# Design and Verification of Stirling Cooler Interfaces Suitable for Long-Lifetime, Spaceborne Sensor Systems

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## ABSTRACT

Interfacing a linear Stirling cooler to a spaceborne sensor system poses some unique challenges with regard to achieving high thermal efficiency within the mechanical and integration constraints of the cooler and sensor system. Interface component development and verification activities performed during the X-ray Spectrometer Cryogenic Subsystem (XRS/CSS) Program established a reliable and high-thermal efficiency interface design. In any large, spaceborne sensor system cooled by a cryocooler that provides a point cooling source, the factors limiting achievable interface efficiency stem from the need for mechanical flexibility and the inherent spreading resistances in the interface. Therefore, the overall interface design approach and components developed for the XRS/CSS are applicable to many other sensor cooling applications. The XRS/CSS design uses split, linear, Stirling-cycle coolers (primary and back-up) to remove heat from the outer vapor-cooled shield of the superfluid helium dewar and greatly reduce the helium loss rate. To maximize the benefit of the cooler, the interface between the coldtip and dewar is designed to provide high thermal conductance with minimal parasitic heating. A key element of the interface is a flexible/thermal link having extremely high efficiency. The thermal impedance of this link is small compared to that inherent in the overall interface, and its use can thus provide nearly the same overall interface efficiency regardless of the type of cooler used. To meet requirements for control of vibration to the sensor, the compressor is mounted with mechanical isolation. Prototype verification was performed to prove the designs of the coldtip interface and compressor mount. In this paper we discuss the thermal/mechanical interface requirements, performance drivers, designs and prototype verification activities.

## INTRODUCTION

Mechanical coolers can be used as the sole source of cooling in a sensor system, or they can be used in a hybrid mechanical/stored cryogen cooling approach to either extend the cryogen lifetime or provide a lower sensor temperature. The XRS/CSS (**Fig. 1**) is a system that uses coolers to extend the lifetime of a superfluid helium dewar about a factor of 3, thereby achieving

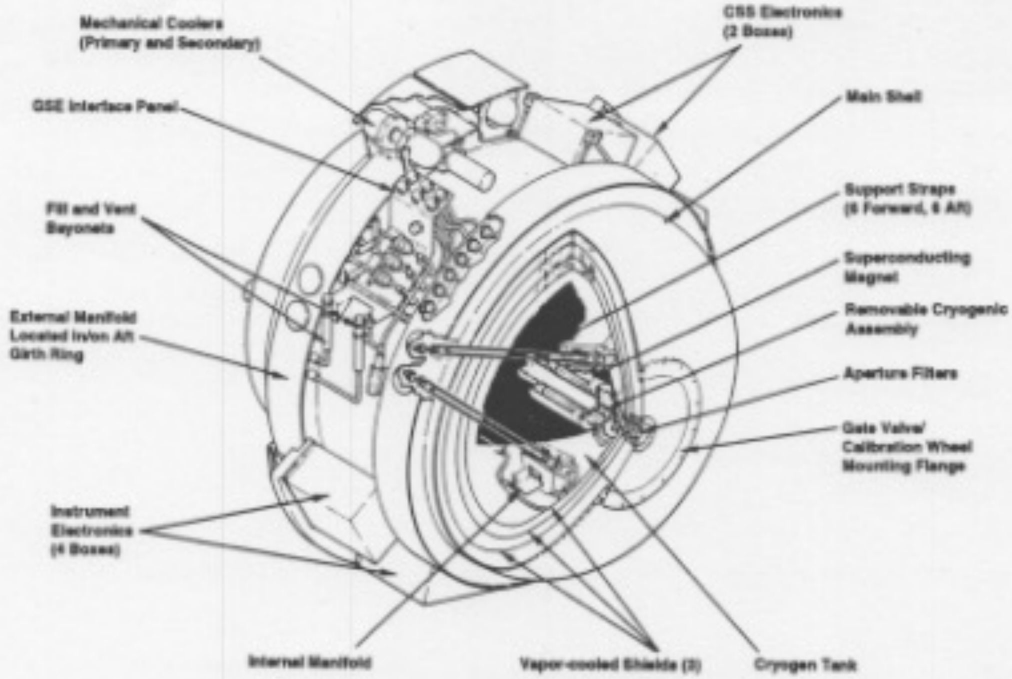


**Figure 1.** XRS Cryogenic Subsystem, showing two Stirling coolers mounted on aft girth ring.

the required 5 year lifetime with minimal overall system mass and envelope[1]. We chose for this application a split, linear, Stirling-cycle cooler built by Ball Aerospace and based on the clearance seal, spiral spring bearing technology developed by the Oxford University and Rutherford Appleton Laboratory. This machine consists of twin compressors and a single, momentum-compensated displacer[2].

The helium tank is supported from fiberglass/epoxy tension straps and surrounded by an insulation system including three vapor-cooled shields (OVCS). The thermally efficient support system has minimal stiffness, and the OVCS moves substantially relative to the vacuum shell during launch. The coolers, primary and back-up, are mounted to the aft girth ring. The two cooler coldtips are connected via a heat-shrink thermal switch to a single cold finger that penetrates the dewar and is attached to the OVCS. Heat removal from the OVCS lowers its temperature to ~90 K and reduces the parasitic heat load to the helium tank. The cooling interface is thermally optimized with regard to the combined degradation of thermal impedance and parasitic heating, while meeting the mechanical and integration constraints inherent to the cooler and dewar[3]. The efficient, reliable thermal switch reduces the heat load from the back-up cooler to nearly zero and has essentially the same thermal impedance and parasitic heat load that would exist in a simple thermal bus connecting the two coldtips to the cold finger. The achievable thermal conductance of the interface is limited by (1) the need for flexibility to accommodate assembly build-up tolerances and relative motions between the OVCS and vacuum shell during launch and (2) the spreading resistances inherent in cooling a large object with a point cooling source. The need for flexibility and the control of spreading resistances are important issues for nearly all applications of large sensor cooling by mechanical coolers. An exception to this would be an approach using a Joule-Thomson cooler[4], where distributed cooling of the cold mass is provided via flexible capillary tubes penetrating the sensor system. The overall interface design and components developed for the XRS/CSS are applicable to any sensor system using mechanical coolers that provide point source cooling.

The cooler compressors are the primary source of microphonic noise in the sensor readout. To achieve the allowable vibration levels at the detector array inside the dewar, disturbances from the compressor are attenuated by a low-frequency spring mount system. Vibration coming from the displacers is much lower, and they are hard mounted to the girth ring.

During the XRS/CSS Program, prototype verification of all components of the cooling interface and the compressor mount designs were performed. The Program was terminated due to budgetary cutbacks in February 1994 just as the detailed system design phase was starting. At that time, all the prototype interface components had been fabricated, and test activities were completed except for final characterization of the thermal switch activation process.

