A 10°K TRIPLE-EXPANSION STIRLING-CYCLE CRYOCOOLER

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ABSTRACT

This paper describes the design of a triple-expansion closed cycle Stirling cryocooler optimized for a cooling load of 50 mW at 10°K. The cooler has been designed with the objectives of low power, low weight, compactness, low mechanical motion, low electromagnetic noise, and low output temperature fluctuations. The design employs a direct drive linear motion piston motor and a triple-expansion free displacer. Piston motion is controlled by feedback from an optical position transducer. Mechanical vibrations are attenuated with a passive resonant counterbalance. Electromagnetic noise is attenuated with layered high permeability magnetic shielding. The regenerators move with the displacer within a thin titanium cold finger. The piston and displacer oscillate at 8.33 Hz on bearings and seals of reinforced Teflon™. The cooler is designed to provide the desired 50 mW of cooling at 10°K with a power input of less than 100 W. The piston can be driven at a greater stroke to produce up to 200 mW of cooling with an input power of 250 W. A lead and copper cold tip heat exchanger will limit temperature fluctuations to within 0.01°K. The cooler weight estimate is 35.9 kg for the thermodynamic components, 17 kg for the vibration absorber, and 8.6 kg for the electromagnetic interference (EMI) shielding. The cooler length without Dewar and vibration absorber is 710 mm.

INTRODUCTION

Development of small, efficient cryogenic refrigerators has been fostered by the requirements of infra-red detection systems. Infra-red system cryocooler requirements typically include high efficiency, compactness, low weight, low vibrations and long, maintainance-free operation in the 20 to 80°K temperature range. Stirling cycle coolers have proven to be well suited to these requirements. More recently, stricter requirements have arisen from increasing interest in superconducting quantum interference devices (SQUID) and magnetic gradiometers. For these devices, it is desirable to have coolers which operate at temperatures below 10°K. In addition to the efficiency, compactness, weight, vibration and reliability considerations, SQUID devices require that the associated coolers produce extremely low magnetic noise and low temperature fluctuations.

This paper describes the design of a Stirling cryocooler which was funded by the Office of Naval Research with the objective of bringing cryocooler technology closer to meeting SQUID requirements in a practical unit. The goals for this design were to produce 50 mW of cooling at 10°K with less than 250 W power consumption in a reliable unit with minimum weight, size, vibrations, temperature variations, and electromagnetic noise. The design presented herein has not yet been fabricated. However, the performance is projected with confidence, since it is based on an earlier low temperature machine, analysis by the Philips Stirling cycle optimization program, and techniques proven in existing (albeit higher temperature) Philips coolers. The cold finger design of the unit described below is similar to an experimental machine which attained a minimum no-load temperature of 7.8°K [refs. 1,2]. In addition, bearing, seal and driver techniques were adapted from various Philips designs, including the MC-80 [ref. 3], the Johns-Hopkins University of Applied Physics Laboratory cooler [refs. 4,5], and the NASA-Goddard unit [ref. 6].

DESIGN APPROACH

THERMODYNAMIC CONSIDERATIONS

The present design employs three stages of expansion and three metal matrix regenerators which oscillate with the displacer within a thin walled titanium cold finger. The choice of metal regenerators was an early pivotal decision since metal in motion near a sensitive magnetometer might produce unacceptable interference. An alternative approach is to use extended gap regeneration and a non-metallic displacer. The extended gap approach typically suffers the disadvantages of an unusally long displacer, low cooling capacity, potential electromagnetic interference (EMI) from static electric charge accumulation on the moving displacer, and short duration of continuous operation due to possible cryopumping of organic vapors toward the cold tip.

Thermodynamically, the metal regenerators were found to be adequate down to the 10°K cooling range, both by experiment and by analysis. The magnetic interference produced by eddy currents induced in the moving metal displacer by the earth's magnetic field were estimated to be extremely low. This is due to the relatively high electrical resistivity and thin construction of the titanium displacer shell, as well as the low displacer amplitude and frequency. Interference originating from the displacer will be dominated by noise generated by even the best, shielded, compressor motors. Further, displacer originating interference has the compensating characteristic, like the motor interference, of being periodic. Thus, it was decided to proceed with metal regenerators.

