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Design, Development and Testing of Moving Magnet Linear Motor Compressor

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Abstract: ***Cryocoolers are special equipment developing lower temperatures. The main part of cryocooler is compressor. Compressor consumes most of the energy of system, so by developing the efficient compressor energy can be saved. The old compressors uses crank mechanism to convert rotary motion to linear motion but it causes friction and loss in the efficiency. Moving magnet or moving coil linear motor actuators can be used in linear motor compressor. Moving magnet compressor uses permanent magnets and coil. In moving magnet compressor magnet moves and coil stays stationary. The coil excites equal and opposite forces on magnets with 10mm axial displacement and at low capacity. The present work aims to design, develop and manufacture a linear moving magnet compressor using flexural bearing.***

Keywords: cryocooler, moving magnet compressor, flexural bearing, FE analysis**.**

1. Introduction

Two main parts of any cryocoolers are compressor and expander. Earlier, the expander used to have moving parts, but after the development of pulse tube coolers, compressor is only moving part. Cryocoolers should satisfy various requirements for any space borne application. These are low input power, high reliability, long life time, maintenance free operation, low weight, minimum vibration/noise, compactness [3]. These diverse requirements affect the type and design of compressor to be used and hence it determines the reliability of compressor.

Due to the limited life of disciplined compressor linear, electromagnetically driven compressors are developed, which utilizes either moving coil or moving magnet type linear motor. A linear compressor runs by the linear motor and not by the crank driven mechanism. If the crank and rotary bearings are removed from the assembly then the frictional losses will be reduced significantly. Because of the oil free system the condenser and evaporator systems in the refrigerator will work properly.

Using the permanent magnets made up of high flux density materials such as NdFeB material the linear compressors can be made compact and efficient .The moving mass can be reduced as compared to other type of compressor by generating the same force using the NdFeB magnets. For the linear motor the resonant system can be generated using simple flexural bearings. Though the price of the permanent magnets is high because of its uses it is in major demand now a days.

An air gap should be as low as possible because increase in the air gap will reduce the field strength. So the inner yoke can be designed differently such as using moving magnet and moving yoke. The piston reciprocates linearly. The air gap should be minimum to avoid the action of rubbing. This dissertation work is concerned with the design and development of linear motor compressor with power input of 100 W and having piston stroke of ±5 mm. The flexure bearings are used for supporting the weight of piston and coil while maintaining the clearance seal of permanent magnet [1]. The radial force and the torque on the moving coil is less the other actuators.

To design the moving magnet compressor, basic parameters have to be considered for optimization. That will include the material of magnet, orientation of magnet, air gap between the translator and yoke, operating frequency, type of flexural bearings. The resonant frequency of the moving magnet compressor depends upon flexural spring, gas spring, magnet spring and total moving mass [3].

2. Analysis of Compressor

For analysis purposes following specifications were developed:

Swept volume: 10 cc, Input power: 100 W, Charge pressure: 10 bar, Pressure ratio: 1.0 to 1.2, Piston stroke: 10 mm, Frequency: 50 Hz, Efficiency goal: 65-75%.

To achieve these specifications following parametric analysis is carried out.

From the parametric analysis the optimum values for the parameters are selected and are used in the calculations of linear motor.

Table 1: Final optimized parameters of all parameters

Parameter	Outer Diameter(mm)	Inner Diameter(mm)	Thickness (mm)
Magnet	35	5	7
Inner pole piece	38	8	9.5
Outer pole piece	112	100	27.9
Coil winding	100	38	22.8

2.1 Magnetic circuit design

Magnet material selected is Nd-Fe-B N52 grade. From the cyclic analysis, the power input to the opposed piston linear compressor is around 100W for a pressure of 10 bar. Thus the power output of the opposed piston compressor would be,
Total input power of the motor = 100 W, Magnet inner diameter D_{mi} = 5 mm, Magnet outer diameter D_{mo} = 35 mm, Length of magnet L_m = 7 mm.

Inner diameter of the magnetic gap is same as the outer diameter of the magnet. Therefore Gap inner diameter D_{gi} = 39 mm, Length of magnetic gap L_g = 30.5 mm, Gap outer diameter D_{go} = 100 mm, Gap height h_g = 19 mm, Mean diameter of gap D_g = 69.5 mm.