## COLDTIP-TO-DEWAR INTERFACE REQUIREMENTS AND DESIGN

The coldtip interface detailed design and component requirements were derived from extensive thermal and mechanical analyses to provide maximum thermal efficiency while being compatible with a Delta II rocket launch and the other mechanical and integration constraints of the cooler and dewar. The helium lifetime depends primarily on the efficiency and size of the dewar, the efficiency and capacity of the cooler, the power available to the cooler, and the efficiency of the coldtip interface. A dewar lifetime model, which accounts for all these factors, was constructed. Based on this model, the lifetime sensitivities were determined to be 2 percent per degree K of temperature drop from the bulk of the OVCS to the coldtip and 2 percent per 100 mW of parasitic heating to the interface. Thermal impedance and parasitic heating can be traded against each other; i.e., either can be improved at the expense of the other. As the design proceeded, we determined that 10 K and 200 mW were achievable within the practical size and mechanical constraints and represented a thermally optimized balance. Thus, the best possible design results in about 14 percent less lifetime than a theoretical, perfect interface.

Coldtip deflection greater than ~0.0005 in. causes undesirable rubbing of the regenerator that could degrade cooler operating lifetime. A lateral force greater than ~3 oz to the unreinforced coldtip would cause unacceptable movement. This could be alleviated by coldtip reinforcement, but that would result in additional parasitic heating. The choice then is between the thermal resistance of a highly flexible attachment to the coldtip or the parasitic heat load of a reinforcing structure. We made a decision early in the program to develop a highly flexible, highly conductive link rather than impose a requirement for coldtip reinforcement upon the yet-to-be-selected cooler.

The coldtip interface design is shown in **Fig. 2**. The displacer bodies, not shown, are mounted and well aligned to the opposite faces of the girth ring. The coldtips are attached to the heat-shrink thermal switch by S-shaped flexible/thermal links that have nearly the same flexibility in all directions to accommodate the random buildup of assembly tolerances. The dual switch can connect or disconnect either or both of the coldtips. A sapphire cold finger is rigidly attached to the thermal switch and attached on the other end to the OVCS via a flexible/thermal link assembly (**Fig. 3**) that uses S-shaped links similar to those which attach to the coldtip. The thermal switch is rigidly attached and well aligned to the girth ring floor with excellent thermal isolation by a folded fiberglass/epoxy tube assembly. A low-emittance radiation enclosure, shown by the dotted line in the figure, surrounds the coldtip flexible links and thermal switch to help minimize parasitic radiation.

Notice that Fig. 2 is out of date in that it shows coldtips of opposing, momentum-compensating displacers, rather than the one coldtip of a single momentum-compensated displacer, which is the selected machine. In the figure the pair of coldtips is connected to the thermal switch ring by an integral dual link so that the gap between the links is minimized to control parasitic radiation. The coldtip of the twin-compressor, momentum-compensated displacer cooler is more resistant to side loading than the single compressor/single displacer cooler which was under consideration. Therefore, we planned to use this same dual link for attaching the single coldtip, thus maintaining the highest possible thermal conductance in the link. An extended coldtip interface block would be used to do this.

