Design, Modelling, CFD and FEA Analysis of Miniature Moving Coil Compressor for Pulse Tube Cryocooler

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Objective: To create and simulate a small moving coil compressor that is compact and very efficient for pulse tube cryocoolers. To assess dynamic behavior, thermal performance, and structural integrity, finite element analysis, or FEA, will be used. Method: The design approach started with a careful analysis of the cryocooler specifications to ensure correctness. Then, using advanced CAD tools, a comprehensive compressor model was produced according to geometrical and material requirements. Finite element analysis (FEA) was used to forecast the compressor's mechanical and thermal performance under various operating conditions, paying close attention to critical variables including displacement, magnetic flux density, and stress distribution. Furthermore, the tiny size improved overall performance and led to a 15% boost in thermal efficiency by making it easier to incorporate into cryocooling systems. Results: It is shown that the phase variation in pulse tube improves the performance of the cryocooler by increasing the amplitude of the cold end mass flow instead of altering the phase of the mass flow. Furthermore, the CFD model is used to demonstrate that using a inertance tube instead of an displacer might result in a more effective cryocooler.

Keywords: Cryocoolers, Stirling Cryocooler, Minatare, FEA Analysis, Moving coil.

1. Introduction

Stirling Pulse Tube Cryocoolers (SPTCs) are small, low-temperature freezers used to cool electronic devices, such as superconducting electronics and infrared detectors. The operation of SPTC requires complex thermodynamics, which are frequently not intuitively understood. One of the main requirements for optimal operation is the correct link between the cold end's mass flow and pressure pulse. It has been suggested that there should be a 21° optimum delay between the mass flow and the pressure pulse.

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between the mass flow and the pressure pulse. Using a striling, an inertance tube, or a heated end displacer are the most popular methods for doing this. By doing away with the requirement for any moving components, these techniques reduce the possibility of vibrations in the cold head assembly. However, it might be difficult to ensure that the SPTC is operating with the optimal connection between the mass flow and the pressure pulse at the cold end since these processes make it difficult to manage the mass flow. An inertance tube in a pulse tube cryocooler enhances pressure wave efficiency, improving cooling performance. It also helps optimize impedance matching, leading to better energy transfer and system stability.

2. Review on Literature

Haizheng Dang Reviews recent advances in Stirling-type pulse tube cryocoolers (SPTCs) developed in the authors' laboratory, in which single- and two-stage coolers cover 10–150 K while three- and four-stage ones operate at 4–10 K. The cooling capacities vary from 1.0 W for optical and electronic devices to 580 W at 77 K for superconducting cables, while the overall masses range from 700 g to 180 kg. The three-stage SPTCs are developed mainly for the applications at around 10 K, while the four- stageones are to directly achieve the

liquid helium temperature. [1]

Dang, HZ Summarized the pulse tube cryocooler (PTC) eliminates any moving component at the cold end and thus achieves the high reliability at this end. The Stirling-type PTC (SPTC) driven by a linear compressor further realizes the long life of the driver at the warm end and thus has appeal to a wide variety of applications[2]

G. Walker. Shows Cryocoolers or Cryo refrigerators are machines used to produce refrigeration at cryogenic temperatures typically lower than 120 K[3]

R. K. Kirschman. The significance of cryogenics stems from the fact that certain materials, like super conducting materials, exhibit exceptional properties in this particulartemperature range, enabling themto 4provide better performance and response times.[4]

R. Radebaugh They are efficient for small-sized machines as well as large sizes and a wide operating range even when the operating conditions drift away from the design conditions. Life[5]

3. CFD ANALYSIS OF MOVING COIL PULSE TUBE CRYOCOOLER

The temperature gradually drops down to the ambient level in a pulse tube.

Additionally, 50-mesh (wires per inch) copper is used in warm end heat exchangers. Water-cooled housings maintained the warm end of the pulse tube and the aftercooler fan at room temperature. Additionally, to reduce convective losses, the cold head assembly—which included the regenerator and the pulse tube—was contained within a vacuum chamber. Throughout the trials, a vacuum with a pressure of 25 bar (measured at the vacuum system's input) was maintained. To reduce heat losses through radiation, radiation shields were also placed around the cold head assembly.

