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DEVELOPMENT AND TESTING OF A LOW TEMPERATURE  
HYDRAULIC SYSTEM (PRESENTATION BROCHURE)

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TITLE: DEVELOPMENT AND TESTING OF A LOW-TEMPERATURE  
HYDRAULIC SYSTEM (PRESENTATION BROCHURE)

MODEL \_\_\_\_\_ CONTRACT NO. NAS8-11722

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ABSTRACT

This document contains the brochure distributed with the final presentation of NAS8-11722 contract results at NASA Marshall Space Flight Center. Other documentation associated with this contract are:

1. D2-90795-1 Developing and Testing of a Low Temperature Hydraulic System - Actuation Applications Trade Study.
2. D2-90795-2 Developing and Testing of a Low Temperature Hydraulic System - Hydraulic System Analysis and Test Program.
3. D2-84075-1 Developing Hydraulic Technology for Space Applications

In the trade study eighty candidate space actuation tasks are considered from which 10 are selected as having the most value. From these 10, one task associated with each of four vehicle types is investigated in detail. The evaluation considers hydraulic, pneumatic, and electromechanical actuation systems to satisfy the requirements of each task. Provisions for power and thermal conditioning for each system are considered. The results of the trade study are in the form of a numerical rating summary showing the comparison of systems, for the same task, on the basis of weight, reliability, cost, availability, and performance margin.

Development preceeding fabrication of a hydraulic system for operation in a space environment of -240 to 275°F and a hard vacuum of  $10^{-6}$  torr included the evaluation of advanced fluids and seal compounds. Testing of the fabricated system showed how reducing fluid temperature affects the operation of the system and the minimum operating temperature for the test fluid selected. Results of operation in a simulated space flight show the system's capability for intermittent use associated with many space actuation duty cycles. The effects of high temperature on system operation are reported, as are the effects of 100 hours of system thermal cycling at temperatures from -240 to +275°F. Environmental effects on seals and system operation following long-term (5-month) hard-vacuum storage of the linear actuator are reported. A thermal analysis is also presented that describes, using parametric data, the protection required for various hydraulic applications in space.

KEY WORDS

Actuation  
Compatibility  
Design  
Development Testing  
Elastomeric Seals  
Hard Vacuum  
Hydraulic System  
Space Application  
System Response  
Temperature  
Thermal Cycling  
Thermal Protection  
Trade Study  
Viscosity  
Wear

- ACTUATION APPLICATIONS TRADE STUDY —
- THERMAL CONDITIONING ANALYSIS —
- HYDRAULIC SYSTEM ANALYSIS AND TEST PROGRAM —

— D 2-90795 VOL.I  
—  
— D 2-90795 VOL.II

## SPACE-VEHICLE ACTUATION TASKS

Man's exploration of space will require additional Earth-orbiting vehicles with and without re-entry capability followed by lunar surface vehicles with and without roving capability. Actuation tasks must be performed during operation of each of these four types of vehicles. The general categories of actuation tasks to be accomplished with each vehicle group are shown on the accompanying figure. Some specific tasks within each general category are:

- 1) Earth re-entry tasks — aerodynamic-surface control, wing-sweep opening, landing-mechanism actuation, and thrust-vector control;
- 2) Erection and retraction of collapsible mechanisms — radar antenna, solar collectors, space radiators, and solar cells;
- 3) Orientation of equipment — solar panels, solar collectors, space radiators, target seekers, photographic apparatus, television cameras, tracking devices, telescopes, and antennas;
- 4) Ejection and separation — stage separations and ejection of expendables before re-entry;
- 5) Life support — artificial gravity and exercise machines;
- 6) TVC in space — gimbaling of maneuver engines and braking engines;
- 7) Landing-mechanism deployment — landing mechanisms and shock absorbers;
- 8) Surface modification — bulldozing, grading, and shoveling;
- 9) Material handling — removal, placement, and assembly;
- 10) Transportation — wheel power.

## SPACE-VEHICLE ACTUATION TASKS

| <u>Space-Vehicle Group</u>                 | <u>Actuation Task Category</u>                       | <u>Duration</u>                 | <u>Type of Actuator</u>                              |
|--------------------------------------------|------------------------------------------------------|---------------------------------|------------------------------------------------------|
| 1) Orbiting with Earth Atmosphere Re-entry | 1) Earth re-entry tasks                              | Single to continuous for 1 hour | Hydraulic, pneumatic, electro-mechanical             |
|                                            | 2) Erection and retraction of collapsible mechanisms | Varied                          | Hydraulic, pneumatic, electro-mechanical, mechanical |
|                                            | 3) Orientation of equipment                          | Intermittent                    | Hydraulic, electro-mechanical                        |
|                                            | 4) Ejection and separation                           | Single to short                 | Pneumatic, mechanical                                |
| 2) Orbiting Without Re-entry               | 5) Life support                                      | Single to long, repeated        | Hydraulic, pneumatic, electro-mechanical, mechanical |
|                                            | 6) Thrust vector control in space                    | Intermittent                    | Hydraulic, electro-mechanical                        |
| 3) Space Travel and Landing                | 7) Landing-mechanism deployment                      | Single                          | Hydraulic, pneumatic, mechanical                     |
|                                            | 8) Surface modification                              | Long, repeated                  | Hydraulic, electromechanical                         |
| 4) Lunar Surface Operation                 | 9) Material handling                                 | Long, repeated                  | Hydraulic, electromechanical                         |
|                                            | 10) Transportation                                   | Long, repeated                  | Hydraulic, electromechanical                         |

## ACTUATION REQUIREMENTS

The list of tasks considered as space-actuation-system candidates was reduced to 10 tasks judged to have higher actuation-power requirements than other candidates. Requirements developed for each of these 10 tasks are shown on the accompanying figure. Descriptions of the tasks selected follow.

- 1) The eleven-surface actuation on a 12-man shuttle version of the M-2 or HL-10 vehicle configurations provides pitch and roll control during atmospheric descent and landing of the vehicle.
- 2) TVC operation of an RL-10 space-propulsion-stage engine assists midcourse velocity corrections during translunar coast and control during orbit injection and ejection.
- 3) A centrifuge aboard a large space station supplies various levels of centrifugal force to relieve the physiological effects of zero g and provides man's re-entry conditioning.
- 4) A large hangar door provides docking access to a large space station by re-entry capsules used for crew rotation.
- 5) Solar panels, recommended to satisfy a major portion of space-station secondary power, require erection from a deployed position and continuous orientation.
- 6) The landing mechanism of a lunar vehicle must engage the lunar surface, absorb the kinetic energy of the vehicle at landing, and support the vehicle.
- 7) The lunar-landing-vehicle thrust-vector control provides vehicle attitude control during braking into equiperiod lunar orbit, descent braking, hover, and landing.
- 8) The material handlers provide hand, hook, wrist, elbow, upper arm, and shoulder motions of manipulators for assembling and moving equipment.
- 9) A surface-modification bucket-loader attachment to a roving vehicle performs excavating, embankment, and clearing of lunar dust primarily to cushion lunar shelters against meteoroid and radiation damage.
- 10) A typical lunar vehicle requires power at six wheels for transportation on the lunar surface.

| Vehicle Type                                  | Actuation Type                                                | Maximum Movement                  | Maximum Rate                   | Type Operation               | Break Frequency cps | Operating Time         | Power hp | Load                                |
|-----------------------------------------------|---------------------------------------------------------------|-----------------------------------|--------------------------------|------------------------------|---------------------|------------------------|----------|-------------------------------------|
| Orbiting with Re-entry                        | <u>Re-entry Shuttle Vehicle</u><br>1. Elevon-surface control* | +40° to - 30°                     | 15°/sec                        | control + space storage      | 3.0                 | 10 min                 | 7        | 144,000 in.-lbs                     |
| Orbiting without Re-entry                     | <u>Space Propulsion Module</u><br>2. Thrust-vector control    | ±7.5°                             | 20°/sec                        | control + space storage      | 3.2                 | 14.2 min               | 2        | 3,900 in.-lbs                       |
|                                               | <u>Large Manned Space Station</u><br>3. Centrifuge*           | ---                               | 3.6 rad/sec                    | control                      | ---                 | 45 min/day             | 6        | 10,750 in.-lbs                      |
|                                               | 4. Hangar-door actuation                                      | 120°/opening                      | 1.8°/sec                       | intermittent                 | ---                 | 60 sec/every 90 day    | 0.6      | 129,500 in.-lbs                     |
|                                               | 5. Solar-cell panel, erect/orient                             | 90°/opening                       | 0.3°/sec/open<br>4°/min/orient | control                      | ---                 | 5 min/open cont/orient | 0.2      | 636 in.-lbs                         |
| Space Travel and Landing                      | <u>Lunar Landing Vehicle</u><br>6. Landing mechanism          | 3 ft                              | 15 ft/sec                      | intermittent + space storage | ---                 | 1 sec/landing          | 3600     | 17,000 lbs                          |
|                                               | 7. Thrust-vector control*                                     | ±7.5°                             | 20°/sec                        | control + space storage      | 3.5                 | 8 min                  | 2        | 3,900 in.-lbs                       |
| Lunar Surface Operation                       | <u>Lunar Roving Vehicle</u><br>8. Materials handling          | 18 in.                            | 2 in./sec                      | control + space storage      | ---                 | ---                    | 8        | 11,500 in.-lbs                      |
|                                               | 9. Surface modification*                                      | 85 ft hor veh.<br>40 ft vert boom | 50 sec/cycle                   | control + space storage      | ---                 | 4 hrs/job              | 5        | 187,000 in.-lbs                     |
|                                               | 10. Motive power                                              | 380 miles                         | 11 mph                         | control + space storage      | ---                 | 40 hrs                 | 6 total  | 1,000 in.-lbs per wheel<br>6 wheels |
| *Tasks on which trade studies were conducted. |                                                               |                                   |                                |                              |                     |                        |          |                                     |

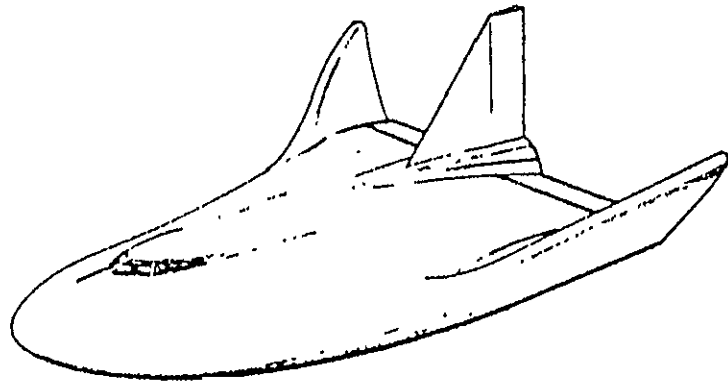
## Actuation Requirements

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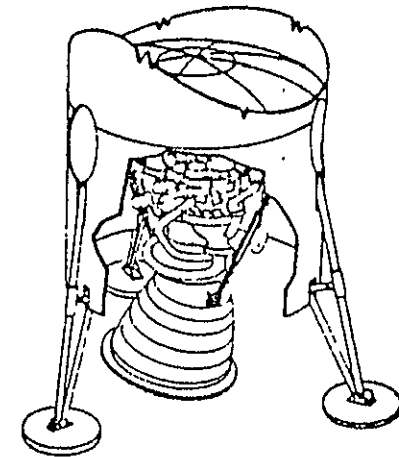
## SELECTED TASKS

As noted on the previous figure, one task associated with each vehicle group was selected for detailed evaluation. These tasks, shown on the accompanying figure, were those considered most competitive for actuation by hydraulic, pneumatic, electromechanical, or mechanical actuation concepts. Actuation tasks requiring less than 1 horsepower were not considered competitive for all concepts. Similarly, single-operation actuation systems were considered not competitive, and a more realistic trade study can be performed by concentrating on continuously operating systems. The surface-modification task using the lunar rover was selected over other lunar-roving-vehicle tasks because funded contract efforts have been awarded in the material-handling and motive-power areas.

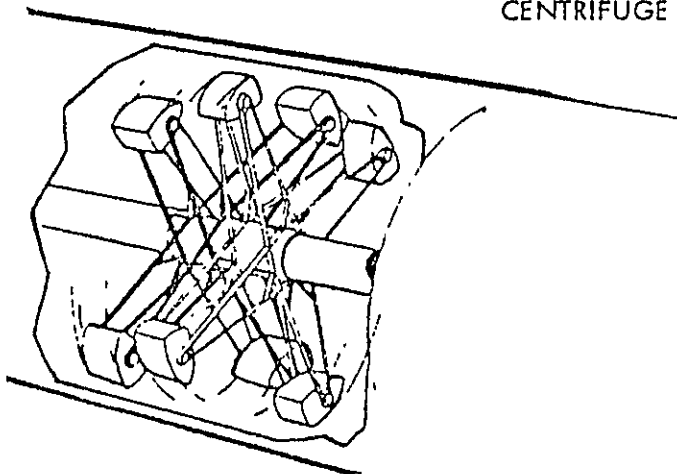
- 1) The elevons are two independently controlled aerodynamic surfaces used during re-entry. They are located on the trailing edge of the delta-shaped vehicle, are powered from a single source, and can be operated by either linear or rotary mechanisms.
- 2) The RL-10 engine is gimbal-mounted and provided with attachments for linear actuators to move the engine in a  $\pm 7.5$ -degree square pattern.
- 3) The centrifuge consists of dual counterrotating assemblies of 15-foot radius, consistent with the maximum diameter of the space station. Each assembly has four personnel carriers.
- 4) The payload attachment to a roving vehicle excavates 1 cubic yard of lunar dust and elevates it to a 40-foot dump height for shelter protection.



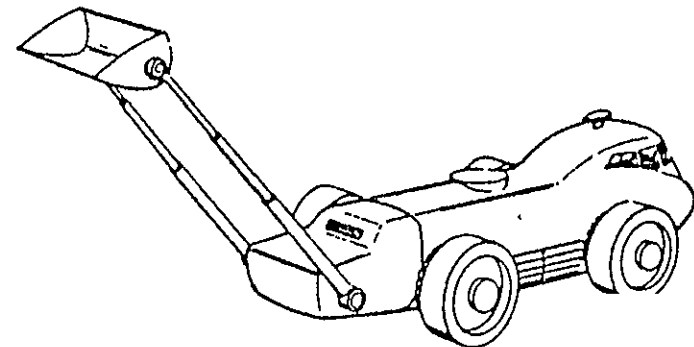
RE-ENTRY VEHICLE  
ELEVON CONTROL



LUNAR LANDER  
THRUST VECTOR  
CONTROL



SPACE STATION  
CENTRIFUGE



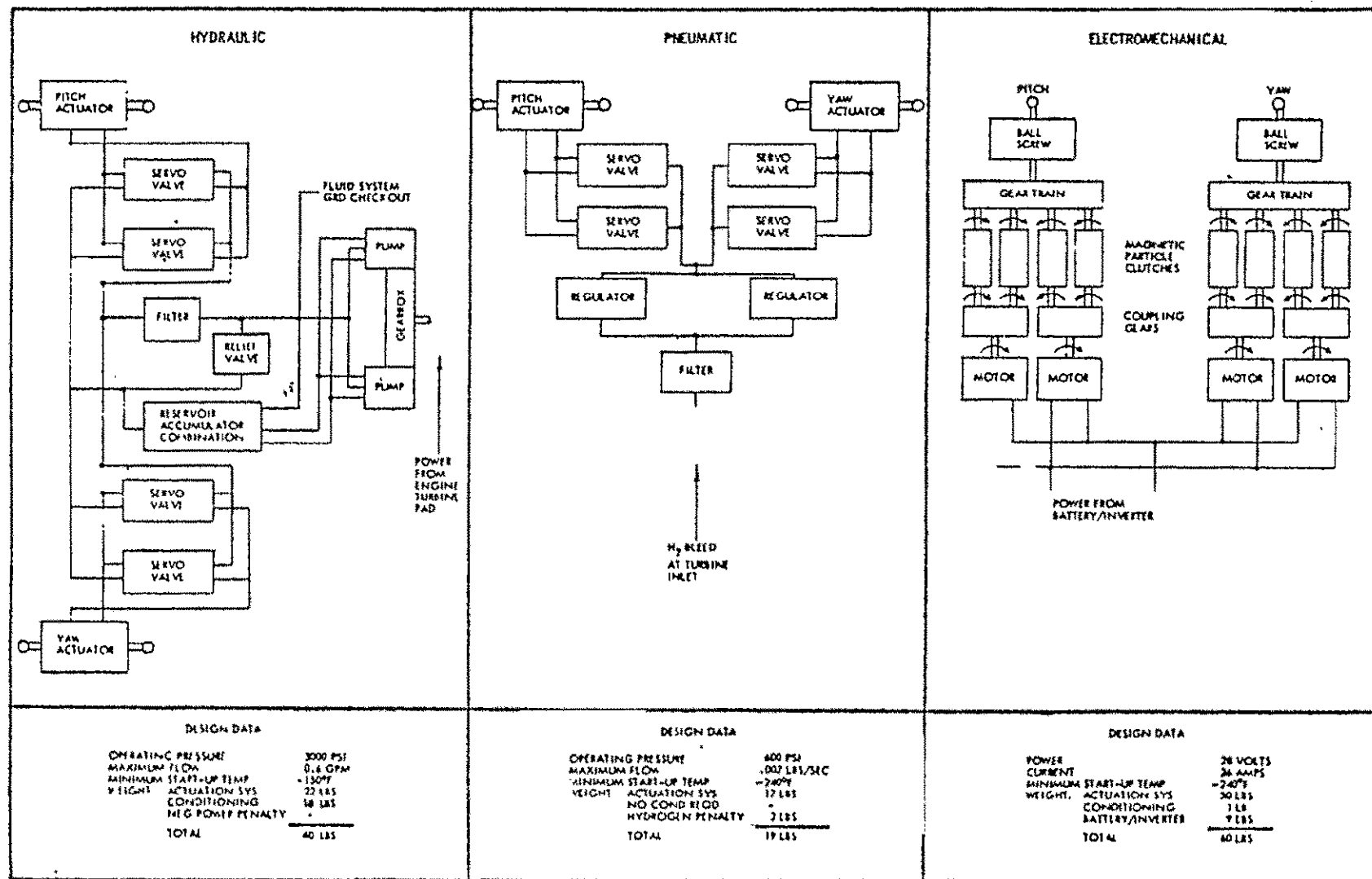
LUNAR ROVER  
PAYLOADER

Selected Tasks

## LUNAR-LANDER TVC ACTUATION SYSTEMS

The detail system designs to satisfy the actuation-task requirements are illustrated on the accompanying figure for one of the four tasks — lunar-lander TVC actuation. Components critical from a failure standpoint were made redundant. These include the pump and servovalves in the hydraulic design, the regulator and servovalves in the pneumatic design, and the motors and clutches in the electromechanical design. A mechanical design was not considered competitive for this task.

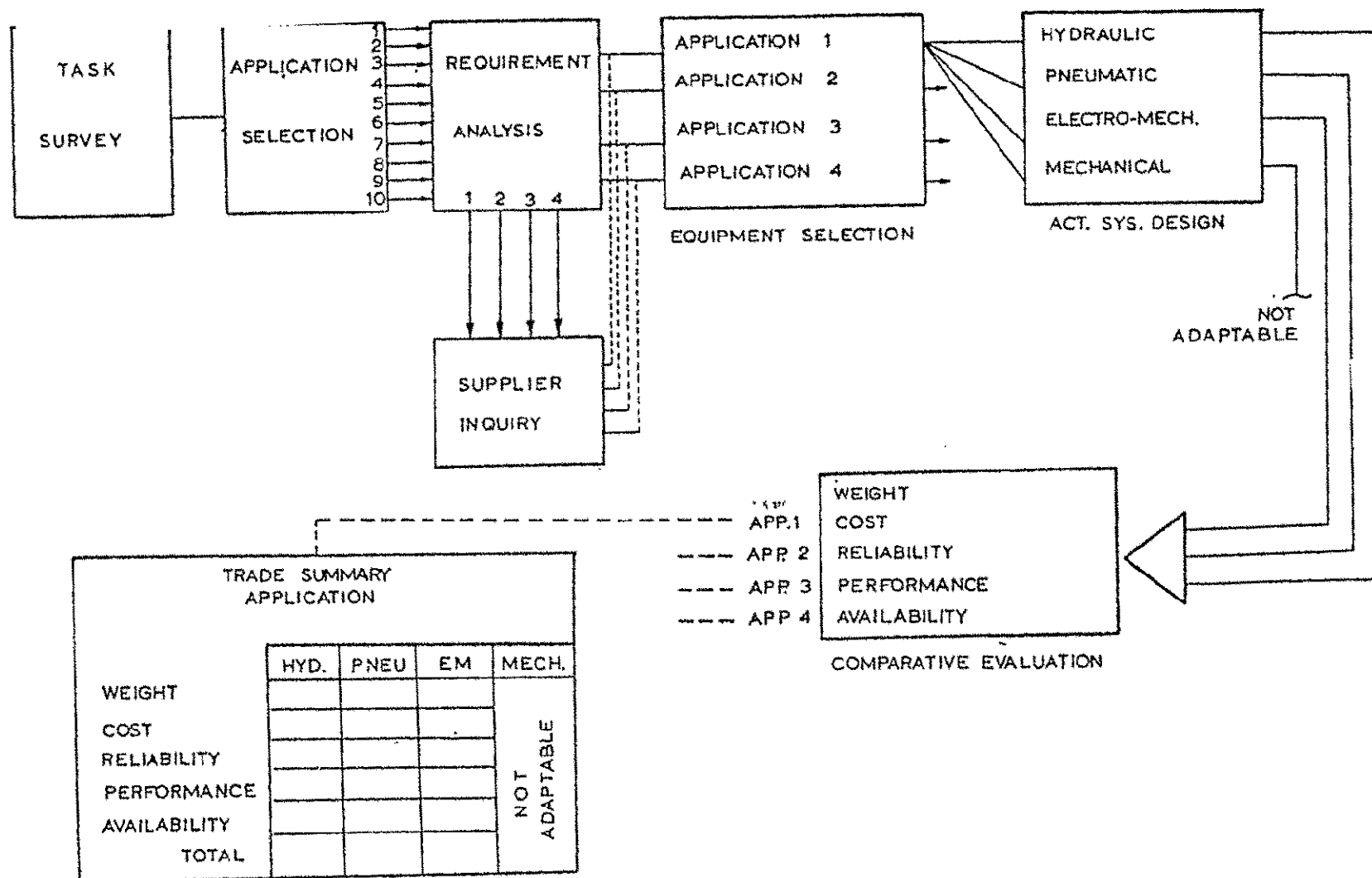
The hydraulic concept requires 18 pounds of conditioning to warm the DuPont E-3 fluid loop before use of the system. With additional development of low-temperature fluids and systems, this conditioning weight could be eliminated. The weight of the hydraulic system would then be competitive with the weight of the pneumatic system. Because of the RL-10 engine construction, shaft power for the hydraulic concept and hydrogen for the pneumatic concept were obtained from the propulsion system at an insignificant penalty. A battery/inverter is used as the lightest-weight power source for the electromechanical concept. The 1-pound conditioning weight is to condition the battery. Similar designs were developed for the elevon, centrifuge, and payload tasks, considering the weight penalties imposed on the actuation systems by power and thermal conditioning.



