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968 Albany-Shaker Road
Latham, New York 12110

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Design and Development of a Resonant Piston Compressor for Hermetically Sealed Platforms

by

P. W. Curwen

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Design a Development of a Resonant Piston Compressor for Hermetically Sealed Platforms

P. W. Curwen

Author (s) P.W. Curwen

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MTI
MECHANICAL TECHNOLOGY INCORPORATED

968 ALBANY - SHAKER ROAD — LATHAM, NEW YORK — PHONE 785-0922

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ABSTRACT

In 1966, under contract NAS 8-11660, MTI delivered a development-prototype Gas Supply System for Hermetically Sealed Platforms to NASA-MSFC. The system was oriented to future applications of the ST-124 stabilized platform for the Saturn V Instrument Unit.

The Gas Supply System consisted of (1) a Recirculation Subsystem which continuously compresses nitrogen from the platform and recirculates it to the gas bearings used in the AMAB-3 accelerometers and the AB-5 gyros, and (2) a Gas Make-up Subsystem for feeding nitrogen into the Recirculation Subsystem at a rate sufficient to match any reasonable leakage rate from the circulating loop to space ambient.

The major accomplishment of contract NAS 8-11660 was development of a unique type of compressor for the Recirculation Subsystem. The compressor, referred to as a Resonant Piston Compressor, was designed for (1) a minimum of 10,000 hours of continuous, maintenance-free operation, (2) zero contamination of the nitrogen gas, (3) hermetically sealable construction, (4) operation during launch, and (5) operation over an ambient temperature range of 0 to 60°C. Over 4,000 hours of testing of the two compressors supplied under contract NAS 8-11660 confirmed that the Resonant Piston Compressor could probably meet the above objectives.

This report describes work performed under contract NAS 8-20720 to achieve the following improvements to the Resonant Compressor design: (1) size reduction via the use of a different type of resonant spring, (2) ability to operate under higher levels of transverse g loading, and (3) use of aluminum castings for the drive solenoid mounting bracket and for the compressor enclosure. One compressor incorporating these improvements was built and tested by MTI, and has been shipped to NASA-MSFC for further evaluation tests.

I. INTRODUCTION

Mechanical Technology Incorporated (MTI), under NASA contract NAS 8-20720, has developed an improved Resonant Piston Compressor for use in a Gas Supply System for Hermetically Sealed Platforms. This work was performed for the Inertial Sensors and Stabilizers Division of the Astrionics Laboratory, George C. Marshall Space Flight Center.

The development-prototype of the Gas Supply System was built by MTI under NASA contract NAS 8-11660 and delivered to NASA-MSFC in 1966. This development was fully documented in the contract final report*.

The Gas Supply System constitutes a means for providing a 10,000-hour supply of pressurized nitrogen to the gas bearings used in the AMAB-3 accelerometers and the AB-5 gyros for the ST-124 stabilized platform. To accomplish this function, the Gas Supply System is composed of two major subsystems as follows:

1. a Recirculation Subsystem which continuously recompresses the nitrogen from the discharge side of the bearings and recirculates it to the inlet side of the bearings;

2. A Gas Make-Up Subsystem comprised of a small high-pressure nitrogen reservoir and a means for feeding gas from the reservoir into the Recirculation Subsystem at a rate sufficient to match any reasonably small out-leakage rate from the circulating loop to space ambient.

The heart of the Recirculation Subsystem, and, in fact, of the whole Gas Supply System, is the nitrogen compressor. The requirements for this compressor are:

1. long, maintenance-free life (in excess of 10,000 hours continuous operation)

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* Curwen, P.W., "Design and Development of a Gas Supply System for Hermetically Sealed Platforms", Mechanical Technology Inc., Report No. 66TR57, Nov. 14, 1966.

2. hermetically sealable construction

3. truly noncontaminating operation (less than one part per million by weight of condensable hydrocarbon content)

4. ability to operate during launch

5. ability to operate over an ambient temperature range of 0 to 60°C

6. minimum possible drive power

7. minimum weight and size consistent with the above objectives.

Under contract NAS 8-11660, a Resonant Piston Compressor was developed to meet the above requirements. The concept and design of the compressor is described in MTI report 66TR57. Two compressors were built and delivered to MSFC (together with the other components of the Gas Supply System) for evaluation testing. Over 4,000 hours of test operation were accumulated by MSFC, including centrifuge testing up to 10g. Throughout this testing there were no failures of the basic compressor mechanism. (Several fatigue failures did occur in the aluminum sheet-metal compressor enclosure.)

In these first models of the Resonant Piston Compressor, the resonant tuning spring was an electron-beam-welded assembly of four U-shaped cantilever springs. Figure 1 shows the spring configuration. Figure 2 shows the spring installed in the assembled compressor. It is evident from Figures 1 and 2 that this first configuration of the resonant spring was quite expensive to make and quite inefficient from the standpoint of envelop size. Although the spring proved to be reliable and trouble free, it had the largest envelop diameter of all the compressor parts and thus dictated overall diameter of the compressor enclosure.

Recognizing the cost and size disadvantages of this first spring configuration, MTI undertook a privately-funded effort to find another resonant spring configuration which would have reduced cost and size. This effort produced a successful demonstration of a parallel-coil helical spring concept having somewhat lower cost and appreciably smaller size. A preliminary layout study indicated that the helical spring concept would significantly reduce the envelop size of the compressor for the NASA Gas Supply System.

Based on the promise of a smaller size, and desiring also several other design improvements, MSFC issued a new RFP for an improved version of the Resonant Piston Compressor. In addition to requesting a size reduction via use of a helical spring, the RFP also requested the following:

1. that the compressor be designed to operate at higher levels (up to 6 g) of transverse acceleration;

2. that the drive solenoid mounting frame and the compressor enclosure be made out of Almag 35 aluminum castings.

Aside from the improvements mentioned above, no other changes were to be made to the design of the Resonant Compressor as initially procured under contract NAS 8-11660.

Contract NAS 8-20720 for design and development of one improved Resonant Piston Compressor was awarded to MTI on March 28, 1967. Shipment of the compressor to MSFC was made on November 29, 1968. The following sections of this report document the improved features of the compressor. Aspects of the compressor design which were unchanged are not discussed here, since they are reported in detail in MTI Report 66TR57.

II. SPECIFICATIONS

The original specifications for the Gas Supply System (and associated compressor) were given in the Scope of Work (Exhibit A) of contract NAS 8-11660. The first two pages of Exhibit A (those containing the technical specifications) are reproduced in Appendix A of this report.

The specifications for the improved Resonant Piston Compressor are given in the Scope of Work (Exhibit A) of contract NAS 8-20720. The first two pages of Exhibit A for this contract are reproduced in Appendix B.

There is only one significant performance difference between the specifications for the improved compressor and those for the original Gas Supply System. The difference occurs in the linear acceleration specification. For the original procurement, linear acceleration was stated to be "10 g [along the flight direction]". For the improved compressor, linear acceleration is stated to be "10 g along the flight direction and 6 g along the other two mutually perpendicular axes". In other words, the improved compressor should be capable of operating when subjected to conditions of constant 6-g transverse acceleration and constant 10-g axial acceleration.*

* Axial acceleration refers to acceleration in the direction of piston reciprocation. Transverse acceleration is acceleration at right angles to the direction of piston reciprocation. The Resonant Compressor should be mounted such that the direction of piston reciprocation is parallel to the flight direction.

III. HISTORY OF IMPROVED COMPRESSOR DEVELOPMENT

Contract NAS 8-20720 for an improved Resonant Piston Compressor was awarded to MTI on March 28, 1967. The first phase of the contract involved analysis and design layout of the improved compressor. The improved design was submitted to MSFC on June 2, 1967, and approval to proceed with detailing, fabrication, assembly, and testing of the compressor was received from MSFC on June 16, 1967.

Initial testing of the improved compressor began in November, 1967. It was quickly found that the reciprocating plunger assembly would go unstable at flow and pressure-rise conditions well below design-point conditions. The instability consisted of large-amplitude vibration in the bending mode of the spring. This motion caused the solenoid plunger to hit against the lateral air-gap pole faces of the solenoid stator.

The bending-mode natural frequency of the reciprocating assembly was experimentally measured and found to be 53 hz. Since the axial stroking frequency of the compressor was 60 hz, it was immediately obvious that the presence of any 60 hz bending-mode excitations would cause significant resonance of the bending-mode displacements. Several types of bending-mode excitation are possible, the most obvious being an imbalance of the solenoid electromagnetic lateral forces acting on the sides of the solenoid plunger.

The problem of bending-mode resonance had been anticipated during the design phase. The resonant spring and reciprocating assembly had supposedly been designed such that there would be no undesirable natural frequency within 30 percent of the 60 hz stroking frequency. Since the experimental data showed that this criterion had not, in fact, been satisfied, the derivations of the various natural frequency equations were reviewed. The review disclosed that a factor of π had inadvertently been dropped from the derivation of the bending stiffness of the spring. This error was confirmed by comparison of the calculated and measured bending stiffness of the spring. The error was reported to MSFC in the December 15, 1967 monthly progress letter, and a study was immediately undertaken to determine how the situation could be corrected.

Using the corrected (and verified) bending-mode natural frequency analysis, a new spring design was evolved which, together with slight modifications to the reciprocating plunger assembly, appeared to be a solution to the compressor instability problem. Fortunately, the corrected design necessitated remaking only two parts: the resonant spring and the connecting rod. The corrected design was forwarded for approval to MSFC on January 19, 1968. A request for additional funding to impliment and test the corrected design was forwarded to MSFC on February 5, 1968.

Approval to proceed with procurement and testing of the corrected design was received from MSFC in May, 1968. A new resonant spring was received from the spring vendor in August, 1968. Receiving inspection revealed that the spring was very much out of tolerance. As a result, the measured axial stiffness of the spring was 395 lb/in. instead of the desired 310 lb/in. Payment for the spring was thus held-up pending determination as to whether the spring would permit satisfactory compressor operation.

Assembly of the compressor with the new out-of-tolerance spring was completed. To achieve the correct axial resonant frequency (57.6 hz) with the 395 lb/in. spring, it was necessary to add 0.25 pounds of mass to the reciprocating assembly. This represented a 22 percent increase in the weight of the reciprocating parts, and was achieved by epoxying a cylindrical piece of Ni-resist iron into the bore of the carbon piston. The bending-mode natural frequency was then experimentally checked and found to be 83.6 hz —— very close to the predicted value of 82 hz. (Because of the manner in which the 0.25 pound mass was added to the reciprocating assembly —— namely, inside the piston —— the mass has no significant influence on the bending-mode natural frequency. This is because of the decoupling effect of the piston connecting rod.)

The assembled compressor was subjected to a series of initial tests to determine optimum values of the three assembly shims (Part Nos. 7, 8, and 47). At the optimum settings, design-point compressor performance could be easily achieved and exceeded. At thermal equilibrium conditions, design-point pressure rise and flow were achieved with approximately 89 v. rms (60 hz) applied to the drive solenoid.* With 96 v. rms applied, 32 percent excess flow was obtained at design-point pressure rise.

At solenoid voltages less than 85 v. rms, the compressor could be safely operated from wide-open to fully shut-off flow. However, above 85 v. rms it was found that overstroking of the piston would occur (causing the piston to contact against the valve plate) as the shut-off flow condition was approached. For instance, with 87 v. rms applied to the solenoid, the plunger would start to contact at flow and pressure-rise values of 0.3 SCFM and 26 psi, respectively. At 90 v. rms solenoid voltage, contact started at 0.43 SCFM and 24 psi.