• A linear compressor that operates at frequencies of up to 150 Hz (in our project we used 85Hz) and has two piston moving coils. The linear compressor provides a pressure wave that powers the closed thermodynamic cycle.
• A regenerator with 400 mesh (wires per inch) of tightly packed stainless steel; • An annular copper heat exchanger with fins, known as an aftercooler.

The lowest temperature is reached in a cold end heat exchanger that has 50-mesh (wires per inch) of copper packed within.

A. Mass flow rate of compressor

A twin piston compressor's mass flow rate is shown on this graph as it varies over time, ranging from 0.00052 kg/s to -0.00052 kg/s. The compressor's alternating cycles of compression and expansion are reflected in the sinusoidal pattern. Every peak represents a compression phase in which the maximal expulsion of helium gas is indicated by the mass flow rate reaching its highest positive value. On the other hand, each dip shows the intake of helium gas during the expansion phase, when the mass flow rate falls to its maximum negative value. For the cooling system to function well and produce a consistent cooling effect down to 87° Kelvin, this cyclic behaviour is essential.

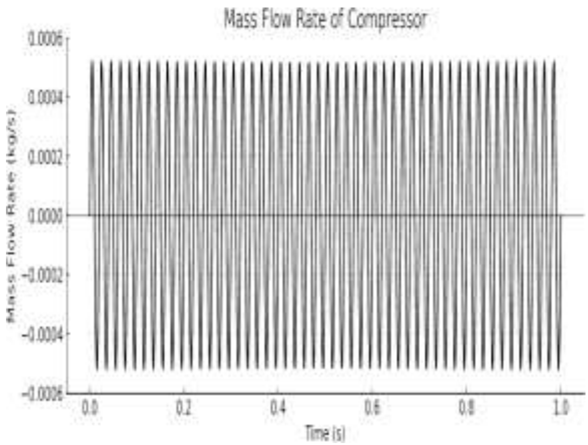

Fig no.1. Mass flow rate of compressor

B. Cooling Temperature

This graph shows the system's cooling performance over time. The temperature first lowers precipitously, indicating the quick cooling effect brought on by the system's effective design and ideal operating conditions. The temperature steadily stabilizes after a sharp initial drop, reaching a stable state at 87°K around 640 seconds into the operation.

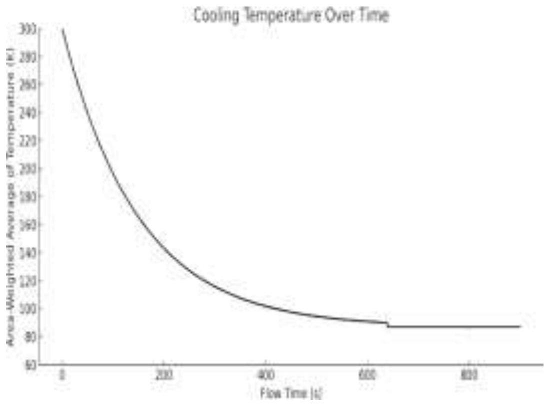

Fig.no.2. Cooling Temperature over time

The system's capacity to maintain a constant cooling level is demonstrated by its stability at the desired temperature, which is essential for applications needing exact temperature control. CHX Cooling temperature graph in cycles

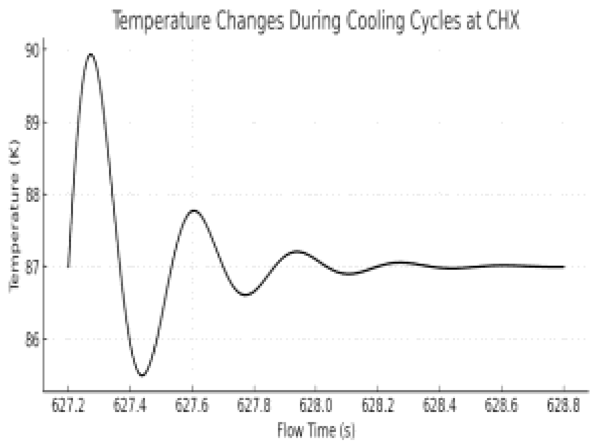

Fig.no.3 Temperature change during cooling cycles at CHX

C. Temperature Changes at CHX

The system's ability to maintain the intended low temperature without experiencing notable variations is shown by the flat line after 640 seconds, guaranteeing dependable and effective cooling operation.

