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THE DESIGN OF A SMALL LINEAR-RESONANT, SPLIT STIRLING CRYOGENIC REFRIGERATOR COMPRESSOR

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For the past two years, Mechanical Technology Incorporated (MTI) has been engaged in the development of a small linear-resonant compressor for use in a 1/4-watt, 78K, split Stirling cryogenic refrigerator. The compressor contains the following special features: 1) a permanent-magnet linear motor; 2) resonant dynamics; 3) dynamic balancing; and, 4) a close-clearance seal between the compressor piston and cylinder. This paper describes the design of the compressor, and presents component test data and system test data for the compressor driving a 1/4-watt expander.

Key words: close-clearance seal; compressor; cryogenic refrigerator; dynamic balancing; linear-resonant compressor; permanent-magnet linear motor; resonant dynamics; split Stirling; 1/4-watt expander

1. INTRODUCTION

The objective for this program was to design, fabricate, and test a linear compressor that could be used in the military's 1/4-watt split Stirling cooler. The compressor was designed to demonstrate that it could ultimately meet the existing specifications for size, weight, and input power when driving an existing expander. The important requirements designed for in the compressor are:

1) high reliability, and MTBF's greater than 3000 hours;
2) the elimination of all lubricants;
3) low mechanical vibration; and,
4) the elimination of contacting seals.

Items 1 and 2 were achieved by using a linear drive system that eliminated mechanical linkages and the need for lubricants. Item 3 was achieved by incorporating into the compressor a balancing technique previously developed and patented by MTI, and Item 4 was achieved by using a close-clearance piston/cylinder seal.

2. COMPRESSOR DESCRIPTION

The prototype linear cooler developed is shown in Figure 1, and the design of the compressor is shown in Figure 2. Beginning at the left end, the compressor consists of the piston/cylinder, with the piston connected to the motor plunger through a flexible rod. The plunger, which is made up of radially energized Sm Co_5 permanent magnets and a back iron, is supported on either end by teflon bearing pads that position the plunger inside the motor stator. The stator contains the electric coils and the end caps rigidly fastened to the stator on either end form the compression cylinder and bearing assemblies described above. On the right end of the compressor is the counterbalance weight, which is connected to the plunger through a mechanical spring and attached to the outer case, along with the stator, through a flexible support configuration. Mechanical springs between the plunger and end caps provide the resonant characteristics. The outer pressure case, gas transfer tube, and displacement sensor located in the center of the plunger complete the design.

Figure 1 MTI Linear 1/4-Watt Split Stirling Breadboard Cooler

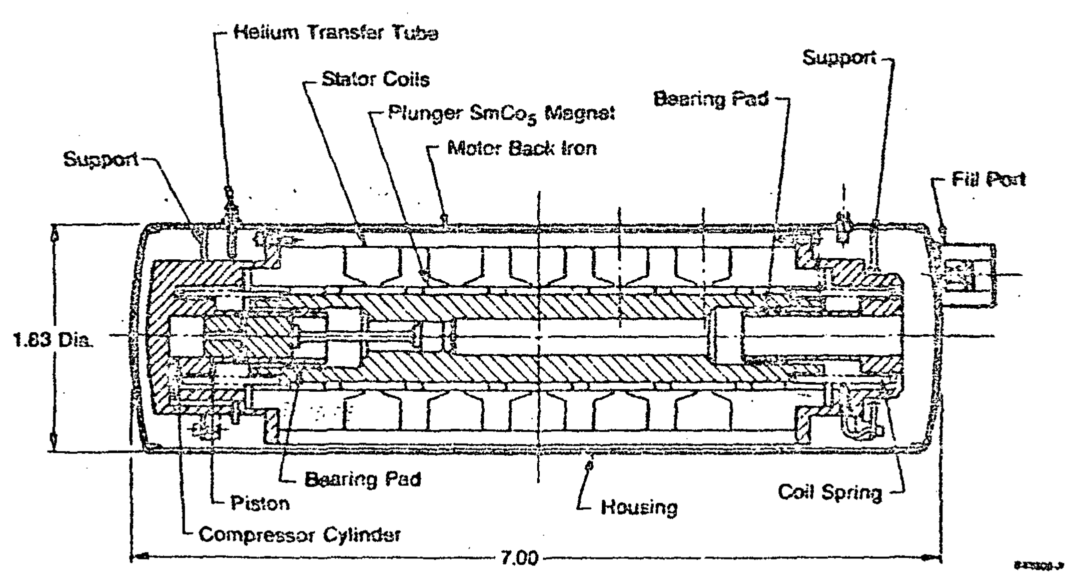

Figure 2 Compressor Layout (Dimensions in inches)

The specifications for the compressor were:

Frequency	22	Hz
Mean Pressure	600	psia
Pressure Differential	400	psia*
Input Power	>30	watts
Weight	>2.5	lb

In the design, it turned out that the critical design trade-off parameters were:

- bore-to-stroke ratio;
- resonant frequency;
- mechanical spring design;
- weight of the motor plunger; and,
- pressure differential.

