho_f E_f + P)] = \nabla.[\varepsilon k_f \nabla T + (\bar{\bar{\tau}}.\vec{v})] + h_{fs}A_{fs}(T_s - T_f) \quad (6)$$

Where T, E_f, and k_f are the temperature, energy, and thermal conductivity of the working fluid.

B. Boundary condition and initialization.

Assumptions used for the simulations are:

- The initial pressure of the system is the charge pressure (3.2MPa) and the temperature is 300K for the fluid and porous region.
- The effect of wall thickness is neglected.
- There is no heat conduction between the regenerator and the pulse tube.

Boundary conditions are:

- The heat exchanges (aftercooler, acceptor and warm heat exchanger) and regenerator are porous regions.
- In porous zone, the effective heat transfer is calculated taking into consideration the effect of difference in temperature difference solid and fluid zone (thermal non-equilibrium condition). The parameters of porous region are shown in table IV.
- The temperature of exterior walls of heat exchanger, inertance tube and reservoir are taken as 300K.
- The regenerator and pulse tube walls are adiabatic.

Table- IV: Values of porous media parameters.

Components	Material	ε	$\alpha\times10^{-10}$ (m^2)	C_2 (1/m)	h_{fs} (W/m^2K)	A_{fs} (1/m)
Regenerator	SS304 (mesh, size #400)	0.786	1.29	35116	39440	34400
Aftercooler	Copper (mesh, size #100)	0.65	6.95	9031	17000	12280
Acceptor	Copper (mesh, size #100)	0.65	6.95	8123	17000	12280
WHX	Copper (mesh, size #100)	0.65	6.95	8646	17000	12280

C. Numerical scheme

The solver uses a pressure-based coupled scheme. For solving density and momentum equations, a second-order upwind scheme was used, and for pressure, spatial discretization PRESTO scheme. The simulations were conducted until the acceptor temperature reached 80K. The time step taken was 1×10^{-4} seconds, with 50 iterations per time step. The convergence criteria for continuity, momentum, and energy were 10^{-3}, 10^{-3}, and 10^{-6}, respectively. The residual monitors of the solution showing the residuals of energy, continuity, and velocity showed that the solution has converged. Three grid systems with mesh numbers 58262, 106662, and 231475 are used to test the mesh independency. The simulation was conducted for a time duration of 100 seconds. The difference in acceptor temperature at the end of 100 seconds for mesh nos. 106662 and 231475 were 0.1K. The grid with 106662 quadrilateral grid elements is used for the simulation. The mesh independence test is shown in **figure 4.**

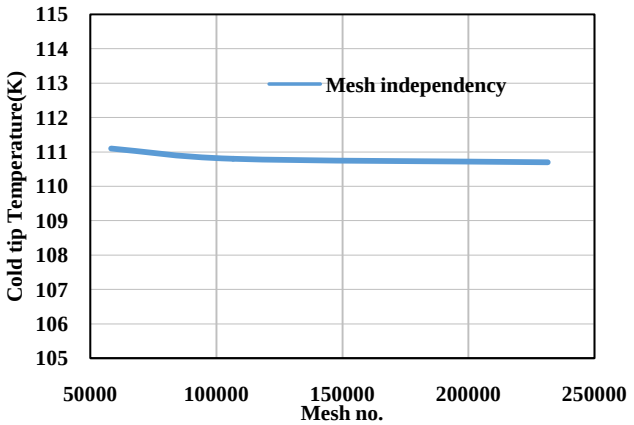

Fig. 4. Mesh independency test

D. Model Validation

The present model is validated with previously conducted experiment by Dang et al.[23] to ensure accuracy. The variation between the simulation and the experiment was found to be less than 20%. The CFD model using ANSYS Fluent model under predicts and the model could be used for further studies.