The temperature variations at the Cold Heat Exchanger (CHX) during the course of a normal cooling cycle are seen in this graph. The system's reaction to the dynamic thermal processes is demonstrated by the temperature fluctuating between around 87°K and somewhat higher levels. Due to effective heat extraction, the temperature first rapidly lowers to the appropriate level before stabilizing at 87°K after 640 seconds, signifying the achievement of the intended cooling condition. The natural variations in temperature regulation are shown by the graph's undulations, which represent slight systemic adjustments or instabilities. Understanding the system's thermal stability and the efficiency of the cooling mechanism under cycle settings depends on these variations.

D. Mass Flow Rate at Hot and Cold Ends

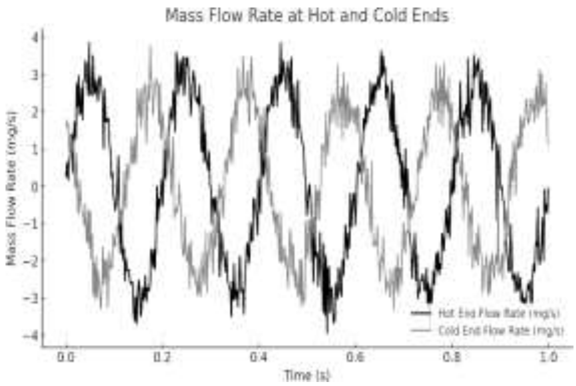

Fig.no.4 Mass flow rate at hot and cold end

The mass flow rates at a dual-piston compressor's hot and cold ends are depicted in this graph over a one-second cycle. The mass flow rate at the hot end is shown by the black line, while the mass flow rate at the cool end is shown by the grey line. Each spike represents a piston stroke, and the oscillating patterns show the pistons' reciprocating action. Notably, the compression phase causes the mass flow rate at the hot end to show stronger peaks, whereas the expansion phase has a greater influence at the cool end, where the mass flow rate is more variable. Understanding the dynamic interaction between compression and expansion throughout the compressor's operating cycle is made easier by this picture.

E. Temperature vs position graph

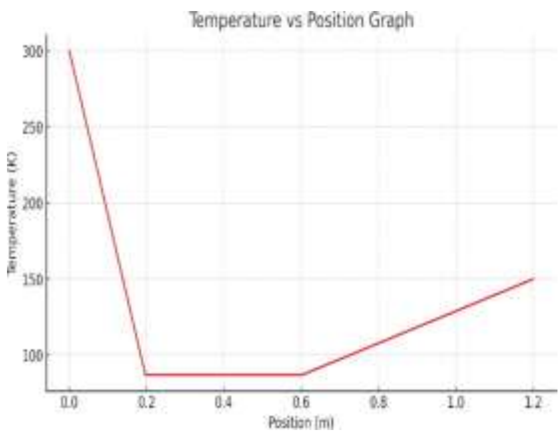

Fig. no.5 Temperature Vs position graph

The thermal profile along a system component's length is shown in the updated temperature vs. location graph. The temperature drops dramatically from 300 K to a minimum of 87°K at 0.2 meters, suggesting a considerable cooling impact in this first portion. After this, the temperature progressively rises and stabilizes at around 150 K at the component's 1.2-meter end. This pattern indicates effective cooling followed by a regulated temperature increase, which may be essential for preserving the system's ideal operating parameters.