After choosing the regenerator material, the cooler parameters were optimized. Optimization is performed by constraining the magnitude and temperature of cooling at the cold tip, then searching for the combination of variables which result in minimum power input. The optimal search is normally halted at the judgement of the designer in order to include considerations of total size and weight. Thus, the total cold finger length and piston displacer volume were chosen as a compromise between total size and efficiency.

DYNAMIC CONSIDERATIONS

After optimizing the thermodynamics, the mechanical dynamics were analyzed. The thermodynamic optimization resulted in specifications on the displacer amplitude, speed and phase shift with respect to the piston. This motion can be produced by a mechanical linkage connected to a crank drive, or by a separate displacer linear motor, or by resonance (alias "free displacer"). Since the mechanical linkage requires extra bearings and seals, and a separate driver requires more space and power, the resonant approach was chosen. When the piston mass and diameter are selected to achieve resonance with the compression pressure wave, the motor load is optimized. When the piston is not in resonance, the motor must also supply substantial reactive power resulting in lower efficiency. The equations for interfacing piston and displacer dynamics with the Stirling cycle thermodynamics for free-piston free-displacer machines have been presented by A. K. de Jonge in reference 3.

The resonance condition for the piston left a single design degree of freedom along a constrained line of piston mass versus amplitude at constant compressor displaced volume. Various linear motor designs were then analyzed to find the combination of piston mass and stroke which permitted the minimum weight motor to supply the necessary compressor power with 250 W electrical input power.

Since attaining the desired cooling at 10°K was essential, weight and size minimization were relaxed somewhat to arrive at a success oriented design. Thus, the cooler was optimized for 200 mW of cooling at 10°K with an input power of 250 W rather than the specified goal of 50 mW at 10°K with the same input power. The resulting design is flexible; power consumpion can be reduced for a reduced cooling load by running at a decreased piston stroke while keeping the piston in resonance. Predicted power consumption for a 50 mW load at 10°K is less than 100 W, including electronic driver losses. Further power reduction is possible for lower cooling loads, and excess cooling capacity can be used to accelerate cooldown time. Most importantly, the design has a high probabiliy of meeting the specified power versus cooling requirement. Temperature fluctuations of the cold tip are compensated by the cold tip heat exchanger, which has a high heat capacity. Mechanical vibrations are reduced with the use of an external passive vibration absorber. EMI attenuation is accomplished by enclosing the motor, the dominant source of low frequency noise, with layered, high-permeability, magnetic shielding.

A layout drawing of the resulting design is shown in figure 1. Table I summarizes the design objectives and table II lists the thermodynamic design values. The basic subsystems of the cooler are described in the following sections, viz., piston/motor, displacer, cold finger, vibration absorber, EMI shielding, and additional peripheral components.

SUBSYSTEM DESIGN

PISTON/MOTOR

The piston assembly includes a cylindrical piston and a concentric, linear-motor armature. The piston is driven directly by a linear motor. The motor and its housing dominate the weight of the thermodynamic components, and thus merit particular attention in their design. These components are shown in figure 2.

The thermodynamic optimization requires that the piston sweep a volume of 153 cc, with a resultant pressure wave of 0.34 to 0.69 MPa. For optimal system efficiency with a linear drive motor, the diameter of the piston and mass of the piston assembly should be selected so that the assembly resonates with the spring force of the compression space, at the cooler speed.

The geometry and dimensions of the motor were optimized with finite-element mappings of the stator and gap flux densities. The finite-element, magnetic-field program permitted the inclusion of analytically intractable effects such as nonlinear hard and soft magnetic materials, saturation, and leakage. Eddy current losses are reduced by laminating the stator with radial cuts. The motor design values are given in table III.

To minimize the side loads on the piston seal and bearings and to meet the mass requirement for the piston assembly, a moving-coil rather than moving-magnet motor design was chosen. Power transfer to the moving coil throughout its relatively large excursion is accomplished with rolling leads consisting of strips of beryllium copper. During operation, the circular deformation of the strips results in an acceptable peak oscillating stress of 275 MPa (40,000 psi), nonreversing.