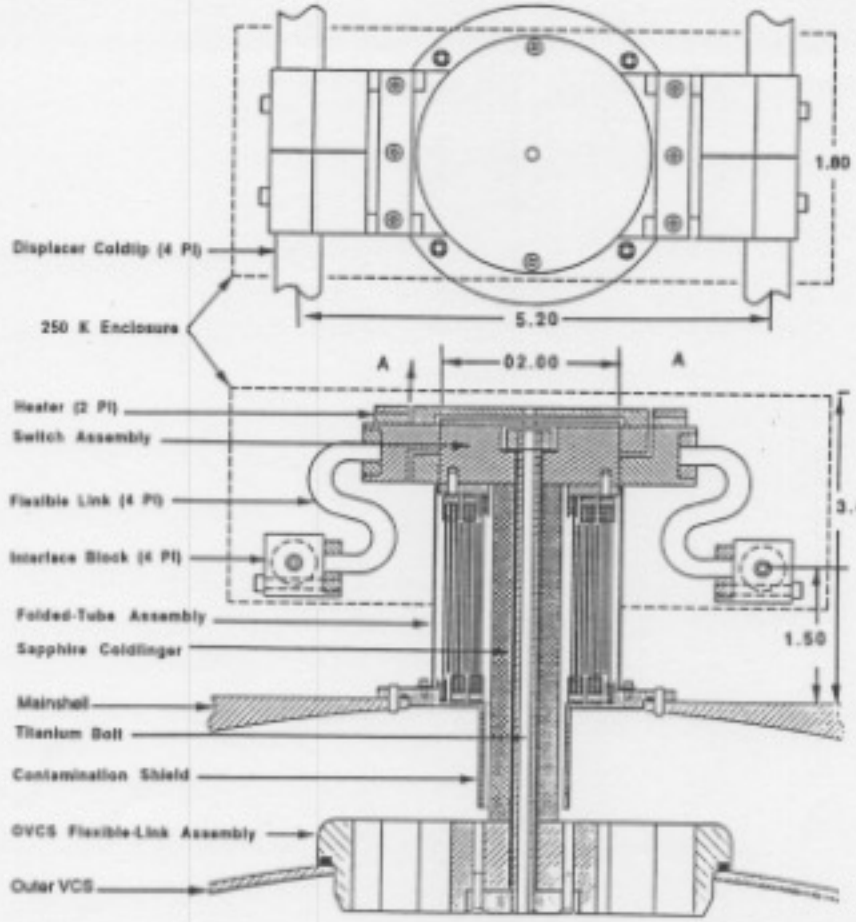


**Figure 2.** Interface between the cooler coldtips and dewar outer VCS; dimensions in inches.

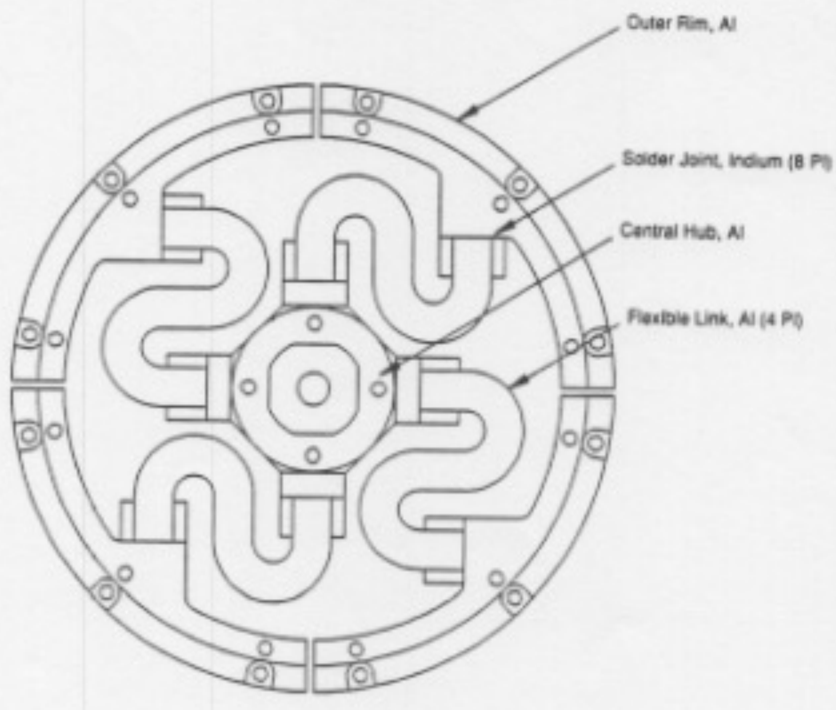


**Figure 3.** Flexible/thermal link assembly between cold finger and outer VCS.

A detailed breakdown of materials, masses, thermal impedances and parasitic heat loads for the elements comprising the interface is given in **Table 1**. The major impedances are in the flexible links needed to accommodate assembly tolerances and OVCS motions during launch and in the spreading resistances of the thermal switch and OVCS itself. Note that the impedance of the coldtip flexible/thermal link is small compared to the overall interface, and the use of a different type of cooler with a more robust coldtip than a Stirling cooler would not provide much improvement in the overall efficiency. Thus, an overall impedance of ~5 K/W can be considered typical of any cooled sensor system, even without a dewar, which uses a point cooling source and has a large cold mass supported with good thermal isolation (i.e., with low stiffness). The table also shows that with the highly efficient designs of the thermal switch support and heater leads, parasitic heating is dominated by radiation.

**Table 1.** Thermal/mechanical performance predictions for coldfinger interface

| Component | Materials | Mass (g) | Resistance (K/W) | Parasitic (mW) Cond. | Parasitic (mW) Rad. | Comments |
|---|---|---|---|---|---|---|
| Coldtip Interface Blocks | Al | 60 | 0.2 | n/a | 10 | Demateable 60 °C solder joint to Coldtip Flexible Link |
| Coldtip Flexible Links | Al | 120 | 1.1 | n/a | 55 | 20 mW rad. parasitic due to edge effects, indium solder to Switch |
| Switch Assembly (on/off) | Al, Mo, Vespel | 415 | 1.3/5000 | n/a | 45 | Zero radial clearance at 170 K, 2 mils at 300 K, -1 mil at 80 K |
| Coldfinger (on/off) | Sapphire, Ti | 120 | 0.4/1.2 | n/a | 15 | 3/4" O.D. x 1/4" I.D. x 4" Long Rod, 1.2 K/W at 170 K, 2.0 K/W at 300 K |
| OVCS Flexible Link | Al | 500 | 1.0 | n/a | n/a | Blind-demateable indium-gasket joint to sapphire coldfinger |
| OVCS Spreader Plate | Al | n/a | 0.2 | n/a | n/a | 1100-H Al, mass considered part of OVCS |
| OVCS | Al | n/a | 1.2 | n/a | n/a | 1100-H Al, 0.032" wall thickness |
| Gasketed/ Soldered Joints | Cerrolow, In | – | 0.1 | n/a | 10 | 5 $cm^2$ at 0.1 emissivity budgeted for joint radiation parasitic |
| Folded Support Tube | Glass/ Epoxy, Al | n/a | n/a | 15 | 15 | 5 mW due to edge effects at outer radiation shield |
| Thermal Switch Heater (2) | Pt/Ceramic, Al | n/a | n/a | 15 | 15 | 30 W at 170 K, 55 W at 300 K → need ~100 W P.S. (80 V, 1.25 A) |
| Contamination Shield | Al | n/a | n/a | n/a | 5 | 250 K radiation from this shield into dewar MLI |
| TOTALS | n/a | 1375 | 5.5/5000 | 30 | 170 | 250 K Al box enclosure reduces parasitic heatload to <150 mW |
| | | | | 200 | | |

**Coldtip Flexible/thermal Link.** Requirements for the link are:

- Thermal conductance ~1.0 W/K (dual, split link)
- Side loading to coldtip <0.2 lb, broken down as follows:
  - Mass <0.13 lb (for dual link) to control gravity loading of the coldtip during horizontal ground operation
  - Spring rate in any direction <1 lb/in. (for dual link) to allow generous 0.1 in. interface assembly tolerance buildup
  - Tolerance for setting null position ~0.01 in.
- Amplitude capability >±0.1 in. all directions (static)
- Low-torque precision demateable joints

The S-shaped link (**Fig. 4**) is made of about 300 layers of 3 in. long by 1 in. wide aluminum foil with clamped end fittings. The configuration provides nearly omni-directional flexibility. Aluminum was chosen because of its inherently low surface emittance, excellent thermal conductivity-to-mass ratio, self-stabilizing oxide coating, and mass-produced availability in consistent form. The ends, including the foils, are milled and coated with multilayer thin films to facilitate soldering. The link is soldered to the thermal switch with indium solder (156 °C) and to the coldtip with Cerrolow 136 solder (58 °C) for the final step of the interface integration process. Use of a low-temperature solder makes this final in-situ procedure easier, ensures that the null position of the link is not perturbed, avoids melting the indium solder joint, and avoids overheating the thermal switch assembly.

**Thermal Switch.** Requirements for the thermal switch are:

- Closed conductance large (~1 W/K) compared to overall interface
- Open conductance small enough (~0.2 mW/K) to reduce parasitic heating from the back-up cooler by at least a factor of 10
- Activation power less than that required to run the cooler (<100 W)
- Minimal parasitic heating by radiation and conduction through the heater power leads
- Launch survival with switch in open or closed configuration

The switch (**Fig. 5**) uses two precision-machined aluminum rings surrounding a molybdenum disk and relies on the materials differential thermal contraction to open and close. The gap at assembly is 0.002 in., resulting in switch closure at a temperature of 180-200 K. Each ring is supported and centered from the disk with thermal isolation by a set Vespel pins. Turning either cooler on causes the corresponding ring to shrink onto the disk. Each ring is equipped with a heater using 100 ohm platinum heating/sensing elements configured as four series pairs to provide redundancy. Activation of the heater with ~50W of power causes the ring to release from the disk. After that, parasitic heating to the ring maintains its open condition.