Fringing flux is the flux bulging out of the magnetic gap is useful flux, which, can be considered by assuming the coil height just greater than 1.3 times the gap height. Therefore

Coil height h_s` = 22.8 mm, Amplitude of piston displacement X_p = 5 mm, Wire diameter (27 SWG gauge, copper wire)= 0.45 mm, Specific wire resistance R_s = 0.1326 Ω/m at 0°C, Temperature coefficient of resistance of copper= $0.00395K^{-1}$ Frequency = 50 Hz, Angular velocity = 2 ×50= 314.16 rad/s,Yoke intermediate diameter D_{ym} = 102 mm, Yoke outer diameter D_{yo} = 110 mm, Number of layers = 22.8/0.45=50, Total number of turns =3388, Wire length Ls = 727.75 m, Total resistance of wire RT = 107.93 Ω, Area of magnet A_m $=\pi/4\ (D_{mo}{}^2 - D_{mi}{}^2)$ = 942.477mm^2, Area of gap = 2074.23 mm^2

The output power to be developed by the motor is evaluated as 27 W from the analysis of the cryocooler. The required impressed voltage amplitude is calculated using the expression of output power for phase angle Φ = 98.1º, Motor efficiency η = 60.05%, Maximum Force generated by motor = 77.8517 N.

The graph of Force generated by the motor with respect to the piston position is calculated by MATLAB program and graph is plotted below.

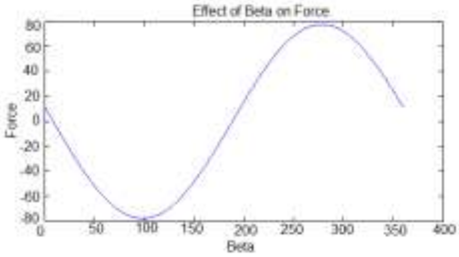

Figure 1: Effect of piston position on the force generated

2.2 Selection of flexural bearing

For present design, beryllium-copper is selected because of its high fatigue strength. Design procedure is used to decide the number of flexures and the thickness of the flexure. For obtained thickness the number of flexure springs is selected using design charts. [2]

Mass of system = 0.338771 kg, the gas spring stiffness is given by Gas stiffness k_g = 24883 N/m, spring stiffness (k_s) = 8553 N/m, Number of flexure discs= 15.

2.3 Force developed by compressor

Force calculated for particular geometry:

Working gas: helium (He), Charging pressure: 16 bar, Cylinder diameter: 36 mm, Sleeve inner diameter: 30 mm, Piston diameter: 35.97 mm, Frequency: 50 Hz, Clearance: 15 µm, Damping coefficient C =49.2686 Ns/m, Flexural spring stiffness (Ks) = 572 N/m, Equivalent stiffness of system, Ke = K_g+K_s = 33436 N/m, Mass of system = 0.338771 kg, Natural frequency = 50Hz, Cc = 212.85 Ns/m, Damping factor = 0.231.

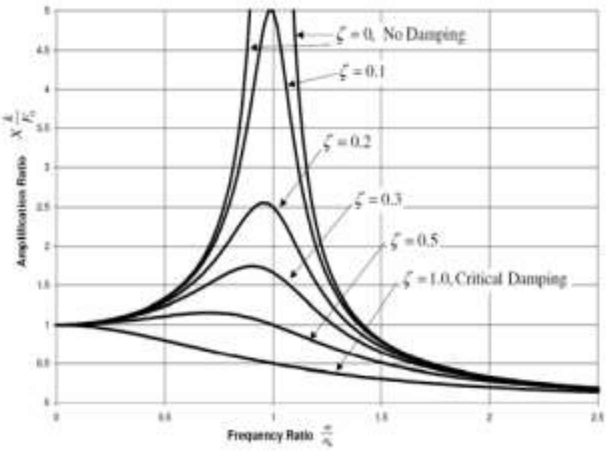

Figure 2: Amplification ratio vs. frequency ratio graph

Magnification Factor = 2.16. Using above values calculated force is F = 77.391 N. This amount of force has to give by motor to satisfactory work of compressor. Because the maximum force generated by motor is 77.852 N.