Design-point pressure rise of the compressor is 15 psi. Accordingly, it appears that overstroking of the piston can easily be prevented in the present compressor configuration by simply assuring that compressor pressure rise never exceeds 20 to 22 psi. A pressure relief valve is the discharge line, set to open at 20 to 22 psi, should provide positive protection against overstroking.

It is also possible that overstroking would not occur if a resonant spring having the intended value of axial stiffness were used. A spring having the 310 lb/in. design value of stiffness would result in greater detuning of the compressor at high pressure-rise conditions relative to the detuning that occurs with the 395 lb/in. out-of-tolerance spring. Accordingly, the out-of-tolerance spring was rejected and the spring vendor was asked to remake the spring.

The improved compressor, fully assembled with the 395 lb/in. out-of-tolerance spring, was shipped to MSFC on November 29, 1968. Total design-point operation on the unit during testing at MTI was in excess of 250 hours.

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* Testing of the compressor at MTI was performed using commercial 60 hz electrical power. To obtain the required 60 hz driving force from the drive solenoid, a diode was connected in series with the solenoid.

Since 60 hz electrical power is not available in the Saturn V Instrument Unit, a special drive circuit has been developed for the compressor by MSFC. The circuit operates from 28 v. d-c and produces a nominal 30 hz solenoid driving current (frequency of the driving current can be adjusted to obtain optimum matching to compressor natural frequency).

Voltage values (rms) measured across the drive solenoid terminals when operating the compressor from a 60 hz power line will be quite different from values measured during operation with the 28 v. d-c inverter circuit.

A replacement spring for the out-of-tolerance spring was received at MTI in March 1969. The axial spring rate was measured and found to be 318 lb/in. Since this value was within 3 percent of the nominal design value, the spring was accepted and forwarded to MSFC on March 30, 1969. At this writing the replacement spring is being installed in the compressor by MSFC personnel.

IV. DESIGN OF THE IMPROVED COMPRESSOR

The following paragraphs document the Resonant Compressor design improvements made under contract NAS 8-20720. The documentation applies to the design as specified by MTI Parts List PL 192E01, a copy of which is included in Appendix C of this report. It should be noted that Parts List PL 192E01 represents the intended design of the improved compressor. Unfortunately, the compressor which was actually tested at MTI (and subsequently shipped to MSFC) did not fully conform to the intended design. The deviations from the intended design were as follows:

1. the resonant spring was excessively stiff (395 lb/in. instead of the nominal 310 lb/in. design value);

2. additional weight, in the form of a 0.25 pound ni-resist sleeve, had to be added (epoxied) to the carbon piston to compensate for the excessive stiffness of the spring.

Although the spring was procured using the correct detail drawing specified on Parts List PL 192E01, manufacturing errors resulted in coil dimensions which greatly exceeded the specified dimensional tolerances. These dimensional inaccuracies, in turn, resulted in the excessively high value of spring stiffness.

As described in the previous section of this report, a replacement spring having the correct dimensions (and hence the correct stiffness) was subsequently received from the spring vendor. However, since the compressor had already been shipped to MSFC (with the out-of-tolerance spring), MTI was not able to experimentally determine whether the intended spring design would, in fact, result in optimum compressor performance. In a very real sense then, the improved resonant spring design documented below has not been experimentally proven. Installation of the replacement spring is underway at MSFC as this report is being written. Evaluation of compressor performance with the intended spring design will likewise be undertaken by MSFC personnel.

Improved Resonant Spring

The improved design of the resonant spring was the major accomplishment of contract NAS 8-20720. Figure 3 shows the configuration of the improved spring, together with the other parts of the reciprocating assembly. Figure 4 shows the assembled arrangement of the improved spring in the improved compressor. Comparison of Figures 2 and 4 immediately shows the much more compact nature of the improved spring design. This compactness results in a considerable overall reduction in envelop size of the improved compressor.

The improved resonant spring is a helical spring having three parallel coils of rectangular cross-section. The pertinent design parameters of the spring are as follows:

Number of parallel coils	3
Number of turns per coils	1.012
Thickness of coils (in axial direction of the spring)	0.128 in.
Width of coils (in radial direction of the spring)	0.352 in.
Mean diameter of spring	3.00 in.
Helix angle of coils	5.18 deg.
Free height of coils	0.8659 in.
Spring material	Almar 362 (Allegheny Ludlum), Precipitation hardened to R_c 37
Properties of spring material	
Density	0.281 lb/in.^{3}
Young's modulus	3.0×10^7 psi
Shear modulus	1.1×10^7 psi
Endurance strength in reversed bending in heat treated condition	93,000 psi
Elastic properties of spring (calculated)	
Axial stiffness	310.8 lb/in.
Restrained lateral stiffness	3,568.8 lb/in.
Torsional stiffness	4,376.6 in.-lb/rad.
Bending stiffness	892.4 in.-lb/rad.
Maximum shear stress in spring coils	106,500 $\frac{\text{psi}}{\text{inch of defl.}}$

The spring is machined from a solid piece of Almar 362 stock. The ends of the coil are thus integrally joined to the top and bottom mounting flanges of the spring. The use of integral flanges, together with generous transition radii at the ends of each coil, eliminates any stress concentration effects at the ends of the coils, such as would be present if mechanical clamps were used to hold the coils.

The maximum allowable design value of shear stress for the spring coils was established by the following formula:

$$\tau_{d_{max}} = \sigma_{eb} \left(F_{es} \times F_{sf} \times \frac{1}{F.S.}\right)$$

where

$\tau_{d_{max}}$ = maximum allowable design value of coil shear stress, psi

σ_{eb} = endurance limit in reversed bending of spring material, psi

F_{es} = shear stress endurance limit derating factor

F_{sf} = surface finish derating factor

F.S. = factor of safety.

Using values of

σ_{eb} = 93,000 psi

F_{es} = 0.57

F_{sf} = 0.80

F.S. = 1.6

gives $\tau_{d_{max}}$ = 27,500 psi.

The maximum shear stress in each spring coil is calculated to be 106,500 psi per inch of spring deflection. Dividing this value into the maximum allowable value of shear stress (27,500 psi) gives a maximum allowable spring deflection of ± 0.258 inches. If the spring is limited to operation about a position of small static compression, the criterion for maximum allowable spring deflection can also be expressed in terms of the total range of spring deflection, which, of course, is the same as piston stroke. Accordingly, maximum allowable piston stroke (assuming small values of static spring compression) for the improved spring design is 0.518 inches.

The calculated design stroke for the improved compressor is 0.42 inches. The calculated design-point value of static spring compression is 0.083 inches (laboratory measurements indicate that the actual static compression is less than the calculated value). Thus the improved resonant spring design can tolerate a 24 percent piston overstroke before the maximum allowable shear stress will be exceeded. In all respects, the improved resonant spring design is quite conservative and should have essentially infinite life in the improved compressor application.

Hydrostatic Piston Bearing

The compressors supplied under contract NAS 8-11660 used two tapered hydrodynamic gas bearings to obtain floatation of the piston within the radial clearance of the cylinder. One tapered bearing was located at each end of the carbon piston. However, these bearings were not able to maintain piston floatation when the compressor was subjected to lateral acceleration loads in excess of several g.

For contract NAS 8-20720, it was specified that the improved compressor should be designed to maintain piston floatation under lateral acceleration loads up to 6 g. To obtain this capability, it was necessary to incorporate a hydrostatic piston bearing into the improved compressor design. While the hydrostatic bearing provides most of the piston load capacity, one of the original hydrodynamic bearings was retained for additional capacity, this being the tapered bearing at the cylinder-head end of the piston.

The hydrostatic piston bearing obtains its pressure and flow from the discharge of the compressor. Consequently, the penalty which must be paid for higher lateral load capacity is somewhat higher pumping power (or, equivalently, somewhat lower apparent compressor efficiency*). To minimize the loss in overall compressor

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* Actual compression efficiency of the NAS 8-20720 compressor is as high, if not higher, as that obtained from the NAS 8-11660 compressors. However, since the hydrostatic bearing flow is not available as useful compressor output flow, it is treated as internal leakage flow and hence is accounted for in the overall apparent efficiency of the compressor.

efficiency, every effort was made (within practical limits) to design a minimum flow hydrostatic bearing.

Based on an analytical design study of the lateral deflection characteristics of the reciprocating piston assembly (which included influences of the helical resonant spring and the hydrostatic piston bearing), the following hydrostatic bearing design, optimized for minimum bearing flow, was evolved:

Nominal Design Conditions for Hydrostatic Piston Bearing*

Length-to-diameter ratio (L/D)	0.70
Diameter (D)	2.00 in.
Radial clearance (C)	0.0006 in.
Number of hydrostatic orifices (n)	15
Diameter of hydrostatic orifices (d)	0.010 in.
Bearing flow (Q)	0.05 SCFM
Radial stiffness (K_R)	17,800 lb/in.
Angular stiffness (K_A)	4,950 in-lb/rad.

Based on the actual dimensional tolerances selected for the piston diameter and the cylinder bore, the radial clearance of the hydrostatic bearing will be between 0.0005 and 0.0007 inches. The corresponding limits of bearing flow at nominal pressure conditions will be 0.03 to 0.067 SCFM. These limits of bearing flow correspond to 5.6 and 12.6 percent of the required 0.53 SCFM of useful compressor output flow. Hence, a 5.3 to 11.1 percent reduction in overall compressor efficiency can be expected due to the hydrostatic piston bearing.

Lateral Load Capacity

An elastic deflection analysis for lateral and angular displacements of the reciprocating piston assembly was derived so that an optimum design of the piston connecting rod could be achieved. The analysis model was based upon a

* Nominal design conditions are based on a 28.5 psia supply pressure and a 13.5 psia ambient pressure for the bearing, these being the nominal inlet and discharge pressures for the compressor.

two-mass representation of the reciprocating assembly. One mass represented the piston mass. The second mass represented the combined masses of the solenoid plunger, the connecting-rod plate, and the effective mass of the resonant spring. These two masses were connected by the flexible connecting rod. Also included in the analysis model were (1) angular and radial stiffness of the hydrostatic piston bearing, (2) lateral and bending stiffness of the resonant spring, (3) lateral and bending stiffness of the connecting rod, and (4) geometry of the complete reciprocating assembly.

The deflection analysis was programmed for digital computer solution and the following design criterion was adopted: Under the influence of a 6-g lateral load, no point on the piston should deflect more than 75 percent of the radial clearance of the hydrostatic bearing. Based on the 75 percent deflection value, the maximum value of lateral acceleration load which could be supported was calculated as a function of connecting-rod diameter. The results of this calculation are shown in Figure 5 for the final design of (1) the resonant spring, (2) the hydrostatic piston bearing, and (3) the geometry of the reciprocating assembly. It is seen from Figure 5 that the lateral load capacity peaks at a connecting-rod diameter of about 0.15 inches. At this value, the reciprocating assembly should be able to tolerate a 12.9 g acceleration load without piston rubbing. However, lateral deflection of the solenoid plunger at 12.9 g would be about 0.014 inches. This amount of plunger deflection might cause problems due to unbalanced electromagnetic forces acting on the lateral pole-faces of the plunger.

It is clear from Figure 5 that connecting-rod diameter has a significant effect on lateral load capacity of the reciprocating assembly, and that there is only a narrow range of rod diameters which is satisfactory for lateral loads up to 6 g. Although the qualitative effect of rod diameter as indicated by Figure 5 is probably correct, there is some question as to the quantitative accuracy of the lateral deflection analysis. The most questionable aspect of the analysis is the modeling of the helical spring. Lacking experimental data, and desiring not to unnecessarily complicate the analysis, it was decided to model the spring as a close-coupled elastic tie having two elastic constants: a lateral and a bending stiffness. However, recently acquired data from another compressor development strongly suggests that it would be better to model the spring as a far-coupled elastic tie; that is, as a beam having four influence coefficients.