**Sapphire Cold Finger.** Sapphire was chosen for the cold finger because of its great thermal conductivity and because its thermal conductivity drops dramatically as the temperature rises during the switch opening process. Since heat is always well conducted from the switch to the outer OVCS via the cold finger, this second characteristic helps minimize the power and time

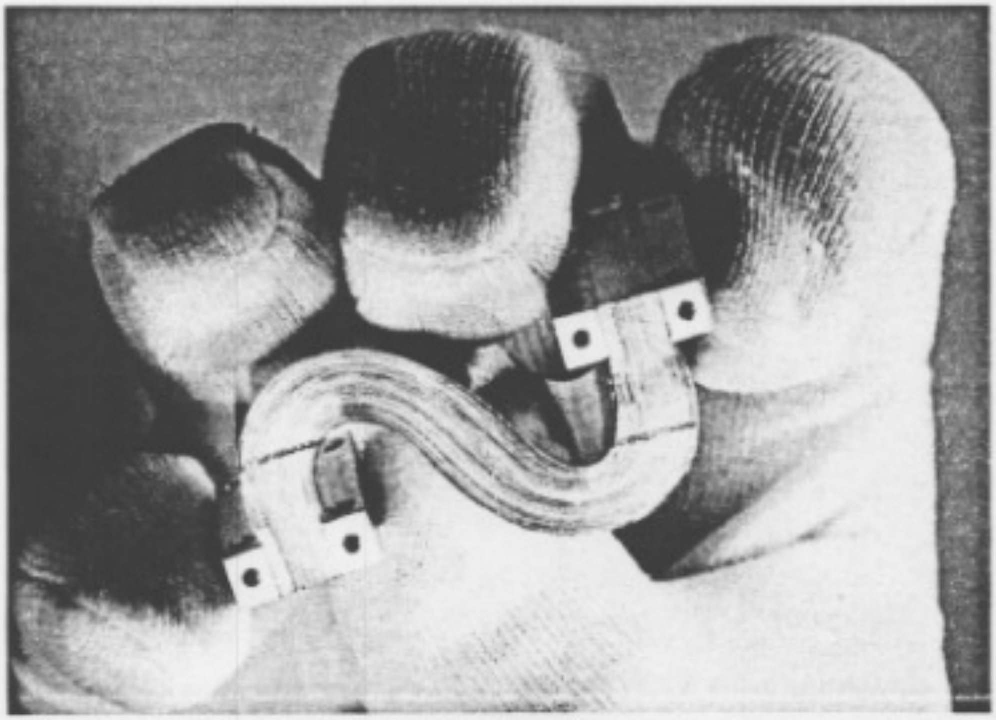

**Figure 4.** S-shaped flexible, high-thermal conductance link.

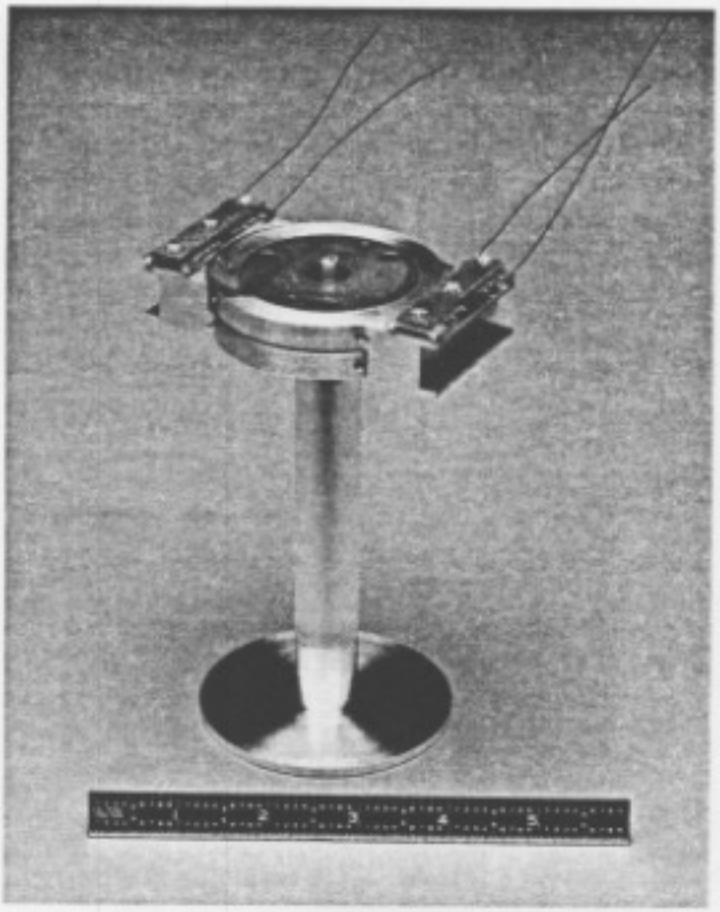

**Figure 5.** Dual thermal switch with attached sapphire cold finger.

needed to open the switch. The sapphire rod, shown in Fig. 5, is ~4 in. long with 0.75 in. diameter, providing a conductance of 2.5 W/K when the switch is closed. The assembly is held together by a high-strength bolt through the center of the sapphire rod, using indium gaskets at each end.

**Folded Fiberglass/epoxy Tube.** The folded tube assembly (**Fig. 6**) holds the thermal switch and accurately registers it to the cooler coldtips while providing excellent thermal isolation from the warm girth ring. The 0.015 in. wall tubes are bonded into trough-shaped joints. It supports a mass is of ~2 lb, resulting in a maximum joint bending moment of 120 in.-lb during qualification loading. Based on thermal modeling, a configuration using five tubes was chosen as the best compromise between complexity and thermal performance. With the warm end at 300 K temperature and the cold end at 80 K, the total predicted heat load through the assembly is ~40 mW, comprised of about half conduction and half radiation.

**Outer VCS Flexible/thermal Link Assembly.** Requirements are:

- Thermal conductance >1.0 W/K
- Side loading to cold finger <1 lb when the OVCS moves 0.25 in. relative to the vacuum shell, and hence the cold finger, during launch; this makes the contribution to the folded tube assembly loading small compared to inertial loads; broken down as follows:
  - Spring rate < 4 lb/in. for assembly (<1 lb/in. for each link)
  - Tolerance for setting null position <0.03 in. for assembly (<0.01 in. for each link)
- Amplitude capability >±0.25 in. at 30 Hz frequency for 300 cycles in vacuum

This assembly (Fig. 3) uses four S-shaped links which connect a central hub to an outer ring, which is in turn attached to the OVCS. The design and fabrication of each link are very similar to the coldtip link.

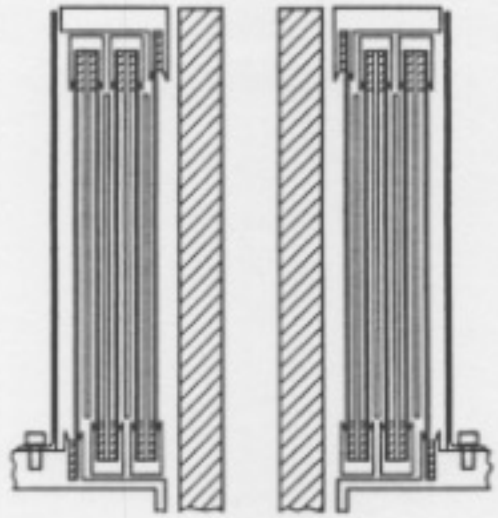

**Figure 6.** Folded fiberglass/epoxy tube assembly for thermal switch/cold finger support.

## COMPRESSOR MOUNT REQUIREMENTS AND DESIGN

The compressor produces residual vibration at its running frequency of ~40 Hz and the first ten or so harmonics. Disturbances at the running frequency are effectively reduced by active electronic control. The significant disturbances are likely to be in the lateral directions and at the harmonic frequencies, with peak forces up to 0.1 lb expected. Based on the allowable vibration at the detector array and the expected amplification factors of the dewar support system, the attenuation requirement for the compressor mount is a factor of at least 10 at 80 Hz and at least 25 above 120 Hz in any direction.