The long piston excursion makes a mechanical spring impractical as a position centering device. Therefore, the piston motion is controlled with position feedback to the motor drive amplifier. An optical transducer was selected as the position sensor.

The piston seal and bearings consist of thin layers of Rulon™ (a reinforced Teflon™) bonded to the contacting surfaces of the piston assembly. The Rulon™ surface, sliding along the cylindrical wall of the compression space, acts as a bearing as well as a seal. A second bearing at the opposite end of the piston assembly also consists of a Rulon™ surface sliding on a guide which prevent rotation of the piston assembly. To facilitate maintenance the bearings are easily removable and replaceable.

The piston assembly is made of titanium to minimize armature weight and to match the thermal expansion of the Permendure motor stator. Shock pads located at both extremes of the stroke prevent damage during transit or in the event of overstroke during operation. The remainder of the motor housing is made of aluminum for low weight, low cost, and ready availability of material in the required dimensions. All motor housing joints use reusable metal C-rings for low leakage and for ease in disassembly.

DISPLACER/REGENERATOR

The displacer and regenerator are the most critical components of a Stirling cycle cooler. The displacer in this cooler has three stages of regeneration with three corresponding expansion spaces. The regenerators oscillate as an integral part of the displacer. The displacer is not independently driven, but oscillates with a mechanical spring in response to the compressor pressure wave. The regenerator must be designed for efficient transient gas-solid heat transfer, yet have low conduction heat leak and low flow losses. The transient heat transfer and thermal energy storage must be compatible with the charge pressure and speed of the cooler. The above requirements dictate an optimal displacer amplitude and phase shift relative to the piston. Furthermore, practical requirements, such as reasonable length, weight, and reliability, must be met. The resulting design values are listed in table IV.

Rulon™ bearing/seal surfaces similar to those of the piston are used at the two link locations and at the warm end of the first (warmest) regenerator. Though not be as easily replaceable as the piston seals. such replacement will be infrequent because the displacer seals will have a lower wear pressure and a much lower wear velocity. Furthermore, some leakage past partially worn regenerator seals will not have a serious effect, since the pressure drop across the seals is low, and the leakage flow will experience some gap regeneration in the displacer clearance space.

Lead was chosen as the third-stage regenerator material because of its ability to maintain its heat capacity at low temperatures.

Annulus-type heat exchangers are used to remove thermodynamic waste heat and to transfer heat from the device to be cooled to the working fluid. A linear variable differential transformer (LVDT) monitors the displacer position. The signal from this transducer can be used to measure the displacer amplitude and phase shift in order to optimize the mass and spring stiffness which are adjustable in the cooler design. The LVDT is packaged concentric with the mechanical spring and thus adds no additional length to the machine.

COLD FINGER

The cold finger and Dewar are shown in figure 3. The cold finger is the thin pressure vessel which contains the displacer. It must be thin to minimize the heat leak from the warm end of the cold tip, yet have the structural integrity to support the mass of the displacer, the cold-tip heat exchanger, and the device to be cooled. Also, it must contain a varying helium pressure without fatigue and not be excited in any vibration modes due to the displacer dynamics. For low heat leak and high mechanical strength, the cold finger is made of titanium. For the design dimensions and materials, a conservative stress analysis predicts:

Lowest mode frequency:	> 100 Hz
Worst case deflection due to all masses:	< 0.01 mm
Hoop stress from peak pressure:	< 27 MPa (4,000 psi)

At the cold tip there is a lead cap for damping temperature variations. The lead provides a large heat capacity for thermal damping. The mass of the lead is 500 g. Its thermal mass will attenuate the temperature variations to less than $\pm$ 0.01°K.

Stainless-steel coaxial signal leads (Uniform Tubes UT-85SS) are provided for connection to the cryogenically cooled device which is attached to the end of the cold finger. The leads are cooled continuously along their length by coiling them about the cold finger in contact with the titanium wall. Further, several turns of the leads are in thermal contact with two titanium disks extending radially from the first and second expansion spaces. These two disks act as heat exchangers for the leads as well as axial thermal radiation shields.