Another aspect which was neglected in the analysis was any unbalanced electromagnetic force acting on the lateral pole-faces of the solenoid plunger. This force would increase as a function of lateral plunger deflection (eccentricity) in the solenoid air gap. Since the electromagnetic force acts in a direction to further increase the plunger lateral deflection, it is basically a destabilizing force. To a first order of accuracy, the electromagnetic force could be modeled as a spring having a negative stiffness.

If test data obtained at MSFC should indicate that piston rubbing does occur at lateral load levels below 6 g, the design of the reciprocating assembly should be reviewed using a more accurate form of the lateral deflection analysis. The deflection analysis should be refined to include a beam (influence coefficient) representation of the helical spring. At the same time, the negative electromagnetic lateral force gradient of the solenoid could be incorporated into the analysis. These two refinements should make the lateral deflection analysis a more accurate design tool for optimizing the connecting-rod diameter and/or other parameters of the lateral deflection system.

Solenoid Design

Design of the electromagnetic drive solenoid is the same as utilized on contract NAS 8-11660 except for the following items:

1. Solenoid Coil

   The solenoid coil for the improved resonant compressor has the following characteristics:

   a) Windings

      107 turns #12 copper wire

      593 turns #13 copper wire

      Taps provided at 0, 107, and 700 turns

   b) Impregnation

      The coil is vacuum impregnated with Hysol C-29F epoxy, which has a thermal conductivity of 7.6×10^{-4} cal./sec.cm.^{2} °C/cm.

2. Solenoid mounting frame
   The solenoid lamination stack is permanently fastened to an Almag 35 cast aluminum frame. The assembly is shown in Figure 6.

Compressor Enclosure

The compressor enclosure consists of a top cover and a bottom base, both of which are Almag 35 aluminum castings. All gas-line fittings and electrical pass-throughs are located in the base casting. The compressor itself is mounted to the base (see Figure 4) and can be operated with or without the cover in place. The enclosure castings are shown in Figure 7. Figure 8 shows the assembled enclosure mounted on omnidirectional high-g isolators.

V. TEST DATA

Drive Solenoid

Figure 9 shows the measured force versus current characteristics of the drive solenoid as a function of plunger axial air gap. Figure 10 shows measured rms volts versus amps, again as a function of plunger axial gap. Figure 11 shows calculated inductance of the solenoid coil versus plunger axial air gap. The inductance calculations were based on the data of Figure 10.

The data shown in Figures 9, 10, and 11 are almost identical to similar data obtained for the drive solenoids built under contract NAS 8-11660 (see Figures 13, 14, and 15 of MTI Report 66TR57). This demonstrates that good reproducibility of performance is obtainable for the basic solenoid design.

Hydrostatic Piston Bearing

A test was performed to determine the hydrostatic bearing flow as a function of compressor discharge pressure. The test was performed under actual compressor operating conditions. A flow versus pressure-rise characteristic for the compressor was first obtained with the hydrostatic bearing operative. The feed line to the hydrostatic bearing was then disconnected and the supply fitting from compressor discharge was sealed. A flow versus pressure-rise characteristic for the compressor was again obtained under exactly the same operating conditions as used previously, except that the piston was now operating with only the single hydrodynamic bearing. By subtracting the initial flow curve from the second flow curve, a measurement of bearing flow versus compressor pressure rise was obtained.

Figure 12 shows the compressor flow versus pressure-rise data, with and without the hydrostatic bearing, and the resulting curve of hydrostatic bearing flow. It is seen that at a pressure rise of 15 psi, the measured bearing flow was 0.055 SCFM. Measured radial clearance of the bearing was very close to the nominal 0.0006 inch value. Hence, the measured bearing flow rate agrees very closely with the predicted flow rate given in the previous section of this report.

Also plotted on Figure 12 is the overall adiabatic compressor efficiency based on measured useful output flow and measured electrical input power to the series connection of drive solenoid and diode. At 15 psi pressure rise, the measured overall efficiency is 30 percent with the hydrostatic bearing and 32.6 percent without the hydrostatic bearing. The reduction in compressor efficiency due to the hydrostatic bearing flow is thus 8 percent, which again agrees very well with the anticipated reduction in efficiency.

Compressor Performance Data

Over 250 hours of compressor operation were obtained prior to shipment of the unit to MSFC. Over 90 percent of this time the compressor was operating at design-point conditions.

The endurance testing was conducted with the compressor sealed within the enclosure. About half of the test hours (roughly 125 hours) were accumulated with natural convection cooling of the enclosure. The remaining test hours were obtained with forced convection cooling of the enclosure (a small fan was used to blow room temperature air over the enclosure cover). Under natural convection cooling conditions, the measured outside surface temperature of the top of the enclosure cover was 129°F (for a compressor pressure rise of 15 psi). Under forced convection conditions, the measured surface temperature was 89°F.

Figure 13 shows measured flow versus pressure rise of the compressor as a function of input voltage under conditions of natural convection cooling. The data were taken after several hours of running at 15 psi pressure rise such that stable temperature conditions had been achieved. Design-point flow and pressure rise (15 psi and 0.53 SCFM) were slightly exceeded at an input of 90 v. rms.

Figure 14 shows measured flow versus pressure rise of the compressor as a function of drive voltage under conditions of forced convection cooling. Again, the data were taken after several hours of operation at 15 psi pressure rise to achieve stable temperature conditions. In this case, design-point flow and pressure rise were appreciably exceeded at an input of 90 v. rms. In fact, design-point conditions were almost achieved at an input of 87 v. rms.

Also shown on Figure 14 is a comparison of the flow versus pressure-rise characteristics at 87 v. rms for both the natural and forced convection cooling conditions. In the vicinity of design-point operation, Figures 13 and 14 indicate that better compressor performance is obtained if the enclosure is cooled. In a space (vacuum) environment, it will probably be necessary to bond a liquid-cooled cooling coil around the outside surface of the enclosure cover to remove the compressor losses. Such a cooling arrangement should be relatively easy to interface with the on-board IU cooling system and should be quite effective.

Measured electrical input power (including diode losses) at design-point pressure rise (15 psi), with natural and forced convection cooling, is summarized below.

Measured Electrical Input Power at 15 PSI Pressure Rise

Method of Cooling	Input Voltage rms	Flow SCFM	Input Power watts
Natural convection	87	0.497	72.8
Natural convection	90	0.544	81.0
Forced convection	87	0.512	72.0
Forced convection	90	0.585	81.6

Interpolating between the above values, it is found that input power for design flow (0.53 SCFM) should be about 78.5 watts with natural convection cooling and about 74 watts with forced convection cooling.

It should again be remembered that all of the compressor test data given in this report applies to the improved compressor design assembled with the out-of-tolerance (excessively stiff) resonant spring.

The compressor could be safely operated from wide-open to fully shut-off flow as long as input voltage was less than 85 v. rms. However, above 85 v. rms it was found that overstroking of the piston would occur (causing the piston to contact the valve plate) as the shut-off flow condition was approached. The pressure rise value at which piston contact begins is a function of input voltage. In Figures 13 and 14, each flow versus pressure-rise curve is plotted for a constant value of input voltage. The maximum pressure-rise data point on each curve represents the compressor operating condition just prior to the onset of piston contact. The data points show that onset of piston contact begins at progressively lower values of maximum pressure rise as input voltage level is increased.

Since piston contacts do not begin until pressure rise is considerably above the 15 psi design-point value, it appears that overstroking of the piston can easily be prevented in the present compressor configuration by simply assuring that pressure rise never exceeds 20 to 22 psi. A pressure relief valve which will by-pass flow from compressor discharge to compressor inlet, set to open at 20 to 22 psi, should provide positive protection against piston overstroking.

Use of a resonant spring having the correct value of stiffness may also alleviate the overstroking condition without the need for a pressure reflief valve. This possibility should be investigated by MSFC.

VI CONCLUSIONS AND RECOMMENDATIONS

Conclusions

An improved version of the Resonant Piston Compressor for Hermetically Sealed Platforms has been built and successfully tested for over 250 hours. Most of the test operation was at design-point flow and pressure rise.

The improvements to the compressor (relative to the compressor design previously developed under contract NAS 8-11660) are as follows:

1. Use of a helical resonant spring which results in a significant reduction in compressor envelop size;

2. Use of a hydrostatic piston bearing to support lateral acceleration loads up to 6 g;

3. Use of an Almag 35 casting for the solenoid mounting frame;

4. Use of Almag 35 castings for the compressor enclosure.

It is concluded that use of a helical resonant spring is technically feasible and results in a much more compact compressor design. The particular spring designed for this application should have an operating life well in excess of the required 10,000 hours.

Application of helical springs to resonant compressors requires very careful mechanical design because of potential problems with undesirable lateral, bending, and torsional mode resonances. In this particular development, the lowest resonant frequency of the coupled bending-lateral vibration mode was the most troublesome. Any future design modifications to the compressor should always be reviewed with respect to their effect on this resonant frequency. In the compressor configuration shipped to MSFC, the measured value of the lowest resonant frequency of the bending-lateral mode was 83.6 hz, very close to the predicted value of 82 hz. This frequency is only 39 percent above the 60 hz operating frequency of the compressor. Every effort should be made to assure that any future modifications to the compressor do not reduce the spread between these two frequencies.

Recommendations

1. The improved compressor, as shipped to MSFC, contained a resonant spring having an axial stiffness which was 27 percent higher than the intended design value. All tests performed at MTI were made with this spring.

   A replacement spring having the correct stiffness was subsequently procured and forwarded to MSFC. It is recommended that the compressor be tested with this spring to assess whether better performance can be obtained with the intended design value of spring stiffness.

2. Overstroking of the compressor piston (as assembled with the excessively stiff spring) will occur at high pressure-rise conditions when input voltage is greater than 85 volts rms. It is believed that the overstroking can be held to safe limits by using a pressure relief valve to by-pass flow from compressor discharge to compressor inlet. The relief valve should be set to open at a pressure rise of 20 to 22 psi. It is recommended that this technique be evaluated as a safe-guard against excessive overstroking at high pressure-rise conditions.

3. The ability of the compressor to operate at the specified condition of lateral acceleration could not be checked at MTI. It is recommended that MSFC perform the lateral g-load test as soon as possible to determine whether satisfactory operation will be obtained under 6-g lateral acceleration conditions.

   As mentioned in Section IV of this report, there is some question about the accuracy of the lateral deflection analysis which was used to optimize the design of the reciprocating assembly. Questions arise because of the manner in which the resonant spring was modeled, and because the lateral electromagnetic solenoid force was neglected. If test operation under 6-g lateral acceleration cannot be achieved, it is recommended that the design of the reciprocating assembly be reviewed using a more accurate form of the lateral deflection analysis. The refinements which should be made to the analysis are (1) a beam representation (influence coefficients) of the resonant spring, and (2) inclusion of the negative electromagnetic force gradient of the solenoid. These two refinements should make the lateral deflection analysis a more accurate design tool for optimizing the design of the reciprocating assembly for operation under lateral g-load conditions.

4. It has been determined by actual test that the compressor will maintain good performance when only one reed of the discharge valve is permitted to open. This means that three of the four discharge valve reeds could be eliminated. If this were done, an inlet valve could be used instead of the present inlet port arrangement. The inlet valve could be mounted on the same valve plate as the discharge valve. An inlet valve could thus be incorporated into the improved compressor design with only minor modifications to the design.