Each twin compressor is mounted to the girth ring with a system of steel coil springs and urethane snubbers, as shown conceptually in **Fig. 7**. The springs are arranged to uncouple the six rigid-body modes and are sized to provide ~14 Hz natural frequency for each mode to provide the required omni-directional attenuation. The snubbers must limit compressor impact loads to <50 g. Gravity causes the assembly to rest on the bottom snubbers. Therefore, to allow system-level measurement of detector array vibration level, the snubbers are removable. In combination the spring and snubber system limits motions to <0.07 in. to avoid damaging the transfer tube

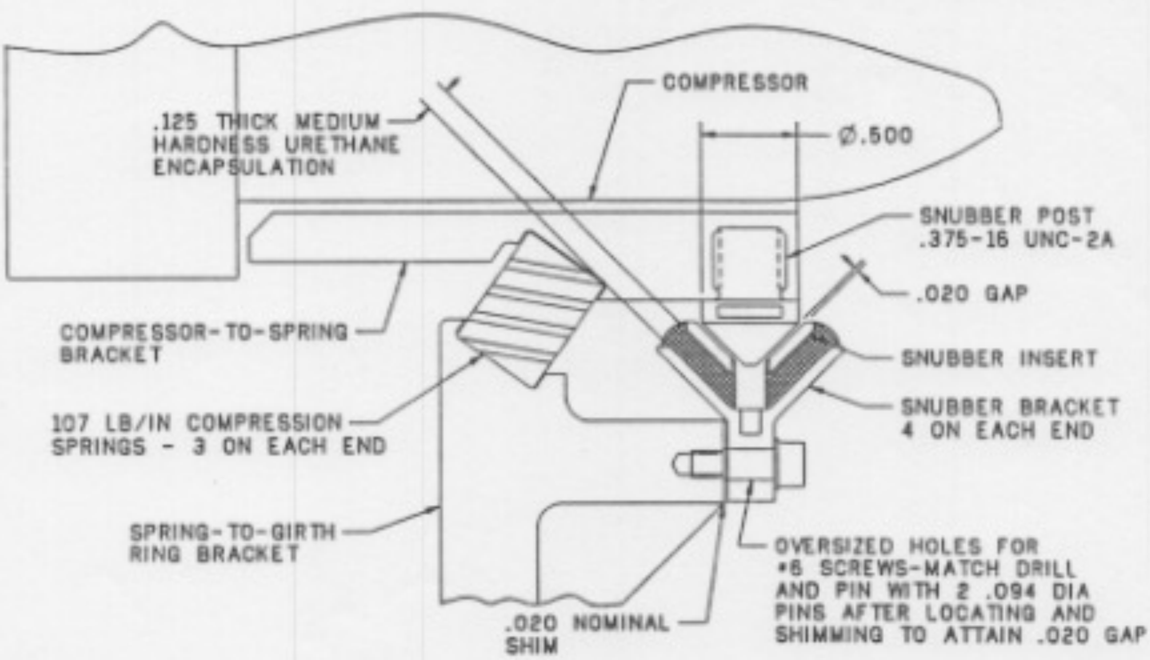


**Figure 7.** Spring and snubber mount approach for compressors.

that connects the compressor to the displacer. The compressor heat rejection scheme includes thermal/flexible links, similar to those discussed above, to conduct heat from the compressor to the girth ring. Their stiffness is small compared to the mounting springs.

## VERIFICATION ACTIVITIES

**Coldtip Flexible/thermal Link.** Because of the tight-tolerance mechanical performance requirements of this link, an apparatus was developed for precisely measuring its null configuration and spring rate with the ends properly constrained against rotation. Without proper end constraint spring rates would appear erroneously low. The apparatus uses a ball-bearing, two-axis slide table (**Fig. 8**) with gravitational force applied to the link by tilting the table. The spring rate is simply and reliably determined by precisely measuring the incremental tilt and deflection and knowing the mass of the moving parts. Measurements on the last several links fabricated showed spring rates and null setting tolerance met the established requirements. Finally, a shake test was performed to show that protoflight qualification load levels did not significantly perturb the spring rate or null position of the link.

**Low-temperature Solder Joint.** Two areas of concern were addressed related to the solder joints. First is the adhesion, stability, solderability and corrosion resistance of the multilayer thin film coatings on aluminum. Second is the reliability, reproducibility and thermal conductance of the fluxless, low-temperature Cerrolow 136 solder joint after repeated mates and demates. A sample joint having approximately the same surface area (0.5 in.$^2$) as the coldtip link was made. A side load of ~250 lb was needed to break it, far greater than would be experienced during launch or could be applied by the S-link. The joint broke cleanly at the solder with no detectable delamination of the thin film coating. It was easily refurbished and resoldered without flux at a temperature of ~60 °C. After repeated breaking and resoldering, there was no loss in strength and no void development in the solder coat. The joint was aged for 2 years at room temperature and again tested. Results showed no form of degradation. Finally, the joint was held at a temperature of 100 °C for a week to determine if corrosion of the thin film coating would occur at the elevated temperature. Testing and inspection again showed no sign of degradation.