The cold finger is enclosed by an evacuated Dewar. Its interior surface is covered with layered superinsulation which provides radial radiation shielding with minimal conduction losses.

VIBRATION ABSORBER

The linearly reciprocating motions of the piston and displacer give rise to an axial momentum imbalance. This imbalance can be compensated with a third mass oscillating in opposition. Such a mass in resonance with a linear mechanical spring will be excited to absorb vibrations at its natural frequency.

Because the piston oscillates against the nonlinear gas spring of the compression space, the resulting piston motion is nonsinusoidal. The imbalance force is, therefore, composed of a large force at a fixed frequency plus superimposed higher harmonic imbalances. A passive absorber will attenuate only the imbalance at the fundamental frequency. For additional compensation, thecooler's position sensors can be used. The higher harmonics of the piston can be attenuated by using the optical sensor in position feedback to the piston motor to enforce pure sinusoidal motion. If further vibration attenuation is required, an active vibration compensator could be built to replace the passive design.

A passive vibration absorber (see fig. 4) was designed for this cooler. It is an external unit which bolts to the back plate of the motor housing, coaxial with the piston and displacer. Resonance tuning is accomplished with a removable mass tuning ring on the moving countermass. Spring linearity is maintained with relieved helical spring mounts. To insure low damping, the countermass is supported only by its springs. Since the countermass is not guided in linear bearings, it must be designed so that its various degrees of freedom do not have resonant frequencies close to the speed of the cooler. Optimization criteria of the countermass included low spring stress, low weight, and separation of modal frequencies. The resulting design values are given in table V.

EMI SHIELDING

The dominant source of electromagnetic interference from this cooler is the piston motor. The motor coil current, which oscillates at 8.3 Hz, induces a large, time-varying, magnetic field. This interference is attenuated by the surrounding motor stator, and further attenuated by shielding that encloses the piston motor. For a conservative calculation of shielding effectiveness, the stator attenuation was ignored. A two-dimensional, finite-element, magnetic field program was used to analyze the static field due to the peak current in the coil within the symmetric enclosure. Since the coil excitation is low frequency, shielding effectiveness is dominated by reluctance considerations rather than by eddy-currents; thus, a quasi-static analysis is valid.

Figure 5 shows a magnetic flux plot for one quadrant of the cross section of the motor coil in free space. There is rotational symmetry, as well as reflected symmetry, in the negative axial direction. The magnetic material chosen, Hipernom (Carpenter Technology Corp.) alloy, has a minimum permeability of 100,000 and a saturation flux density of 0.4 Tesla. Figure 6 shows the effect of enclosing the coil with a Hipernom cylindrical can connecting the two end plates. The peak field within the enclosure is 0.02 Tesla. The peak field within the 0.76 mm (0.030") thick shield is 0.24 Tesla. The peak field at the cold tip location is less than 0.1 micro Tesla. Additional layers of shielding material at the end plates produce further attenuation, but their influence is beyond the numerical accuracy of the field program. Multiple radial shielding layers for the cylindrical can will provide no additional attenuation of EMI at the cold tip.

The shielding design consists of a single layer of 0.76 mm (0.030") Hipernom supported by a shell of aluminum for the radial shielding, and multiple layers of Hipernom at the end plates for axial shielding. The end plates consist of three disk layers of 0.76 mm (0.030") Hipernom separated by two layers of 5 mm (0.200") copper. The separating copper disks provide an optimal reluctance gap as well as high-frequency eddy current shielding. Both end plates experience some leakage through bolt holes. The hole locations are placed off axis, as far as possible, to reduce the contribution of this leakage to the magnetic noise at the cold tip. Also, gas passages for helium flow between the compressor and displacer have been designed to minimize the effects of magnetic leakage.

The rear end plate and cylindrical can are external, and thus easily removable from the cooler. The end plate which forms the manifold between the displacer and piston can be removed if it is replaced with a similar plate to form the gas manifold and seat the C-seals. Thus, the shielding can be increased, if necessary, or removed, if not needed.