   Preliminary calculations indicate that the use of an inlet valve would reduce piston stroke to about one-half of the present value. This means that vibration amplitudes of the compressor enclosure, and force amplitudes transmitted through the compressor mounts, would be reduced by about one-half. Reduction of transmitted forces would also reduce vibration-excited noise from the compressor mounting panel. If these improvements are desirable, it is recommended that a modified version of the improved compressor be designed, built, and tested using an inlet valve instead of an inlet port.

5. The height of the compressor enclosure could be reduced by 1 to 2 inches, and compressor efficiency increased, by eliminating the tapered hydrodynamic bearing. Elimination of this bearing would permit the cylindrical extension of the piston, which presently extends beyond the piston face, to be cut-off. Besides the obvious reduction in compressor height which could thus be achieved, this would eliminate the internal leakage path (the largest leakage path) which presently exists between the ID of the piston and the OD of the cylinder head which projects into the piston. Elimination of this leakage path should significantly improve compressor efficiency.

APPENDIX A

SCOPE OF WORK FOR CONTRACT NAS 8-11660

CONTRACT NAS8-11660
MECHANICAL TECHNOLOGY INC.

EXHIBIT "A"

SCOPE OF WORK

PART I - STATEMENT OF WORK

SATURN V research and development - gas supply system for hermetically sealed platforms.

1. Design a gas supply system including compressor to the specifications outlined below.

2. Manufacture and deliver one complete prototype system.

3. Manufacture and deliver one compressor complete with fittings suitable for use as a spare compressor for the system.

4. System Operation:

a. Figure 1 is included to illustrate a possible component arrangement in order to clarify the type of system desired. It is not intended to fix the design. This figure will be used as a reference to briefly describe the system operation.

b. Gas of the desired specifications is to be supplied to the stabilized platform. The gas, after flowing through the gas bearing components, will be ejected into the void inside the platform cover; and the system loop is closed by returning this ejected gas to the input of the compressor.

c. If the pressure inside the platform cover drops below a specified limit, replenishment of gas from the gas reservoir into the platform cover is activated and is deactivated when a specified upper limit of pressure is reached.

5. Physical Specification (System):

Size - 28" square panel (approx. - no larger)
Weight - minimum to achieve other specifications
Mounting - optional
Ambient temperature operating range - 0°C + 60°C
Ambient pressure - 10^{-9} Torr (space vacuum)

Scope of Work (Cont'd)

6. Compressor Specifications:

   P_1 = input pressure range 12 to 15 psia
   Gas flow = 14 to 15 liters/min
   Output to system = capable of monitoring 15 psig = ½ at 0.50 scfm, as measured across platform

7. Gas Supply Specifications:

   Gas reservoir - cut in at 12 = ½ psia absolute pressure
   cut out at 15 = ½ psia absolute pressure

   A ½ cu. ft. volume titanium sphere shall be used for the gas reservoir. Sources for these spheres include:

   (a) Menasco Mfg. Co., 805 S. Fernando Blvd., Burbank, California

   (b) Airite Products, Div. of Electrada Corp., 3518 East Olympic Blvd., Los Angeles, California

   Output temperature of gas: 15°C ± 5°C
   Dew point: less than minus 54°C
   Dust particles - no greater than 8 microns
   Condensible hydrocarbon content - less than 1 part/million by weight

8. System Vibration Requirements:

   Linear acceleration - 10g
   Log sweep - 15 minutes
   0-30 cps at 0.2 inch double amplitude
   30-2,000 cps at 5g
   Shock - 50g for 11 milliseconds

9. Electrical Power:

   Voltage - either 26 volts, 400 cycle ac or 28 V dc
   Power - 150 watts maximum

10. Operating Life - 10,000 hours.

APPENDIX B

SCOPE OF WORK FOR CONTRACT NAS 8-20720

CONTRACT NAS8-20720
MECHANICAL TECHNOLOGY INC.

EXHIBIT "A"

SCOPE OF WORK

PART I - STATEMENT OF WORK

A. The Contractor shall design, develop, fabricate, assembly, test, and deliver one resonant piston compressor. The compressor design shall generally conform to the compressor developed and delivered in accordance with MTI Drawing 141-F-02 dated November 17, 1964.

PHASE I

1. Redesign the resonant piston compressor as follows: Integrate the helical resonant spring design as shown in MTI drawing SK-E-2265 dated March 29, 1966.

2. Perform the necessary redesign to improve the side loading characteristics of the compressor piston. A limited effort should be exerted toward improving the present hydrodynamic gas bearing principle used in centering the piston. If design studies so indicate, an alternate approach shall be undertaken which uses the hydrostatic gas bearing principle for piston centering.

3. Improve the solenoid to cylinder mechanical connection scheme by using a casting as a mounting structure. Aluminum alloy, "Almag 35" is suggested as a material to consider.

4. Design a casting for the compressor enclosure. Aluminum alloy, Almag 35, is suggested as a material to consider. The enclosure design shall reflect the reduction in volume gained by the relocation of the heat exchangers and the integration of the helical spring into the compressor. The enclosure design should also incorporate three mounting surfaces, located in a plane perpendicular to the stroke of the piston and passing through the center of gravity of the compressor, compatible with the vibration isolator shown on outline drawing GC326520 dated October 31, 1966, which is incorporated herein by reference.

5. Outline drawing GC625026 dated October 1966, of the compressor is incorporated herein by reference, which is intended to be used as a guide by the contractor. This drawing is included to illustrate a possible design scheme in order to clarify the type design desired. It is not intended to fix the design.

6. Sumbit to MSFC technical representative for approval, a design layout of the resonant piston compressor incorporating a helical resonant spring, cast metal enclosure, and other design improvements as specified in this work statement.

Scope of Work (Cont'd)

PHASE II

1. After receipt of approval of Phase I, the contractor shall fabricate, assemble, test and deliver one (1) resonant piston compressor in accordance with the MSFC approved design layout. Testing shall include flow vs pressure rise, input power characteristics, and 150 hours continuous operation bench test.

B. The following specifications are applicable to this procurement:

1. Input pressure range: 12 to 15 psia

2. Gas flow: 15 liters per minute at 15 psig.

3. Linear Acceleration: 10g along the flight direction and 6g along the other two mutually perpendicular axes.

4. Resonant spring safety factor: The factor of safety (endurance limit) for resonant spring shall be no less than 1.6.

5. Operating life goal - 10,000 hours.

C. The following documentation is required:

The contractor shall deliver one (1) reproducible of each drawing generated in the fabrication and production of the item called for under this contract and shall deliver on or before the expiration date of the contract.

PART II - REPORTS OF WORK (SEPTEMBER 1962)

A. Monthly Progress Reports: The Contractor shall submit separate monthly progress reports of all work accomplished during each month of contract performance. Reports shall be narrative form, and brief and informal in content. Monthly reports shall include:

1. A brief description of all work performed during the current reporting period, including test procedures, test results, pertinent diagrams and schematics.

2. An indication of any current problems which may impede performance and proposed corrective action.

3. A discussion of the work to be performed during the next monthly reporting period.

4. Monthly reports shall be submitted in ten (10) copies and distributed by the Contractor as directed by administrative letter from the Contracting Officer.

APPENDIX C

PARTS LIST FOR IMPROVED COMPRESSOR

MECHANICAL TECHNOLOGY INCORPORATED

PL 192E01 sheet 1 of 3

Title: ASSEMBLY RESONANT PISTON COMPRESSOR

Quantity Per Group 6	5	4	3	2	1	Item No.	Part No.	Description
					1	1	192D03P1	Container Cover
					1	2	192D04G1	Container Base
					1	3	192B23G1	Reciprocating Assembly
					1	4	192C13G1	Cylinder Assembly
					1	5	192C35P1	Spring
					1	6	192D08G1	Solenoid Assembly
					1	7	192B20P3	Spacer
					1	8	192B21P3	Spacer
					AR	9	Ecobond #104	Epoxy-Emerson, Cummings or Equiv.
					1	10	192B19G1	Valve Assembly
					AR	11	MS20995C20	Lockwire
					3	12	192B25P1	Support Plate
					4	13	192B24P1	Spacer Post
					2	14	Cat.#6F5BX-SS	St. Thd. Conn., Parker or Equiv.
					1	15	Cat.#6C5BX-SS	Elblow, Parker or Equiv.
					1	16	Cat.#6S5BX-SS	Tee, Parker or Equiv.
					2	17		Tube, 3/8 x .035 Wall x 8.0 lg. Stn. Steel
					4	18	Cat. #3-6	"O" Ring, Parker or Equiv. (Buna N)
					6	19		Screw, 1/4 - 28 x 3/4 lg. Hex Socket Head
								St'l - Cad. Pl.
					1	20		Tube, 1/4 x .028 wall x 7.0 lg., Stn. Stl.
					1	21	Cat.#KPT02H14-12P	Receptacle, Cannon or Equiv.
					1	22		Gasket for KPT02H14 Cannon Receptacle
					1	23	Cat.# 2-167	"O" Ring, Parker or Equiv. (Buna N)
					1	24	Cat.#P/N53-587	Connector, Microdot or Equiv.
					1	25	Cat.# 2-22	"O" Ring, Parker or Equiv. (Buna N)
					4	26		Screw, #6-32 x 3/8 lg. Hex Socket Head
								Steel-Cad. Plate, Drilled Hd.

Rev.	Change	Date/By
1	Added Sh. 3 Item 5 Desc. Was 192C06	5/1/68 F.E.B.
2	Added Item 47 (Sh. 3)	H.J. 5/8/68

Approved H. Jones 8/8/67
Checked R.S. 8-8-67
Drawn RV Newell 2 Aug '67

Rev. No. 2

PL 192E01 sheet 1 of 3

MECHANICAL TECHNOLOGY INCORPORATED

PL192E01
sheet 2 of 3

Title
ASSEMBLY
RESONANT PISTON COMPRESSOR

Quantity Per Group

6	5	4	3	2	1	Item No.	Part No.	Description
					12	27		Screw, #10-32 x 1/2 Hex Socket Head Steel-Cad
								Plate, Drilled Head
					6	28		Screw, 1/4-28 x 1.00 lg. Hex Socket Head
								Steel Cad Plate
					4	29		Screw, 1/4-28 x 2.25 lg. Hex Socket Head
								Steel - Cad Plate
					4	30		Washer, for 1/4 Screw-Steel Cad Plate
					6	31		Screw, 1/4-28 x 1.25 lg. Hex Socket Head
								Steel - Cad Plate
					6	32	Cat.#52NTE-048	Locknut, Elastic Stop Nut
					6	33		Washer, for 1/4 Screw - Stl. Cad. Plate
					12	34		Washer, for #10 Screw- Stl. Cad. Plate
					6	35		Screw, 1/4-28 x 3/4 lg. Hex. Socket Head
								Steel Cad. Plate, Drilled Head
					1	36	192B31P1	Cap. Probe (.010 Range)
					2	37	192B29P1	Cap. Probe (.003 Range)
					3	38	192B30G1	Cable Assembly
					3	39	192B30G2	Cable Assembly
					1	40	Cat.#Electro 3055-A	Magnetic Pickup, Electro Products Labs, Inc.
					1	41	192B26P1	Retainer
					X	42	192B27P1	Cover
					1	43	141B15P1	Diaphragm
					4	44		Screw #4-40 x 1/4 lg. Hex Socket Hd.
								Steel Cad. Plate Dr. Hd.
					3	45		Screw Hex. Head #10-32 x 1/2 lg. Steel Cad
								Plate Drilled Head
					3	46		Screw Hex. Head #10-32 x 3/8 lg. Steel Cad
								Plate Drilled Head