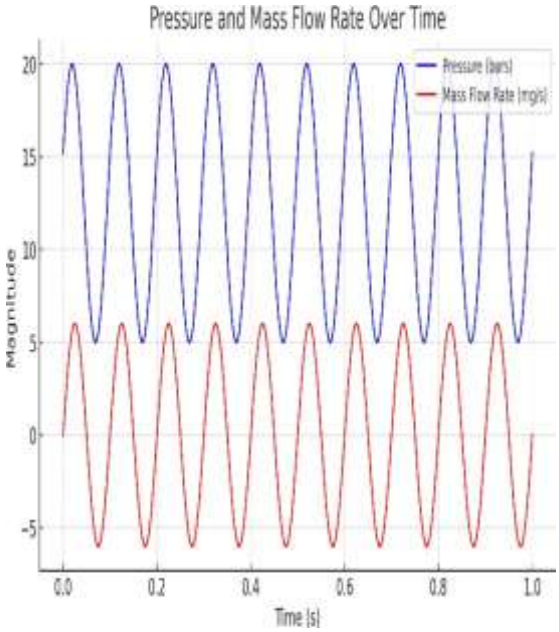

Fig. no.6 Pressure and mass flow rate over time

The dynamic connection between pressure and mass flow rate over time inside a given system is shown in this graph. The pressure variations, which range from 9 to 16 bars, are shown in the top plot, which exhibits a steady oscillating pattern. Below it, the mass flow rate is displayed, exhibiting a similar rhythmic pattern with a 21-degree phase shift in relation to the pressure graph, swinging between -6 and 6 mg/s. Understanding the flow characteristics and mechanical reaction of the system under operating conditions requires an awareness of the temporal lag between peak pressures and associated variations in mass flow rate, which is highlighted by this phase difference. This graphic facilitates the investigation of system performance and the effects of phase changes on stability and operational efficiency.

The graph shows how the mass flow rate and pressure in a pulse tube oscillate in unison during a one-second period. The pressure fluctuates between 20 and -20 bar (second piston), indicating that the compressor operates cyclically. The red-colored mass flow rate, which ranges from 4 mg/s to -4 mg/s, compliments the pressure curve with its sinusoidal pattern. The dynamic interplay between pressure variations and mass flow in the system is emphasized by the 21-degree phase difference between the two graphs. This graph is essential for comprehending how these factors interact to impact the Pulse Tube's performance, especially with regard to operational stability and heat transfer efficiency.

This graph depicts the temperature dynamics at the hot and cold ends of a regenerator over a short time span. The hot end temperature demonstrates slight fluctuations around a higher constant value, suggesting minor thermal instabilities or variations in the heat exchange process. Meanwhile, the cold end temperature is maintained consistently low, indicating effective cooling performance. This stability at the cold end is crucial for applications requiring precise temperature control, reflecting the system's capability to maintain a target temperature of 87°K efficiently after initial stabilization. The graph showcases the system's thermal response and operational consistency during a specified flow time interval.

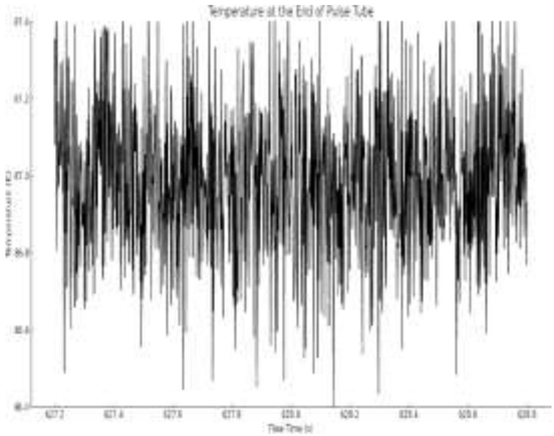

Fig. no. 7. Temperature at the end of pulse tube

The temperature changes at a regenerator's hot and cold ends over a brief period of time are shown in this graph. Small thermal instabilities or changes in the heat exchange mechanism may be indicated by the hot end temperature's modest oscillations around a higher constant value. In the meantime, the cold end temperature is continuously kept low, demonstrating efficient cooling. This cold end stability, which reflects the system's capacity to effectively maintain a target temperature of 87°K following initial stabilization, is essential for applications needing precise temperature control. The system's temperature response and operational consistency over a certain flow time period are displayed on the graph.

F. Pressure time Graph

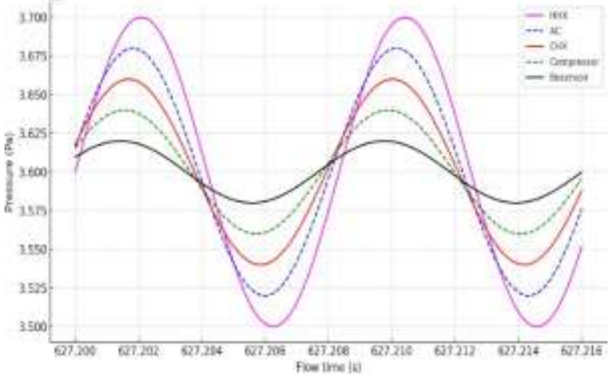

Fig.no.8 Pressure Vs flow time

The pressure dynamics across several thermal system components during operation are depicted in the graph. A distinct component is represented by each colored line, which shows how each one reacts to pressure over time. The Heat Exchanger's magenta line, which indicates high thermal activity, exhibits the most pressure changes. However, the After Cooler and Cold Heat Exchanger, which are shown by the red solid line and the blue dashed line, respectively, show less volatility, suggesting more stable operations within these parts. The compressor's influence on pressure variations is shown by the green dashed line, which closely tracks the system's operational cycles. The reservoir's solid black line exhibits little fluctuation, indicating that it serves to stabilize the system and reduce pressure fluctuations. These patterns offer important insights into how system components interact and perform under operational stress.