PERIPHERAL EQUIPMENT

Peripheral support equipment for the cooler consists of motor driver electronics and an ambient, forced-convection cooling system. The piston motor requires a dc power supply of at least 200 V, 250 W. The motor driver is a pulse-width-modulated amplifier which can achieve a power transfer efficiency of 85%.

The cooling system is required to remove the heat of compression and the motor losses from the ambient heat exchanger. This can be accomplished with a closed-cycle recirculating bath, or with an open-cycle flow of any available liquid at ambient temperature. The flow requirements of 3 liters per minute through a pressure drop of 0.2 MPa (3 psi) can be satisified with a small water pump consuming less than 5 W.

The peripheral equipment is commercially available, although custom design would reduce power, weight, and size. This support equipment can be easily detached and transported separately.

DISCUSSION OF RESULTS

The given design meets or betters the goals for cooling and power consumption. Extensive effort was expended to minimize vibrations and EMI; the resulting noise, however, may not meet the demanding design goals. The total system weight is substantially higher than that of the goal; a weight breakdown is given in table VI. The limiting factors and trade-offs are discussed below.

The thermodynamic efficiency of the cooler is primarily limited by the regenerator efficiency. The responsible effect is the loss of specific heat of all materials at low temperature - a problem that has long been recognized. An improvement in regenerator efficiency would require an impractical geometry for an operational cooler (excessively long displacer) or a breakthrough in materials development. Nonetheless, the current displacer design meets the specified thermodynamic objectives.

The cooler will provide cold in excess of the prescribed 50 mW; specifically, 200 mW of cooling at 10°K at the maximum allowed input power of 250 W. By reducing the piston amplitude, the machine would provide the 50 mW of cooling with less than 100 W input power. For lower, steady-state cooling loads, the piston amplitude and corresponding input power requirements can be decreased even further. Conversely, at full power, the reserve cooling capacity can be used to accelerate the cooldown time. At 50 mW net cooling, the cooldown time is expected to be about six (6) hours.

The vibrations and EMI of the cooler were attenuated as far as practical with reasonable weight, size, power, and complexity. Additional attenuation could be achieved with added mass, active vibration compensation, more EMI shielding, active field compensation, or with a motor design stressing noise reduction rather than weight and efficiency. Each of these approaches involves penalties in system weight or power, for only a marginal reduction of noise.

Further noise attenuation could be accomplished through a major design revision. The cooler could be redesigned as a split system, allowing the compressor, the primary offending component, to be separated from the displacer. This approach also would involve some penalties in efficiency and weight.

An option, if permissible to the operational constraints of the superconducting device, would be to run the cooler intermittently. Assuming 50 mW of excess cooling capability at 10°K decreasing to 0 W cooling at 8.5°K, the cooler could lower the temperature of the cold-tip thermal mass from 10°K to 9°K in roughly 60 seconds. Upon shutting off the cooler power, the vibrations and EMI noise generated by the cooler would cease. The heat leak through the regenerator matrix and stagnant helium, coaxial leads, cold finger wall, plus the Dewar losses, would cause the cold tip temperature to rise at a rate of roughly 2°K/minute. This would clearly not meet the desired temperature regulation of $\pm$ 0.01°K. However, a known, repeatable, smooth monotonic temperature ramp may be an acceptable disturbance subject to electronic compensation at the device output. This approach would satisfy the desired noise requirement and result in considerable weight savings, since the vibration absorber and EMI shielding could be eliminated.

DISCUSSION

Question by S. Shtrikman, Weizmann Institute of Science, Rehovoth, Israel: What is the phase angle between the motion of the displacer and that of the pressure pulse? To what is this angle optimized?

Answer by author: The phase angle is estimated to be 30°. For designs in which the displacer is driven by the pressure wave created by the piston motion. The optimized phase angle is the one that maximizes cold production for a given piston stroke.

Question by D.E. Daney, National Bureau of Standards, Boulder, Colorado: In computing the refrigeration capacity, do you see $\oint Vdp$ or $\oint \alpha Vdp$ where α is the dimensionless thermal expansivity?