Lunar Lander TVC-Actuation Systems

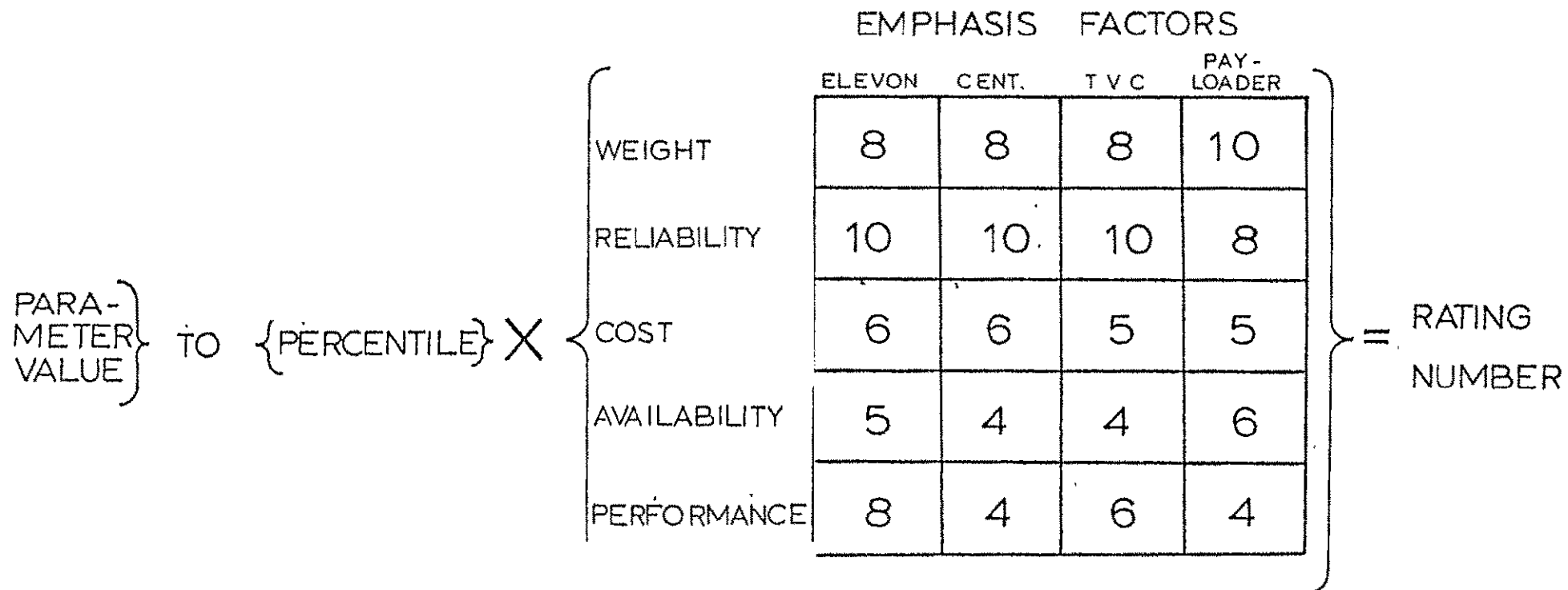
## TRADE-STUDY FLOW CHART

The data flow for the parametric trade study is shown on the accompanying figure. Task requirements were furnished to selected equipment suppliers for the four tasks analyzed in detail, and comments on available systems or components were requested. Some valuable information was gained and used in the actuation designs prepared for each task. Each design was rated on five parameters: weight, reliability, cost, availability, and performance. Weight was important as a trade parameter because all tasks were associated with the payload portion of the vehicle. Actuation-system reliability may determine mission success or failure. The importance of cost arises from development of new hardware or modification of off-the-shelf equipment. Availability was used in considering how much development was required to provide flight-quality systems. Performance was measured according to how well each system could maintain a desired performance level under abnormal operational or environmental conditions. Ratings on all parameters were summed to obtain the final rating given to each system, and the final rating was compared to similar ratings for other actuation designs used to perform the same task.



Trade-Study Flow Chart

# RATING EXAMPLE



## TRADE-STUDY SUMMARY

Ratings for each parameter are shown on the accompanying figure. The sum of all parameter ratings for each concept is the final rating for the concept. The ratings clearly show the pneumatic concept's superiority for the lunar-landing-vehicle thrust-vector-control task, mainly because of the insignificant penalty for thermal conditioning. The electromechanical actuation system is superior for the centrifuge task and has a slight advantage for the payload task. The hydraulic system shows an advantage for the elevon task and is competitive for the payload task.

The evaluation method does not allow direct numerical comparison among ratings for a single system (such as hydraulic) on different tasks. Comparisons can be made only among different systems (e.g., hydraulic and pneumatic) for a single task.

The overall space-actuation-system trade study provides sufficient evidence to conclude that hydraulic, pneumatic, and electromechanical concepts all have application to space tasks. Hydraulic systems were shown to be competitive in two of the four tasks evaluated. Because only four tasks from a possible 80 were evaluated, it is not immediately evident which concept should be selected for any of the remaining 76 tasks. Trade studies should be performed for tasks as they are conceived.

Space-environment effects, particularly low temperature, must be considered when selecting a space actuation system. In the next discussion, thermal-conditioning analyses conducted during the contract are reviewed.

| <u>Trade Parameters</u> | <u>Elevon<br/>Re-entry Vehicle</u> |             |           | <u>Centrifuge<br/>Space Station</u> |                 |           | <u>TVC<br/>Lunar Landing</u> |             |           | <u>Payloader<br/>Lunar Rover</u> |             |           |
|-------------------------|------------------------------------|-------------|-----------|-------------------------------------|-----------------|-----------|------------------------------|-------------|-----------|----------------------------------|-------------|-----------|
|                         | <u>Hyd</u>                         | <u>Pneu</u> | <u>EM</u> | <u>Hyd</u>                          | <u>Pneu</u>     | <u>EM</u> | <u>Hyd</u>                   | <u>Pneu</u> | <u>EM</u> | <u>Hyd</u>                       | <u>Pneu</u> | <u>EM</u> |
| Weight                  | 4.2                                | 7.7         | 2.9       | 6.0                                 | Not Competitive | 7.2       | 4.6                          | 7.7         | 1.6       | 7.6                              | 9.5         | 5.9       |
| Reliability             | 7.2                                | 4.1         | 4.1       | 4.4                                 |                 | 9.4       | 4.9                          | 9.1         | 4.9       | 5.1                              | 4.0         | 7.4       |
| Cost                    | 5.1                                | 1.7         | 5.6       | 2.6                                 |                 | 5.6       | 1.7                          | 4.5         | 2.4       | 2.6                              | 2.0         | 4.6       |
| Availability            | 4.6                                | 1.8         | 4.5       | 1.4                                 |                 | 3.6       | 1.1                          | 3.7         | 1.8       | 4.7                              | 2.4         | 5.5       |
| Performance             | 3.0                                | 4.6         | 6.8       | 1.2                                 |                 | 3.7       | 3.7                          | 5.6         | 1.4       | 2.9                              | 0.8         | 3.8       |
| Rating                  | 24.1                               | 19.9        | 23.9      | 15.6                                |                 | 29.5      | 16.0                         | 30.6        | 12.1      | 22.9                             | 18.7        | 27.2      |

System Selected

## Trade Study Summary

D2-84076-1

## THERMAL PROTECTION FOR ACTUATION SYSTEMS

Contract NASS-11722 included an analytical study to determine the weight penalty for thermal protection of an actuation system. The work was done in two major areas.

- 1) Because this contract emphasized low-temperature performance, a parametric study was conducted to determine the weight penalty associated with protecting components in a -240°F environment. Parameters included several heat sources, two minimum allowable component temperatures, and two insulation materials. Mission time was selected as the independent variable. This work will be discussed in detail.
- 2) Thermal-protection systems were designed and weighed for each actuation system in the trade study. Both heating and cooling systems were studied. This work will also be discussed in detail.

The analytical study indicated several areas requiring additional analysis or developmental work. These areas will also be discussed.

## Thermal Protection for Actuation Systems

- ANALYTICALLY DETERMINE WEIGHT PENALTY FOR LOW TEMPERATURE PROTECTION
  - HEAT SOURCES
  - MINIMUM ALLOWABLE COMPONENT TEMPERATURES
  - INSULATIONS
  - MISSION TIME
- DESIGN THERMAL PROTECTION SYSTEMS FOR TRADE STUDY
  - HEATING
  - COOLING
- RECOMMEND FUTURE THERMAL PROTECTION WORK
  - ANALYTICAL
  - DEVELOPMENTAL

## HEAT SOURCES

In the low-temperature parametric study, both electric heaters and isotope heaters were evaluated. Power sources for the electric heaters included silver-zinc batteries, solar cells, and fuel cells. The battery penalty assumed was 90 watt-hours per pound of battery, and the solar-cell penalty assumed was 1 watt per pound. Three fuel cells were studied:

- 1) Dependent fuel cell — reactants and tankage penalty only (500 watt-hours per pound);
- 2) Semidependent fuel cell — assumes a fuel cell installed in vehicle, but an installed weight penalty is included for reactants and tankage (500 watt-hours per pound plus 0.11 pound per watt);
- 3) Independent fuel cell — a fuel cell is installed for the thermal-protection system (500 watt-hours per pound plus installed weight of 10 pounds for 1 to 10 watts, increasing to 13 pounds for 40 watts).

For isotope heaters, both polonium 210 and plutonium 238 isotopes were considered.

The parametric study was separated into the following vehicle categories because of the lack of competition from batteries when a vehicle uses a fuel cell as its main power source:

- 1) Vehicles with fuel-cell power units;
- 2) Vehicles without fuel-cell power units.