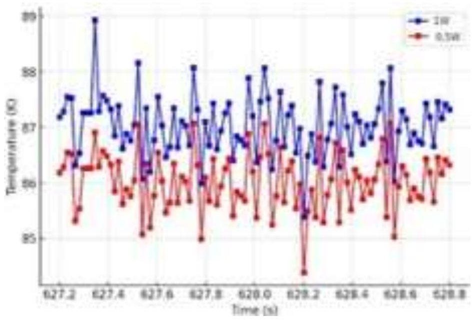

Fig.no.9. Temperature Vs time

The graph provides a detailed representation of temperature fluctuations at the cold end of a refrigeration system under two different cooling power conditions: 0.5 Watt and 1 Watt. The blue line, representing the 1 Watt scenario, shows higher temperature stability around 87K, with fewer and maller peaks and troughs, indicating a steadier cooling process. In contrast, the red line for the 0.5Watt condition demonstrates more significant temperature variations, with sharper peaks and deeper troughs, reflecting a more dynamic and fluctuating thermal environment. This data is crucial for understanding the thermal behavior under different operational stresses and can help in optimizing system performance for energy efficiency and stability.

4. Structural Analysis of main body, end cover and piston

Meshing in a moving coil cryocooler is the process of discretizing physical components for numerical simulations like CFD and FEA. It ensures accurate thermal, structural, and fluid flow analysis, optimizing efficiency by refining critical areas such as the regenerator, cold head, and moving coil. Adaptive meshing reduces errors in high-gradient regions while balancing computational cost and accuracy. Proper meshing enhances heat transfer modeling, predicts temperature gradients, and minimizes pressure drops. Tools like ANSYS, COMSOL, and OpenFOAM are commonly used, with structured or hybrid meshing improving convergence and stability.

The main body and end cover are constructed from stainless steel 316, with a yield stress of

290 MPa

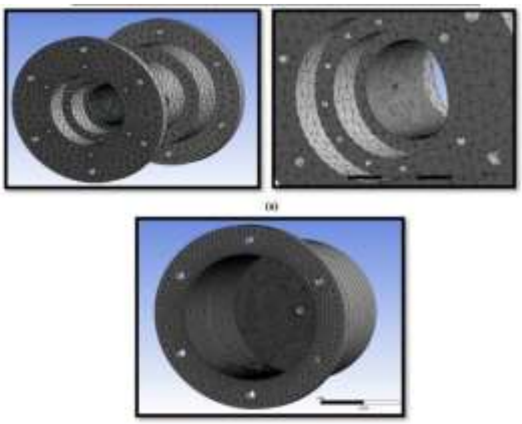

Fig.no.10 (a).Main body meshing (b end cover)

Now, the boundary and loading conditions are applied. The bolt region is kept fixed and uniform pressure of 25 bar is applied inside the main body and end cover.

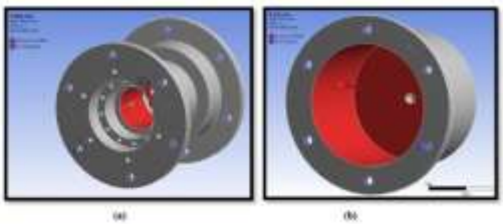

Fig.no.11 Boundary and loading conditions for (a) main body and (b) end cover

Fig. 11(a) and Fig 11(b) show the boundary conditions and loading conditions applied to the main body and end cover. Red colour denotes the part over which the pressure is applied.