Answer by author: In calculating the gross cold production, the expression $\oint pdV$, which is based on the isothermal model, is used. Non-isothermal effects are included in the calculation of net cold production.

TABLE I. DESIGN OBJECTIVES.

Cold production:	50 mW @ 10°K
Input power:	< 250 W
Weight:	minimize (25 kg goal)
Magnetic noise:	minimize (1.0 micro Gauss goal)
Vibrations:	minimize (goal of 0.1 micro-radian rotation of cold tip)
Cold tip temperature regulation:	fluctuations < 0.01°K
Cooldown time:	< 24 hours

TABLE II. THERMODYNAMIC DESIGN VALUES.

Mean pressure:	0.49 MPa (4.84 atm)
Speed:	8.33 Hz (500 cpm)
Piston amplitude:	45 mm (1.77")
Piston diameter:	46.5 mm (1.83")
Piston mass:	2.6 kg
Displacer amplitude:	3.73 mm (0.147")
Displacer diameters:	18, 25, 40 mm
Displacer phase shift:	52°
Displacer mass:	0.55 kg
Displacer spring stiffness:	14.5 kN/m (82.8 lbf/in.)
Expansion space temperatures:	10, 53, 105°K
Maximum pressure:	0.69 MPa (6.83 atm)
Minimum pressure:	0.34 MPa (3.37 atm)
Cold Production:	200 mW @ 10°K
Mechanical input power:	180 W

TABLE III. MOTOR DESIGN VALUES.

Stator

Magnets:	Material		samarium cobalt
	Energy product		160 kJ/m^3 (20 MG-Oe)
	Magnetic ring		
	Dimensions:	Axial length	72.5 mm
		Inside diameter	105.3 mm
		Outside diameter	127.3 mm
Yoke:	Material		vanadium Permendure
	Stator flux density		2.1 Tesla
	Gap flux density		0.45 Tesla

Armature

Coil:	No. of turns		336
	Resistance		1.54 ohms
	Inductance		71.0 mH
	Peak current		4.4 A
	Peak voltage		87 V
	Electrical input power		192 W
	Mechanical output power		175 W
	Efficiency		91%
	Dimensions:	Axial length	72.5 mm
		Inside diameter	98.2 mm
		Outside diameter	105.0 mm

TABLE IV. REGENERATOR DESIGN VALUES.

Regenerators

Temperature (°K)	10	55	105
Type	lead spheres	phosphor bronze mesh	phosphor bronze mesh
Length (mm)	50.0	27.2	50.0
Area (sq. mm)	227.0	452.0	1163.0
Fill factor	0.63	0.42	0.34
Sphere or wire dia. (mm)	0.135	0.041	0.041

TABLE V. VIBRATION ABSORBER DESIGN VALUES.

Springs: Combined pair	
Combined axial stiffness	32.6 kN/m
Combined lateral stiffness	52.2 kN/m
Moving Mass:	
Mass	11.9 kg
Polar inertia	0.029 kg-m^2
Rocking mode moment of inertia	0.050 kg-m^2
Modal Frequencies:	
Axial translation	52.3 rad/sec
Lateral translation	66.2
Rotation about axis	22.5
Rocking mode	80.2
Operating Conditions:	
Peak force	350 N
Peak amplitude	11.0 mm
Peak spring stress	122.0 MPa (18,000 psi)

TABLE VI. WEIGHT SUMMARY.

Compressor	27.2 kg
Expander	8.7 kg
EMI Shielding	8.6 kg
Countermass	17.0 kg
Total	61.50 kg