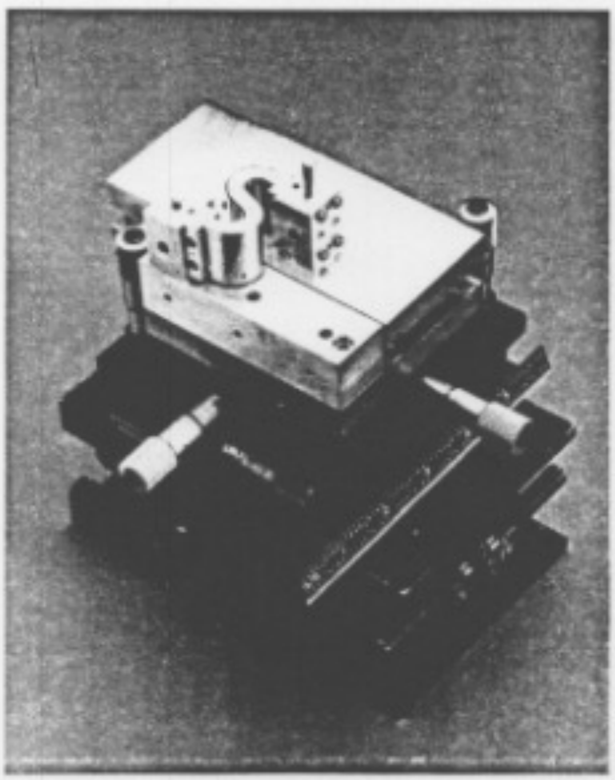

**Figure 8.** S-shaped link on spring rate and null position measuring apparatus.

**Thermal Switch.** The thermal switch design uses no new fundamental technology. However, because of the tight tolerances and special materials involved, we decided to build a full-up prototype to verify the fabrication process, the switch performance, and launch survivability. We first tested the heater alone to show that the platinum elements and power leads had sufficient current-carrying capability. We also showed that temperature gradients within the heater were acceptably small. Also, unable to find adequate data, we performed tests to verify that Vespel has good fracture toughness at a temperature of 77 K. The switch was thermally cycled 25 times from 300 to 77 to 300 K, and making and breaking of switch contact at a temperature of ~180 K was reproducible. Testing in a simulated thermal/vacuum environment to further characterize the switch and determine the exact activation control requirements is still planned. We decided the switch would be closed at launch, and a shake test is not necessary.

**Folder Fiberglass/epoxy Tube.** We fabricated a partial prototype to establish the bonding process and to verify the strength of the thin-walled fiberglass/epoxy tube and the two types of joint design used. The prototype was fixtured (**Fig. 9**) for testing on a shake table. It was subjected to a random qualification level of 14.1 $g_{rms}$ at temperatures of 300 K and 80 K without damage. A sine sweep was then used to provide increasing levels of lateral load at a temperature of 80 K until failure resulted. The bending moment at failure was over 50 percent greater than required.

**OVCS Flexible/thermal Link Assembly.** We were concerned about the potential for cold welding between foils of this link due to the large amplitudes of motion it would experience in vacuum during launch and testing. Substantial cold welding would compromise the flexibility of the link. We installed a link in a vacuum chamber with a vacuum-rotary feedthrough to allow the link to be flexed by a motor (**Fig. 10**). The link could be positioned within the chamber to allow flexing in any direction, and liquid nitrogen lines were included so that it could be exercised cold (**Fig. 11**). We flexed the link 28,000 cycles each at amplitudes of 0.2 and 0.3 in., at temperatures of 300 and 80 K, in all directions, and frequencies of 10, 20 and 30 Hz. Even after this severe routine, inspection showed no substantial change in null position or spring rate. We then cycled it to an amplitude of 0.4 in., which eventually resulted in a minor change of the null position and tearing of two outer foils.

**Compressor Mount.** Since the performance of a soft snubber of this type is somewhat unpredictable, we first tested the snubber alone to verify it had suitable stiffness. We then assembled a complete prototype of the mount (**Fig. 12**) using a full-size mass model of the twin compressor and a simulated transfer tube. Testing verified: (1) natural frequency of the spring system, (2) survivability of the mount system, especially the snubbers, (3) survivability of the

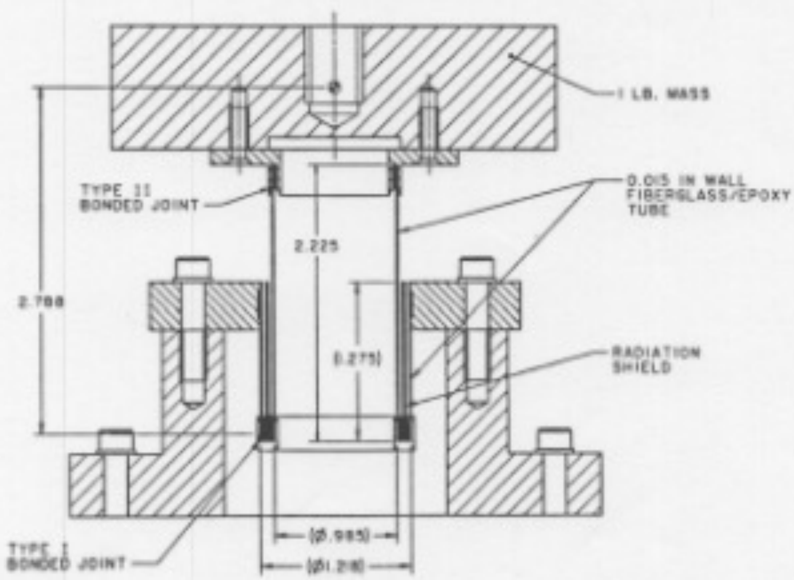


**Figure 9.** Prototype folded tube in strength test fixture.

transfer tube, and (4) level of impact loading to the compressor. Measured frequencies of the six modes ranged from 13.8 to 17.2 Hz, which agrees with predictions to within 10 percent. During exposure to 14.1 $g_{rms}$ random qualification-level vibration (**Fig. 13**), the load input to the compressor was less than 30 g, close to the predicted level, and well within the 50 g goal. Post-test inspection showed no change in compressor alignment and no significant damage to the snubber system. Microscopic inspection of the transfer tube critical areas showed no damage.