The bore-to-stroke ratio was important in establishing the size of the motor, the compression-space spring component, the stress in the mechanical springs, and the piston leakage and frictional losses. A trade-off analysis of these parameters with the bore-to-stroke ratio showed (Figure 3) that an upper limit on stroke of 0.6 inch was set by the stress in the mechanical springs, and a lower limit of 0.3 inch was set by the outside diameter of the motor and the amount of gas spring developed in the compression space. The mechanical springs were limited to a stress of 60,000 psi, the gas spring was limited by a maximum plunger weight of 0.5 lb, and the motor size was limited by the maximum diameter specification of 1.750 inch. As for the bore-to-stroke ratio, the other four items also relate to the dynamic operation of the compressor. For this design, the operating frequency of the compressor was specified as 22 Hz. Achieving resonant operation in a high-pressure differential, low-stroke compressor at this low frequency is difficult because of the large spring component derived from the compression space. Balancing this spring with the inertia component of the moving assembly required a large plunger weight. To minimize this weight, a maximum stroke of 0.6 inch was selected.

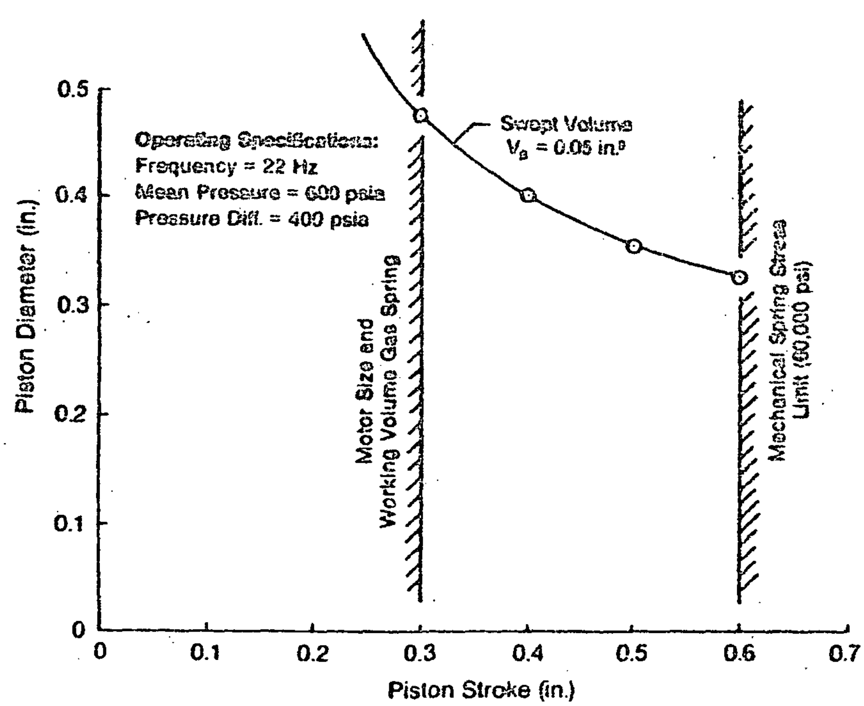

Figure 3 Compressor Bore-Stroke Relationship

*When connected to a 1/4-watt split Stirling cooler expander or equivalent volume.

The final design criteria for the compressor were:

Frequency	-	22 Hz
Swept Volume	-	0.82 cm^3
Stroke	-	15.24 mm
Bore	-	7.9 mm
Motor Diameter	-	44.45 mm
Plunger Weight	-	0.25 kg

2.1. Motor Characteristics

The linear motor used is one of a family of permanent-magnet ($Sm\,Co_5$) linear motors developed at MTI. Figure 4 schematically shows a section of the motor with the plunger in the center position. The motor consists of stationary outer electrical coils wound on a steel bobbin, and an inner permanent-magnet plunger. The magnets are radially magnetized, with alternate magnets having north polarity on the outside diameter and south polarity on the inside diameter. The design characteristics for the motor are:

Frequency	22 Hz
Stroke	15 mm
Static Force	35.6 nt @ 2.0 amp D.C.
Electrical Input Power	<30.0 watts
Voltage	17.5 V.A.C.
Weight	< 1.0 Kg

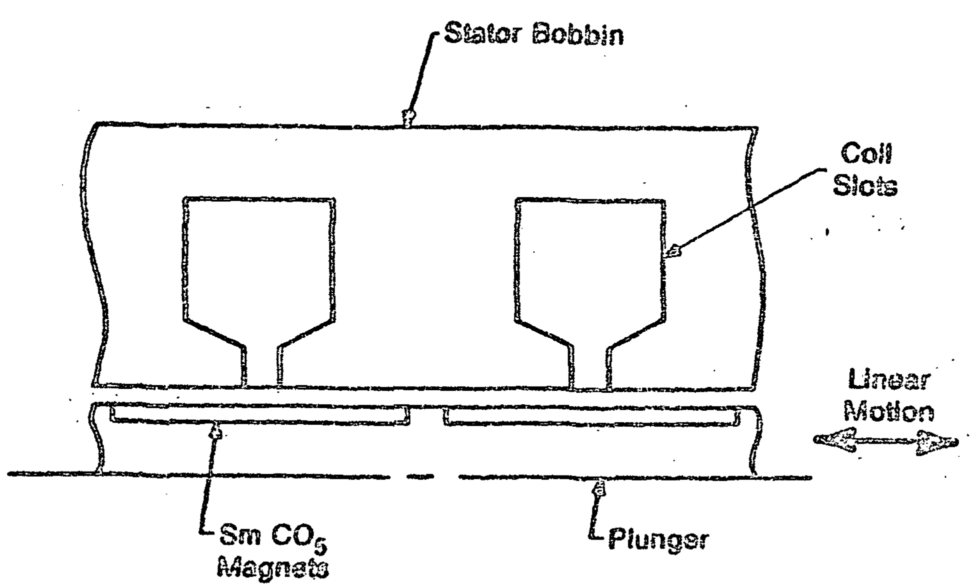

Figure 4 Permanent-Magnet Motor

2.2. Vibration Isolation

Vibration isolation is achieved by employing a counterweight system with the isolation characteristics described in Figure 5. For this isolation technique, the counterweight is attached to the stator through a mechanical spring, and both the stator and counterweight are supported from the housing by mechanical springs. Through the proper design of this spring mass system, the counterweight (M_m) is tuned so that the transmitted force to the case through the two springs (K_3 and K_4) may be designed to cancel one another, producing zero transmissibility. The advantages of this system are that the counterweight mass and amplitude are kept small, and good force isolation can be obtained over a broad operating range of frequencies (ω) equal to or near ω_o. The characteristics of the isolation system are:

$$
\begin{aligned}
K1 &= 343 \ \text{nt/m} \\
K2 &= 820 \ \text{nt/m} \\
K3 &= 491 \ \text{nt/m} \\
M_m &= 0.09 \ \text{kg} \\
\omega_n &= 18 \ \text{Hz} \\
\omega_o &= 22 \ \text{Hz}
\end{aligned}
$$