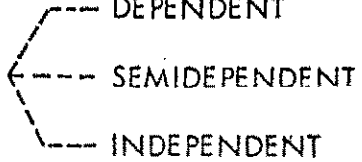
## Heat Sources

- ELECTRICAL HEATERS

- BATTERIES

- SOLAR CELLS

- FUEL CELLS



```
graph LR; FC[FUEL CELLS] --- D[DEPENDENT]; FC --- SD[SEMIDEPENDENT]; FC --- I[INDEPENDENT];
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- ISOTOPE HEATERS

- PLUTONIUM 238

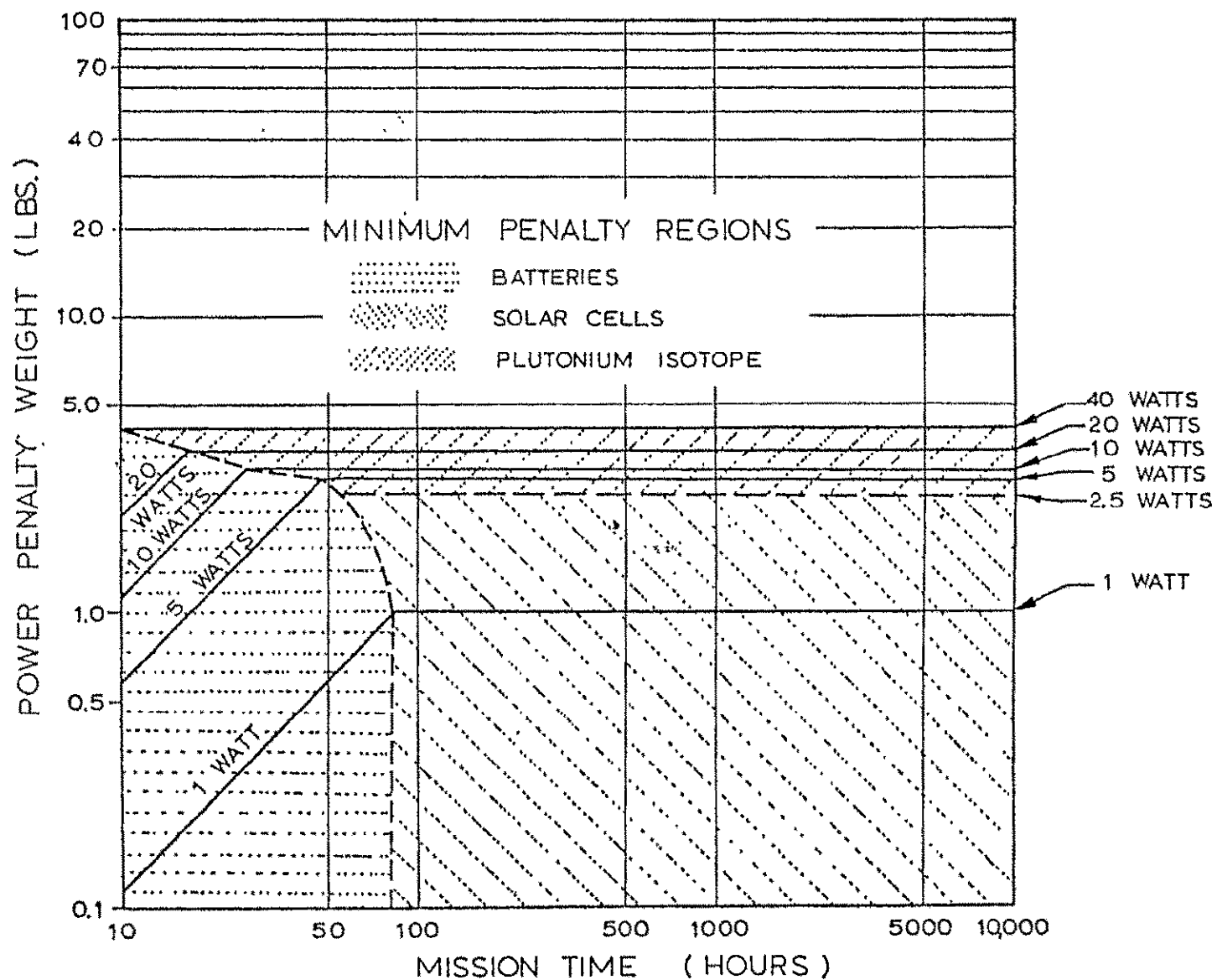
- POLONIUM 210

POWER PENALTY SUMMARY FOR VEHICLES WITHOUT  
FUEL CELL POWER SOURCE

For vehicles without a fuel cell, this figure shows the regions of minimum penalty for power levels from 1 to 40 watts. The power sources considered were:

- 1) Silver-zinc batteries;
- 2) Solar cells;
- 3) Polonium isotopes;
- 4) Plutonium isotopes;
- 5) Independent fuel cells.

The region of minimum power penalty for batteries is limited to short missions. The isotope power source is a lower-weight system than solar cells, if the total power exceeds approximately 3 watts. The polonium isotope and the independent fuel cell are always heavier and do not have a region of minimum power penalty, as do the batteries, solar cells, and plutonium isotopes.



POWER PENALTY SUMMARY FOR VEHICLES WITHOUT FUEL CELL POWER SOURCE

POWER PENALTY SUMMARY FOR VEHICLES  
WITH FUEL CELL POWER SOURCE

For vehicles with a fuel cell, this figure shows the regions of minimum penalty for power levels from 1 to 40 watts. The power sources considered were:

- 1) Silver-zinc batteries;
- 2) Solar cells;
- 3) Polonium isotopes;
- 4) Plutonium isotopes;
- 5) Dependent fuel cells;
- 6) Semidependent fuel cells

The fuel cells are generally limited to missions of 3 weeks or less. The isotope power source is a lower-weight system than solar cells, if the total power exceeds approximately 3 watts. Both dependent and semidependent fuel cells are shown. Batteries and polonium isotopes are always heavier and do not have a region of minimum power penalty, as do the fuel cells, solar cells, and plutonium isotopes.



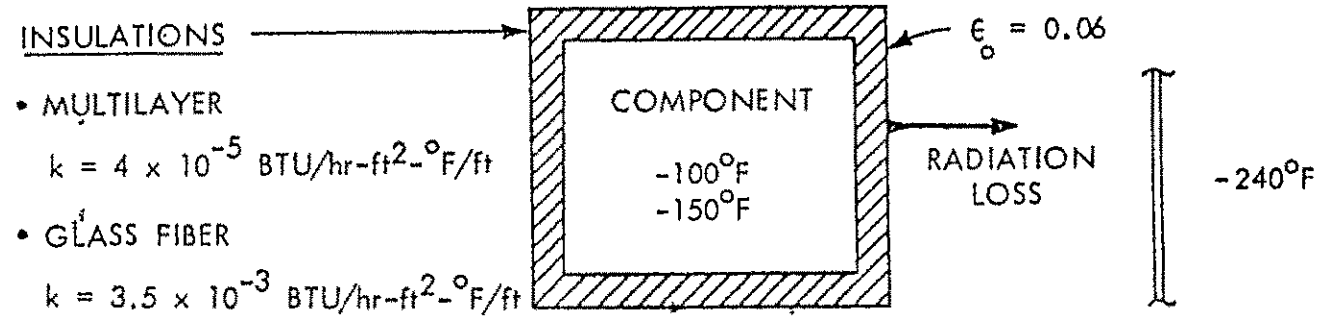
## HEAT TRANSFER ANALYSIS

The heat-transfer analysis considered the heat radiated from the component and the conduction losses. An environmental temperature of  $-240^{\circ}\text{F}$  was assumed. Parametric data included component temperatures of  $-100$  and  $-150^{\circ}\text{F}$  and both glass fiber and multilayer insulations.

Conduction losses result primarily from the supports on the components and the actuator bearings. The heat loss from the actuator bearing, which could be 5 watts or greater, was predicted analytically. Because the radiation heat loss can be reduced to less than 0.1 watt per square foot with insulation, the major heating requirement might be conduction.

## Heat Transfer Analysis

- RADIATION LOSS



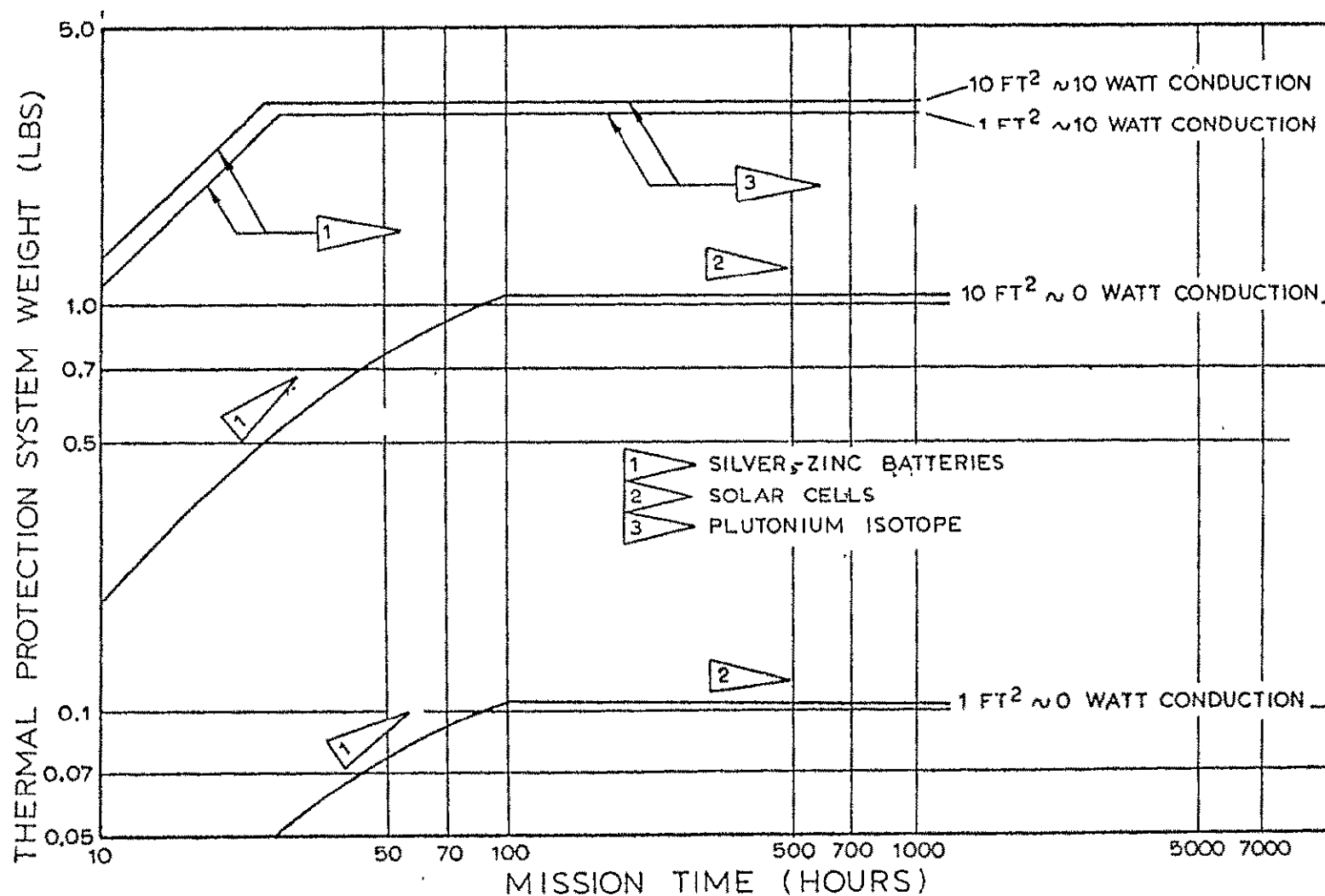
- CONDUCTION LOSS

- ACTUATOR BEARINGS
- EQUIPMENT SUPPORTS

THERMAL PROTECTION SYSTEM WEIGHT INCLUDING CONDUCTION LOSSES  
FOR VEHICLES WITHOUT FUEL CELL POWER SOURCE  
 $T_c = -150^{\circ}\text{F}$        $\epsilon = 0.06$   
MULTI-LAYER INSULATION

This chart and the following chart are examples of the parametric data available in the contract report documents.

For vehicles without a fuel cell, the accompanying chart shows the total weight penalty for a component requiring thermal protection to  $-150^{\circ}\text{F}$ . Total weight penalty is shown for both 1 and 10 square feet of surface area and for conduction losses of 0 and 10 watts. The battery and solar-cell systems require an additional 0.2 pound for each temperature switch and 0.4 pound for each square foot of heater area. The isotope system requires additional weight for conduction rods from the power source to the components.



THERMAL PROTECTION SYSTEM WEIGHT INCLUDING CONDUCTION LOSSES  
FOR VEHICLES WITHOUT FUEL CELL POWER SOURCE

$T_c = -150^\circ\text{F}$

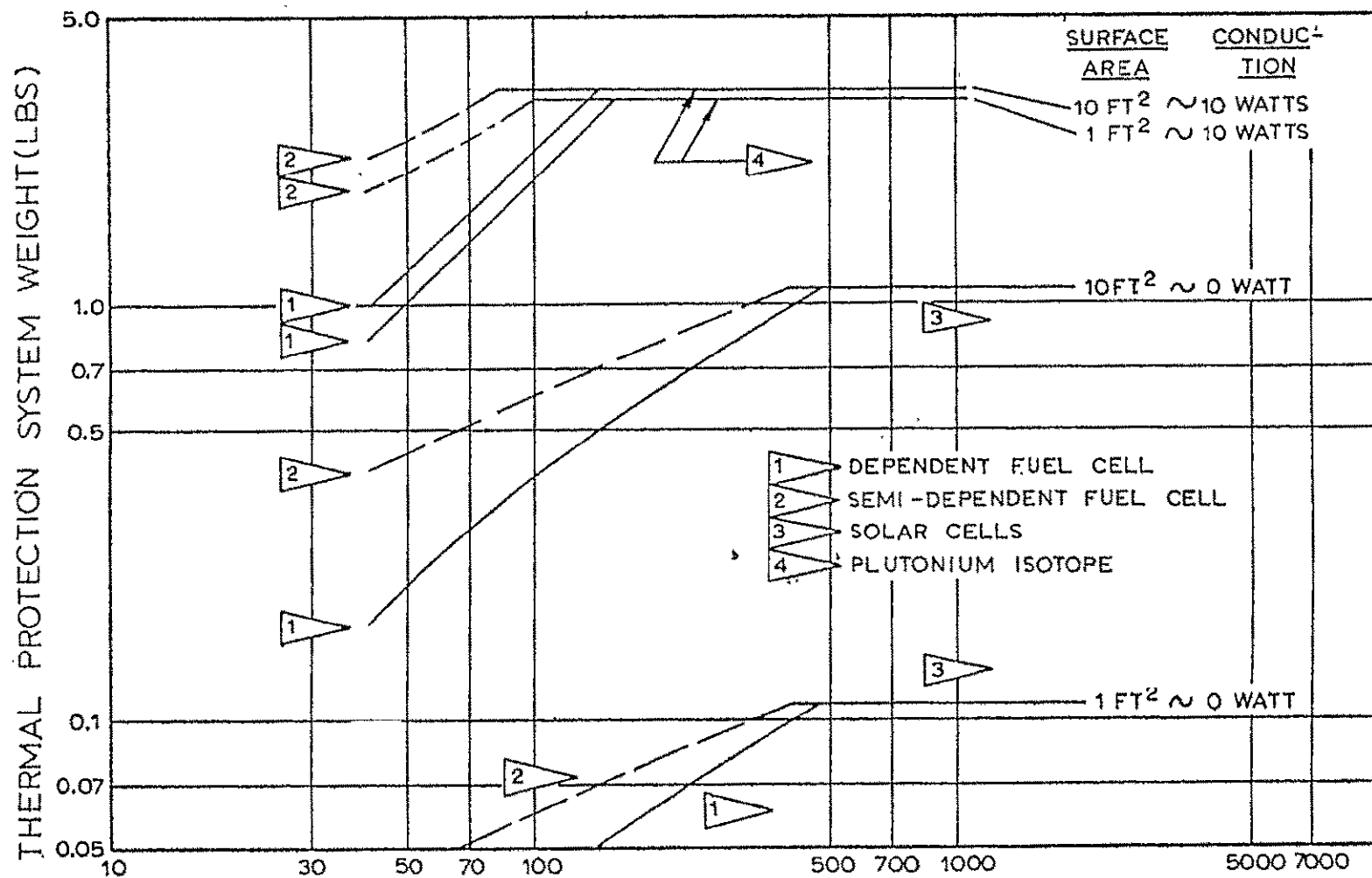
$\epsilon_o = 0.06$

MULTI-LAYER INSULATION

D2-84076-1

THERMAL PROTECTION SYSTEM WEIGHT INCLUDING  
CONDUCTION LOSSES FOR VEHICLES WITH FUEL  
CELL POWER SOURCE  
 $T_c = -150^\circ\text{F}$   
MULTI-LAYER INSULATION  
 $\epsilon = 0.06$

For vehicles with a fuel-cell power source, this figure shows the total weight penalty for a component requiring a minimum temperature of  $-150^\circ\text{F}$ . Total weight penalty is shown for both 1 and 10 square feet of surface area and for conduction losses of 0 and 10 watts. The fuel-cell and solar-cell systems require an additional 0.2 pound for each temperature switch and 0.4 pound for each square foot of heater area. The isotope system requires additional weight for conduction rods from the power source to the components.



THERMAL PROTECTION SYSTEM WEIGHT INCLUDING  
CONDUCTION LOSSES FOR VEHICLES WITH FUEL  
CELL POWER SOURCE

$$T_c = -150^{\circ}\text{F}$$

MULTI-LAYER INSULATION

$$\epsilon_o = 0.06$$

D2-01076-1

## TRADE STUDY SUPPORT

The four actuation tasks included in the trade study and the environmental temperatures used for the trade study are shown. One or more of the actuation systems associated with the thrust-vector-control (TVC) task requires a heating system. The elevator task requires a cooling system, and the payload requires a heating and cooling system. Thermal protection for the hydraulic TVC system will be discussed in more detail.

## Trade Study Support

- TVC TASK                    -240°F TO 100°F
  - HEATING SUBSYSTEM
  
- ELEVON TASK               -60°F TO 550°F
  - COOLING SUBSYSTEM
  
- CENTRIFUGE                CONDITIONED COMPARTMENT
  - LOCATED IN SPACE STATION — USES ACTIVE COOLING LOOP OF STATION
  
- PAYLOADER                 -240°F TO 275°F
  - HEATING SUBSYSTEM
  - COOLING SUBSYSTEM

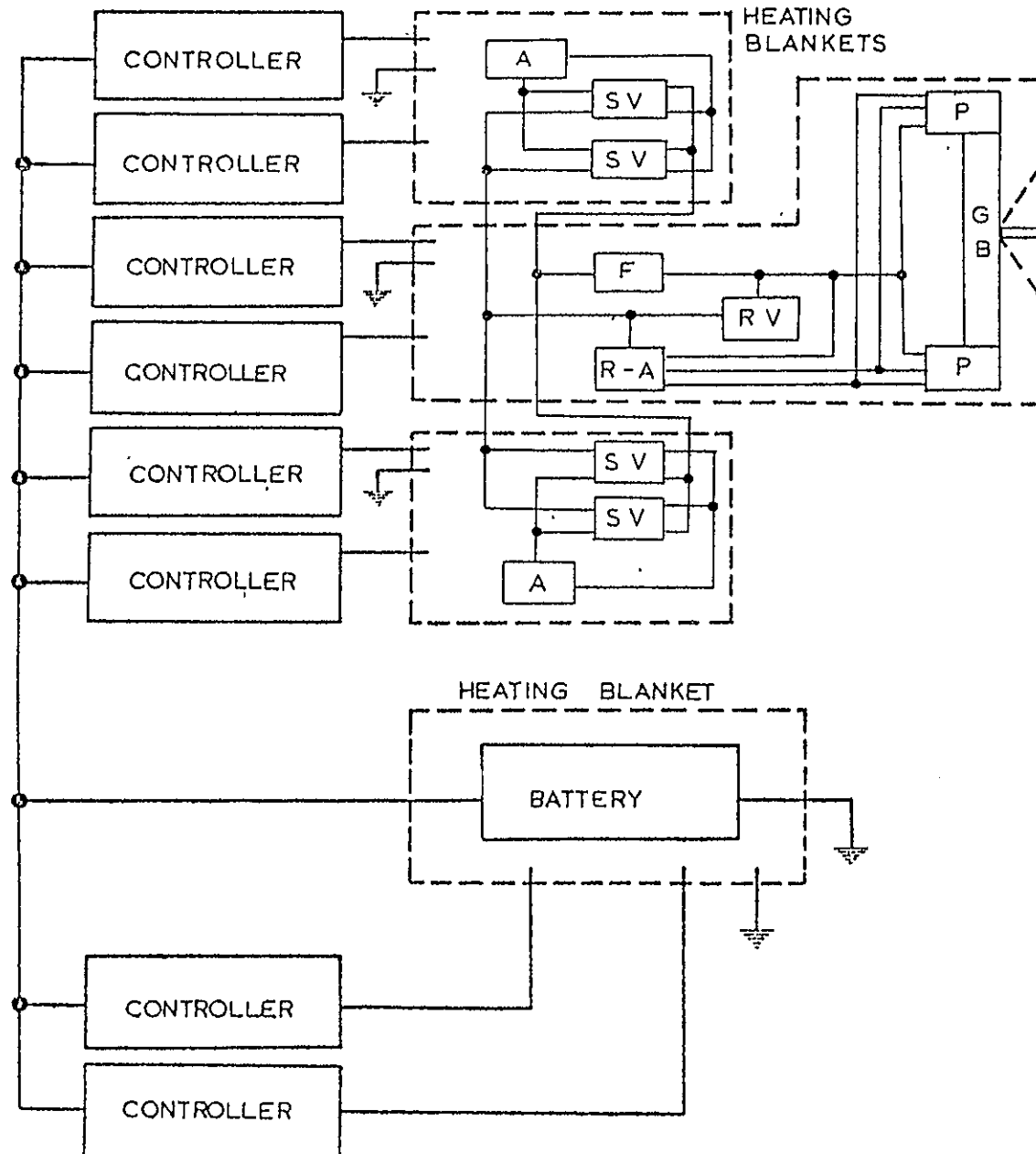
### THERMAL PROTECTION: HYDRAULIC TVC ACTUATION SYSTEM

This figure shows the RL-10 TVC hydraulic-actuation system and thermal-protection system. A thermal-protection system was required to provide a minimum operation temperature of  $-100^{\circ}\text{F}$ . Three heating systems were considered:

- 1) Battery-power source for maintaining the components at  $-100^{\circ}\text{F}$  for the 14-day assumed mission;
- 2) Isotope-power source for maintaining the components at  $-100^{\circ}\text{F}$ ;
- 3) Battery system for preheating the components from  $-240^{\circ}\text{F}$  to  $-100^{\circ}\text{F}$  before each operation — two preheats were required, one for midcourse correction and one for lunar landing.

The isotope and the battery preheat systems weighed approximately the same because three isotope sources were required (two actuators and one pump package). The battery preheat system was selected because of less development effort, and because of the possible high-temperature problems with the isotope system in the  $-100^{\circ}\text{F}$  environment.

# THERMAL PROTECTION:HYDRAULIC TVC ACTUATION SYSTEM



## FUTURE WORK

Additional effort is required in three areas.

- The first is to continue determining component limitations. This effort involves both the minimum operating temperature and component cooling below this minimum if the components are preheated before use.
- Isotope heating is promising for long missions and larger heating requirements. Both analytical and development work are required. Conduction losses must be determined, but low heat leakage might not have to be developed. The conditions at maximum environment must be considered because the isotope system cannot be shut off.
- Electrical heating requires low-heat-leak bearings and other conduction losses. Efficient use of multilayer insulation is also required.

## Future Work

- DETERMINE COMPONENT TEMPERATURE LIMITATIONS
- DEVELOP ISOTOPE HEATING
  - ANALYTICAL
    - SAFETY
    - OPERATION AT MAXIMUM ENVIRONMENTAL TEMPERATURES
    - DESIGN CONDUCTION RODS TO CONDUCT HEAT FROM SOURCE TO COMPONENT
  - DEVELOPMENTAL
    - DEVELOP UNIFORM HEATING METHODS
    - HEATING FOR HYDRAULIC AND COOLING LINES
    - DETERMINE CONDUCTION LOSSES
- DEVELOP ELECTRICAL HEATING
  - DEVELOPMENTAL
    - DEVELOP LOW HEAT LEAK ACTUATOR BEARINGS AND SUPPORTS
  - DEVELOP INSULATION INSTALLATION TECHNIQUES

## HYDRAULIC-SYSTEM ANALYSIS AND TEST PROGRAM

The effective use of hydraulic systems for space applications depends largely on the compatibility of these systems with space temperature and pressure environments. In previous test experience, hydraulic-system limitations have been associated mainly with materials and component evaluations rather than complete system performance analyses. For example, state-of-the-art hydraulic fluids limit low-temperature testing of a system to approximately  $-65^{\circ}\text{F}$ . Therefore, advanced fluids are necessary for evaluation conducted in simulated space environments. In this section, the evaluation of low-temperature fluids and seals to be used in the breadboard test system, the design of the test system and supporting equipment, and the tests performed with that system will be reviewed.

HYDRAULIC-SYSTEM

ANALYSIS

AND

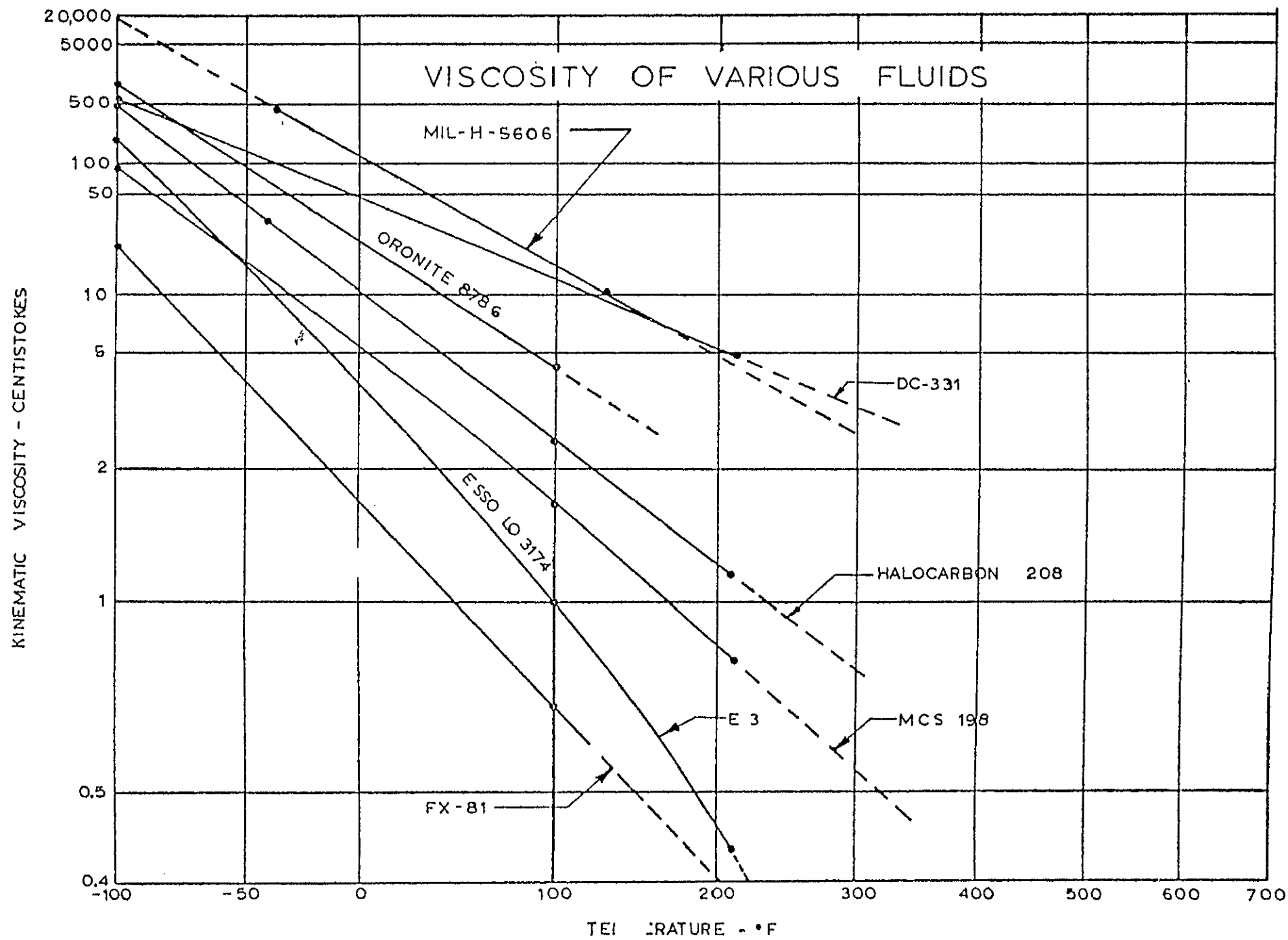
TEST PROGRAM

## VISCOSITY OF VARIOUS FLUIDS

The temperature effects on the viscosity of various advanced low-temperature hydraulic fluids are shown on the accompanying figure and compared with the viscosity of MIL-H-5606. Samples of five of these fluids were obtained for evaluation of the properties of interest in high-pressure hydraulic actuation systems. The fluids were:

- 1) Dow Corning DC 331, a low-viscosity dimethyl silicone;
- 2) Du Pont E-3, a fluorinated ether;