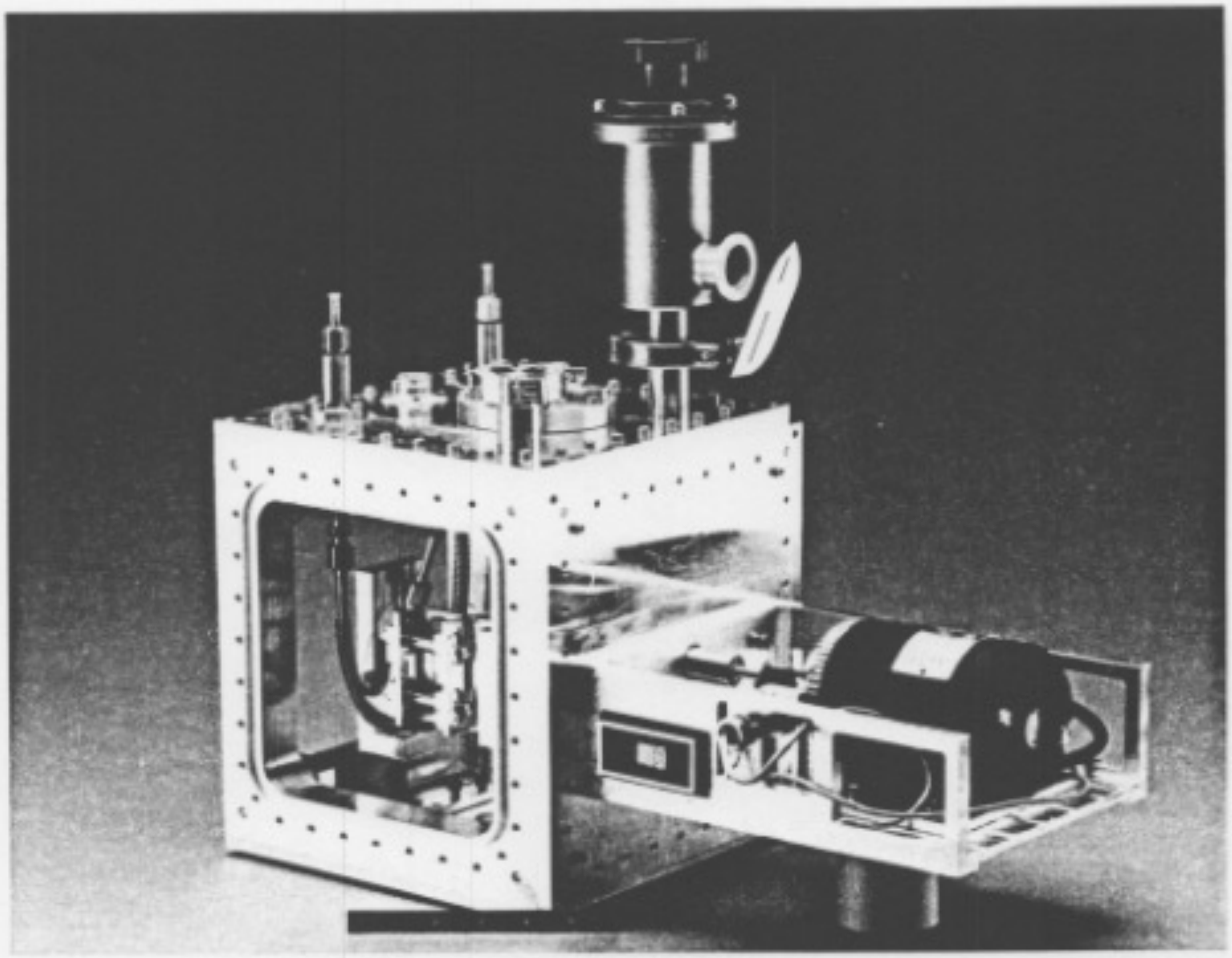

**Figure 10.** Apparatus for motion testing S-shaped flexible/thermal link in vacuum.

## CONCLUSIONS

Detailed designs for interfacing of Stirling coolers to a long-lifetime, spaceborne sensor system were established by extensive analyses and verified by prototype testing. The coldtip interface design provides an optimum combination of low thermal impedance and low parasitic heating while accommodating generous assembly tolerance buildup and large relative motions between the vacuum shell and cooled sensor during launch. A critical element of the design is the S-shaped flexible/thermal link that has an extremely large ratio of thermal conductance-to-stiffness. The thermal impedance of this high-performance link is small compared to the overall interface and makes it possible to interface a Stirling cooler coldtip with nearly the same efficiency as other types of cooler, such as a pulse tube, whose coldtips are not as sensitive to side loading. The simple, highly efficient thermal switch nearly eliminates the parasitic heat load of the back-up cooler with hardly more thermal impedance and parasitic heating than would result from a simple thermal bus connecting the two cooler coldtips to the cold finger.

## ACKNOWLEDGMENTS

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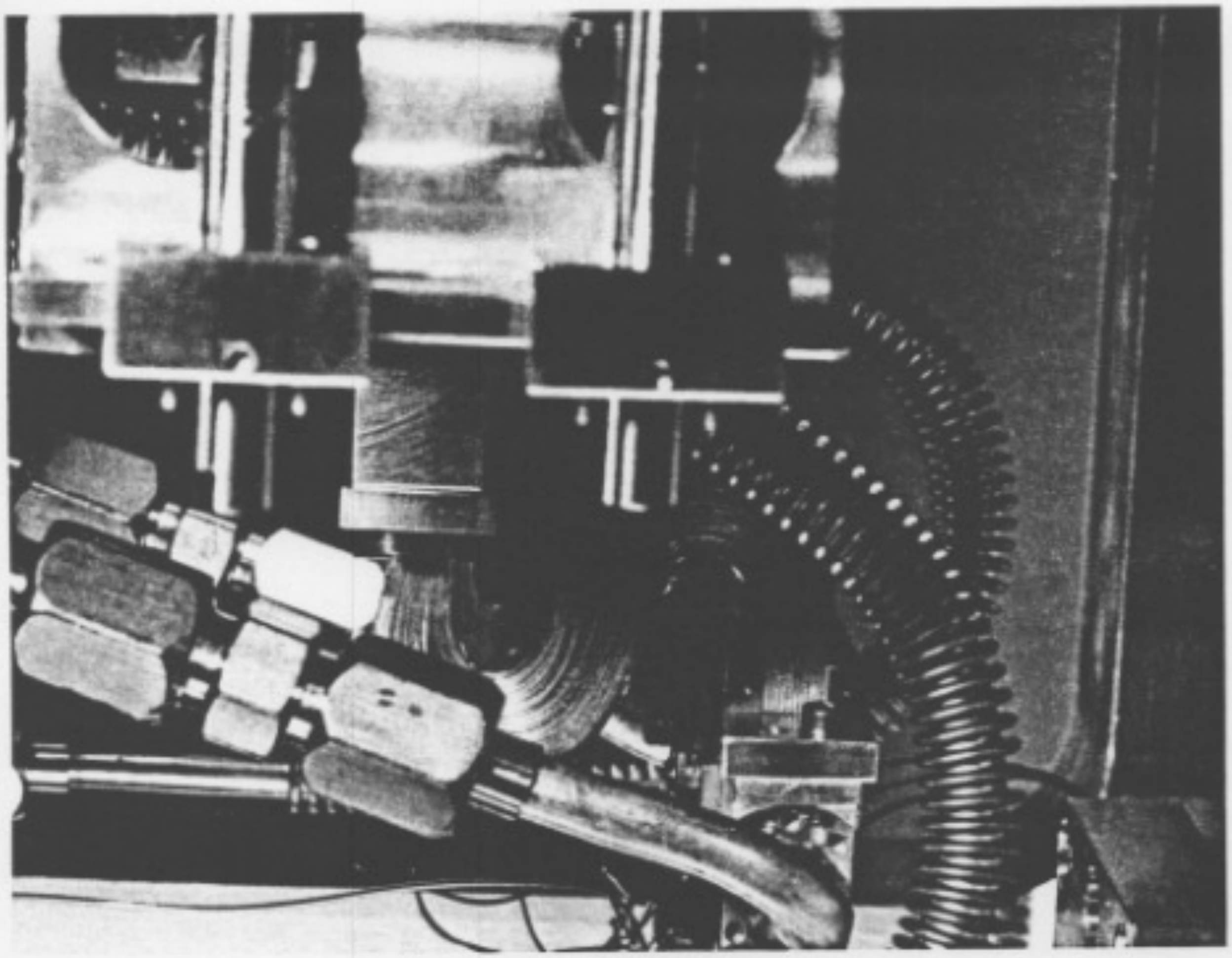