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Table 2: Summary of the linear motor design for the compressor

Sr. No.	Content	Value
1	Operating frequency, F, Hz	50
2	Power input of motor, Po, W	60.74
3	Magnet material	Nd-Fe-B N52 grade
4	Pole piece material	Soft iron
5	Saturation flux density, T	0.42
6	Inner diameter of magnet, Dmi,, mm	5
7	Outer diameter of magnet, Dmo, mm	35
8	Length of the magnet, Lm, mm	7
9	MMF loss factor, K2	1.45
10	Magnet flux density, Bm, T	0.2241
11	Gap flux density, Bg , T	0.1721
12	Coil height, hs, mm	22.8
13	Diameter of the copper wire, Dw mm	0.45
14	Total length of the coil winding, LS , m	564.7
15	Total resistance of coil, RT, Ω	83.71
16	Power input, PI, W	100
17	Efficiency of the motor, %	60.74

3. FE Analysis of Compressor Body

The analysis on the outer body of the linear motor compressor is done using commercial ANSYS software. The pressure equal to 18 bar is supplied. Using pressure vessel theory following are the calculations done to select the dimensions of the main body layout.

The input parameters are p = pressure inside main body=18 bar, d = diameter of the cylinder=36 mm.By doing calculations, t_{design} = 0.11 mm, t_{actual} = 25 mm. For d=58.9 mm, t_{design} = 0.57 mm, t_{actual} = 13.55 mm. For d=77.9 mm, t_{design} = 0.91 mm, t_{actual} = 8 mm.

Similarly, following are the calculations done to select the dimensions of the end cover layout.
For d = 116 mm, t_{design} =1.13 mm, t_{actual} = 7 mm. For d = 30 mm, t_{design} =0.19 mm, t_{actual} = 5 mm

In preprocessing of the specimen, a 10 node tetrahedral 187 of element type is given to the specimen and loading is done. To check the strength of the specimen von-mises stresses are considered as it is based on the distortion energy required to yield the specimen. Figure 3 shows the CAD model of the compressor layout and figure 4 shows the meshed model of the compressor layout.

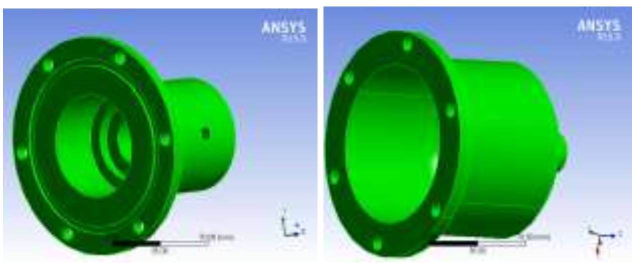

Figure 3: CAD Model of main body and end cover

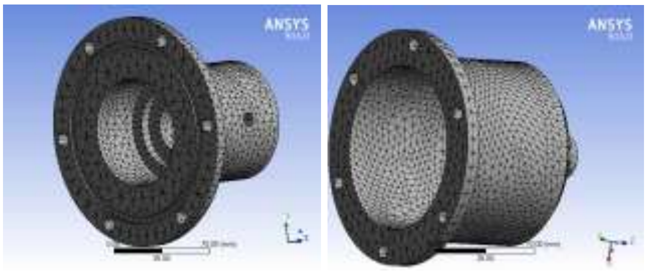

Figure 4: Meshed model

While doing meshing, 10 node tetrahedral 187 of element type is given.

Table 3: meshing details

Sr. No.	Part Name	Number of Nodes	Number of elements
1	Main Body	25418	14411
2	End Cover	30830	17106

Figure 5 shows the boundary conditions applied. The bolt region is fixed in this case.