- 3) Halocarbon 208, a halogenated branched-chain aliphatic hydrocarbon;
- 4) Minnesota Mining and Manufacturing Company FX-81, a cyclic fluorinated ether;
- 5) Monsanto MCS-198, a silicate ester.

The viscosity temperature slope of the DC 331 curve was the most productive for testing, although its low-temperature capability with an ASTM pour point of  $-130^{\circ}\text{F}$  was not as good as some other fluids. MCS-198, with a pour point of  $-142^{\circ}\text{F}$ , was the most highly developed fluid, having been used in the heat-transfer loop of the Gemini vehicle. FX-81 had the lowest pour point at  $-172^{\circ}\text{F}$ , with E-3 at  $-155^{\circ}\text{F}$ , and Halocarbon 208 at  $-118^{\circ}\text{F}$ .



## LOW-TEMPERATURE-FLUID THERMAL STABILITY AND MATERIAL COMPATIBILITY

Some of the properties of the evaluated fluids are shown on the accompanying figure. These results show the high-temperature acceptability of each fluid and its compatibility with various metals that might come in contact with the fluid in a hydraulic system. None of the fluids evaluated was entirely acceptable for space-system use. Although it has low viscosity at 275°F, which would make its use in a recirculating system questionable, Du Pont E-3 was selected for use in the test system. FX-81 had low viscosity at high temperature and poor lubricity even at room temperature. DC 331 solidified at -90°F after 8 hours of exposure, disqualifying it as a competitive fluid. MCS-198 had poor lubricity and caused serious swelling of elastomers. Halocarbon 208 had very destructive effects on phenyl-silicone elastomers.

As shown in the accompanying figure, compatibility with metals was not a serious problem for any of the evaluated fluids.

| Fluid   | Time (hrs)      | Temp (°F) | ① Viscosity Change (%) | Change in Flash (°F) | Change in Solids (mg) | Weight Change in Metals (mg/cm <sup>2</sup> ) |                |        |        |              |                  |        |                     |        |
|---------|-----------------|-----------|------------------------|----------------------|-----------------------|-----------------------------------------------|----------------|--------|--------|--------------|------------------|--------|---------------------|--------|
|         |                 |           |                        |                      |                       | 4130                                          | Bearium Bronze | 440C   | M-10   | Naval Bronze | Titanium 6AL-4V  | Be-Cu  | 30435               | 32155  |
| MCS-198 | 20              | 275       | -4.2                   | -5                   | +0.7                  | -0.177                                        | -0.276         | -0.079 | -0.256 | -0.336       |                  |        |                     |        |
| DC-331  | 20              | 275       | -0.33                  | +15                  | +0.6                  | 0.00                                          | 0.00           | -0.02  | 0.00   | -0.02        |                  |        |                     |        |
| E-3     | 20              | 275       | 0.0                    | —                    | +0.05                 | -0.177                                        | -0.059         | -0.198 | -0.198 | -0.079       |                  |        |                     |        |
| H-208   | 18 <sup>①</sup> | 275       | +0.3                   | —                    | —                     | —                                             | —              | -0.019 |        | -0.059       | 0.0 <sup>②</sup> | —      | 0.0 <sup>②</sup>    | -0.019 |
| E-3     | 48 <sup>①</sup> | 275       | +21.0                  | —                    | —                     | —                                             | —              | 0.0    |        |              | -0.019           | -0.059 | +0.039 <sup>②</sup> | -0.019 |

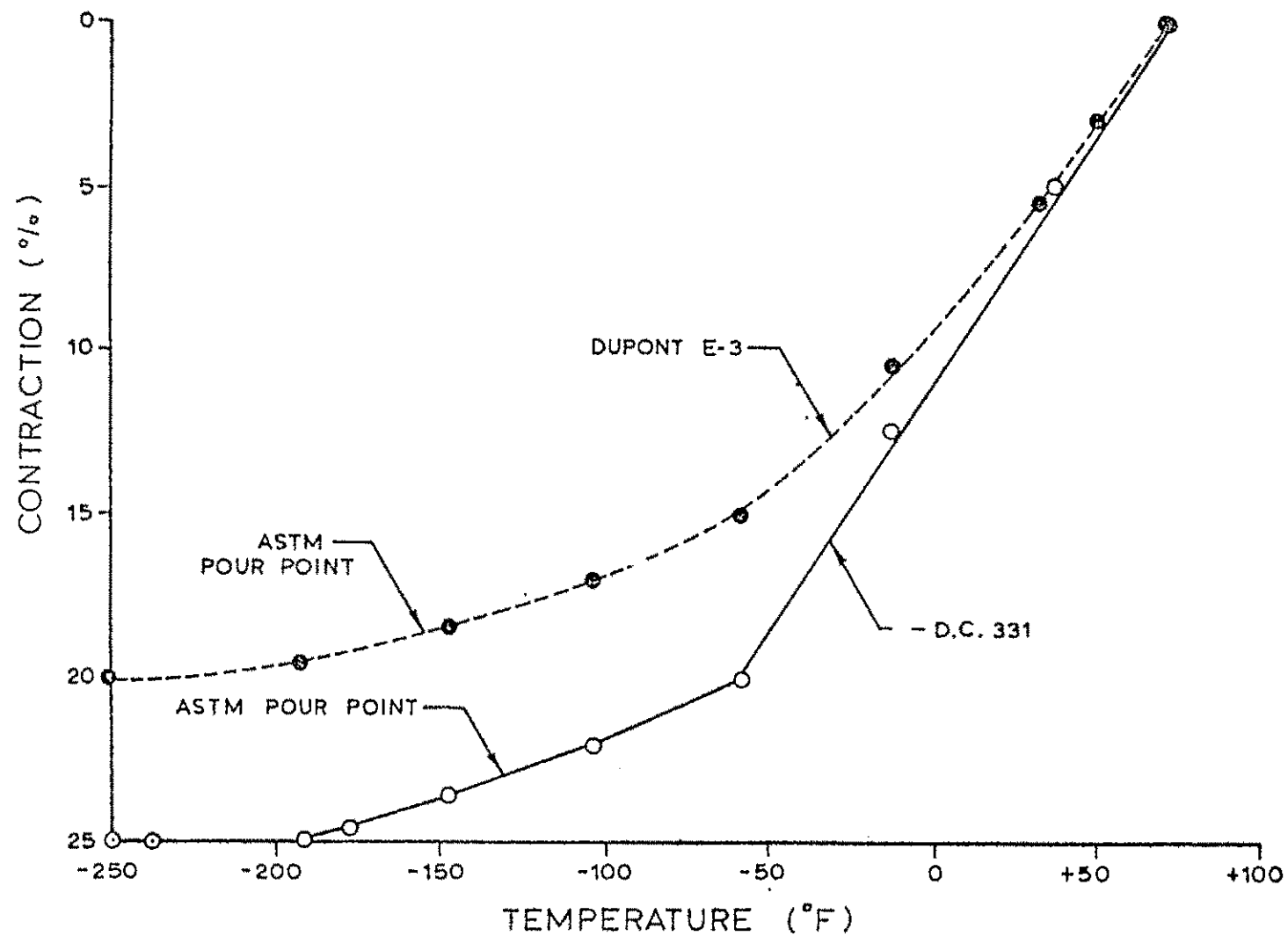
① DC 675 O rings present in bomb during run.

② These metals sectioned for intergranular corrosion inspection. None found.

## Low-Temperature Fluids Thermal Stability And Material Compatibility

## CONTRACTION OF LOW-TEMPERATURE HYDRAULIC FLUIDS

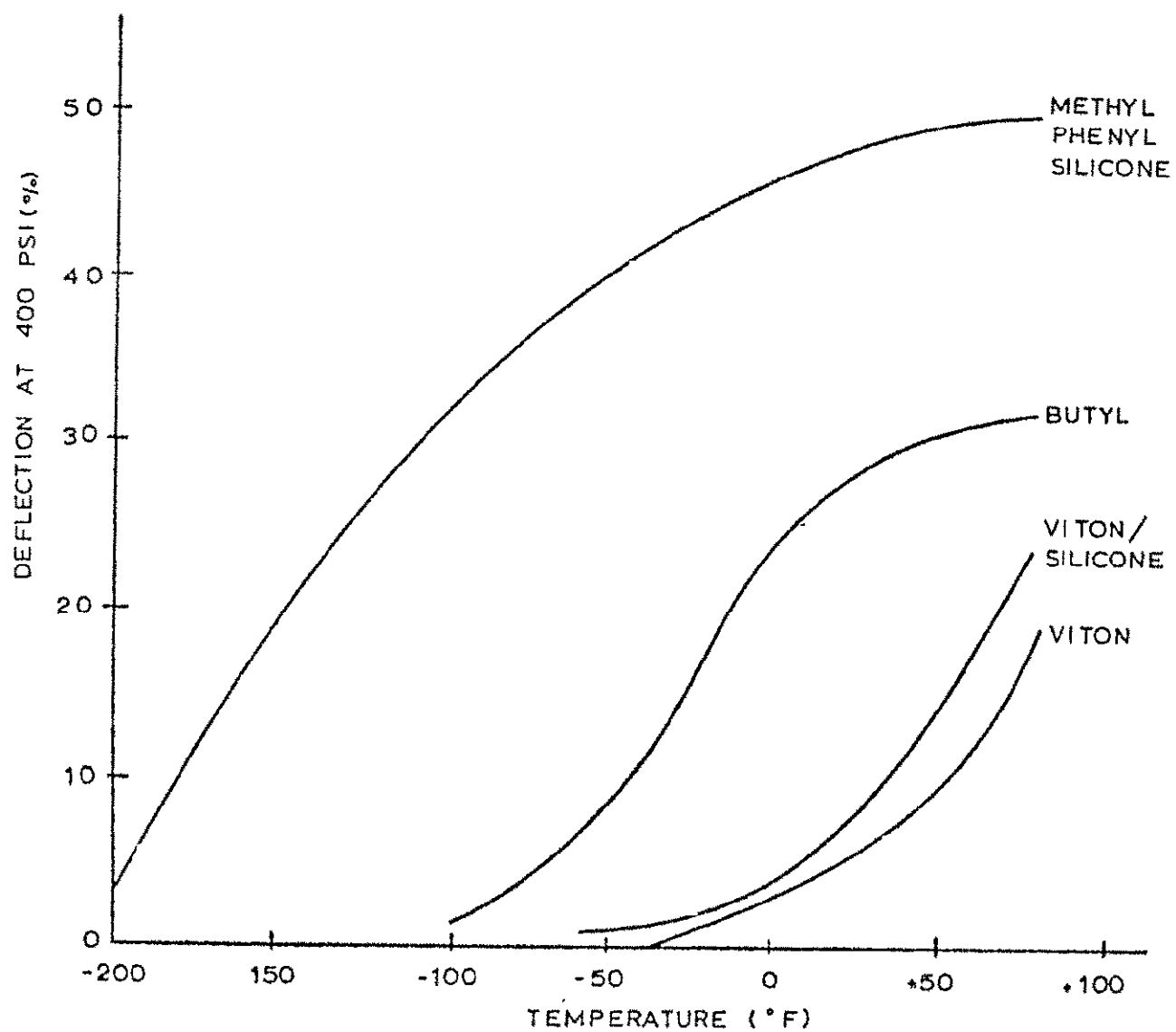
The acceptability of Du Pont E-3 for low-temperature testing depended on whether it contracted with decreasing temperature. The accompanying curve shows the contraction that occurred during tests of both E-3 and DC 331 fluids. The major contraction from room temperature to  $-50^{\circ}\text{F}$  is shown, as is the continued contraction beyond the pour point. This large amount of contraction at low temperature and the subsequent expansion at high temperature means that the reservoir design problem must be thoroughly evaluated.



Contraction Of Low-Temperature Hydraulic Fluids

## SEAL-MATERIAL DEFLECTION CHARACTERISTICS

The results of Boeing research on seals for space applications are shown on the accompanying figure and indicate the superiority of phenyl-silicone seals at low temperature. Using the criterion that adequate high-pressure hydraulic-system sealing can be attained with elastomers at 10-percent deflection, the curves show the adequacy of phenyl-silicone seals below the pour point of the E-3 fluid, selected for use in the test system.



Seal Material Deflection Characteristics

## FLUID-ELASTOMER COMPATIBILITY

Five phenyl-silicone elastomers were evaluated for compatibility with the fluid samples obtained from the various suppliers. The results of these compatibility tests are shown on the accompanying figure. Parker S-424-7 compound was selected for use with the E-3 fluid in the test system. The seals showed little change in tensile strength and no change in hardness after exposure for 72 hours at 160° F. No reaction of the seal material with the E-3 fluid occurred during this test. Seals for all static and dynamic locations within the components to be tested were purchased from the Parker Seal Company.

| Elastomer         | Fluid   | Before Exposure  |              |     | After Exposure<br>of 72 hours @ 160°F |              |     | Volume<br>Change<br>(%) | Remarks                               |
|-------------------|---------|------------------|--------------|-----|---------------------------------------|--------------|-----|-------------------------|---------------------------------------|
|                   |         | Tensile<br>(psi) | Elong<br>(%) | Dur | Tensile<br>(psi)                      | Elong<br>(%) | Dur |                         |                                       |
| DC 675            | MCS-198 |                  |              |     |                                       |              |     | +183.5                  |                                       |
| DC 675            | DC-331  |                  |              |     |                                       |              |     | + 80.5                  |                                       |
| DC 675            | H-208   | 632              | 200          | 74  | 245                                   | 60           | 60  | +191                    |                                       |
| DC 675            | E-3     | 752              | 131          | 72  | 739                                   | 128          | 72  | + 0.9                   | White precipitate<br>in fluid         |
| DC 651            | E-3     | 1180             | 473          | 53  | 1079                                  | 418          | 53  | + 1.2                   | Compound soft<br>for best O-<br>rings |
| DC 960            | E-3     | 1392             | 543          | 59  | 1384                                  | 500          | 62  | + 0.6                   | Compound soft<br>for best O-<br>rings |
| Parker<br>S-383-7 | E-3     | 1096             | 360          | 66  | 1112                                  | 353          | 68  | - 7.5                   | No precipitate                        |
| Parker<br>S-424-7 | E-3     | 892              | 200          | 73  | 915                                   | 170          | 73  | + 1.1                   | No white pre-<br>cipitate             |
| 3-2013<br>Buna-N  | H-208   | 2228             | 220          | 70  | 2084                                  | 180          | 60  | + 14.5                  |                                       |

Tensile and elongation by ASTM D412.

Durometer by ASTM D676.

Volume Change by ASTM D471.

## Fluid-Elastomer Compatibility

## DYNAMIC NONELASTOMERIC SEALS

A test requirement to measure fluid leakage past the primary dynamic gland seals at each end of the double-acting linear actuator established a need for a secondary seal in each rod gland. An investigation was conducted to select a nonelastomeric seal configuration for use in the secondary seal installation. The seal configurations evaluated are shown on the accompanying figure. Metal seals were considered undesirable in leakage because of the hard-vacuum environment to be encountered. Little test data was available on the various nonelastomeric seals investigated. The "omniseal" was selected for the application because of acceptable previous use and what appeared to be an easier installation in the internal seal grooves.

| Seal Trade Name | Vendor                             | Dynamic Seals Proposed for use with E-3            |                                                                                      |                                                                                                                 |