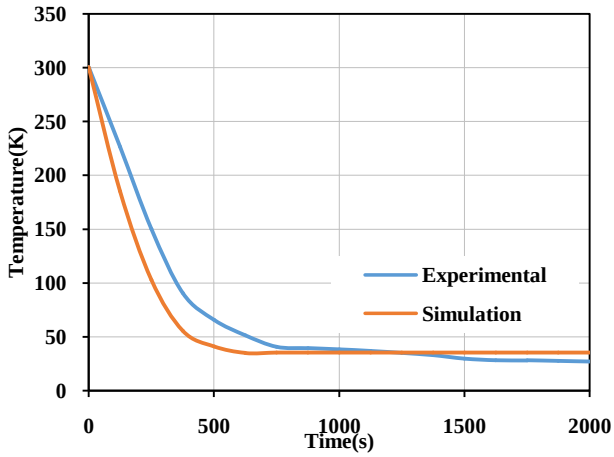

Fig. 5. Simulation Validation

E. Simulation results

The simulation of the numerical model proposed to simulated cool down curve for cases 3 and 4 and is shown in figure 6. In Figure 6 plots the temperature measured at the acceptor to the time in seconds for the experimented. The simulation were conducted at the no-load condition at the acceptor. The cryocooler using inertance tube-bounce space combination attained 80K in 165sec where the simulation took only 142 seconds. As in the case of inertance-reservoir combination, the experiment took 182 seconds while simulation took 152 sec.

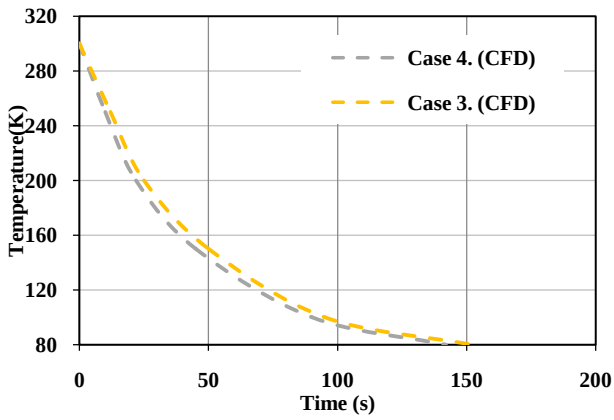

Fig. 6. Simulated Cooling curve comparison

A time-varying temperature contour of the cold head is shown in figure 7. The simulation is done for the cryocooler with inertance tube- reservoir as the phase shifter, with reservoir volume of 400 cm^3 (Case 3). The heat load at the acceptor is zero i.e. the acceptor wall is adiabatic condition (no load).

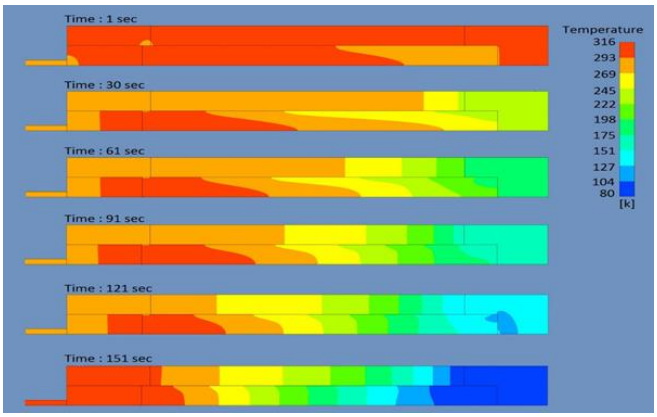

Fig. 7. Temperature contour of the cold head at no load.

IV. EFFECT OF RESERVOIR AND BOUNCE SPACE

To study the influence of volume on performance, six cases were simulated for different reservoirs and bounces space volumes as listed in table III. Heat load was applied to the acceptor wall (in CFD simulations) which was obtained from Sage simulations.

A. Simulation of cryocooler with the heat load

Figure. 8 and 9 show the temporal variation of the dynamic pressure and mass flow rate at acceptor for cases 1- 4 with the heat load. It is found that the phase of mass lags the pressure wave at the acceptor. The phase shift obtained at acceptor for the cases 1-6 are given in table V along with the mass flow at the inlet of the acceptor. The mass flow rate decreases for cryocooler using inertance tube-reservoir and inertance tube-bounce space individually. The phase shift obtained in table V could be calculated by measuring the phase difference from figure 8 and 9. Phase shift obtained for cryocooler with inertance tube- bonce space combination (400 cm^3) is -45.47º.

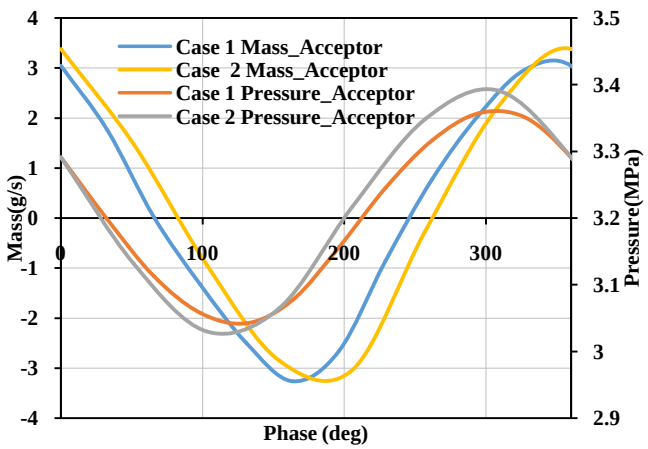

Fig. 8. Pressure oscillation at Acceptor, for bounce space and reservoir (case 1&2)