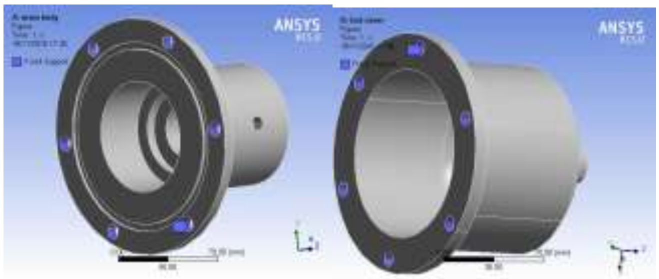

Figure 5: Boundary conditions

Figure 6: Loading conditions

The red colour shows the part of the compressor body which will be under pressure while working of the compressor. Generally the pressure inside the compressor body is 10 bar. But considering the factor of safety the pressure of 18 bar is applied inside surface of the compressor body.

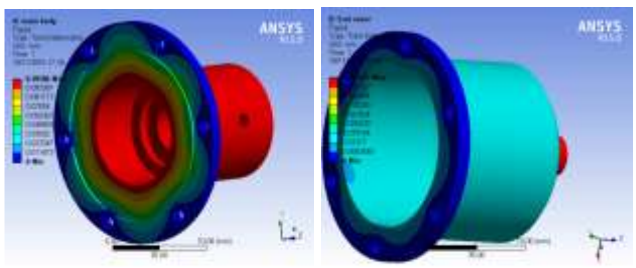

Figure 7: Total deformation

Figure 7 shows the total deformation of compressor layout due the applied internal pressure. The deformation values are shown in the following table.

Table 4: maximum deformation

Sr. No.	Part Name	Maximum deformation(mm)
1	Main Body	0.10506
2	End Cover	0.05746

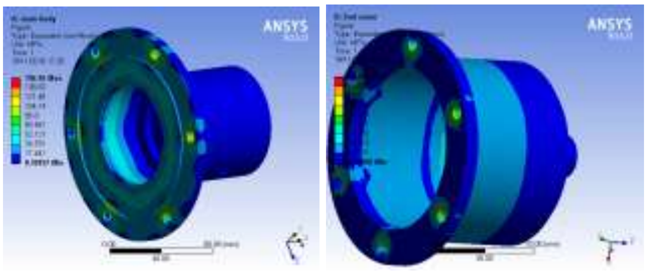

Figure 8: Equivalent stress

Figure 8 shows the equivalent stresses in the compressor body. Maximum stress generated inside the compressor body is shown in the following table.

Table 5: Maximum stress

Sr. No.	Part Name	Maximum stress generated (Mpa)
1	Main Body	156.16
2	End Cover	84.159

The yield stress of this material is 250 Mpa at room temperature. The max value of equivalent stress is below yield stress. So the design is safe.

4. Development of Compressor

We designed and selected parts for the compressor. All the components required for the compressor assembly are listed out here.

Table 6: Part List

Sr. No.	Name	Material	Quantity
1	Sleeve	C.I.	1
2	Magnet	NdFeB	1
3	Outer pole piece	Soft iron	1
4	Inner pole piece	Soft Iron	2
5	Main Body	Aluminium	1
6	End cover	Aluminium	1
7	Flexure support ring	Structural steel	1
8	Flexures	Beryllium copper	25
9	LVDT attachment	Structural steel	1
10	Coil supporter	Delrin	1
11	inner spacers	Aluminium	40
12	outer spacers	Aluminium	40
13	Piston and shaft	Structural steel	1
14	Pin	Structural steel	1
15	End cover O- ring	Neoprene	1

5. Testing and Results

The assembled compressor runs at atmospheric pressure, 1 bar, 2bar, 3bar, 4bar, 5bar charging pressure. The clearance volume was set up at 7cc while working. The input voltage was 160 volts. The fluid used for the experimentation is air. The compressor runs for some time for each charging pressure. The pressure drop due to leakage through the compressor body was 0.01 bar per minute.

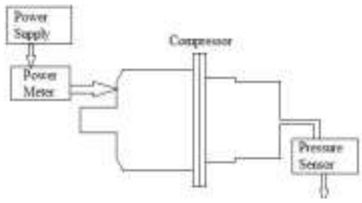

Figure 9: experimental set up

The instrument used to check the pressure variation is given below:

Pressure Variation Measurement:
(a) Charging Pressure
Bourdon Pressure Gauge
Range: 0-21 bar, least count: 0.5
(b) Dynamic Pressure:
Model: Dynamic pressure indicator – KI-KIT-P-100
Make: KAPTL INSTRUMENTATION- DEHRADUN
Range: 0 – 35 bar, least Count: 0.01 bar.