|-----------------|------------------------------------|----------------------------------------------------|--------------------------------------------------------------------------------------|-----------------------------------------------------------------------------------------------------------------|
|                 |                                    | Type                                               | Rod Seal Configuration                                                               | Comment                                                                                                         |
| Omniseal        | Aeroquip Corp.                     | Teflon with dual 304 springs.                      |   | Information indicates that the seals would perform satisfactorily.                                              |
| Bal-Seal        | Bal-Seal Engrg Co.                 | Teflon with a single stainless steel spring.       |   | Similar to the "omni-seal," but contains only one load spring.                                                  |
| Delta-Seal      | R. Krueger Co.                     | Teflon-graphite with a spring-loaded wedge ring.   |   | Relatively new. Difficult to design into the actuator.                                                          |
| Pacific         | Pacific Piston Ring Co.            | Step-gap teflon with 17-4PH contracting ring.      |   | No information available on seal performance.                                                                   |
| Raco            | Raco Engineering                   | Teflon cap strip with O-ring.                      |   | The swell characteristics of the elastomer may result in excessive rod friction.                                |
| Cook            | Dover Corp.,<br>Cook Airtomic Div. | Outer ring AMS 5643 inner ring. Teflon and O-ring. |  | Four sets of seals were required for each rod gland.<br><br>The elastomer may result in excessive rod friction. |

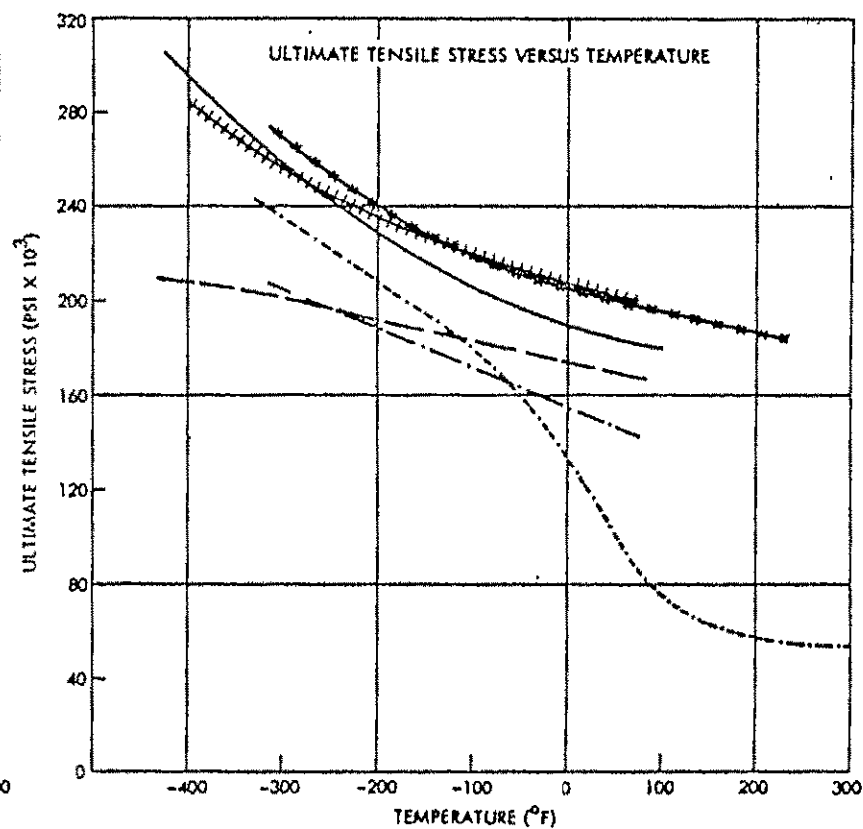
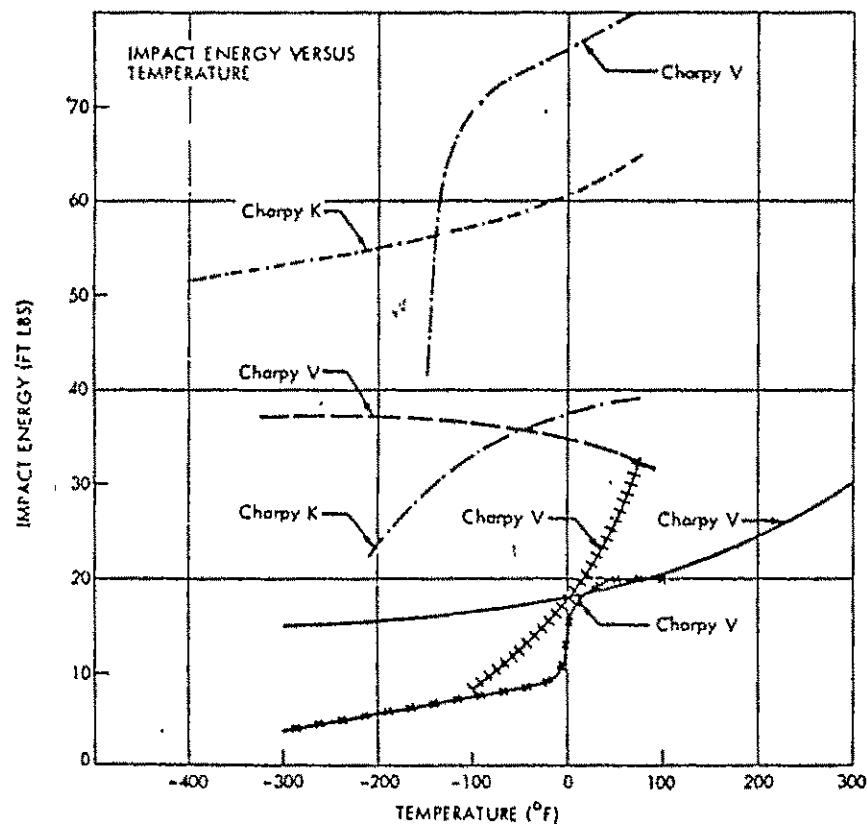
## DYNAMIC NONELASTOMERIC SEALS

## MATERIAL CHARACTERISTICS AT LOW TEMPERATURE

Because the hydraulic system to be tested is the first of its type used in extremely low-temperature and hard-vacuum testing, it was important to select versatile materials with capabilities somewhat beyond the operating temperature range of the fluid selected. An analysis was conducted to select materials for use in the manufacture of the actuator and reservoir. Strength properties of the various materials considered are shown on the accompanying figure.

The tensile strength of materials considered for the high-pressure-fluid power system presents no problem. Moreover, tensile strength data actually shows a significant improvement at low-temperature operation. When impact strength is also important, as it is for the actuator, the fact that the impact strength of some materials decreases rapidly at low temperatures is a principal consideration. The material selected for use in manufacturing the actuator was 304 LC stainless steel; however, this material has low tensile strength at room temperature, which necessitates extreme care in handling surfaces with fine finishes. Inconel X, the second choice, would have alleviated this problem, but would have resulted in higher material and fabrication costs. Titanium was selected for the reservoir pressure vessel, which did not have to withstand impact loads.

— Titanium 6Al-4V  
 - - - Inconel X, Age Hardened  
 ••••• 17-4PH Steel  
 - - - 304L C Annealed Steel  
 - - - 4340 Steel HT to 150 PSI  
 + + + + 416 Steel, HT at 1800°F for 1 Hr, OQ; Temper at 700°F for 4 Hrs, AC



Material Characteristics At Low Temperature

## TEST-SYSTEM DESIGN REQUIREMENTS

The space actuation task selected for simulation in the test program was the thrust-vector-control gimbaling of an RL-10 propulsion engine, which would be required on a typical lunar-landing vehicle. This task is one of the four tasks evaluated in the trade-study portion of the contract.

The TVC requirements are  $\pm 7.5$  degrees motion of the RL-10 engine in a square pattern at a maximum rate of 20 degrees per second and an acceleration of 250 degrees per second<sup>2</sup>. A total operating time of 8 minutes is required, and the desired servo system break frequency is 3.2 cycles per second.

The actuation-system requirements for the recirculating-system flight application and single-pass-test-system simulation are compared on the accompanying figure. The maximum force requirement of 559 pounds for the selected task requires a piston area of 0.279 square inch when using supply fluid at 3000 psig and a two-thirds pressure drop at the load. To ensure actuation performance at all temperatures, the force capability of the test actuator was increased to 780 pounds by using a piston area of 0.39 square inch. This increase in force capability provides a design safety factor of 1.4.

The piston velocity of 6 inches per second, which accompanies the 4-inch stroke for the selected task, requires a fluid flow of 0.435 gallon per minute to satisfy the actuation task in the flight application. To explore the actuation frequency range of 0.5 to 6 cps during the test, system-operation fluid flow between 0.39 and 2.2 gpm was required, using an actuator with a 0.39-square-inch piston area. Dry-run tests with the serv analyzer and an electrical-load simulation indicated that 2.25 minutes would be adequate for the 0.5- to 6.0-cps frequency scan without exceeding the maximum allowable speed of the analyzer. This duration was used in establishing the required test reservoir volume. Before attempting to investigate pump operational problems, a single-pass test system was selected to eliminate the difficulties of maintaining low stabilized temperatures with a pumping system and to evaluate fluid and seal problems.

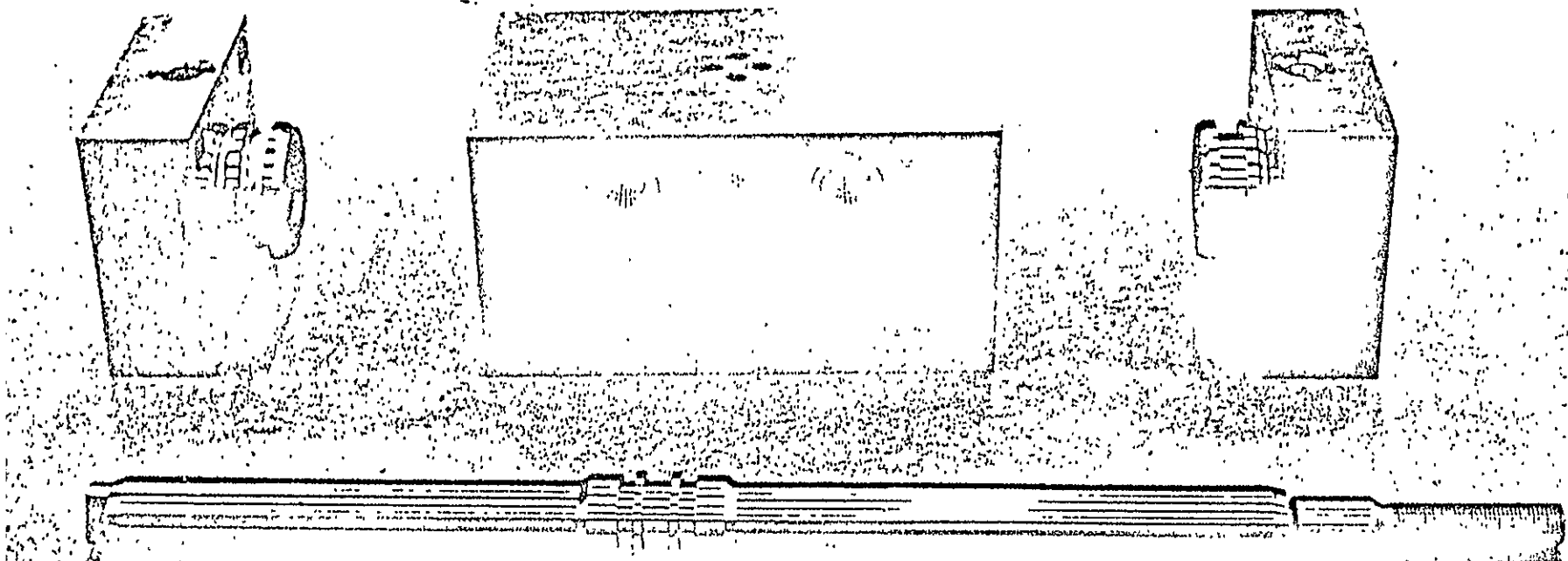
## TEST-SYSTEM DESIGN REQUIREMENTS

| <u>Characteristic</u>               | <u>Flight<br/>Application</u> | <u>Test<br/>System</u> |
|-------------------------------------|-------------------------------|------------------------|
| Frequency Range (cps)               | 0.4 to 6.0                    | 0.5 to 6.0             |
| Actuator Stroke at 0.5 cps (inches) | ±2.00                         | ±2.00                  |
| Actuator Stroke at 6.0 cps (inches) | ±0.90                         | ±0.90                  |
| Servo Break Frequency (cps)         | 3.2                           | 3.0                    |
| Actuator Force (lb)                 | 559                           | 780                    |
| Actuator Piston Area (sq in)        | 0.279                         | 0.39                   |
| Maximum Flow Rate (gpm)             | 0.435                         | 2.2                    |
| Operation Time/Test (min)           | 8.0                           | 2.25                   |
| Fluid Volume/Test (gal.)            | 3.48                          | 2.91                   |

## TEST ACTUATOR PARTS

The major actuator parts are the housing, piston, and two seal glands, which are shown on the accompanying figure. In addition, the complete actuator incorporates an adjustable rod-end bearing and a head-end cap with bearing. These are the parts for attaching the actuator to the inertia load and the load fixture structure. All actuator parts were machined from 304 LC stainless steel.

The removable-seal-gland design used to provide easy seal configuration changes is an important feature of the actuator. A second design feature is the use of redundant seal installations on the piston and in the seal glands. Grooves for two seal configurations were placed on the piston to provide additional sealing in the event that the single elastomeric O-ring allowed an undesirable amount of leakage. Dynamic primary and backup rod seal grooves were also provided for this reason and to measure any leakage passing the primary seal.



Test Actuator Parts

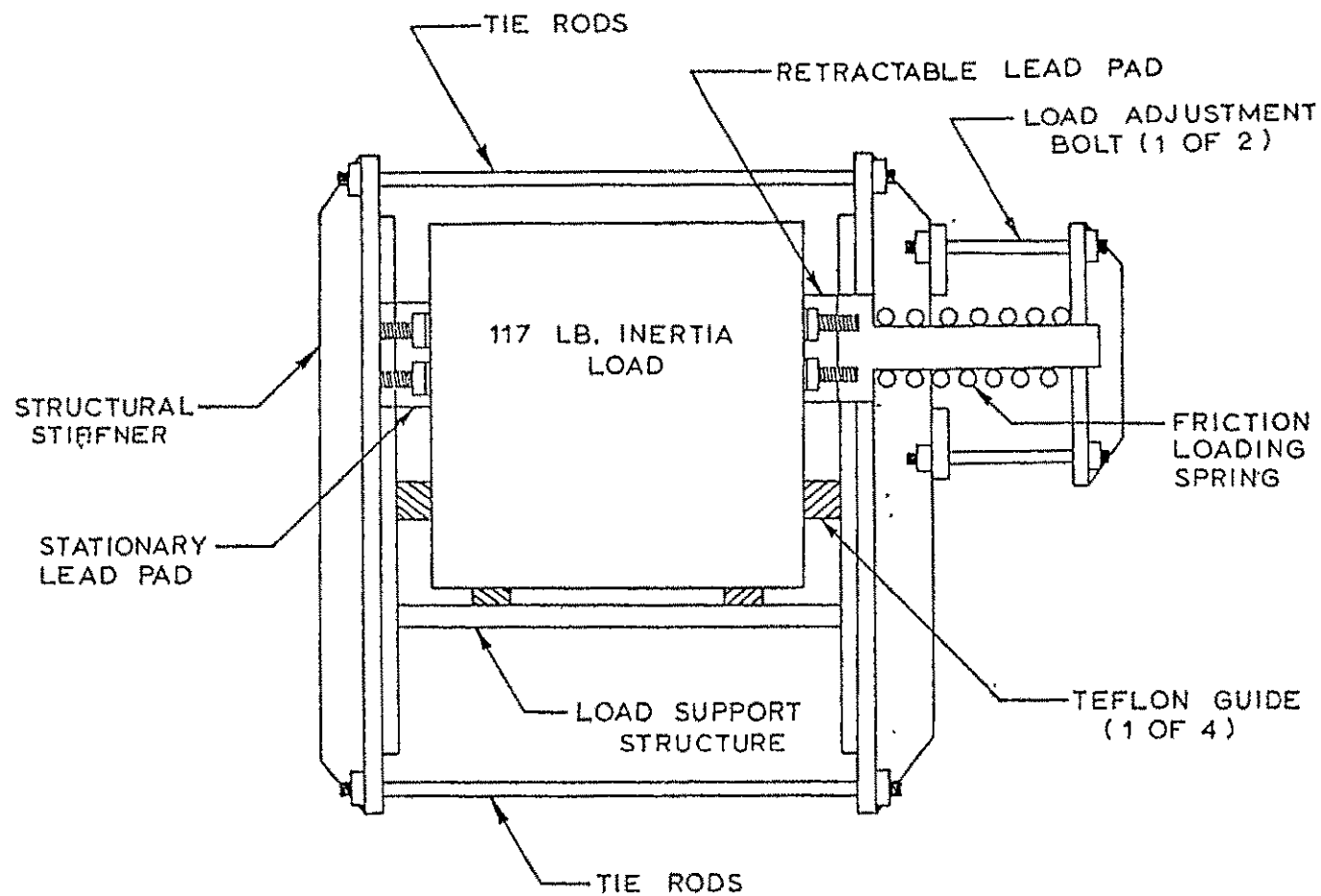
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## LOADING-DEVICE CROSS SECTION

The relationship of flight load torque to the equivalent test load in pounds is shown below.

| <u>Hinge Moment Component</u> | <u>Flight Torque (in.-lb)</u> | <u>Test Load (lb)</u> |
|-------------------------------|-------------------------------|-----------------------|
| Longitudinal Acceleration     | 1260                          | 180                   |
| Coriolis Damping              | 120                           | 17                    |
| Rotational Inertia            | 135                           | 19                    |
| Gimbal Friction               | 2400                          | <u>344</u>            |
|                               |                               | <u>560</u>            |

The accompanying figure shows the loading mechanism for simulating these conditions during the test. The rotational inertia load was represented during the test by the inertia of a sliding block weighing 117 pounds. The remainder of the application load requirement was simulated with a friction-loading device, and the friction load was applied to the sides of the mild steel inertia load by a spring-loading mechanism. A single spring loaded one of two lead pads against the inertia load, and a second pad was rigidly installed on the opposite side of the load in line with the spring. The choice of lead on steel provided a friction factor of 0.95, which was constant in both static and dynamic conditions.