Approved H. Jones 8/8/67
Checked R.S. 8-8-67
Drawn RV Newell 2 AUG '67

Rev. No. 2

PL192E01
sheet 2 of 3

MTI-72 (11-67)

MECHANICAL TECHNOLOGY INCORPORATED

PL192E01
sheet 3 of 3

Title
ASSEMBLY
RESONANT PISTON COMPRESSOR

Quantity Per Group

6	5	4	3	2	1	Item No.	Part No.	Description
					1	47	192B37P1	Shim

Approved H. Jones 8 MAY 68
Checked R.L. 5-8-68
Drawn J. Bartnecki 8 MAY '68

Rev. No. 2

PL192E03
sheet 3 of 3

MTI-72 (11-67)

FIGURES

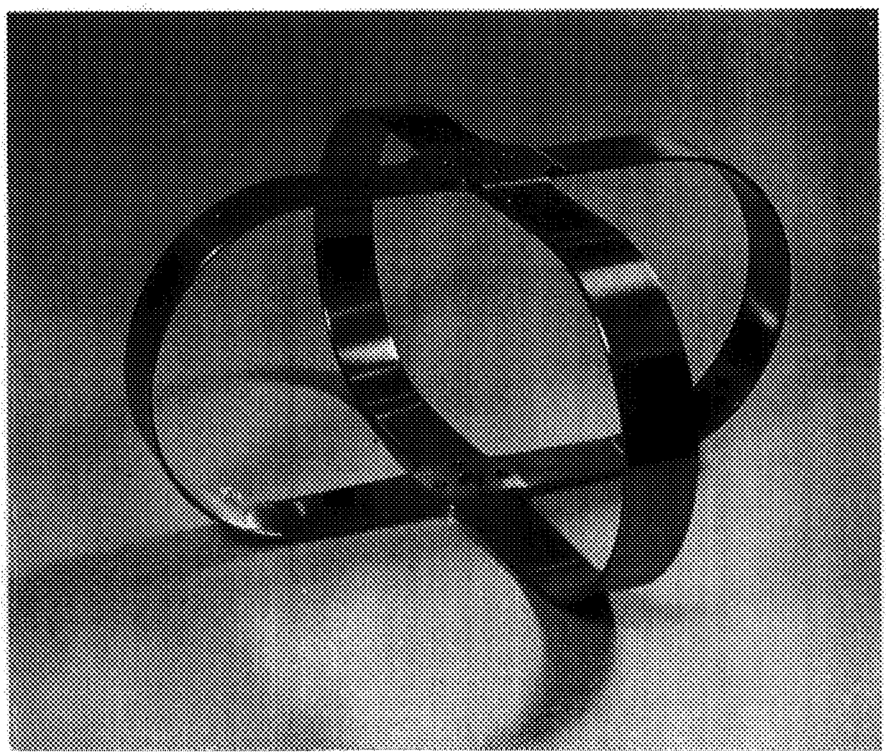

Fig. 1 Resonant Spring Configuration Used in the Two Resonant Piston Compressors Developed Under Contract NAS 8-11660.

MTI-6642

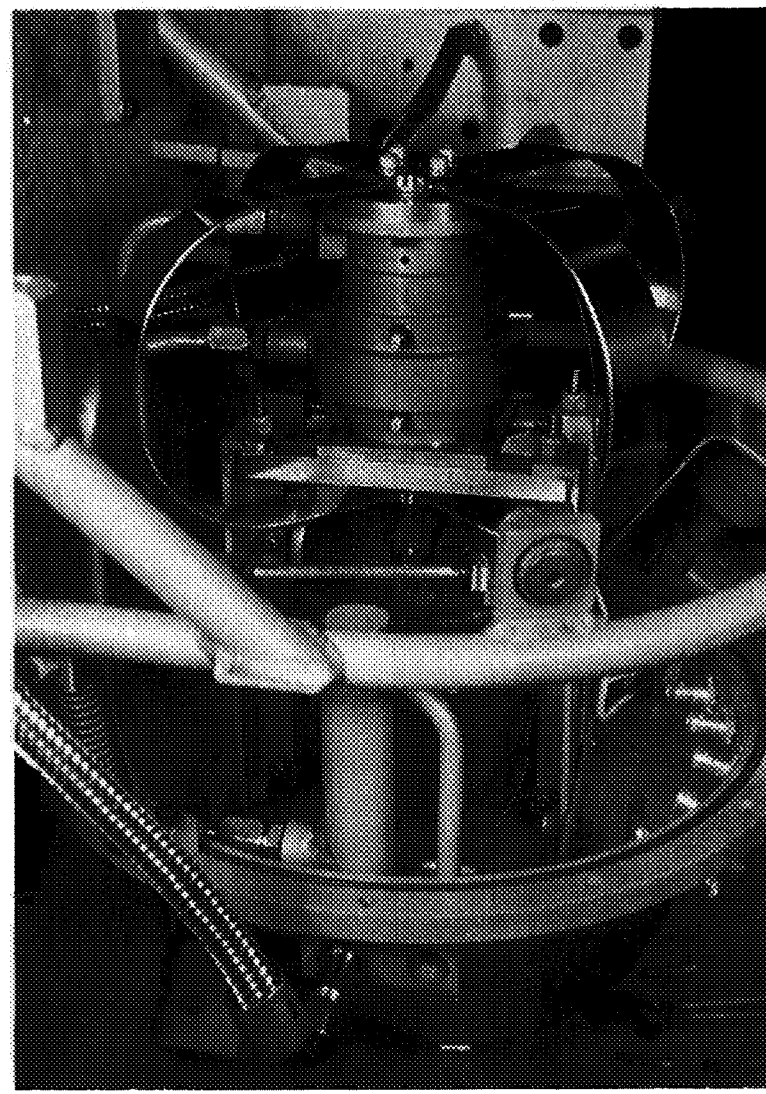

Fig. 2 View Showing Assembled Arrangement of the Resonant Spring in the Compressors of Contract NAS 8-11660.

MTI-6643

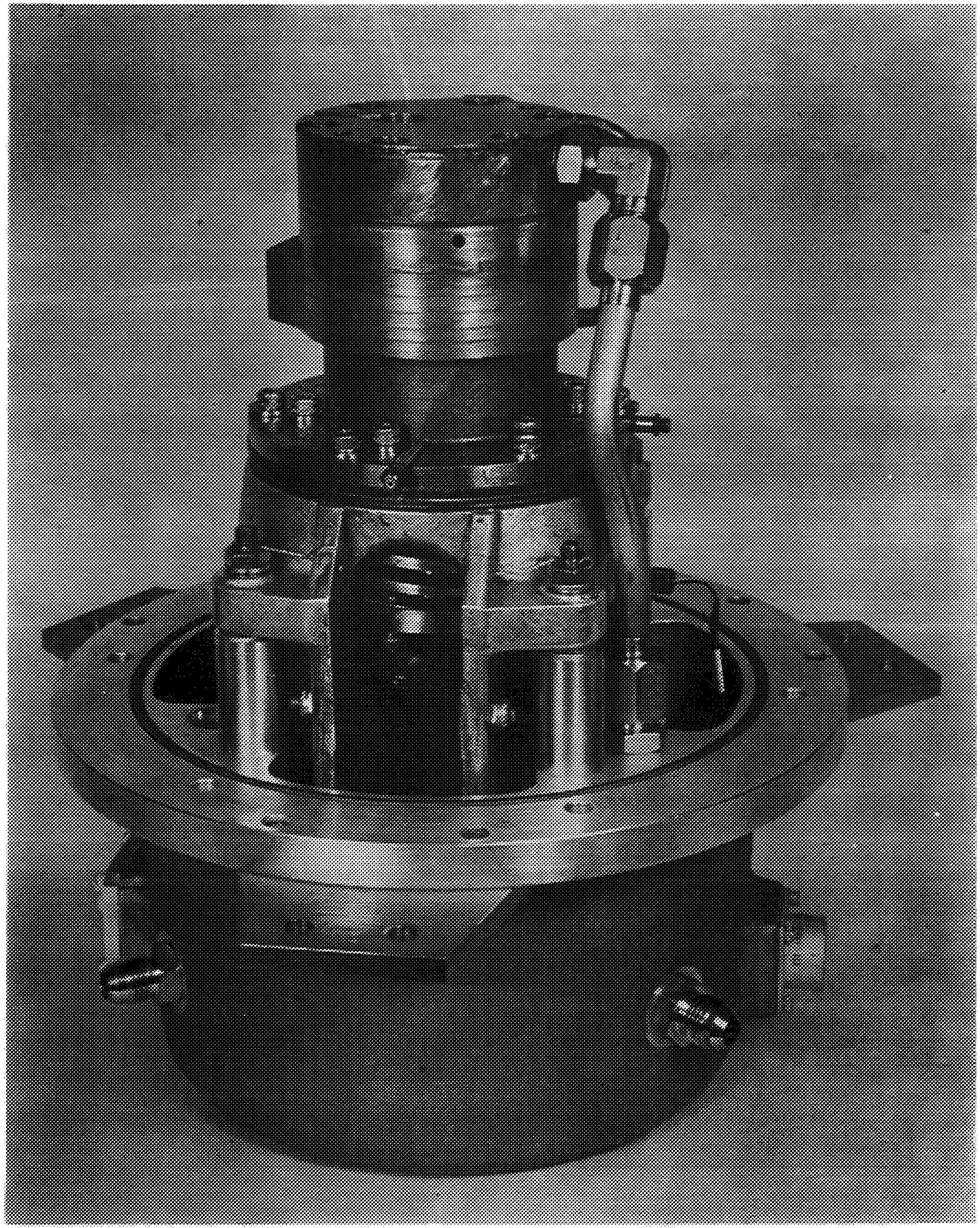

Fig. 4 View Showing Assembled Arrangement of the Improved Resonant Spring in the Improved Compressor of Contract NAS 8-20720.