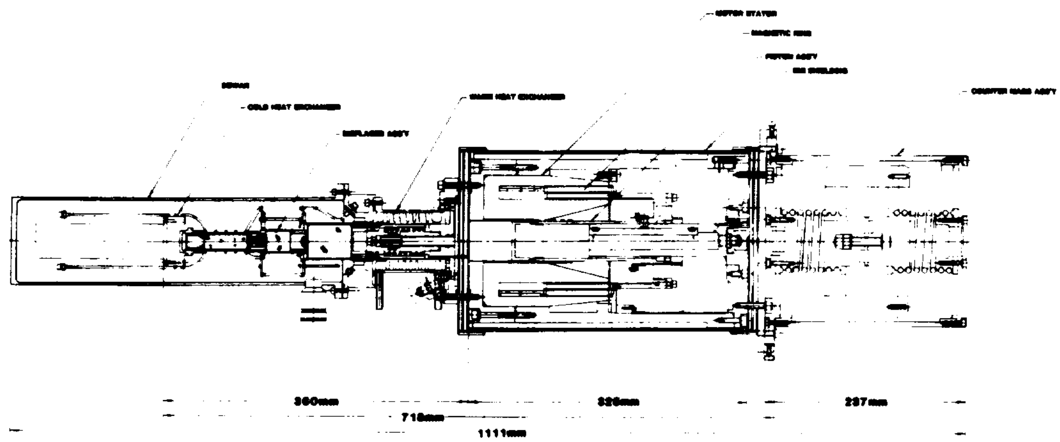

Fig. 1. Layout drawing of 10°K Stirling cycle cooler.

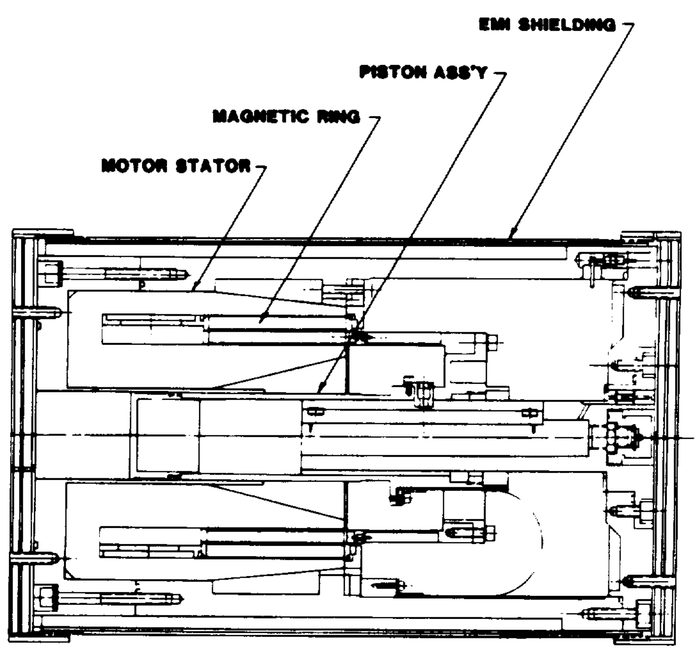

Fig. 2. Layout drawing of piston/motor section with piston at mid-stroke.

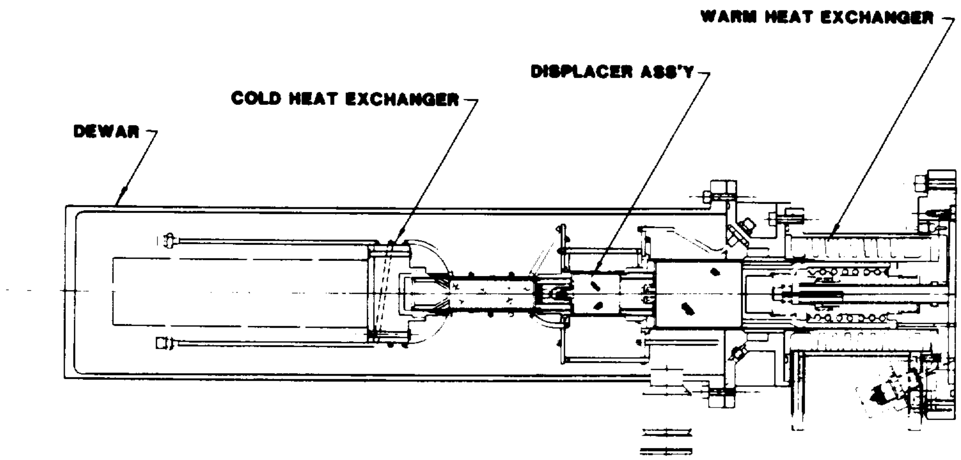

Fig. 3. Layout drawing of displacer section.