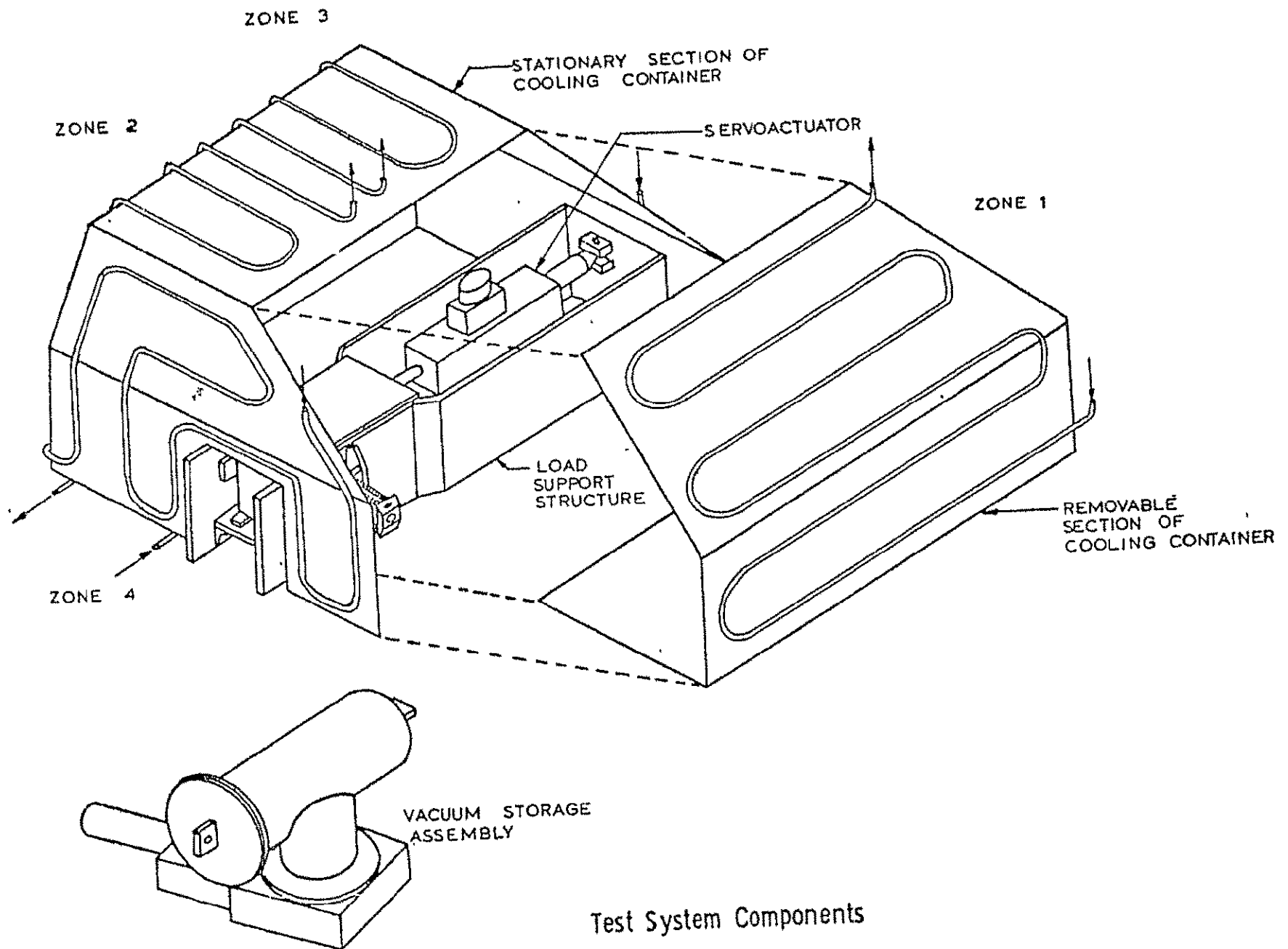
LOADING DEVICE CROSS SECTION

## FACILITY COMPONENTS

Special facility items constructed to support test operations were the cooling container, vacuum storage assembly, and load-support structure (see accompanying figure). A cooling container was constructed to isolate the test system and load apparatus within a temperature environment controlled by the container. Use of the container was particularly important when a hard-vacuum environment and low temperature were desired, where radiation was the primary means of thermal control.

The load-support structure was provided to support the inertia load and to provide attachment points for the friction spring-loading mechanism. This structure is the shape of a modified U-section. Extensions from the above structural members hold the servoactuator in correct alignment with the inertia load and carry the weight of the reservoir and filter in the operational test system — only the sides of the U-section are extended. The space between the extended sides is wider than the space at the inertia load-support section to accept installation of the vacuum container for operational testing of the vacuum-stored servoactuator.

The vacuum container was constructed to store a hydraulic servoactuator in a hard-vacuum environment for 5 months and subsequently to operate the actuator as part of a system in the vacuum environment. The container is an 8-inch inside-diameter cylinder with one end sealed and long enough to contain the test servoactuator in the retracted position. A 6-inch-diameter port with a connecting flange was provided as an outlet to the vacuum pumps. The vacuum container is separated from the pumps by a gate valve used to seal the container during transfer from the pumping station.



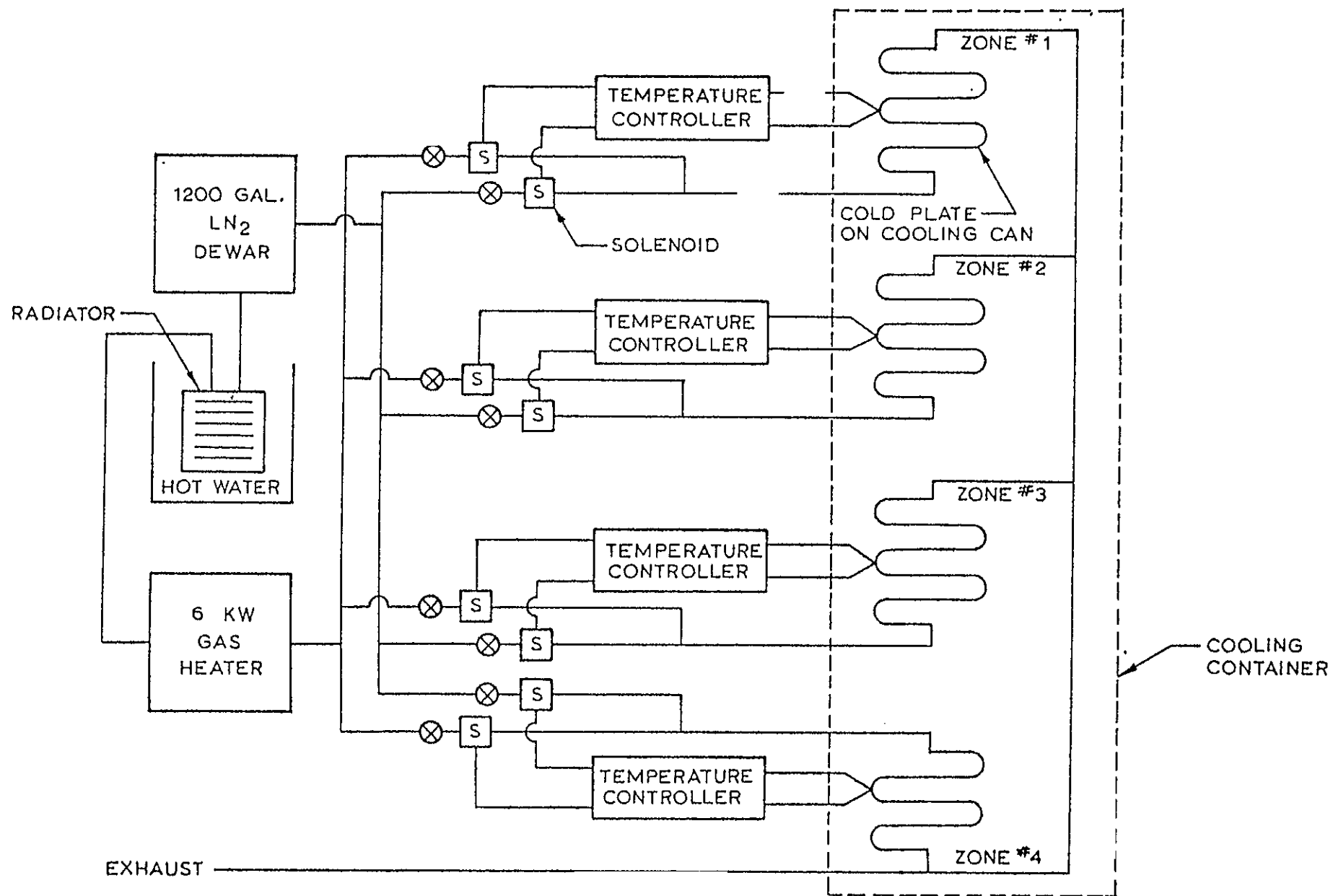
Test System Components

D2-04076-1

## AUTOMATIC COOLING SYSTEM

The controls for the automatic cooling system are shown schematically on the accompanying figure. The controls were assembled to assist in temperature stabilization of the test system.

A copper-constantan thermocouple sensed the temperature of each cooling zone on the cooling container. These temperatures were compared to a desired temperature for each zone by the controllers. The desired zone temperatures were established to maintain near-constant temperature distribution inside the container. Each temperature controller actuated solenoids that allowed either liquid nitrogen or warm gaseous nitrogen to be delivered to the cooling coils of a particular cooling zone. Liquid nitrogen was allowed to flow when additional cooling was required and gaseous nitrogen when heating was required. The outlets from the cooling coils for all zones were brought into a common line and exhausted from the space-simulation chamber.



AUTOMATIC COOLING SYSTEM  
DP-84076-1

## TEST PROGRAM OUTLINE

The purpose of this hydraulic-system test evaluation was to establish the ability of hydraulics to operate in low-temperature and vacuum environments. The test program was also a means of determining the need for or extent of the future development to provide flight-quality hydraulic-actuation space systems. Secondary objectives of the program were to investigate the ability of a hydraulic system to function at the high temperatures encountered in space and to withstand nonoperational thermal cycling between high and low temperatures. The chronology of testing accomplished during the program is shown on the accompanying figure, which outlines the basic tests, the environment present during the tests, the test duration, and system operational sequences performed during the tests. The major tests conducted were:

- 1) Low-temperature performance test;
- 2) Hard-vacuum storage test;
- 3) Simulated flight test;
- 4) High-temperature test;
- 5) Thermal-cycling test.

Table 8 - Test Program Outline

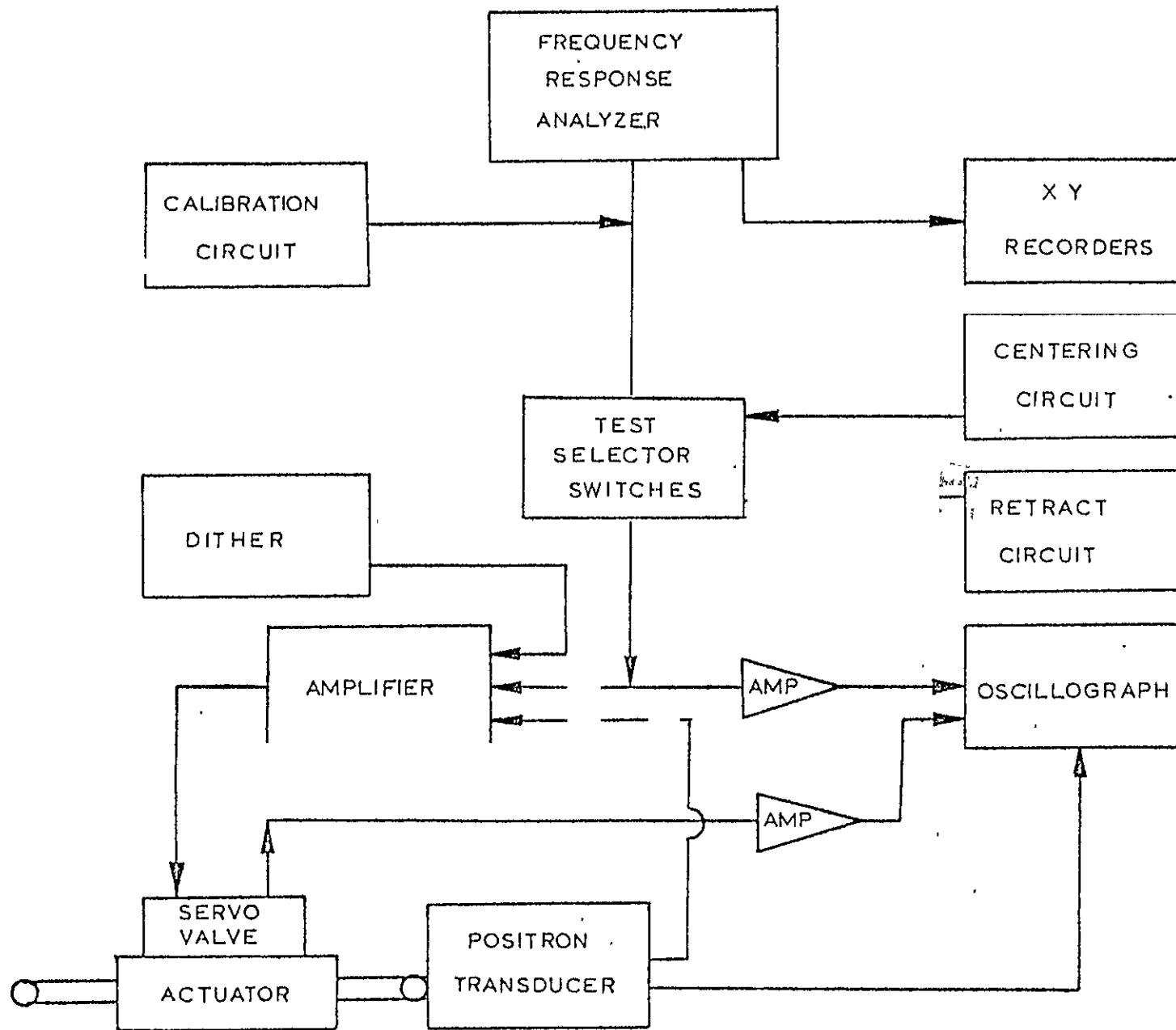
| Test                                             | Temperature Environment (*F) | Pressure Environment     | Test Duration | Actuation Duration (minutes) | Number Test Runs | Fluid Pressure (psi)  | Actuation Frequency (cps) | Fluid Temperature (*F) |
|--------------------------------------------------|------------------------------|--------------------------|---------------|------------------------------|------------------|-----------------------|---------------------------|------------------------|
| <u>Low-Temp Perf</u><br>Servoactuator perf       | 75 to minimum fluid temp     | Atm                      | 2.5 min       | 2.5                          | 10               | 3000                  | 0.5-6                     | 75 to minimum operable |
| <u>Hard-Vacuum Effects</u><br>Storage            | 75                           | Less than $10^{-6}$ torr | 5 mo. max     | None                         | 1                | 20-0                  | ---                       | 75                     |
| Servoactuator perf                               | 75                           | Less than $10^{-6}$ torr | 2.5 min       | 2.5                          | 1                | 3000                  | 0.5-6                     | 75                     |
| <u>Combined Environ Perf</u><br>Simulated flight | Min fluid operation          | Near $10^{-8}$ Torr      | 6 days        | 2 @ 2.5                      | 1                | 10 Store<br>3000 act. | 0.5-6                     | Minimum operable       |
| High-temp test                                   | 275                          | Atm                      | 2.5 min       | 2.5                          | 1                | 3000                  | 0.5-6                     | 275                    |
| <u>Thermal Cycling</u>                           | -240 to 275                  | Atm                      | 100 hrs       | None                         | 1                | 10                    | ---                       | -240 to 275            |

## Test Program Outline

## DATA-ACQUISITION SCHEMATIC

The accompanying figure shows the basic instrumentation common to all tests. The instrumentation was used either to control operation of the test system or to measure the operational characteristics of the test system. System operation was controlled through the circuitry associated with the servoloop operational amplifier and the frequency-response analyzer. The sinusoidal signal generated by the analyzer was modified by the load-position feedback signal from the actuator, the dither signal, and the amplifier gain to provide the command input to the test system. The frequency-response analyzer also computed the amplitude ratio and phase angle between the command and actuator outputs, transferring them to the X-Y recorders. The command output signals were duplicated on an oscillograph to provide backup instrumentation to the X-Y recorders. Special circuits were used (through a switching mechanism) to retract the actuator during environmental soaks, to center the actuator at the initiation of an operational sequence, and to calibrate all instruments before a test, without disturbing the actuator position.

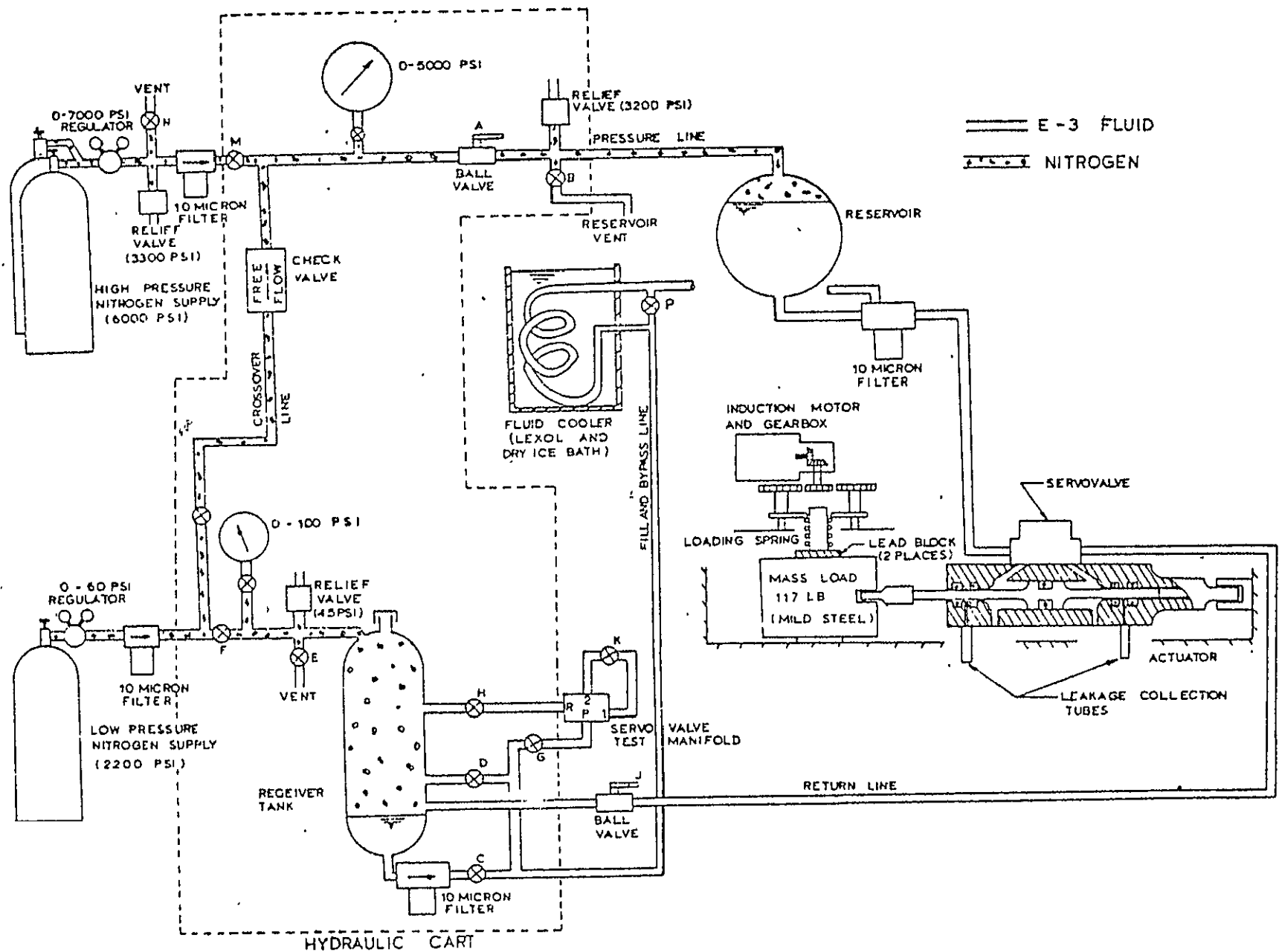
# DATA ACQUISITION SCHEMATIC



## TEST-SYSTEM SCHEMATIC

The schematic of the fluid loop used to perform the operational portions of the test program is shown on the accompanying figure. The equipment at the right, including the reservoir, servomotor, filter, and loading mechanism, comprise the single-pass system under test. This equipment was placed in an environmental chamber during the tests. The remainder of the equipment was used to assist blowdown of the reservoir, to collect the E-3 fluid after blowdown, or to assist refill of the reservoir. The fluid cooler was added to the system to precool the E-3 fluid during the reservoir refilling operation to shorten the time required to stabilize the reservoir at low environmental temperatures.

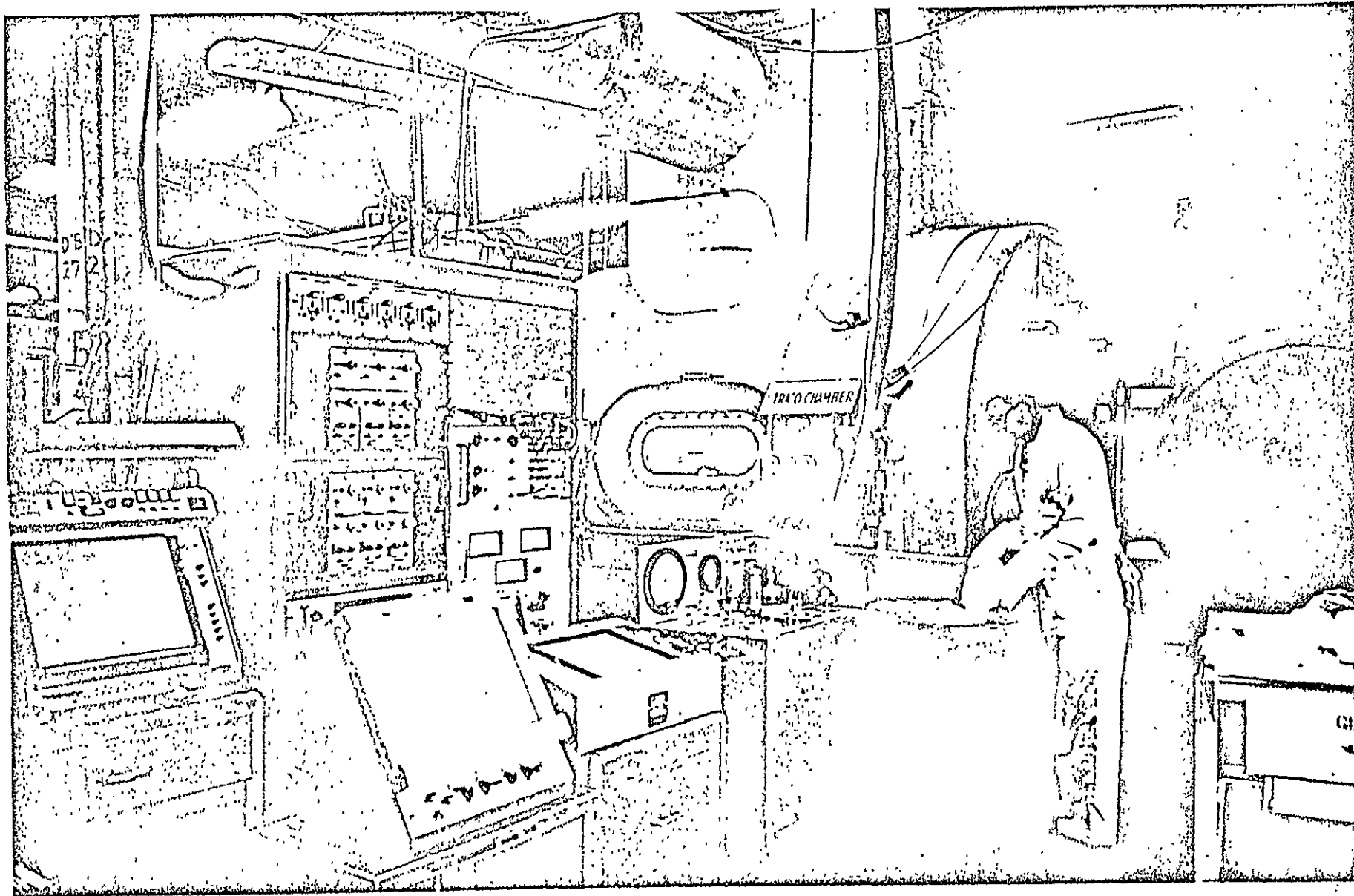
Each blowdown operation was conducted by regulating the pressure from the 6000-psig dry-nitrogen storage bottles to provide 3000 psig to the reservoir. The pressurized E-3 was used to move the actuator, as commanded by the servovalve, and was then discharged into the receiver tank. To refill the reservoir, the receiver tank was pressurized with low-pressure nitrogen and the E-3 was passed through the fluid cooler to the reservoir. The servo checkout loop was used to evaluate the servovalves; E-3 was used as the test fluid. Pressure to the servovalve test manifold was supplied through the bypass line.



Test System Schematic

## LOW-TEMPERATURE TEST FACILITIES AND INSTRUMENTATION

The accompanying photograph shows the oscillograph, X-Y recorders, instrument racks, and the supporting portion of the hydraulic loop described previously. This installation, as shown, was used during the low-temperature test series. The blowdown portion of the hydraulic loop was installed in the large chamber in the background. Dry ice is being added to the fluid cooling bath in preparation for a test.

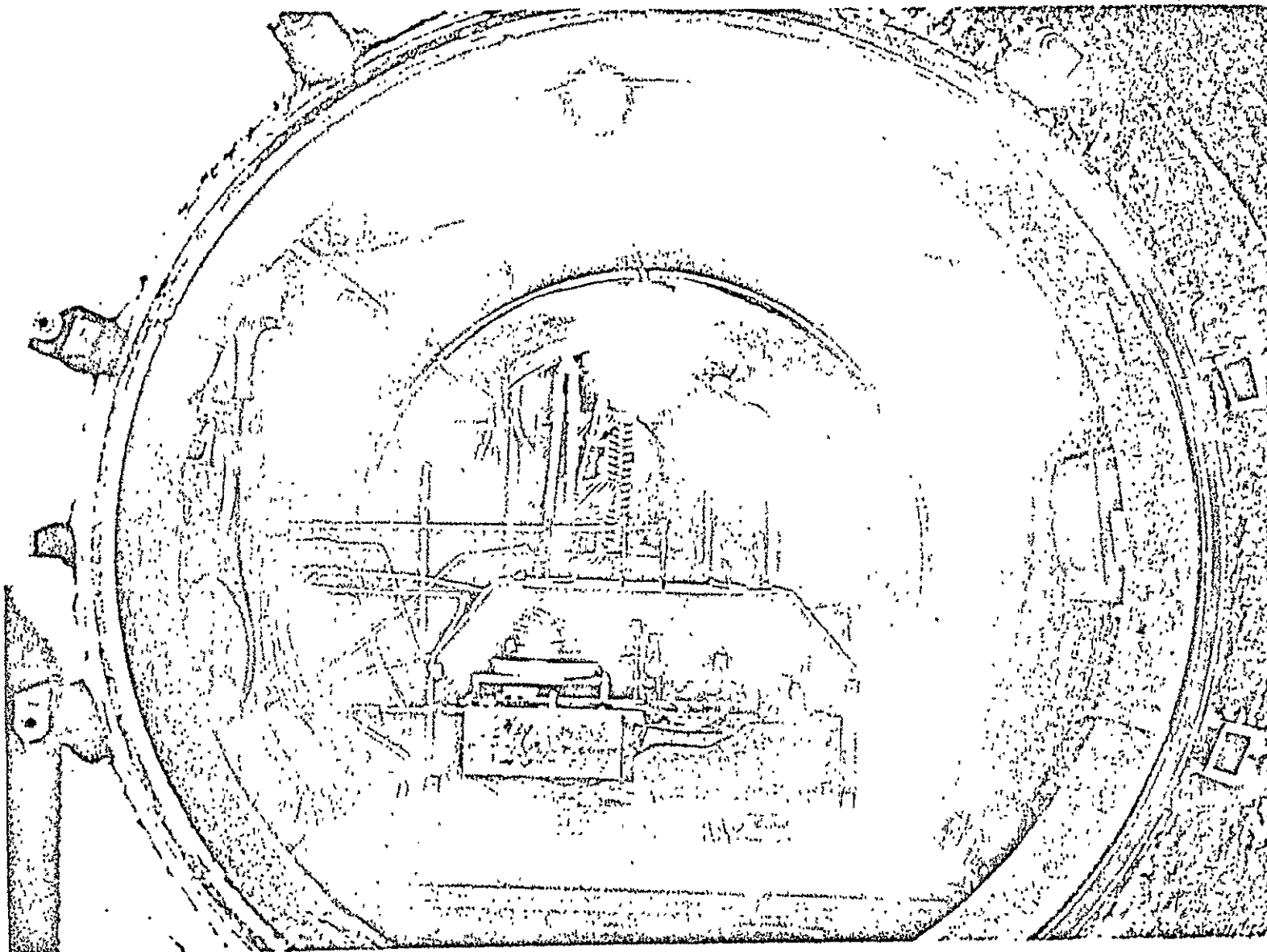


Low Temperature Test Facilities & Instrumentation