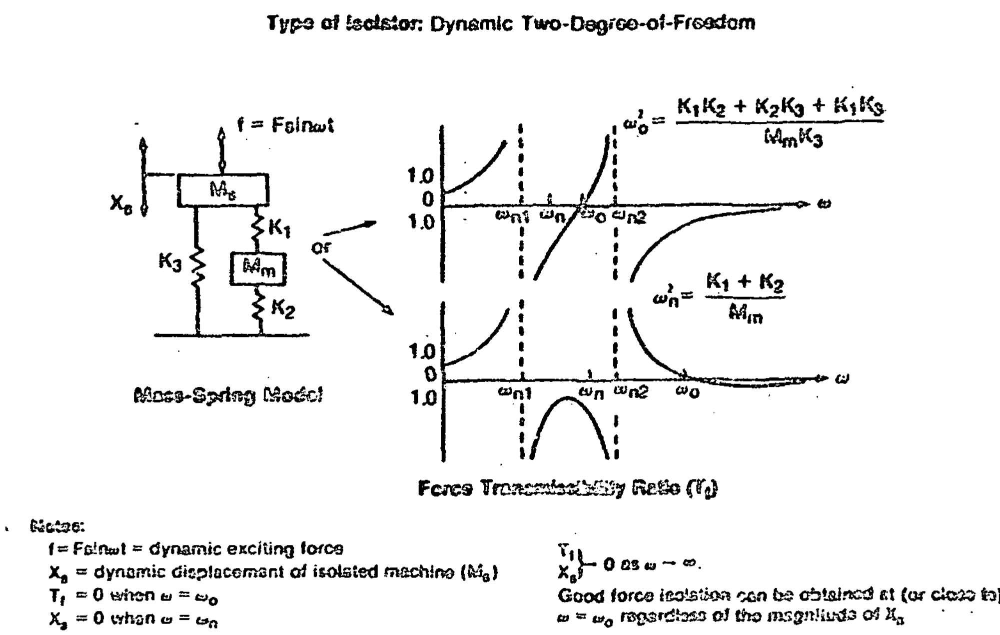

Figure 5 Dynamic Force Transmissibility Characteristics
--Vibration Isolation Technique--

2.3. Test Results

Testing of the compressor was performed using both a 1/4-watt split Stirling cooler expander and a simulated test volume that matches the expander and transfer tube volumes at the design operating point of 80°K. Tests conducted on the compressor showed that the compressor performed better at 30 Hz than at 22 Hz. Rather than change the plunger mass to reduce the operating resonant frequency, it was felt that at this point in the development, testing at the higher frequency should continue, and the inefficiences from the mismatch between the compressor and expander should be accepted.

During the initial checkout tests of the compressor, it took significantly more than 30 watts to develop the required pressure differential of 400 psi. The problem was diagnosed as too large a clearance between the compressor piston and cylinder. By inspecting the hardware, it was found that instead of a radial clearance of .0002 to .0004 inch, the actual clearance was .0006 inch. After refurbishing this hardware and achieving a .0003-inch radial clearance, the machine performed much better. The pressure difference generated by the machine is given in Figure 6. As shown, at the design specification of 400-psi pressure difference, the input power was less than 30 watts.

The cooling capacity of the compressor driving the 1/4-watt expander (shown in Figure 1) is given in Figure 7. The results of these tests showed that the lowest temperature achieved was 35°K; at 77°K, the machine produced 0.35 watts of refrigeration for an input power of 34.5 watts. The cool-down time to 80°K was four minutes.

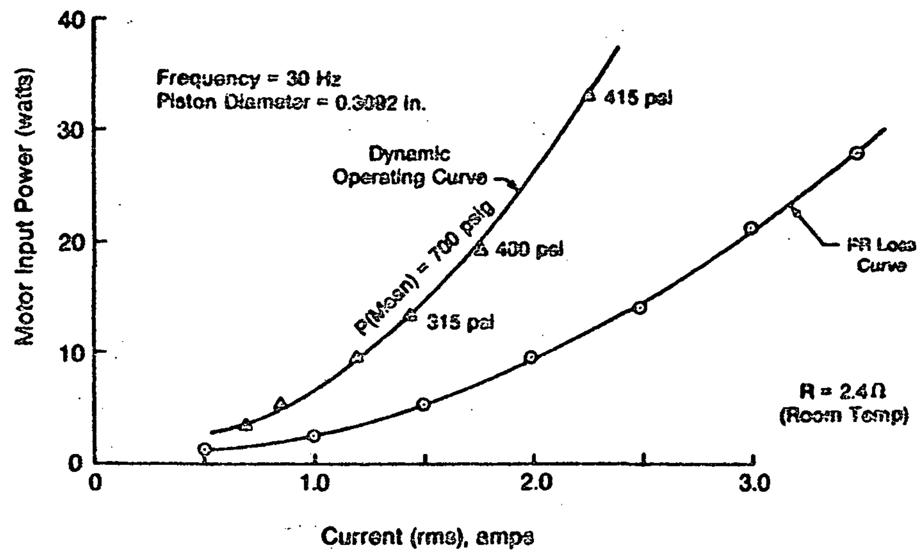

Figure 6 Locked Plunger Power Curve and Dynamic Operating Curve for the Compressor