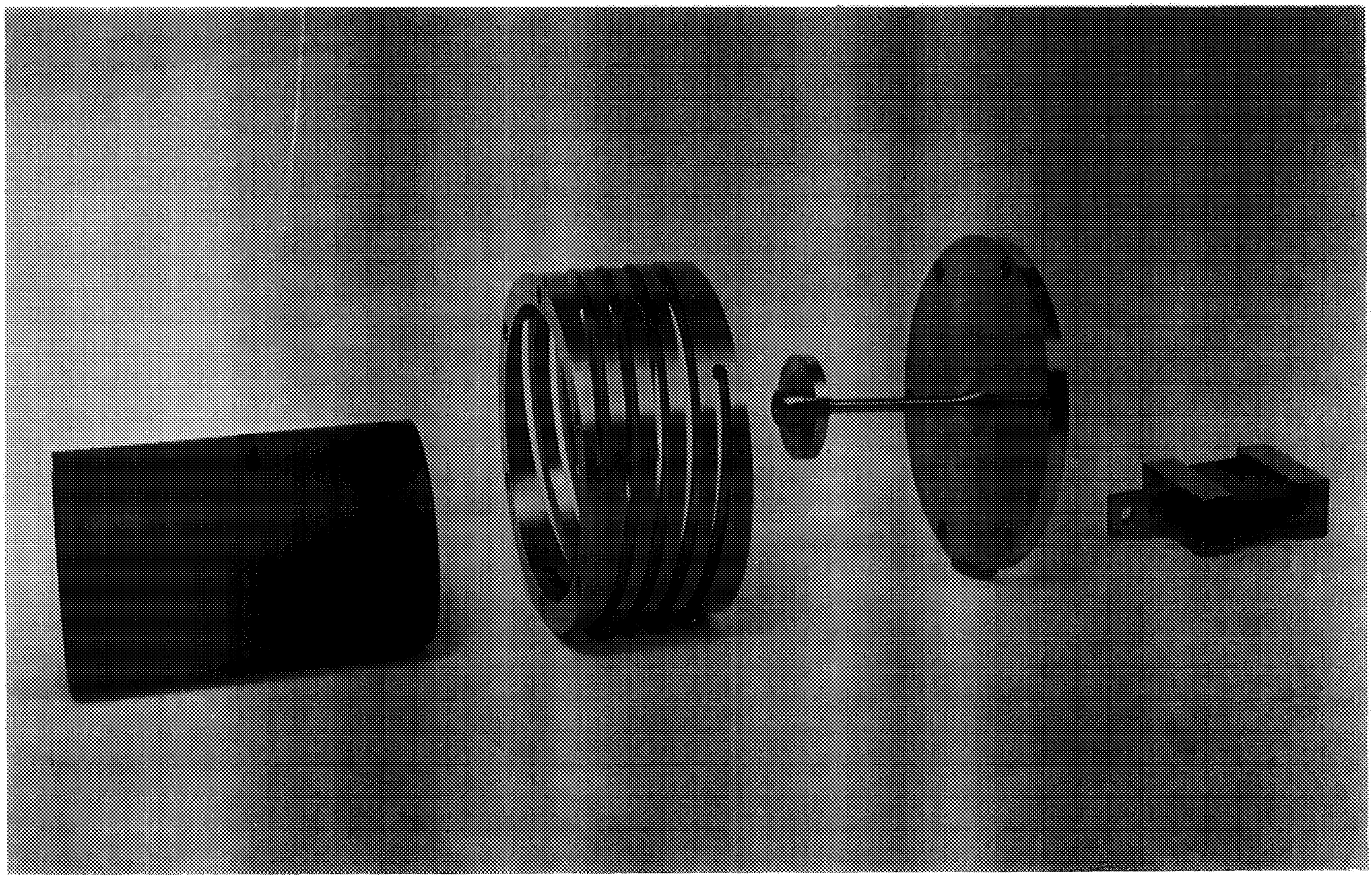

Fig. 3 View of the Reciprocating Parts for the Improved Resonant Piston Compressor Developed Under Contract NAS 8-20720.

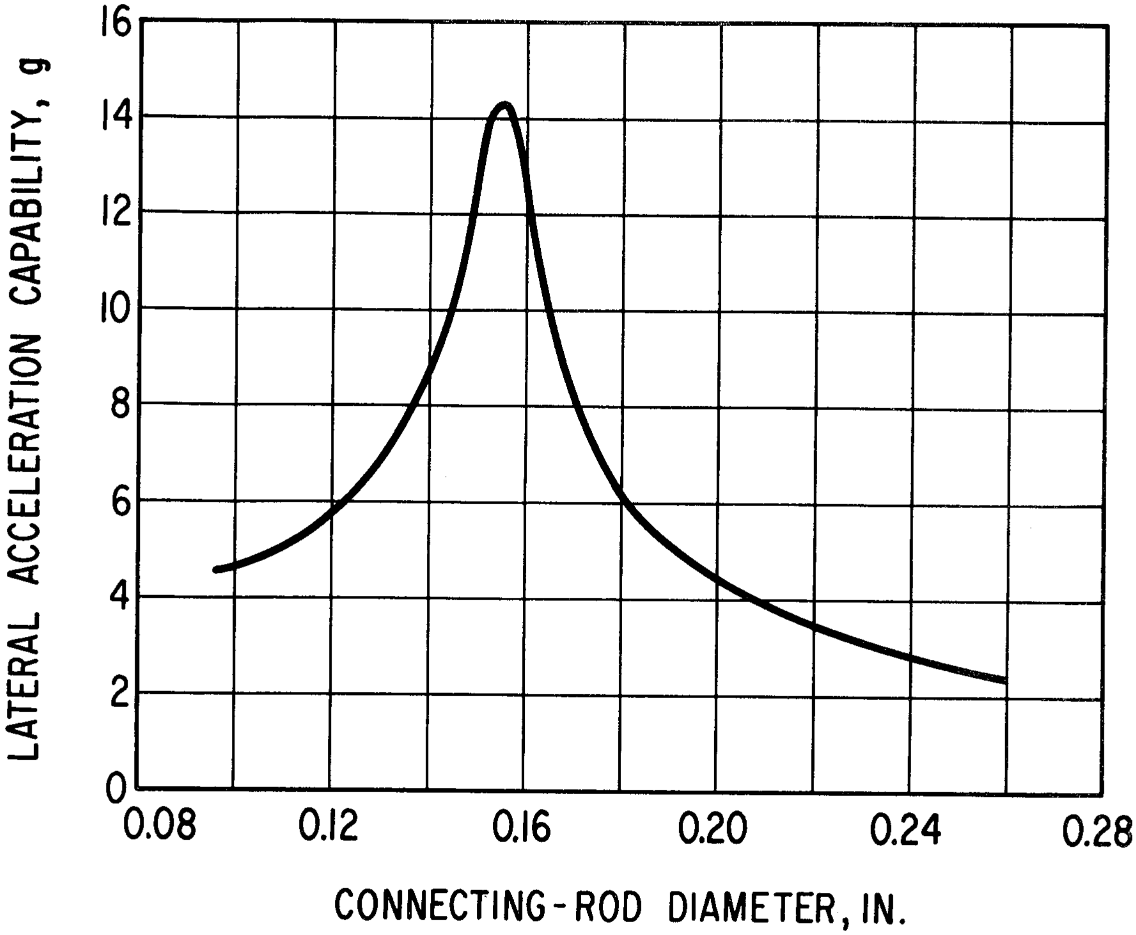

Fig. 5 Calculated Lateral Load Capacity of the Reciprocating Piston Assembly as a Function of Connecting-Rod Diameter.

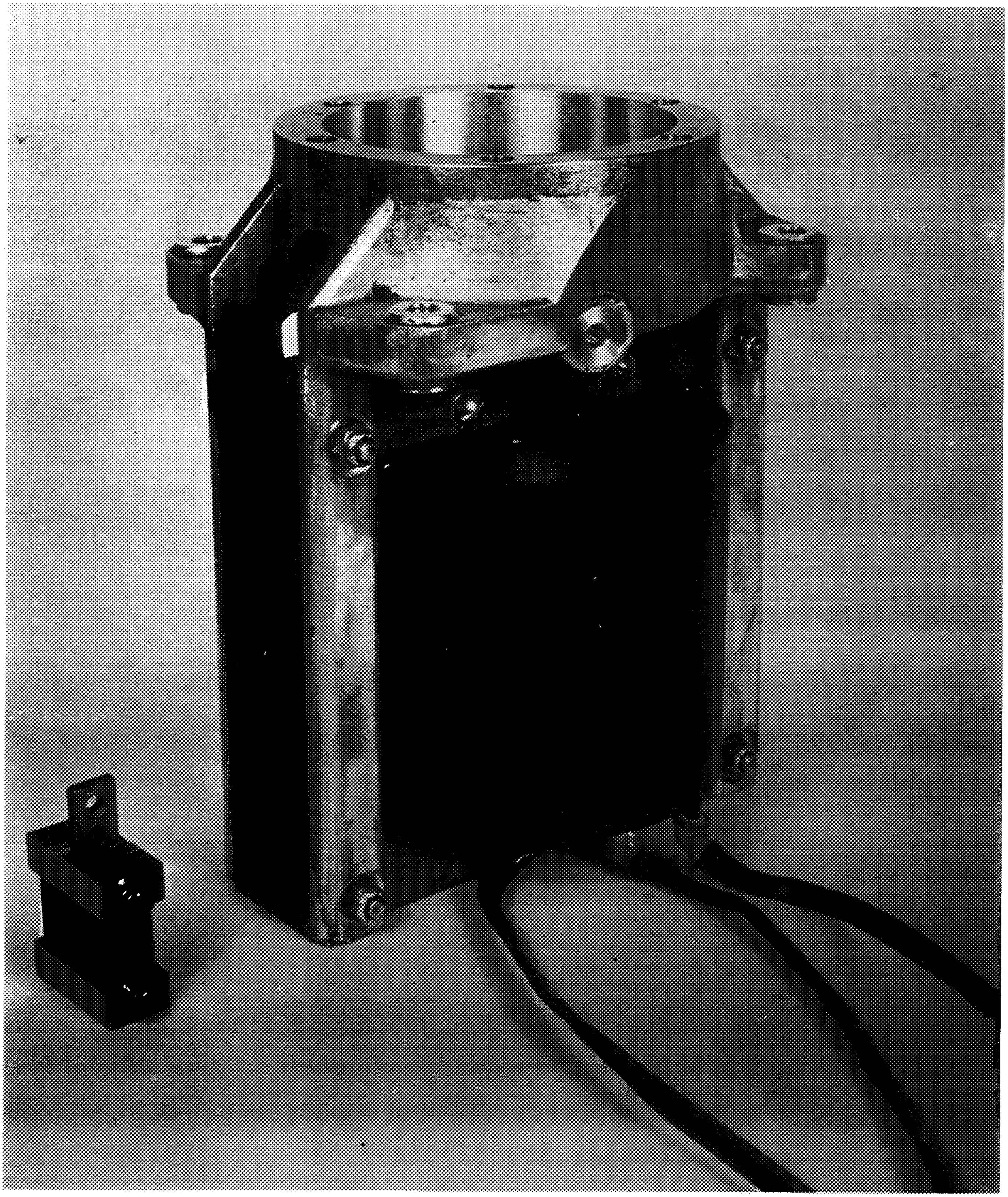

Fig. 6 Assembly of Drive Solenoid and Almag 35
Cast Aluminum Frame.