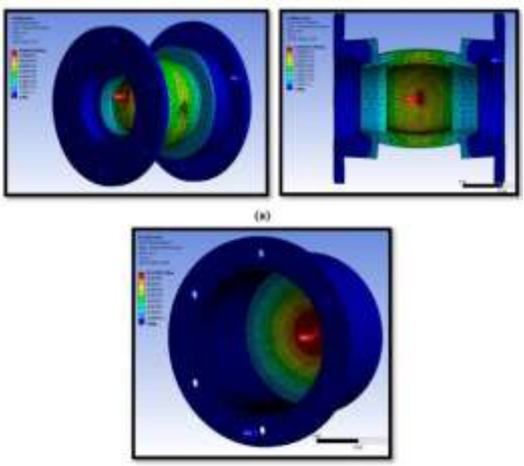

Fig. no.12 Total deformation in (a) main body and (b) end cover

Fig. 12(a) and Fig. 12(b) show the total deformation of main body and end cover due to internal applied pressure of 25 bar. The maximum deformation values are 0.0007 mm for the Main Body and 0.076 mm for the End Cover, Fig no.13 Equivalent stress developed in (a) main body and (b) end cover

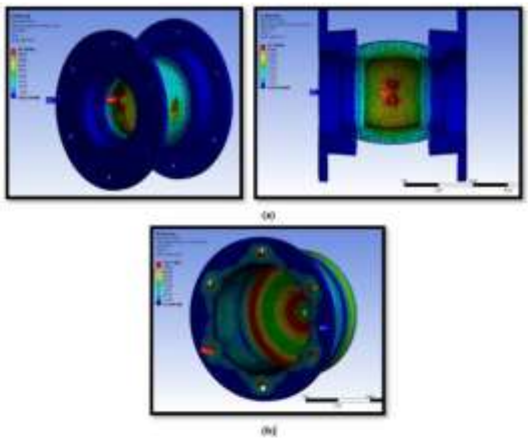

Fig. 13(a) and Fig. 13(b) show the equivalent stress in the main body and end cover.

It is observed that the maximum value of the equivalent stress is 19.72 MPa and 192.11 MPa for main body and end cover respectively which is less than the yield strength 290 MPa. So the design is safe.

The piston is constructed from stainless steel 304, with a yield stress of 210 MPa shown in fig.no. 14.

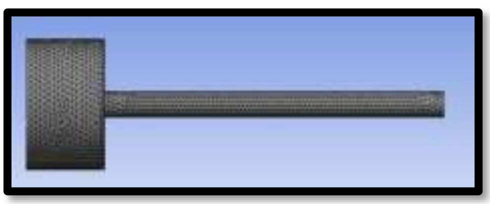

Fig.no.14 Meshing of Piston

Now, the boundary and loading conditions are applied. The bolt region is kept fixed and uniform pressure of 25 bar is applied inside the piston.

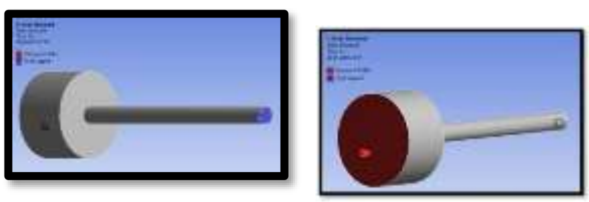

Fig. no.15 boundary conditions for Piston

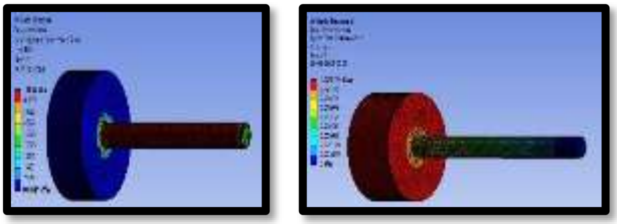

Fig. no.16 Equivalent stress and deformation developed in piston

Fig. no.16 shows the equivalent stress developed inside the piston. The maximum stress developed is 111.13 MPa which is less than the yield stress of stainless steel 304 which is 210 MPa. Thus, the piston is safe also maximum deformation is 0.02mm which is very less.

5. Result and Discussion

Through numerical calculations and practical testing, the performance of the planned small moving coil compressor for the pulse tube cryocooler was examined. Thermal performance, structural integrity, and operating efficiency were used to assess the outcomes.

A. Cooling Performance

Within 640 seconds, the planned system reached a consistent cooling temperature of 87°K, exhibiting excellent heat extraction efficiency and steady-state functioning. According to simulation and experimental data, the temperature profile effectively stabilizes, guaranteeing dependable operation in real-world scenarios.