**Figure 11.** S-link and cooling coils inside motion testing apparatus.

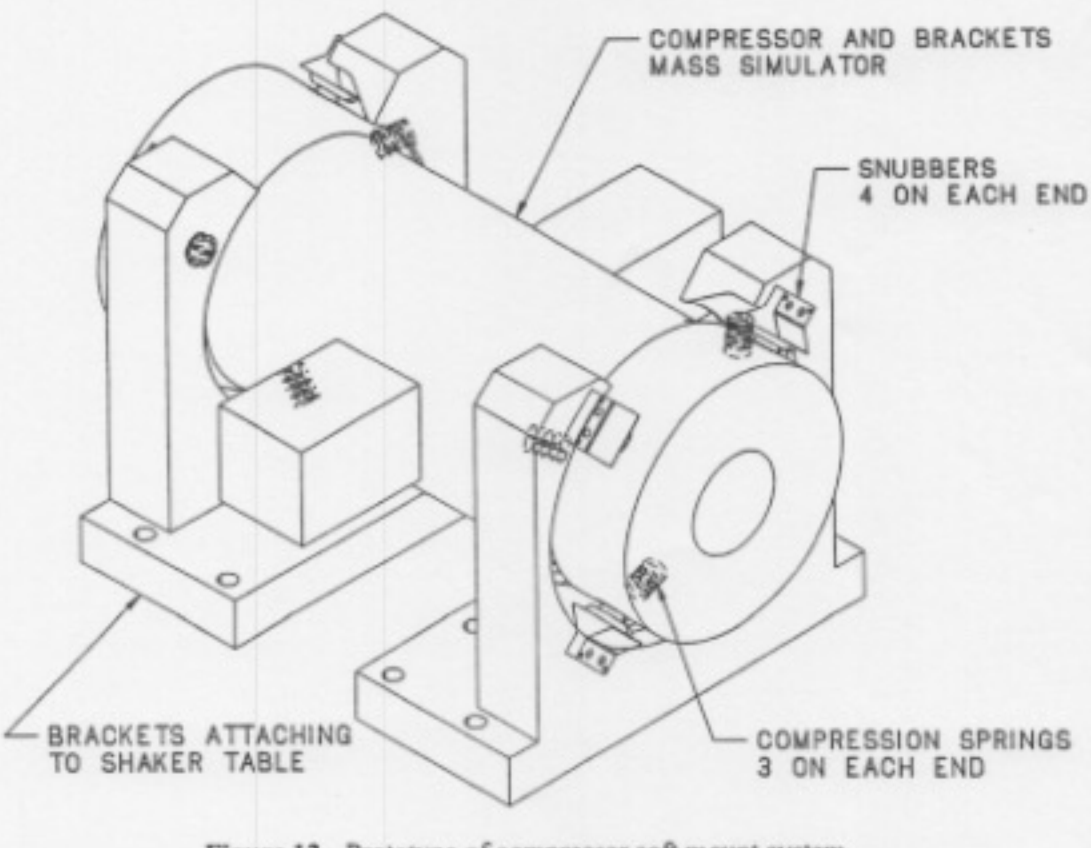


**Figure 12.** Prototype of compressor soft mount system.

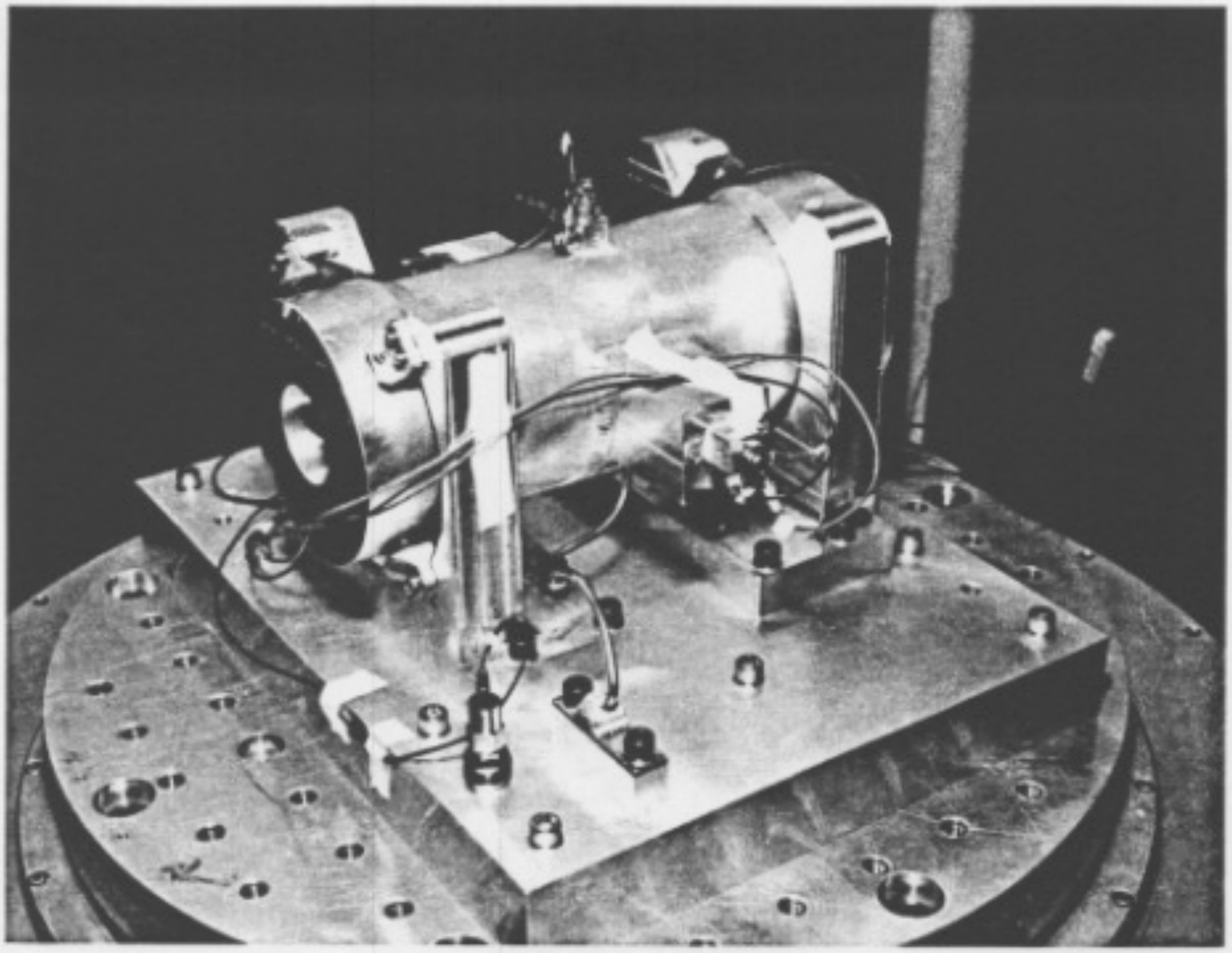

**Figure 13.** Compressor mount prototype on shaker.

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