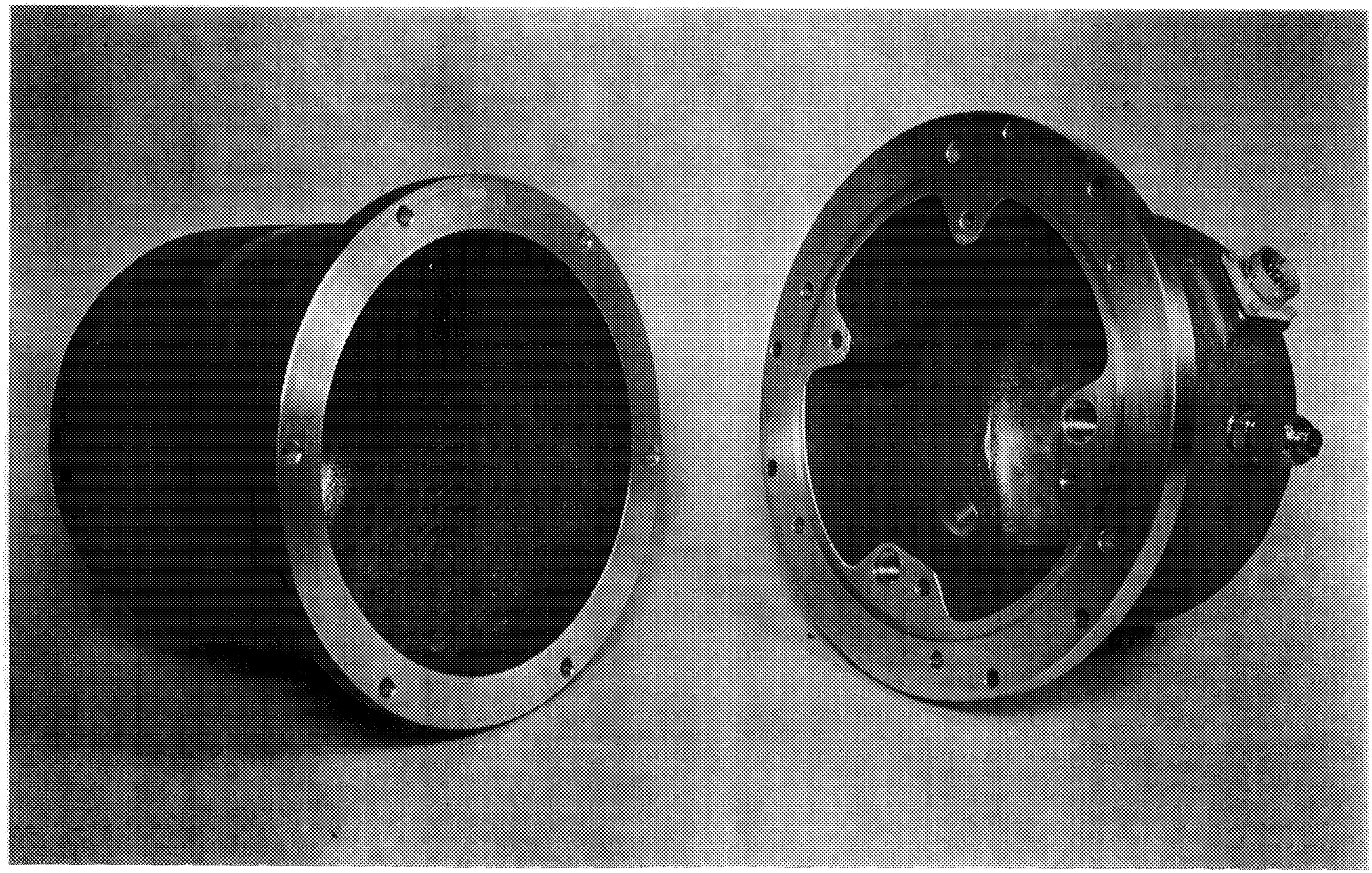

Fig. 7 Almag 35 Enclosure Castings.

MTI-6597

Fig. 8 Complete Compressor Assembly Mounted on Omnidirectional High-G Isolators.

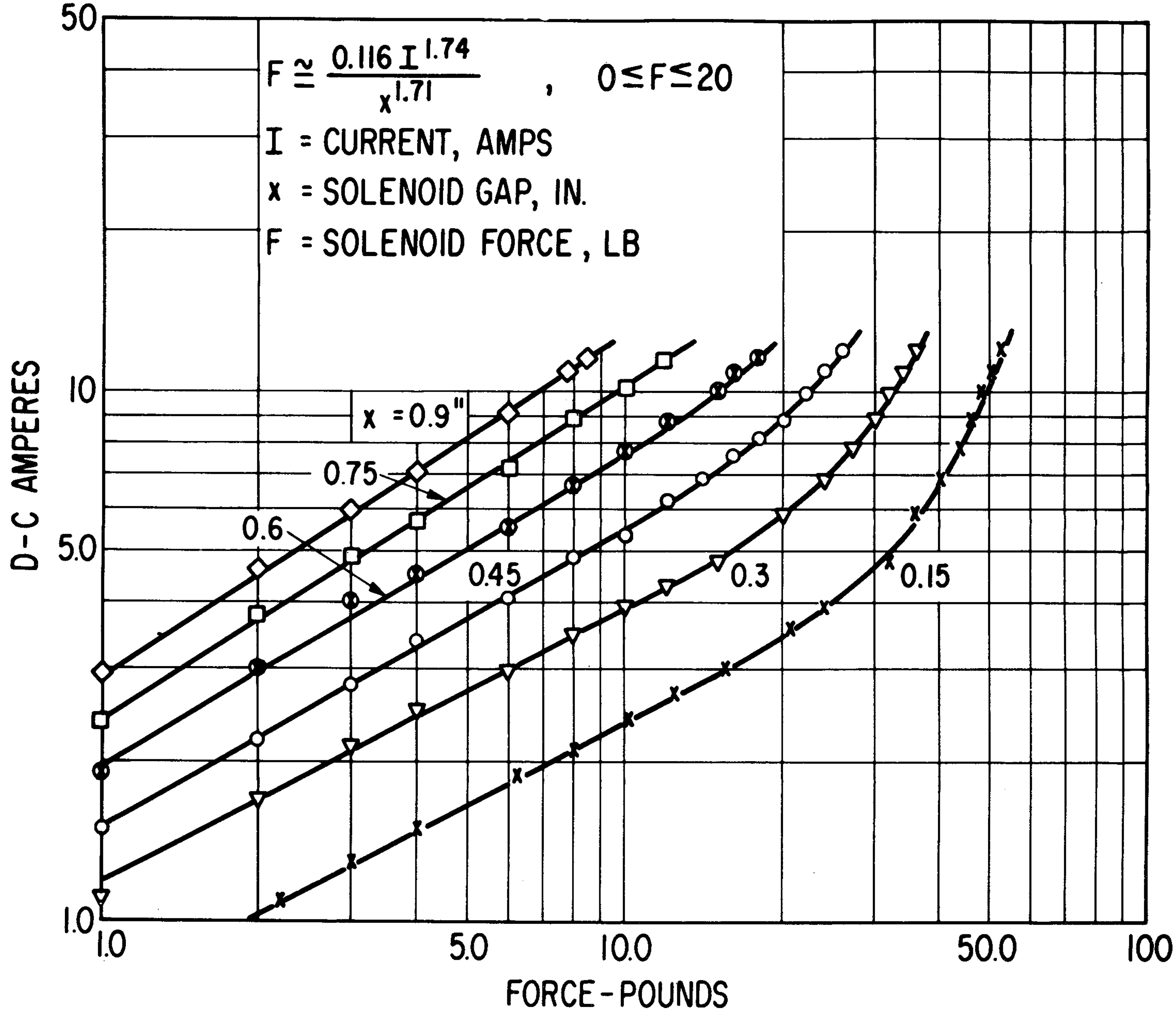

Fig. 9 Measured Force Versus Current Characteristics for the Drive Solenoid Built Under Contract NAS 8-20720.

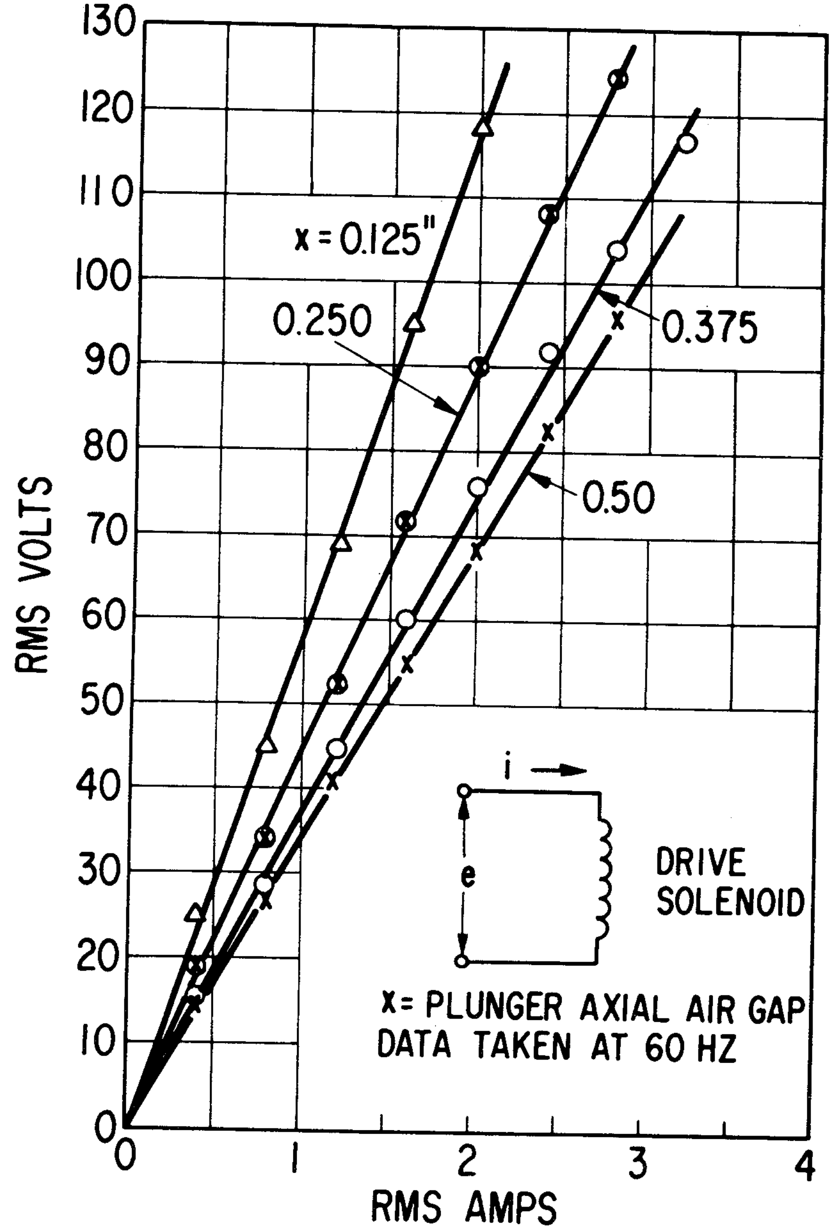

Fig. 10 Measured 60 Hz RMS Current Versus Voltage for the Drive Solenoid Built Under Contract NAS 8-20720

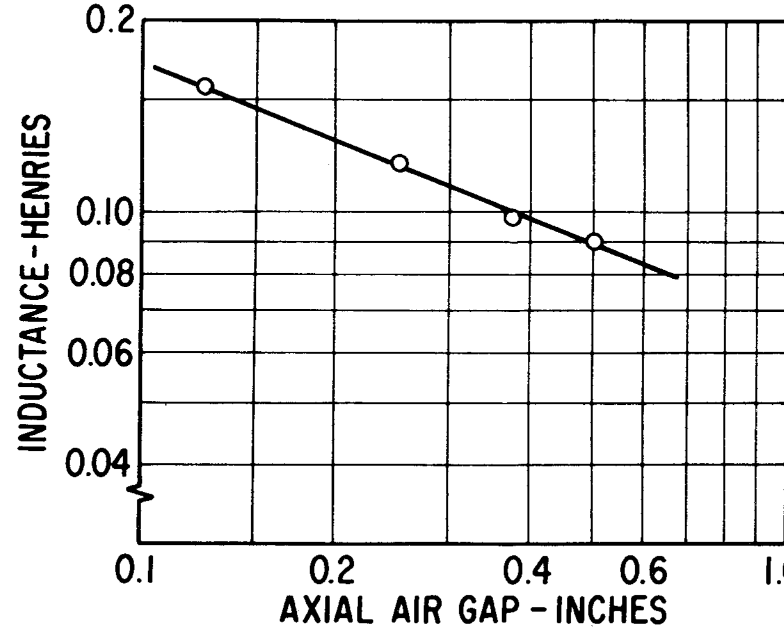

Fig. 11 Calculated Inductance Versus Plunger Axial Air Gap for the Drive Solenoid Built Under Contract NAS 8-20720.