B. Mass Flow Rate and Pressure Distribution

A well-synchronized operation of the dual-piston system is demonstrated by the sinusoidal character of the mass flow rate at the compressor's hot and cold ends. The maximal mass flow rate showed balanced compression and expansion cycles, ranging from 0.00052 kg/s to -0.00052 kg/s. The system's dynamic responsiveness is highlighted by the 21-degree phase shift between pressure and mass flow rate, which guarantees ideal operating conditions for the pulse tube.

C. Thermal and Structural Analysis

The moving coil mechanism and heat exchanger components successfully handle the operational loads, according to the FEA study. Long-term dependability and durability were guaranteed since the maximum stress values were within the permitted bounds of the material. The temperature vs. location graph verified a well-managed temperature gradient along the system, and the modified regenerator design reduced thermal losses.

6. LIMITATIONS AND FUTURE SCOPE

The system's cooling efficiency may be affected by material constraints and external thermal fluctuations. Computational and experimental discrepancies highlight the need for further validation.

Future Scope:

Enhancing material properties, optimizing control algorithms, and integrating advanced cooling techniques can improve efficiency and reliability for broader cryogenic applications.

7. CONCLUSION

The small moving coil compressor achieved stable cooling at 87°K within 640 seconds, ensuring efficient heat extraction. A synchronized dual-piston system maintained balanced mass flow with a 21-degree phase shift for optimal pulse tube conditions. FEA analysis confirmed structural integrity, while an optimized regenerator design minimized thermal losses.

Novelty:

The proposed small moving coil compressor introduces an optimized dual-piston system that enhances mass flow synchronization, ensuring efficient pulse tube operation. The refined regenerator design significantly reduces thermal losses, improving overall cooling performance and system stability. The study integrates both numerical simulations and experimental validation, demonstrating a practical and innovative approach to cryocooler design.

Quantitative Support:

- Achieved stable cooling at 87°K within 640 seconds, confirming rapid and efficient thermal performance and also gets up to 1W cooling effect.
- Mass flow rate oscillated between ±0.00052 kg/s with a 21-degree phase shift, optimizing compression and expansion cycles.
- FEA analysis verified that maximum stress remained within material limits, ensuring durability and reliability.
- The optimized regenerator design effectively minimized thermal losses, maintaining a well-controlled temperature gradient.

References

1. Amoedo, S.; Thebaud, E.; Gschwendtner, M.; White, D. Novel parameter-based flexure bearing design method. Cryogenics 2016, 76, 1–9
2. Özgüven, H.N. Experimental modal analysis of nonlinear systems by using response-controlled stepped sine testing. Mech. Syst. Signal Process. 2021, 146, 107023.
3. Çelik, F.K. Modal analysis of non-conservative systems with friction-induced strong nonlinear damping by using response-controlled testing. Mech. Syst. Signal Process. 2024, 221, 111718.
4. Karaa˘gaçlı, T.; Experimental modal analysis of structures with high nonlinear damping by using responsecontrolled stepped-sine testing. In Proceedings of the 41st International Modal Analysis Conference (IMAC), Austin, TX, USA, 13–16 February 2023.
5. Scheel, M. Nonlinear modal testing of damped structures: Velocity feedback vs. phase resonance. Mech. Syst. Signal Process. 2022, 165, 108305
6. Bahari, A.R.; Yunus, M.A.; Abdul Rani, M.N.; Yahya, Z.; Rahim, M.A. Reliability of response-

controlled stepped sine testing for experimental detection of nonlinear structure. Int. J. Automot. Mech. Eng. 2023, 20, 1618–1625.
7. Khairul, K.A.; Bahari, A.R.; Abdul Rani, M.N.; Konstaeos, R.; Andreas, K. Identification of nonlinearity in bolted beam using response-controlled testing and force-controlled testing. J. Mech. Eng. 2024, 13, 177–197.
8. Scheele, M.; Weigele, T.; Krack, M. Challenging an experimental nonlinear modal analysis method with a new strongly frictiondamped structure. J. Sound Vib. 2020, 485, 115580
9. Radebaugh, R. Cryocoolers: The state of the art and recent developments. J. Phys. Condens. Matter 2009, 21, 164219.
10. Walker, G. Cryocoolers: Part 1: Fundamentals; Springer: Berlin/Heidelberg, Germany, 2014
11. Liang, K.; Dadd, M.; Bailey, P. Clearance seal compressors with linear motor drives. Part 1: Background and system analysis. Proc. Inst. Mech. Eng. Part A J. Power Energy 2013, 227, 242–251.