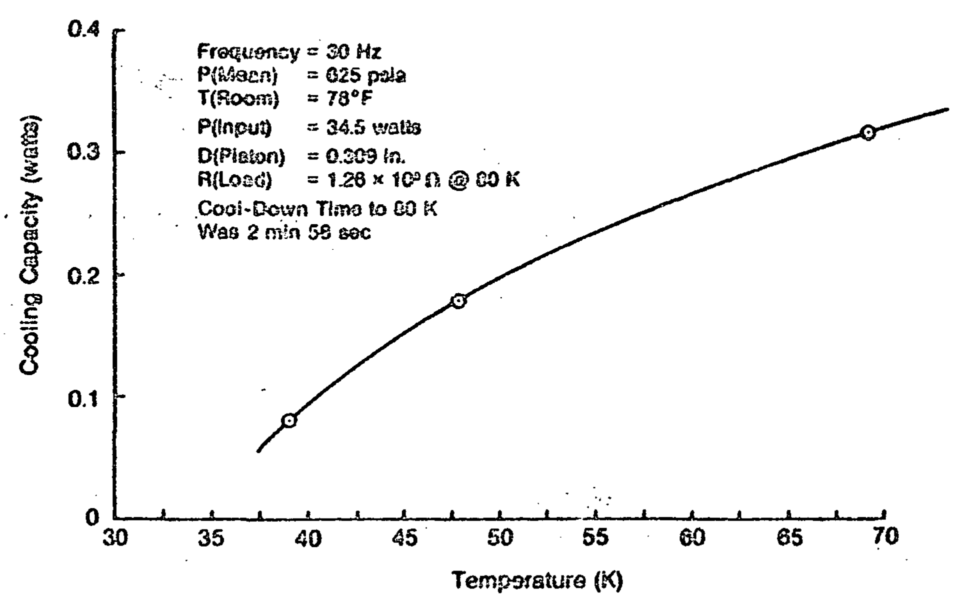

Figure 7 1/4-Watt Cryocooler Cooling Capacity

Another interesting test result was that the number of electrical coils in the stator could be reduced without affecting compressor performance. In the fabrication of this machine, the five coils were wound individually, and separate power leads were brought out of the machine, enabling the coils to be disconnected and the effect studied. The result (shown in Figure 8) was that the two outer coils could be eliminated without a reduction in the pressure differential (the right-hand ordinate), or an increase in input power (the left-hand ordinate). The implication of this is that the motor length can be reduced by two coil segments (1.44 inches), and the overall length of the compressor can be reduced from 7.00 inches (Figure 1) to less than 5.5 inches.

3. CONCLUSION

The concept of using a linear drive for the military 1/4-watt split Stirling cooler compressor is practical, as demonstrated by the program; in particular, the program showed the following:

1. It is difficult to design for a low frequency of 22 Hz and a large pressure difference because of the large spring force produced by the compression space and the large moving weight required to balance the spring component. Increasing the frequency will eliminate this problem.
2. A linear compressor can be developed for the 1/4-watt split Stirling cooler that will meet all of the present cooler specifications.
3. Effective vibration isolation can be incorporated and still meet the design specified envelope.

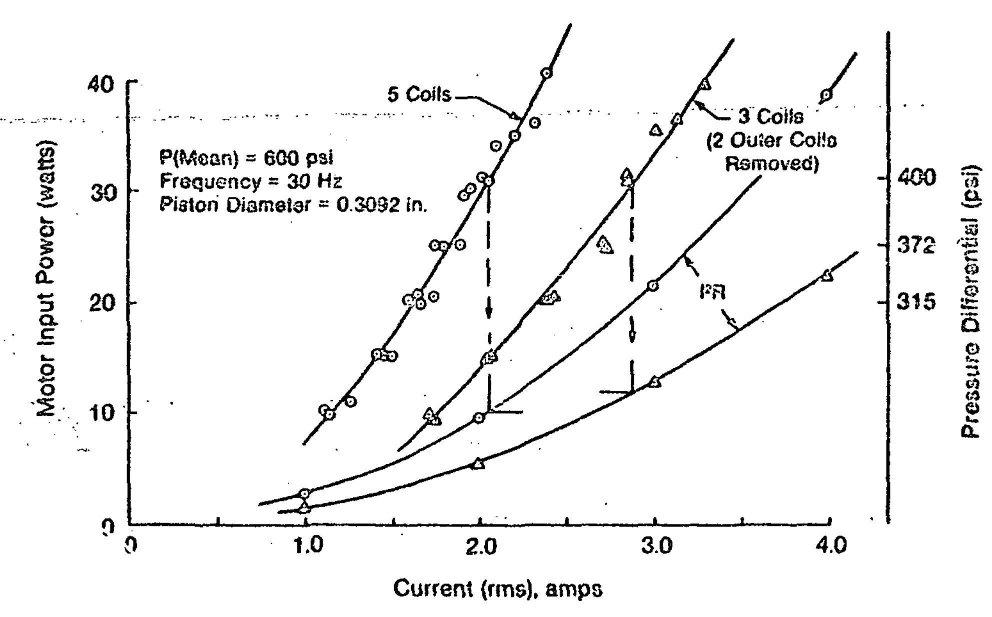

Figure 8 Effect of Eliminating Coils