The dynamic pressure measuring instrument is used to measure the cyclic variation of the pressure generated inside the cylinder. Following graphs shows the pressure generated inside the compressor at various charging pressures:

5.1 Observations of cyclic pressure fluctuation at 1.08 bar

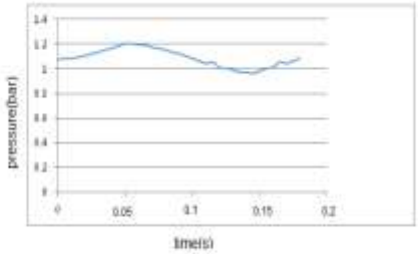

Figure 9: readings at 1.08 bar

From the graph it is observed that the pressure ratio is 1.237.

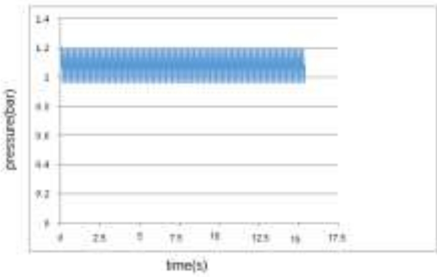

Figure 10: readings at 1.08 bar

From figure 9 it can be seen that the maximum achievable pressure is 1.2 bar while the minimum pressure is 1.07 bar. From figure 10 the thick band of blue line can be seen. This is due to the number of fluctuations per second. The pressure measurement instrument take 200 samples per second and the input frequency is 50 Hz. So it will take 4 samples each cycle.

5.2 Observations of cyclic pressure fluctuation at 2.07 bar

The compressor is charged at 2.07 bar with air. Then the compressor runs for some time. The readings are taken using the dynamic pressure indicator. The following graph shows the pressure fluctuations with respect to time.

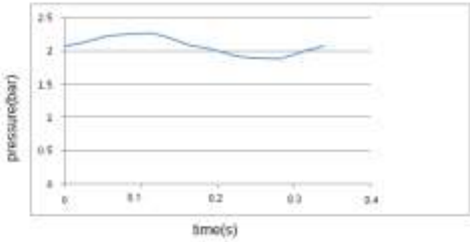

Figure 11: readings at 2.07 bar

From the graph it is observed that the pressure ratio is 1.2021.

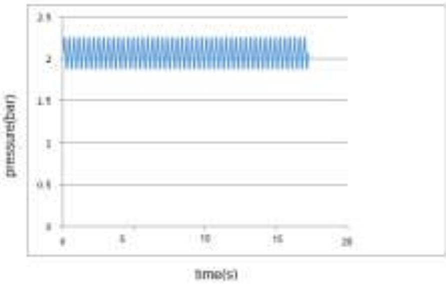

Figure 12: readings at 2.07 bar

5.3 Observations of cyclic pressure fluctuation at 3.07 bar

The compressor is charged at 3.07 bar with air. Then the compressor runs for some time.

The readings are taken using the dynamic pressure indicator. The following graph shows the pressure fluctuations with respect to time.

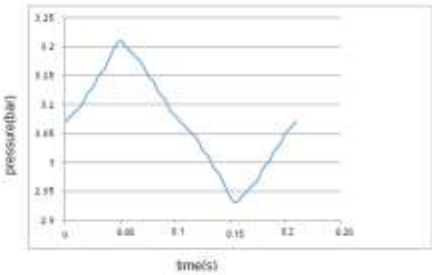

Figure 13: readings at 3.07 bar

From the graph it is observed that the pressure ratio is 1.09.

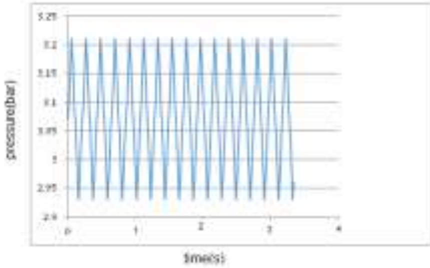

Figure 14: readings at 3.07 bar

5.4 Observations of cyclic pressure fluctuation at 4.05 bar

The compressor is charged at 4.05 bar with air. Then the compressor runs for some time. The readings are taken using the dynamic pressure indicator. The following graph shows the pressure fluctuations with respect to time.