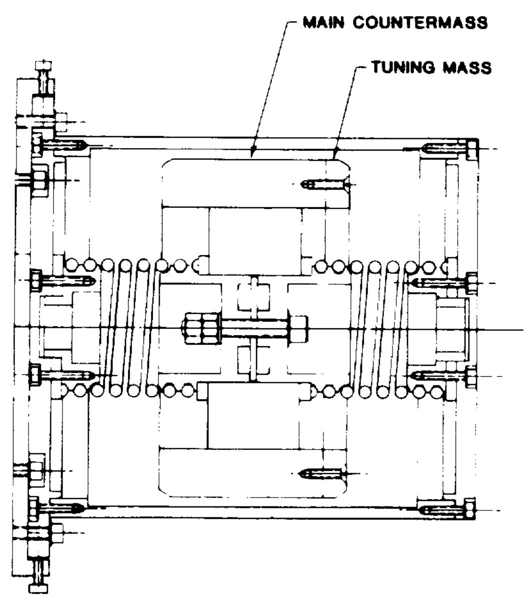

Fig. 4. Layout drawing of vibration absorber.

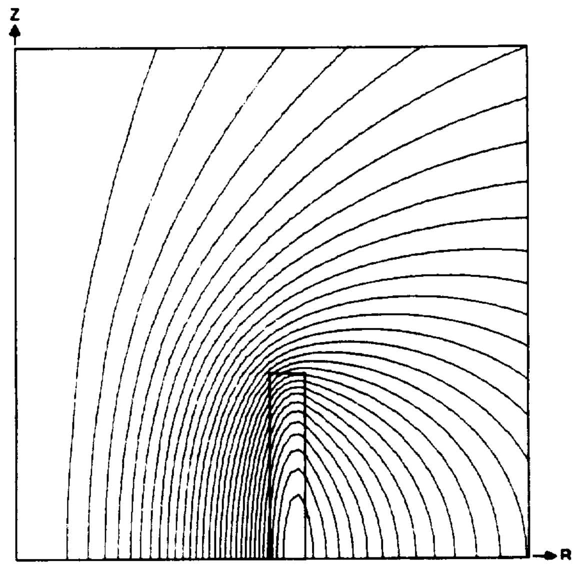

Fig. 5. Flux plot of cylindrical coil in free space.

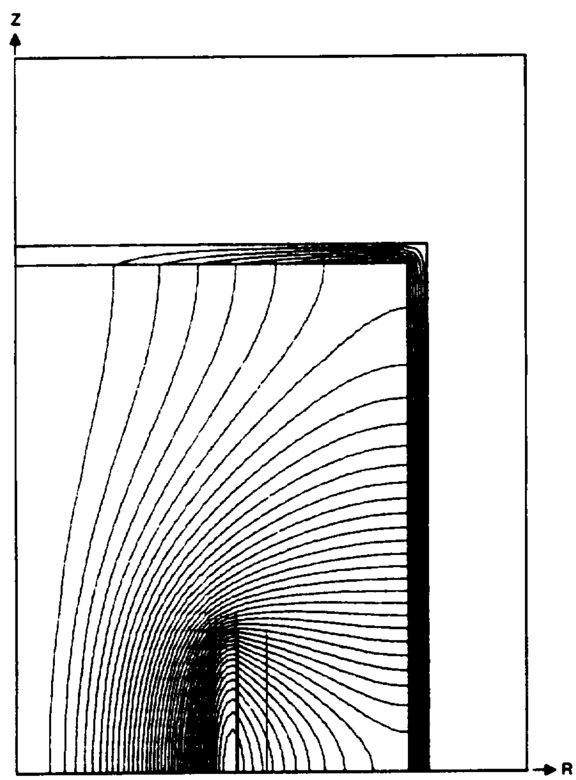

Fig. 6. Flux plot of coil enclosed by magnetic shielding.

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