Figure. 10 shows the cooldown curve for cases 1-6 with the heat load. The cooling curves prove that the designed coolers using either of the kinds of phase shifters (case 1-6) are capable of attaining cryogenics temperature(80K) with the applied heat load within a time gap between 195-229 seconds, well within 4 minutes. The applied heat load and input power and the performance of the cryocoolers are shown in table 6.

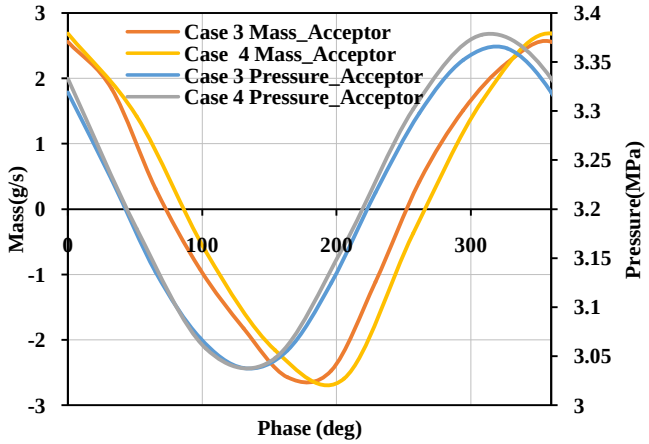

Fig. 9. Pressure oscillation at Acceptor, for bounce space and reservoir (case 3&4)

Table- V: Comparison of mass flow and phase shift at acceptor.

Case no.	Acceptor Mass (g/s)	Phase shift (deg)
Case 1:	3.077	-36.03
Case 2:	3.232	-58.45
Case 3:	2.536	-32.69
Case 4:	2.602	-45.47
Case 5:	2.339	-31.00
Case 6:	2.353	-40.50

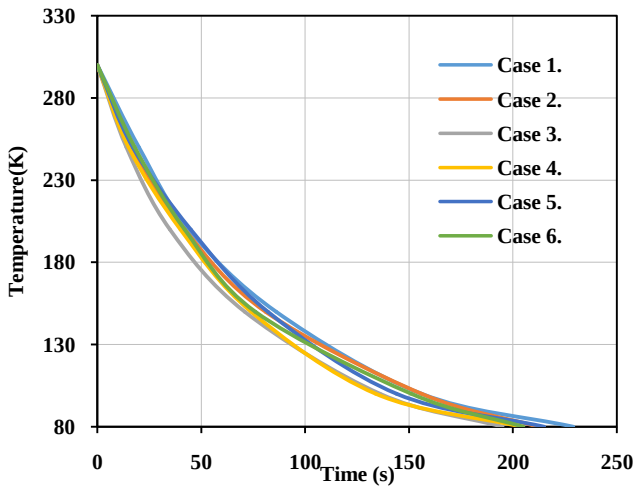

Fig. 10. Cooldown curve simulation with heat load using CFD simulation.

$$COP = \frac{Q_r}{W_{in}} \quad (11)$$

$$COP_{ideal} = \frac{T_c}{T_w} \quad (12)$$

Table VI provides a comparison of performance for the cases 1-6. The performance is compared as the ratio of Coefficient of Performance (COP) to that of the Ideal COP for pulse tube cryocooler. The COP of the cryocooler is given by the ratio of cooling load to that of the input power as given in eq. 11.

The ideal COP of Pulse Tube Cryocooler is the ratio of the temperature of warm end to that of the temperature of the cold end as in eq. 12[13]. The performance of cryocooler using inertance tube-bounce space and inertance-reservoir is compared and is shown as performance improvement. At 200 cm^3 volume, the difference in performance is 60.2%. The difference in performance decrease to 10.3 % and 4.99% for volumes 400 and 600cm^3. Figure 11 shows the variation of cooling load with the increase in reservoir and bounce space volume. The performance of the cryocooler with inertance tube- bounce space as phase shifter is superior.

Table-VI Performance comparison of Cryocooler.