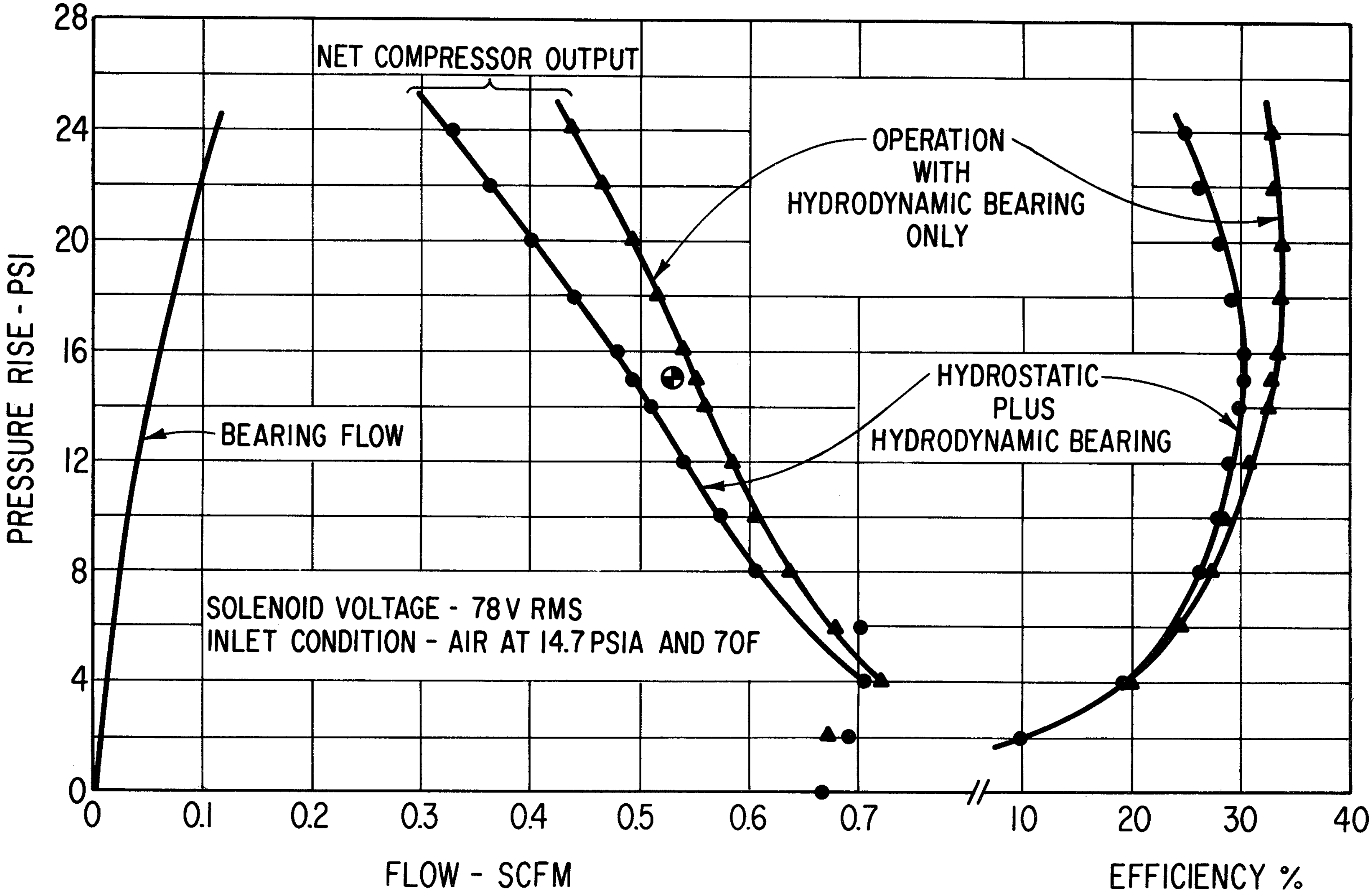

Fig. 12 Measured Flow and Efficiency Versus Pressure-Rise Data to Show the Effect of the Hydrostatic Bearing Flow on Compressor Performance.

MTI-6594

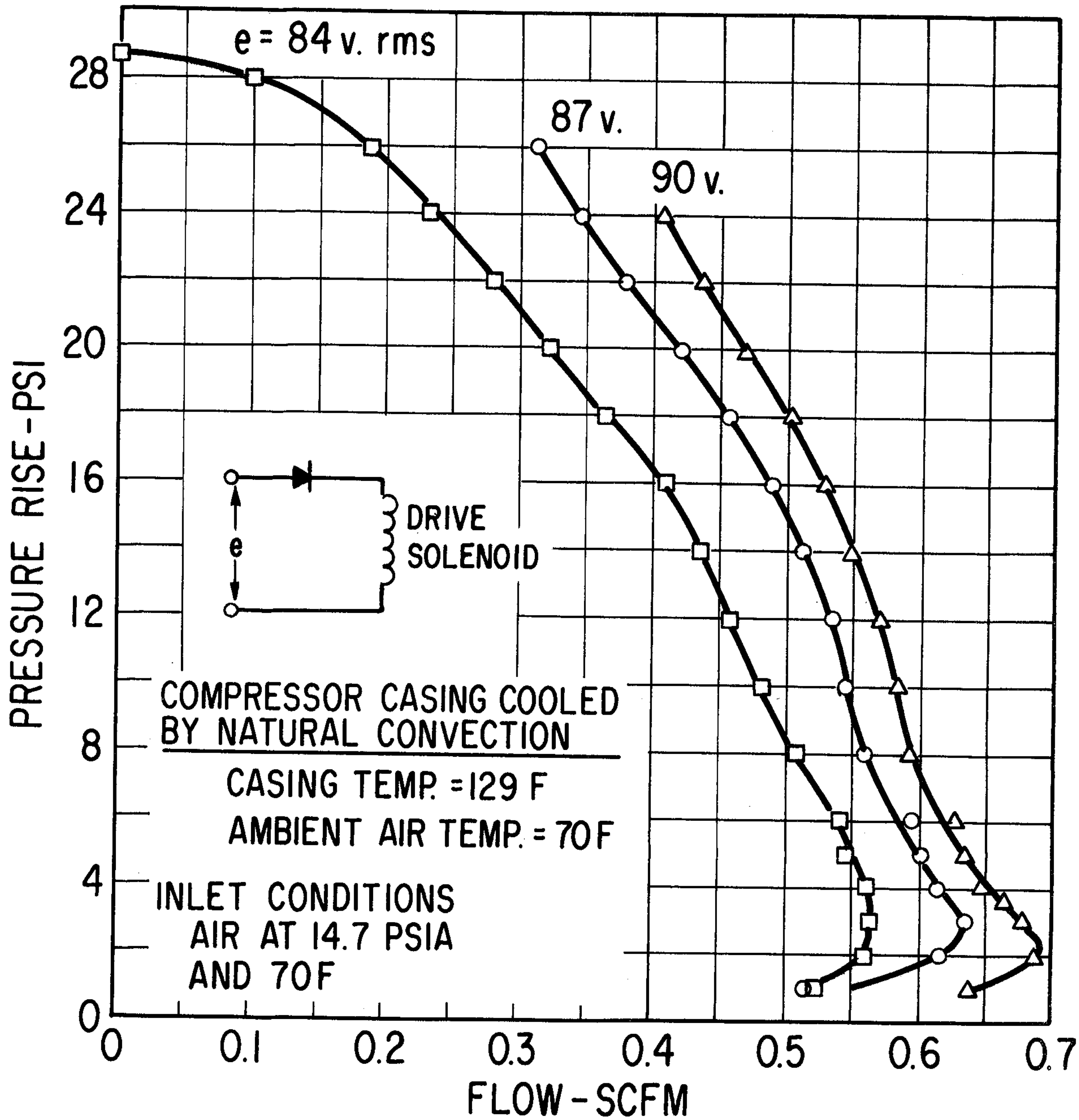

Fig. 13 Measured Flow Versus Pressure Rise for the Improved Compressor as a Function of Input Voltage; Compressor Enclosure Cooled by Natural Convection

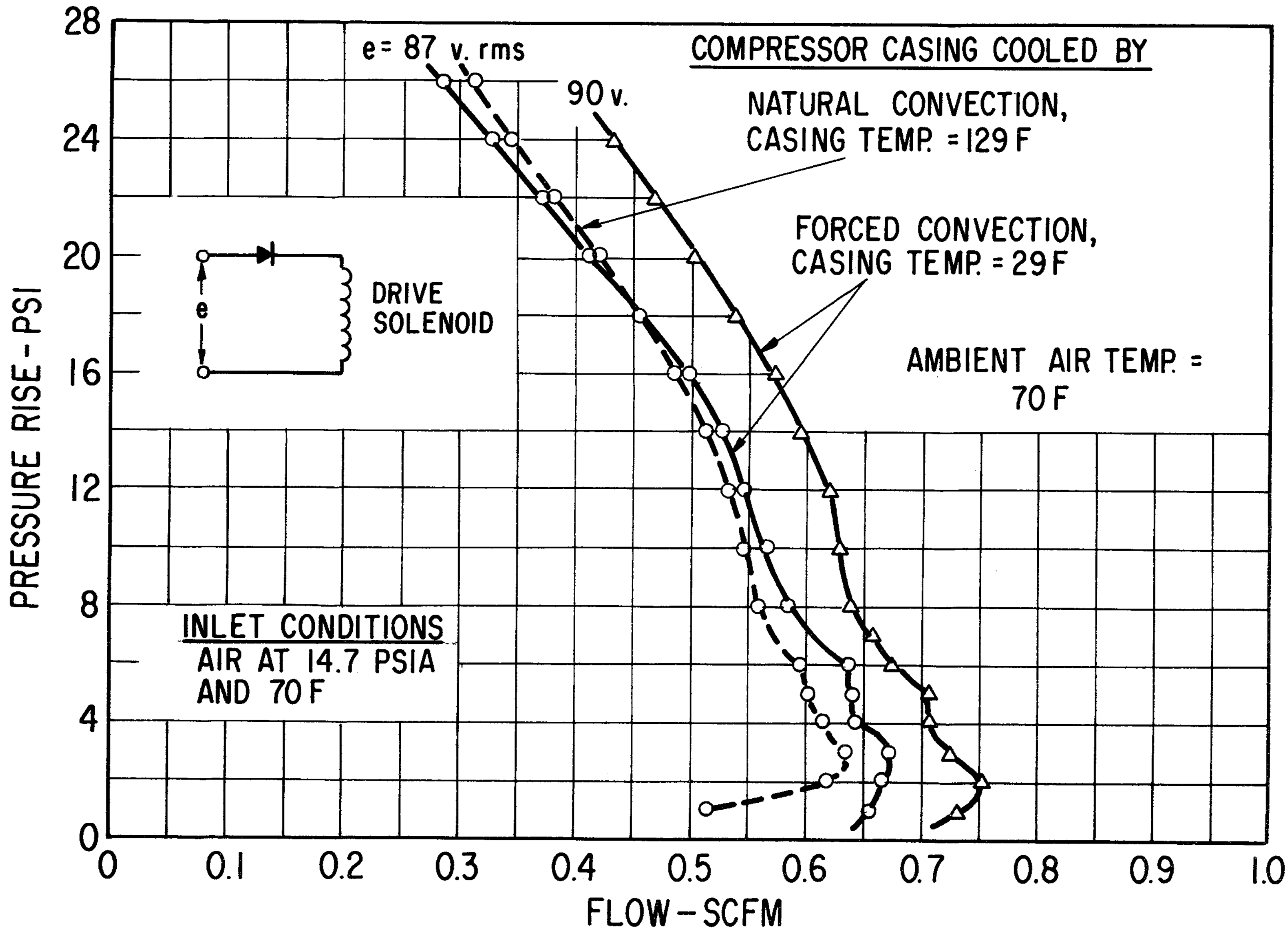

Fig. 14 Measured Flow Versus Pressure Rise for the Improved Compressor as a Function of Input Voltage; Comparison of Natural and Forced Convection Cooling of the Compressor Enclosure.

MTI-6592