## TEST-SYSTEM INSTALLATION FOR LOW-TEMPERATURE TESTS

The low-temperature tests were conducted in the Boeing strato chamber shown on the accompanying figure. This chamber is 7 feet in diameter by 12 feet long and has a vertical section 5 feet in diameter extending 4.5 feet above the roof of the horizontal section of the chamber. The entrance door to the chamber is 5 feet in diameter. The chamber has a low-temperature capability of  $-105^{\circ}\text{F}$ , which is obtained by circulating chamber air through refrigeration coils. Temperature within the chamber is regulated automatically through set temperature controllers that turn on and shut off airflow, as needed, through the refrigeration coils. The refrigeration system consists of four stationary compressors that have a total output capacity of 11.4 tons of refrigeration using Freon 12. The low-temperature hydraulic system is shown installed in the strato chamber, with the refrigeration coils and air blower in the background.

Temperatures below  $-105^{\circ}\text{F}$  were stabilized by feeding liquid nitrogen from outside the chamber to the coils of the cooling container designed to cool the test system. The stationary portion of this container is shown surrounding the test equipment.



Test System Installation Low Temperature Tests

## OPERATIONAL TEST SYSTEM

The accompanying photograph shows the detailed installation of the servoactuator, filter, load, and reservoir inside the copper cooling container. The reservoir is at the upper left. The filter, upper center, is instrumented for fluid temperature and pressure measurements, the pressure transducer being wrapped with heating tape for use at extremely low temperatures. The servoactuator is instrumented for fluid-temperature measurement at the return port and for differential-pressure measurement across the piston. The fan at the lower right provided convection currents within the closed cooling container.

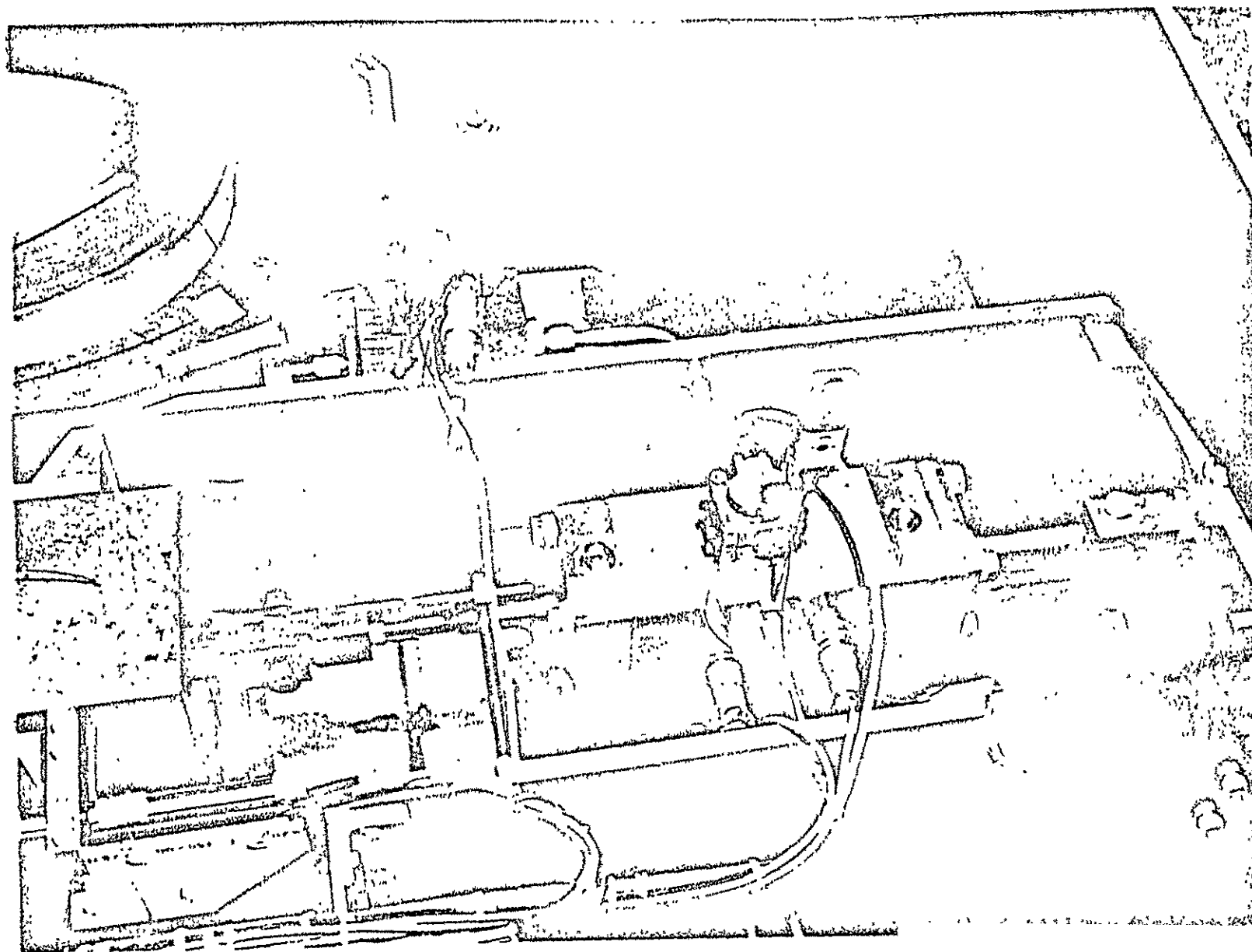


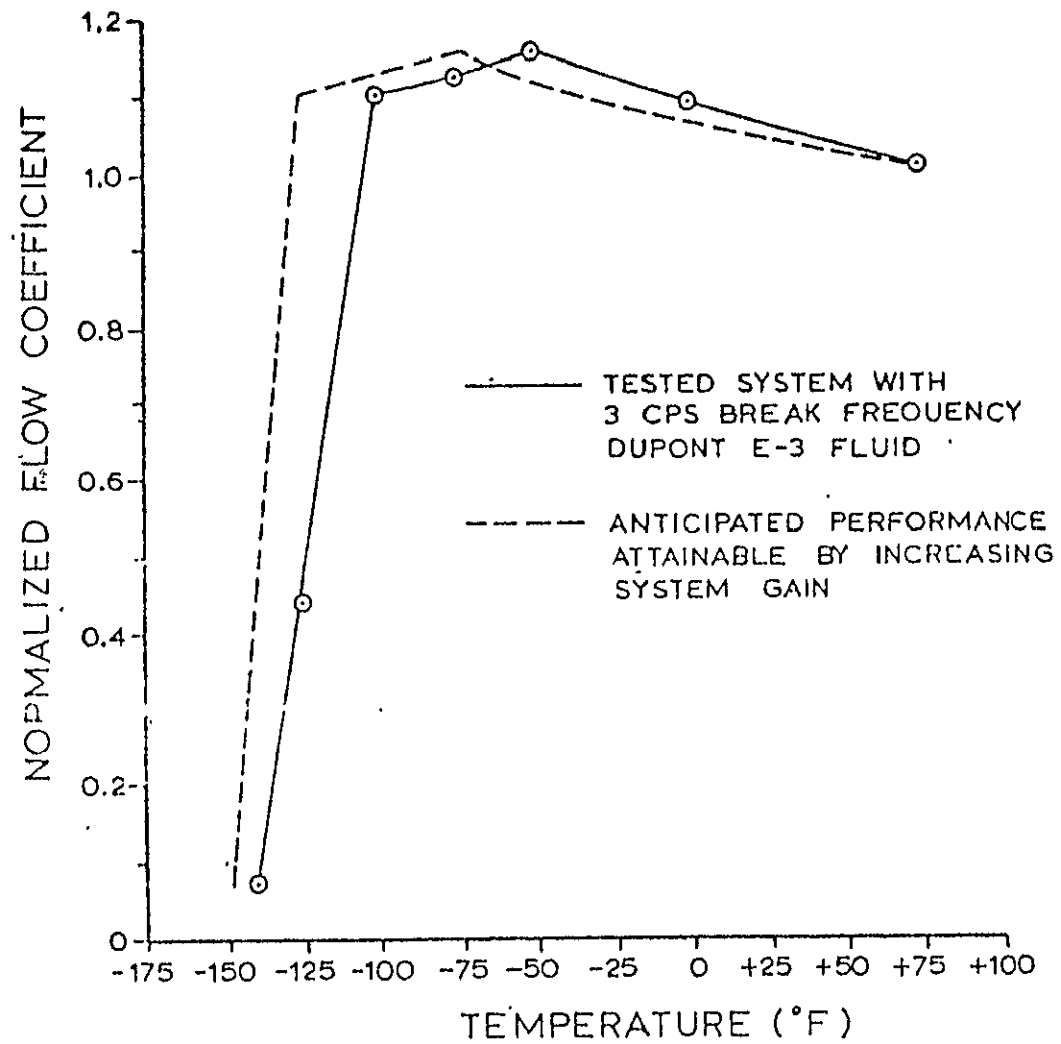
Figure 42: Operational Test System.

## FLOW COEFFICIENT VERSUS TEMPERATURE

The low-temperature test results show that the DuPont Freon E-3 fluid flowed at temperatures as low as  $-165^{\circ}\text{F}$  under high pressure, although the data also shows a marked change in fluid performance at temperatures below  $-100^{\circ}\text{F}$ . A curve illustrating the effect of temperature on fluid performance is shown on the accompanying figure.

The flow coefficient (K) was computed from the test data recorded on the oscillograph. The figure shows that as the fluid temperature is decreased, K becomes larger than the value at  $+75^{\circ}\text{F}$  until the temperature is lower than  $-100^{\circ}\text{F}$ , at which point K decreases very rapidly. This variation in K is also seen in the test system closed-loop frequency-response data. The sudden break in the plot of K versus temperature is attributed to fluid viscosity, which increases rapidly as the temperature decreases. By increasing the system gain, thereby increasing the break frequency, acceptable response at temperatures down to approximately  $-125^{\circ}\text{F}$  would be expected throughout the frequency range evaluated in the test program. This is shown by the dotted line on the accompanying figure.

# FLOW COEFFICIENT VS. TEMPERATURE

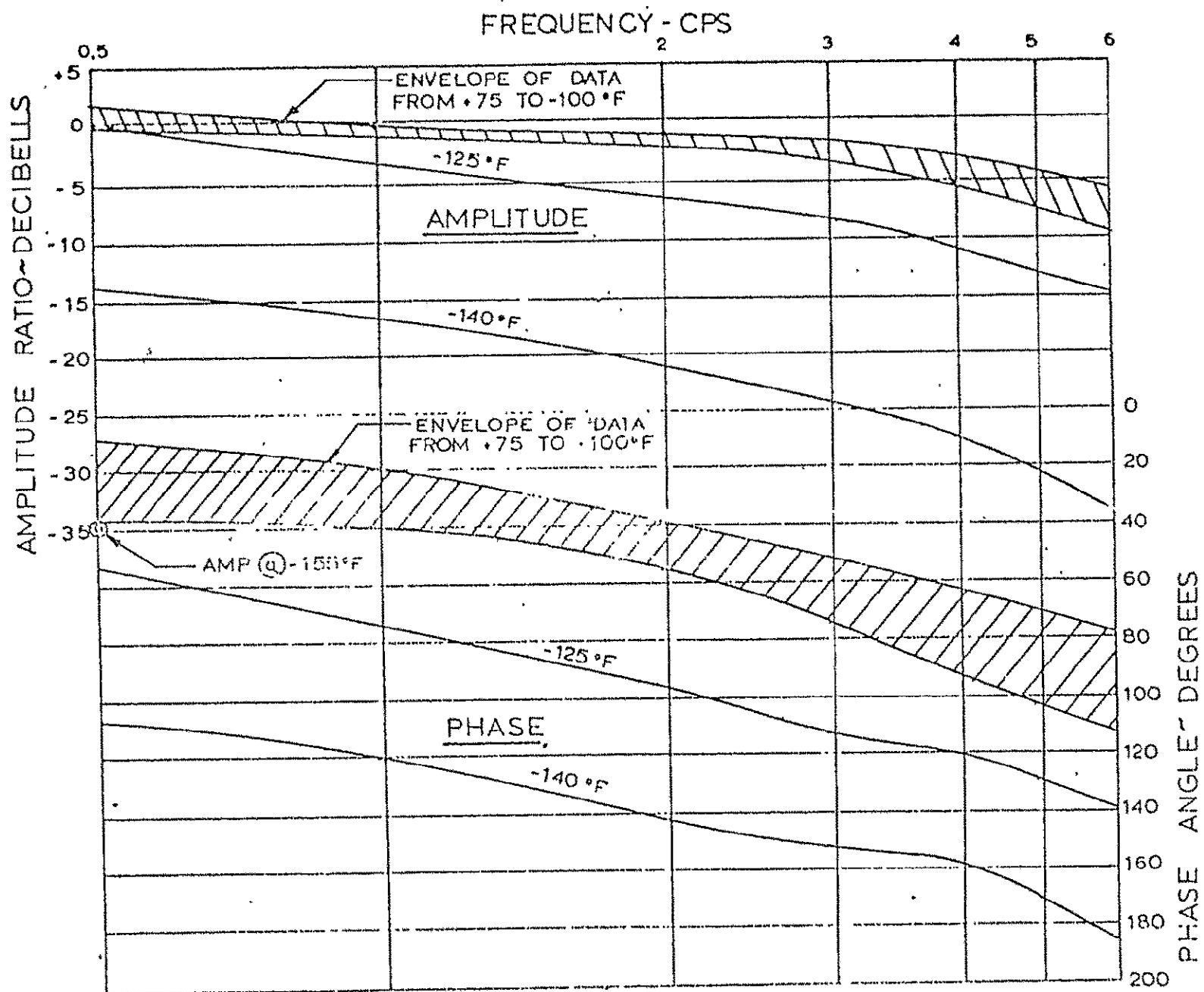


$$\text{FLOW COEFFICIENT} = K = \frac{Q}{I \sqrt{\frac{p_s - p_L}{\rho}}}$$

## SYSTEM RESPONSE (+75 TO -140°F)

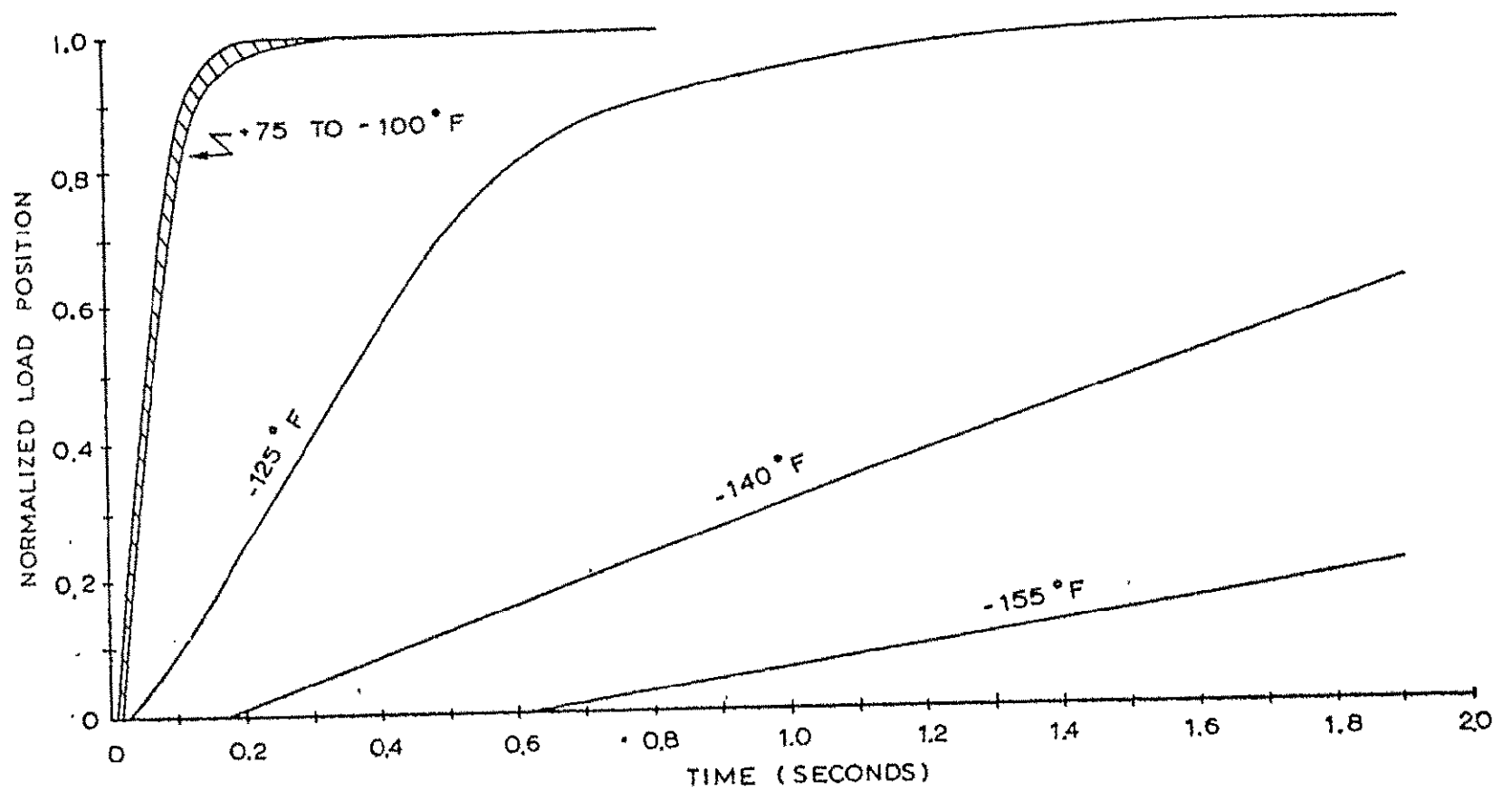
The system closed-loop frequency-response data from +75 to -140°F illustrates the conclusions drawn from the flow coefficient relationship to temperature discussed previously. The improvement in system response expected at these tested frequencies, using an E-3 fluid system with a higher gain, is that the -125°F curves would fall within the data envelopes shown for the range of +75 to -100°F. It was not expected that the improvement would provide much more than a 5- to 10-db increase in amplitude ratio or a 30- to 40-degree improvement in phase angle at 6 cps. This improvement still would not allow use of the E-3 fluid system at -140°F except for those tasks that do not have critical performance requirements.

# SYSTEM RESPONSE, + 75 TO -140°F



## ACTUATION-SYSTEM LOW-TEMPERATURE STEP RESPONSE

The system step response at low temperature is shown on the accompanying figure. This data was obtained during an initial operational command to the system to center the actuator from a retracted position. The time required for the actuator to reach the centered position was a function of the viscosity of the fluid and the effects of viscosity on fluid flow through the critical orifices of the servovalve. The correlation between the E-3 and MIF-II-5606A system test data indicates that further study should be devoted to analyzing the possibility of a general relationship between fluid viscosity and system performance. In conjunction with such an evaluation, the effects of fluid viscosity and temperature on servovalve performance, separate from a system, should be investigated. Such an investigation could lead to design techniques that would extend the low-temperature capability for flight control applications.



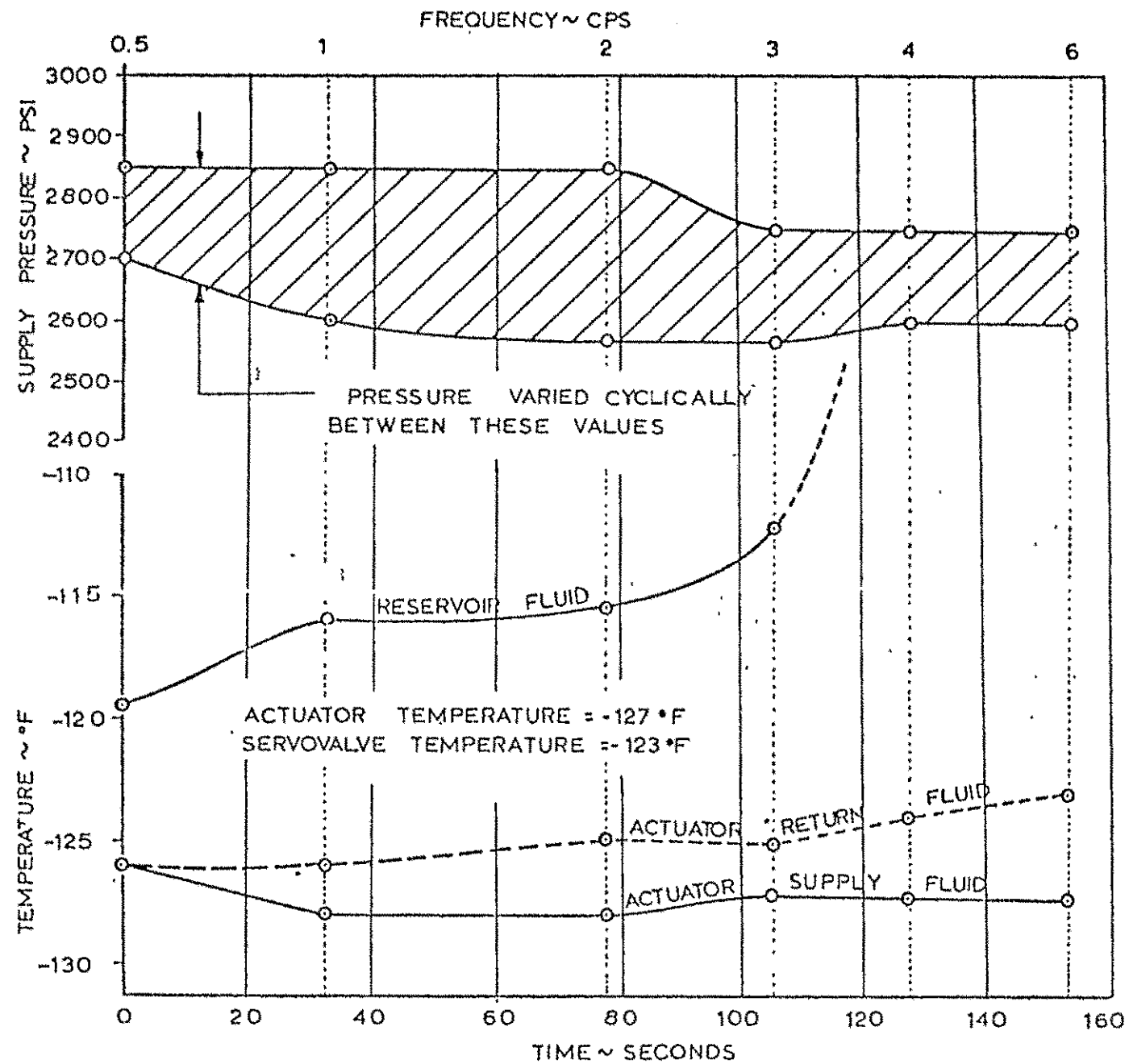
Actuation System Low Temperature Step Response

## FLUID PRESSURE AND TEMPERATURE DURING TYPICAL LOW-TEMPERATURE TEST RUN

The accompanying figure shows that fluid throttling during performance of the operational sequence tests produced little temperature rise within the system. The fluid temperature in the reservoir, measured by an immersion probe, showed some temperature rise induced by the warm high-pressure nitrogen used to blow down the reservoir. The temperature rise in the reservoir did not produce a significant rise in the temperature of the fluid passing through the actuator. The measurements of reservoir fluid temperature after approximately 100 seconds were considered invalid because the thermocouple was no longer immersed in oil.

The variation in supply pressure during the typical operational sequence is shown as a shaded band. This indicates a cyclic variation in the supply pressure observed during all tests below  $-100^{\circ}\text{F}$ . The pressure variations cycled at the same frequency as the actuator movement. Peak pressure coincided in time with the end of the actuator stroke, when the velocity was zero, and low pressure occurred when the actuator velocity was maximum. The supply pressure variation was caused by pressure drop through the micronic filter located upstream of the supply pressure transducer. Because the supply pressure did not show this effect at temperatures of  $-100^{\circ}\text{F}$  and above, the fluid viscosity at low temperature is concluded to be the cause. This establishes a serious problem that conflicts with the main purpose of filters in the system, that of contamination control. Further investigation of pressure drop in filters with high-viscosity fluids is advised before selecting flight application hardware.

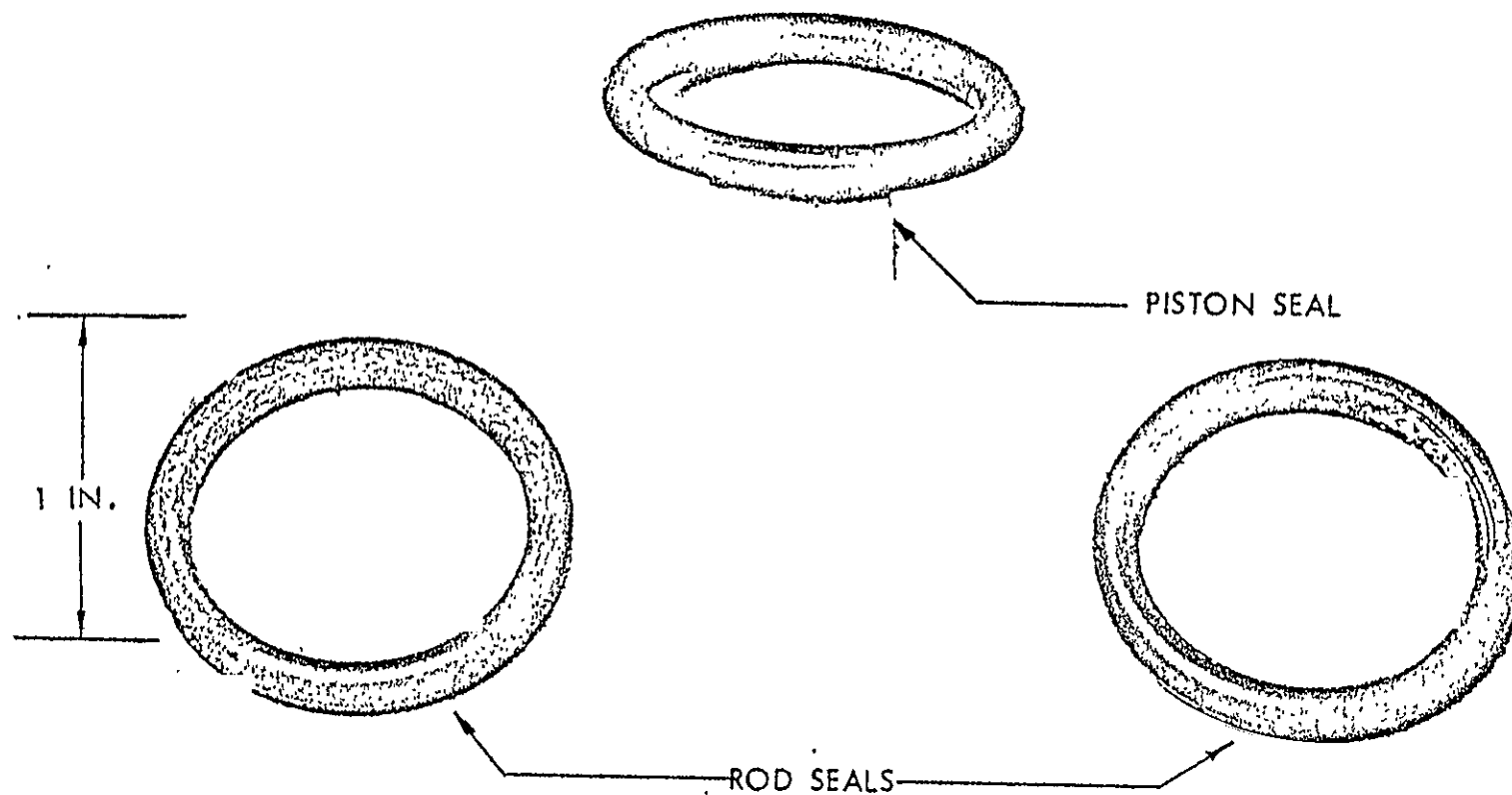
# FLUID PRESSURE AND TEMPERATURE DURING A TYPICAL LOW TEMPERATURE TEST RUN



## SEAL WEAR

Test servomotors were disassembled for seal changes or inspection three times during the test program. Serial Number 1 actuator was disassembled at the completion of low-temperature tests and at the completion of thermal cycling, and Serial Number 2 actuator was disassembled at the completion of vacuum testing. Wear on the seals (see accompanying figure) shows the wear discovered at each inspection. The black ring on the piston seal was a deposited chrome and nickel compound. The abrasion of the rod seals was judged to be partly caused by the sand contaminants found in the system.

Seal-evaluation results of the entire program reveal that potential fluid leakage past the seals and deterioration of seal materials in space environments were not as severe as expected. Although seal installations in the test actuator were designed with extra squeeze (20 percent), this compression was not the maximum that could be incorporated. No leakage was encountered from the 105 hours of thermal cycling between +275°F and -240°F, indicating compatibility of seals with a wide range of temperatures during inoperative storage. Leakage during storage at vacuum environment was well within a tolerable limit for elastomeric dynamic seals, but reduced the ability to test to the desired hard vacuum. It is concluded that the sealing conditions required to test at hard vacuums, greater than  $10^{-6}$  torr, will demand near-zero leakage past the elastomeric seals. This condition is not required for flight applications; therefore, the necessity for seal development will depend on the desire to test the effects of harder vacuums before flight usage.



Seal-Wear Actuator Number 1

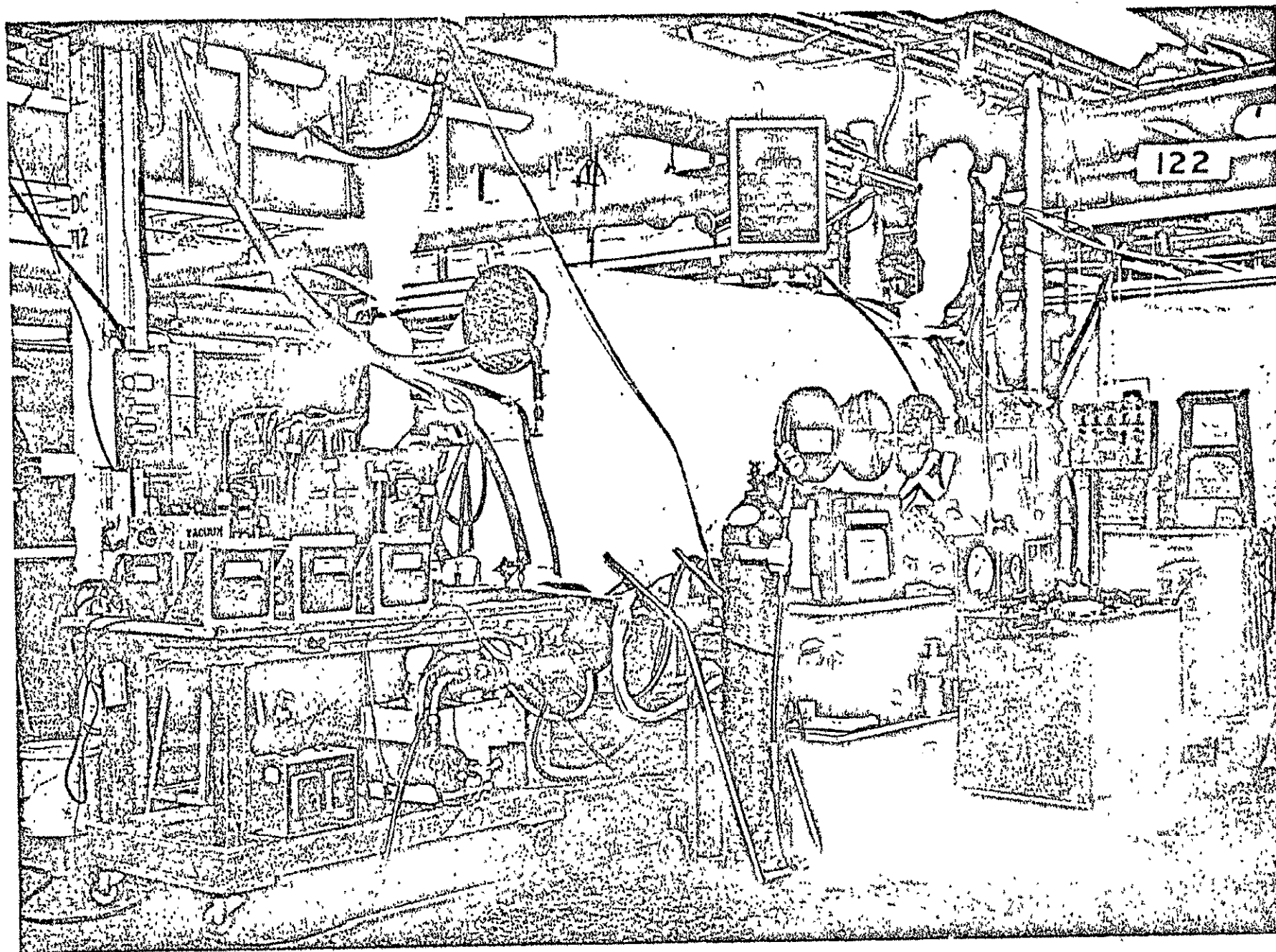
D2-64076-1

## SPACE-SIMULATION FACILITY

All of the system tests subsequent to the low-temperature series were performed in the space-simulator facility shown in the accompanying photograph. This chamber is capable of a vacuum of  $8 \times 10^{-9}$  torr when it is clean, dry, and empty. Vacuum is provided by a 35-inch-diameter, 50,000-liter-per-second, oil-diffusion pump. A liquid-nitrogen-cooled baffle prevents diffusion-pump oil from entering the test chamber.