Volume of reservoir/ bounce space	Case 1: 200cm^3 reservoir	Case 2: 200cm^3 Bounce space	Case 3: 400cm^3 reservoir	Case 4: 400cm^3 Bounce space	Case 5: 600cm^3 reservoir	Case 5: 600cm^3 Bounce space
Cooling capacity at 80K (W)	0.8	1.557	2.51	2.684	2.573	2.823
Compressor input (W)	71.6	87	64.9	62.9	63.6	66.5
COP/COP_{ideal} (%)	4.190	6.711	14.503	16.002	15.171	15.919
Performance Improvement(%)	60.2		10.3		4.99	

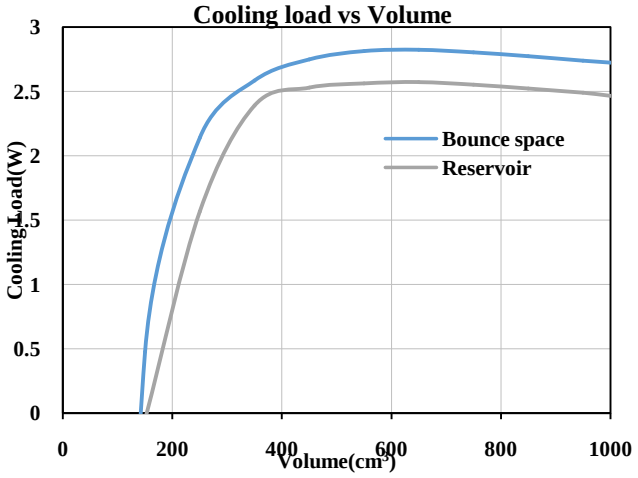

Fig. 11. Performance comparison for the cryocooler

V. CONCLUSION

An simulation model of cryocooler was developed based on the dimensions obtained from Sage software. To study the effect of volume on the performance of cryocooler an ANSYS Fluent model was developed. The proposed cryocooler has the option of operating with inertance tube-bounce space or inertance tube-reservoir as a phase shifter. The volume of bounce space or reservoir is as 400 cm^3 for experiments. The model was validated with experimental results and the model under predicts the outcomes with a maximum deviation of 20%. CFD simulations were conducted for three different volumes(200, 400 and 600 cm^3) of reservoir and bounce spaces with heat load. The cool down curve was plotted with the simulation, and all the combination gained 80K in less than 4minutes.

The study on the different reservoir and bounce space shows that the cryocooler could work effectively with inertance tube-bounce space as phase shifter. On increasing the volume of bounce space, there was a steep rise in performance till 400 cm^3 and a marginal increase henceforth. The performance increase in varying the bounce space volume from 200 to 400cm^3 was about 1.38 times. The comparative performance analysis based on Table VI shows that the cryocooler using inertance tube-bounce space as a phase shifter has better performance. The difference in performance between inertance tube-reservoir type and inertance tube-bounce space type was 60.2% while the difference in phase shift obtained was 22.42º. The difference in phase shift was 12.78 º and 9.5º for 400cm^3 and 600cm^3 respectively. The decrease in variation in phase shift obtained with the increase in volume resulted in a reduced difference in improvement. The cryocooler with 400cm^3 bounce space capacity is the best choice concerning performance and compatibility.

ACKNOWLEDGMENT

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ABBREVIATIONS

CFD	Computational fluid dynamics
IPTC	Inertance Pulse Tube Cryocooler
o.d	Outer Diameter (mm)
i.d	Inner Diameter (mm)
f	Frequency (Hz)
$P_{inlet\ 1}$	Inlet Pressure (Pa)
t	Time (m/s)
P_{charge}	Charge pressure (Pa)
$P_{inlet\ 2}$	Bounce space pressure (Pa)
$\Delta P_{\ inlet\ 1}$	Inlet pressure amplitude (Pa)
$\Delta P_{inlet\ 2}$	Pressure amplitude in bounce space (Pa)
ε	Porosity
α	Permeability
C_2	Inertial resistance factor(m^{-1})
ρ_f	Density of fluid (kg/m^3)
$\bar{\bar{\tau}}$	Stress tensor (N/m^2)
$\vec{v}$	Velocity (m/s)
S_i	Momentum source term (N/m^3)
T	Temperature (K)
E_f	Specific energy of fluid (J/kg)
k_f	Thermal conductivity of fluid (W/m.K)
h_{fs}	Heat transfer coefficient(W/m^2K)
A_{fs}	Interfacial area density (m^{-1})
τ	Period (s)
T_c	Temperature at Cold End (K)
T_w	Temperature at Warm end (K)
Q_r	Cooling load (W)
W_{in}	Input power (W)

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