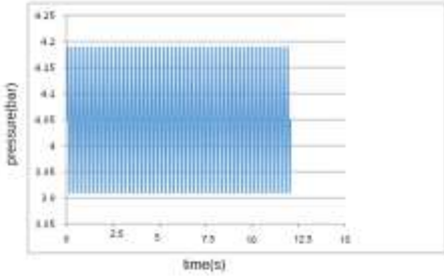

Figure 15: readings at 4.05 bar

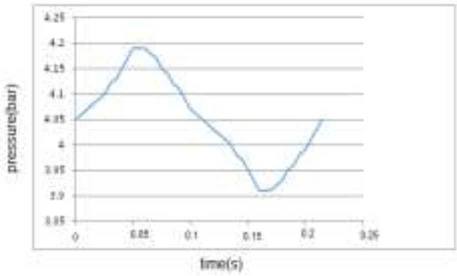

Figure 16: readings at 4.05 bar

From the graph it is observed that the pressure ratio is 1.07.

5.5 Observations of cyclic pressure fluctuation at 5.03 bar

The compressor is charged at 5.03 bar with air. Then the compressor runs for some time. The readings are taken using the dynamic pressure indicator. The following graph shows the pressure fluctuations with respect to time.

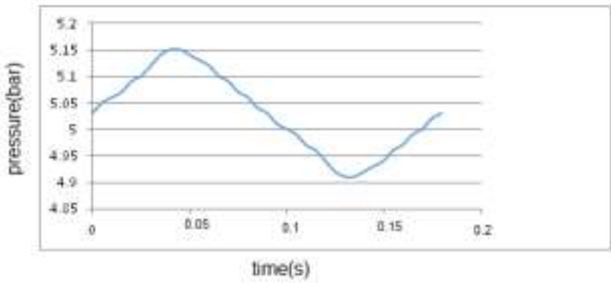

Figure 17: Readings at 5.03 bar

From the graph it is observed that the pressure ratio is 1.06.

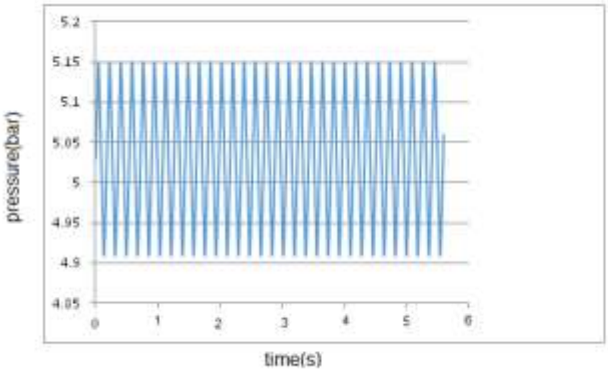

Figure 18: readings at 5.03 bar

Figure 19: Temperature measured using thermocouple

The compressor was run for 30 minutes with different charging pressures. The temperature of the magnet was checked with the help of thermocouple. The room temperature was 27 degree Celsius and after working for 30 minutes the temperature 37.3 degree Celsius.

Table 7: Pressure fluctuation with respect to charging pressure

Sr. No.	Pressure (bar)	Pressure Ratio
1	1.08	1.237
2	2.07	1.2021
3	3.07	1.09
4	4.05	1.07
5	5.03	1.06

6. Conclusion

The MATLAB program is developed to optimize the dimensions of all the components for maximum efficiency keeping the total moving mass as small as possible and input voltage less than 230V. The magnet is selected as per the availability. Effect of all the components is observed theoretically. The single piston, moving magnet compressor needs to be designed using complex equations which have mutually interactive effect. Hence the parametric analysis has been done to study the effect of different geometric parameters on the efficiency of motor. The damping force is calculated for the working of the compressor. Proper functioning of the linear motor demands high precision machining and very careful handling and assembly of the unit. Narrow clearances are required to be maintained between mating parts such as piston and cylinder, pole piece assembly and coil. Hence special care is taken during machining process.

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