The test system is cooled by radiation to the cold walls of the cooling container. The equipment positioned behind the simulator is the automatic cooling system, and the equipment to the right of the chamber is the hydraulic support equipment for operational tests. Tests conducted in this facility were the simulated flight test, high-temperature test, thermal-cycling test, and the operational portion of the vacuum tests.

The simulated flight test was performed following completion of the low-temperature tests. The simulated flight test investigated intermittent use of the system during exposure to low-temperature and hard-vacuum environments. The results of the simulated flight test demonstrated that a hydraulic-actuation system using Du Pont E-3 fluid with thermal protection to  $-140^{\circ}\text{F}$  could be used a number of times in space environments, with periods of inactive system storage between usage. The hard vacuum and low temperature caused only normal seal leakage, and the amount of fluid lost did not affect the simulated mission. The effects of the simulated flight test did not produce a permanent change in the operational capability of the system.



Space Simulator Facility

D2-84076-1

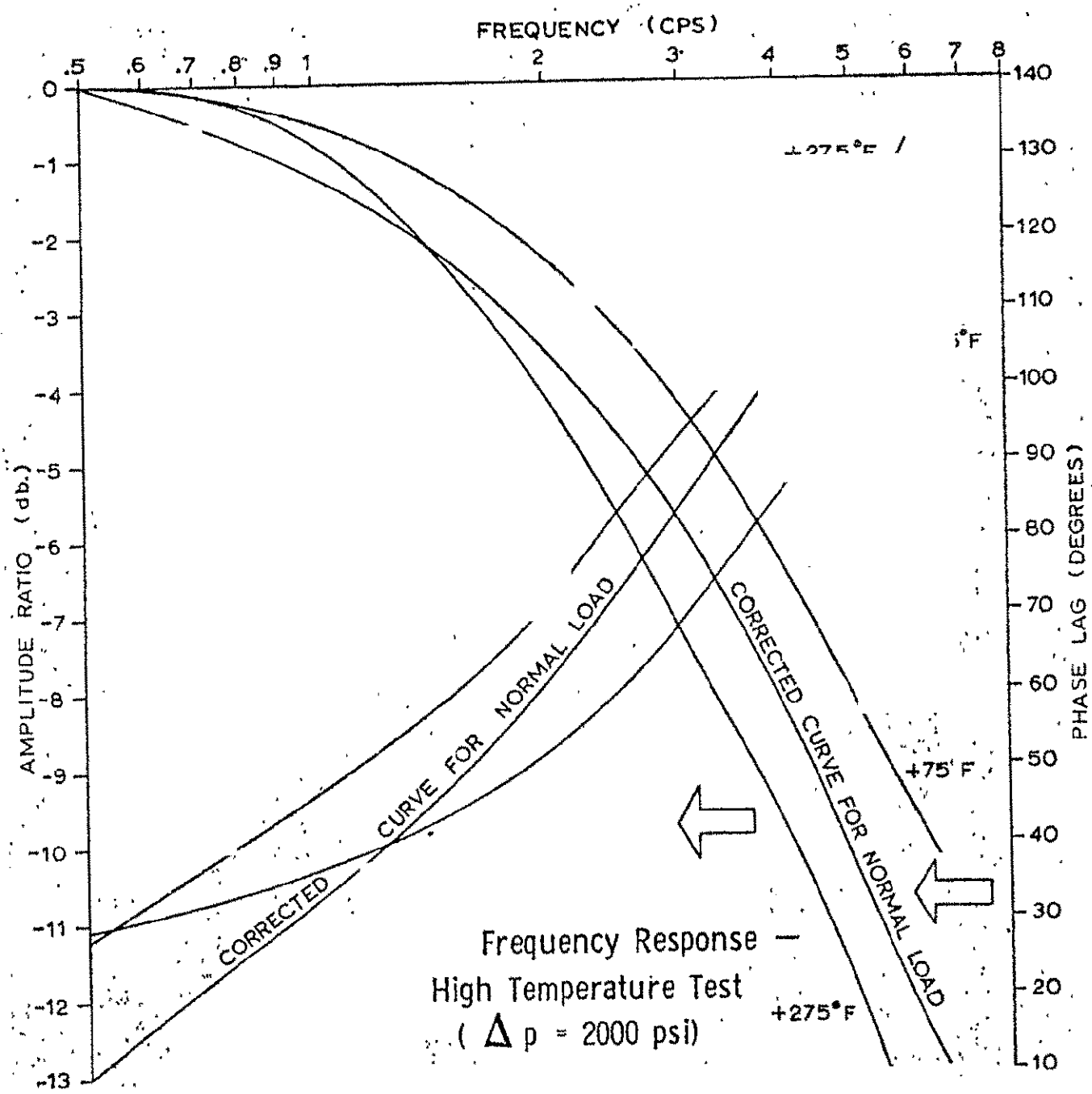
## HIGH-TEMPERATURE TEST

The high-temperature portion of the combined effects test was performed to determine the effects of a simulated maximum lunar-surface temperature environment on the hydraulic test system. The results of this test are shown on the accompanying figure. The requirement for hard vacuum is not imposed during this test, because exposure for short durations has no effect on system performance.

The test was conducted by operating the system and obtaining data points at discrete actuation frequencies, rather than by scanning the frequency range. This procedure was used because the scanning test could not be completed with the fluid capacity in the system, which was due to the low viscosity at high temperature.

Because of the differential expansion of components in the loading system at high temperature, the load, desired to be 1400 psi across the actuator piston, could not be reduced below 2000 psi. The high-load and normal-load conditions were compared during tests at room temperature, and these relationships were used to correct the high-temperature test data to provide a curve for the normal-loading condition. The system amplitude ratio and phase lag curves at the normal-loading condition are comparable to low-temperature test performance characteristics for a temperature between -100 and -125°F.

Performance of the test system during the operational sequence at high temperature was satisfactory. Seal leakage was 0.8 cubic centimeter at the rod-end gland and 0.6 cubic centimeter at the head-end gland during the 3-minute test. This amount of leakage is insignificant for the simulated application.

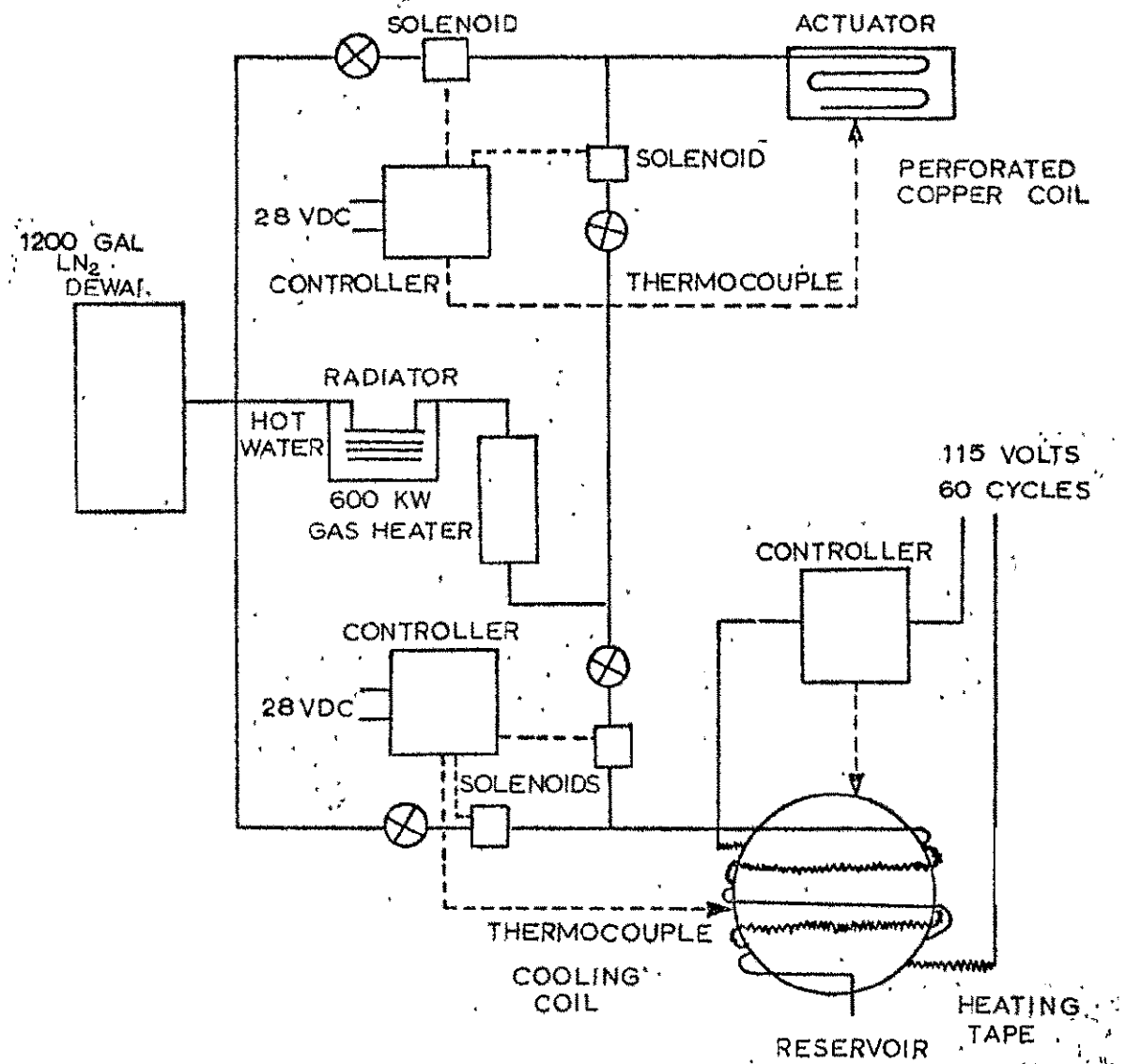


## THERMAL-CYCLING TEMPERATURE CONTROL

The thermal-cycling test required both high cooling and high heating rates to control the desired temperature fluctuations. The temperature-regulation system used for the thermal-cycling test is shown on the accompanying figure. The servoactuator was wound with a coil of perforated copper tubing and wrapped with a blanket of insulating material. Liquid nitrogen or hot gaseous nitrogen is supplied to the copper tubing by direction of a temperature controller. Temperature sensing for the controller is provided by a copper-constantan thermocouple on the actuator. Thermal control of the hydraulic-fluid reservoir was accomplished by winding the reservoir with alternate coils of copper tubing and electrical-resistance heating tape. The heating tape was used because the desired temperature rise rate could not be attained using only gaseous nitrogen, due to the large mass of fluid requiring heating. Temperature controllers were used to direct the flow of liquid or gaseous nitrogen in the tubing and to control current flow in the resistance heating tape. The short lengths of tubing connecting the reservoir to the filter and the filter to the actuator acted as conductors to heat or cool this portion of the test system.

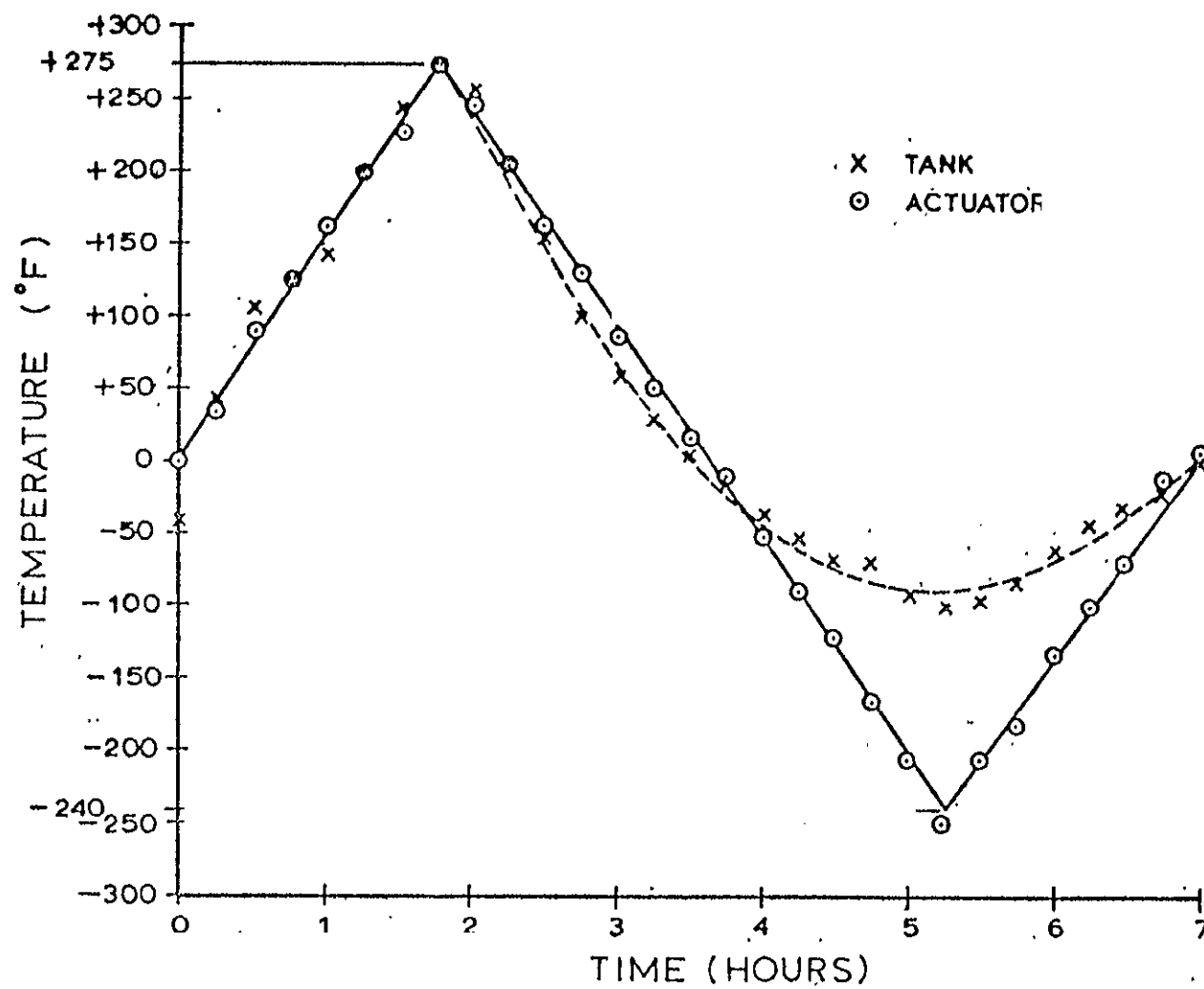
## TYPICAL THERMAL CYCLE

Some experimenting was necessary to determine the proper temperature controller settings to achieve the time-temperature relationships shown on the accompanying figure. The actuator body temperature was used as the control point for the test, and the reservoir was programmed to follow this control as accurately as possible. Temperatures throughout the system were monitored every 15 minutes on a multiple-point stamping recorder. Twenty thermal cycles were performed; the first five cycles were used to adjust the temperature controllers and to determine temperature reversal times. The last 15 cycles were performed in a continuous test. The operational characteristics of the system, both before and after thermal cycling, indicated that no permanent changes in the system resulted from the thermal cycling. No fluid was found in the leakage-collection tubes at the completion of the thermal cycling test.



THERMAL CYCLING TEMPERATURE CONTROL

## TYPICAL THERMAL CYCLE

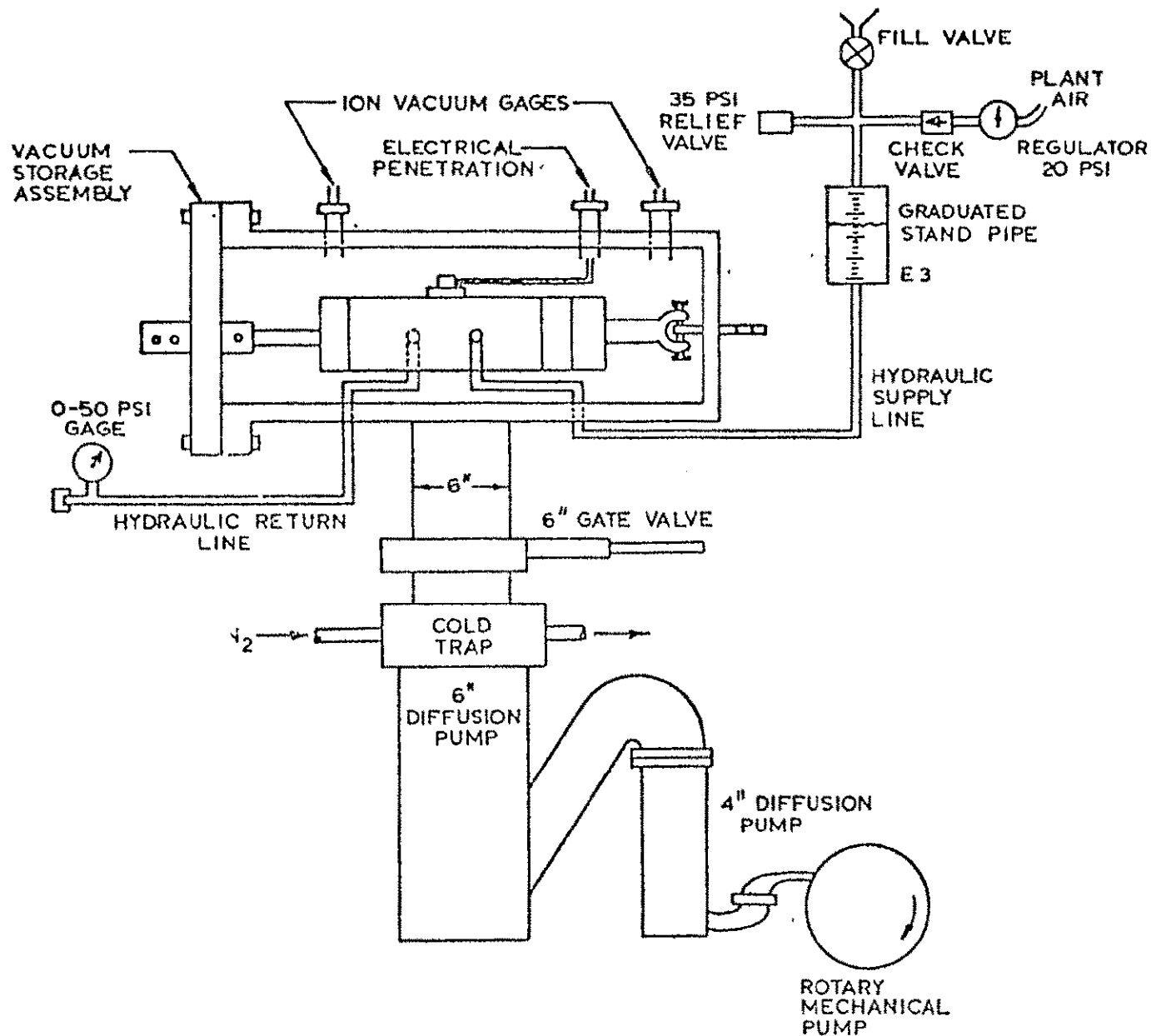


## VACUUM-STORAGE-STATION SCHEMATIC

The objective of the vacuum tests conducted in this contract were to determine the deteriorating effects of continuous long-term hard-vacuum exposure on elastomeric seals and other outgassing materials used in a space hydraulic system. The seriousness of the hydraulic-fluid leakage during such long-term storage was also evaluated. The vacuum-storage station used in the test is shown schematically on the accompanying figure.

The vacuum container previously described provided the means for isolating the servoactuator at hard vacuum during storage and during transfer and installation of the vacuum-storage assembly into the operational test-system load fixture. The vacuum-storage tests were performed using a small pumping system to maintain a hard-vacuum environment.

# VACUUM STORAGE STATION SCHEMATIC



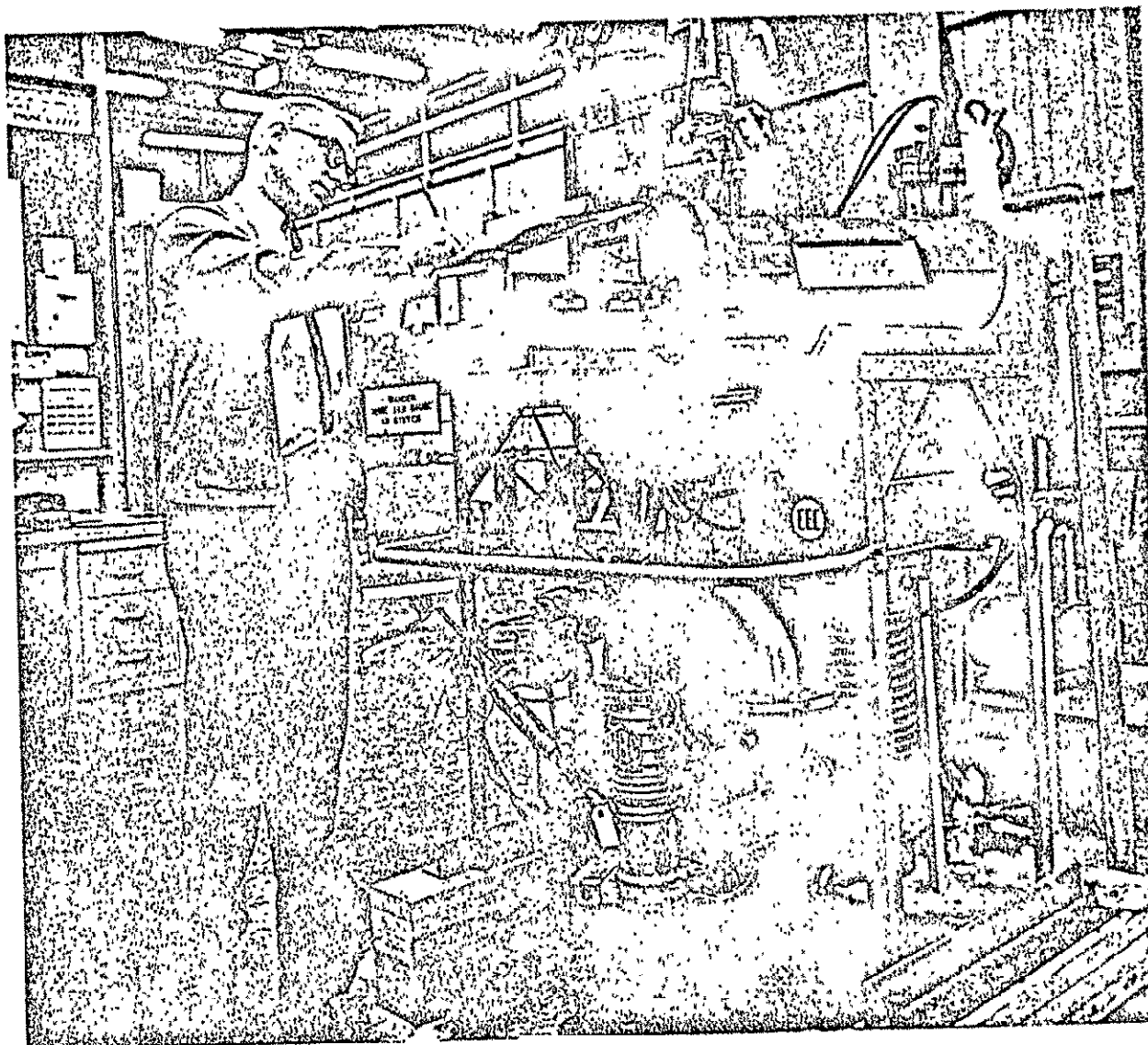
## VACUUM-STORAGE STATION

Vacuum pressure measurements were read from the readout instrument for the hot-cathode ionization gage installed in the assembly, as shown on the accompanying figure. The fluid standpipe was pressurized to 20 psi.

After 48 hours of storage, an increase in environmental pressure indicated saturation of the cold trap with condensed E-3 leakage. At the end of 5 days, the storage test was suspended to clean the cold trap and investigate this leakage problem. Pressurization of the fluid in the stored actuator was discontinued to improve the ability to maintain a hard vacuum. The E-3 fluid in the standpipe was vented to the atmosphere by leaving the cap on the pipe loosely threaded to ensure that the fluid would remain at atmospheric pressure.

The vacuum-storage test was restarted and the maximum possible vacuum maintained for 134 days. After completion of the storage portion of the test, the gate valve was closed and the storage assembly installed on the load fixture in the CA-4 test chamber in preparation for system operational testing. Extension of the actuator rod and attached storage container lid during the initial operational sequence equalized the vacuum environment in the CA-4 chamber with that of the storage container.

The long-term storage of an inoperative space hydraulic-actuation system in a hard vacuum of  $10^{-6}$  torr did not damage elastomeric seals or some insulating materials and did not permanently affect system operational characteristics.



Vacuum Storage Station

DP-84076

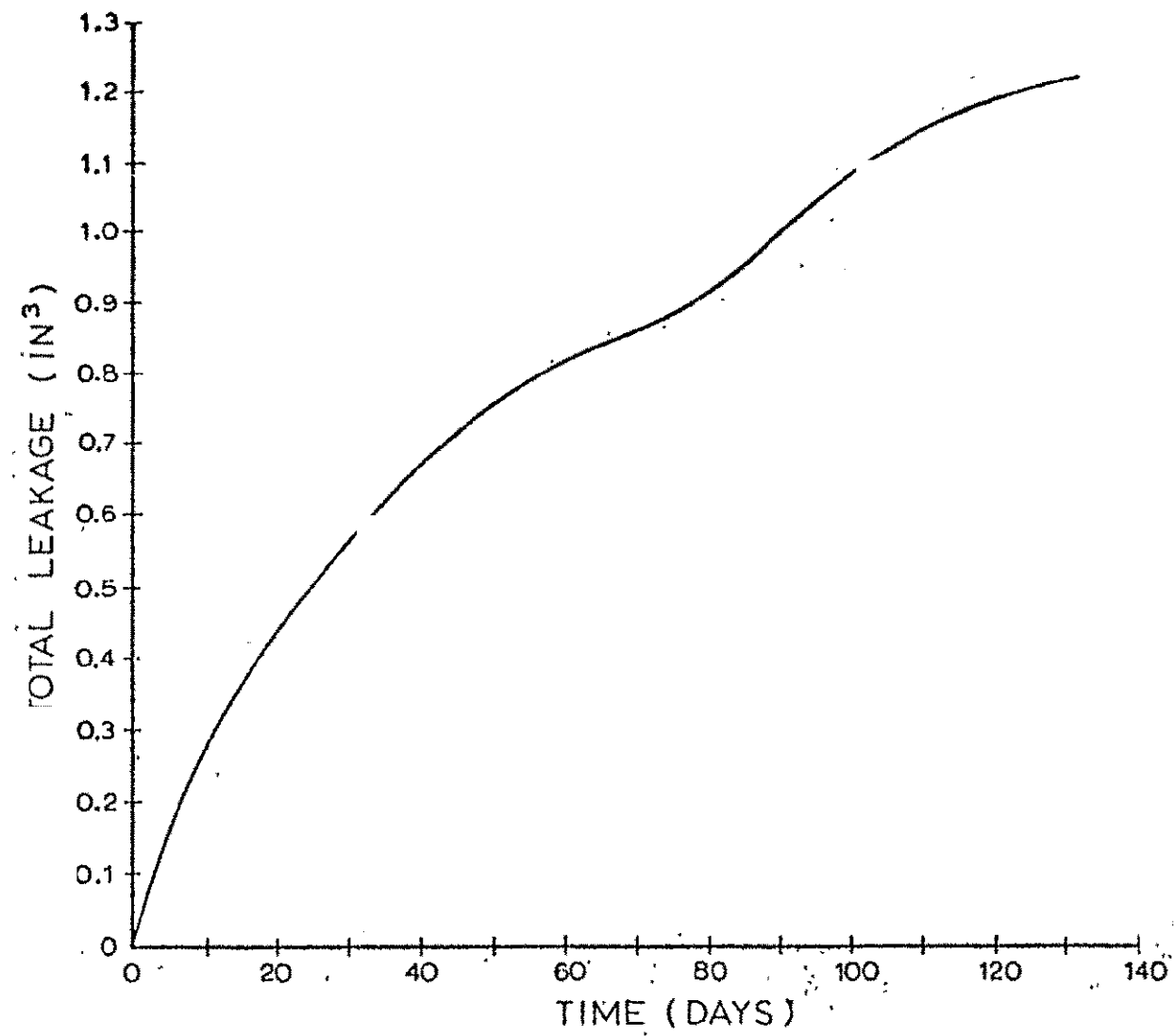
## VACUUM-STORAGE TEST FLUID LEAKAGE

This figure is a graph of the fluid lost during the continuous vacuum-storage period. Total fluid loss during the 134 days of vacuum storage was 1.23 cubic inches. Leakage rates at various times in the vacuum testing are shown in the table below.

| <u>Leakage</u>                                                       | <u>Fluid Pressure</u> | <u>Leakage Rate</u><br><u>(in.<sup>3</sup>/day)</u> |
|----------------------------------------------------------------------|-----------------------|-----------------------------------------------------|
| During 5-day checkout                                                | 20 psig               | 0.205                                               |
| Before adding redundant rod seal<br>to improve sealing               | 2100 nsig             | 2.05                                                |
| After adding redundant rod seal<br>(first day of continuous storage) | 14.7 psia             | 0.032                                               |
| Average for 134-day storage period                                   | 14.7 psia             | 0.0093                                              |

The ability to further evaluate hydraulic systems at harder vacuums will depend on the development of zero-leak dynamic seals. Such seals do not appear to be necessary for flight applications because the amount of leakage evidenced in the vacuum test is considered to be well within a tolerable limit. The vacuum-seal design requirements for vehicle-system PFRT chamber testing are, therefore, more severe than the design requirements for a flight application. Consequently seal development efforts must be related to the need for chamber testing at higher vacuums.

# VACUUM STORAGE TEST FLUID LEAKAGE



## FUTURE EFFORTS

### THERMAL PROTECTION ANALYSIS AND TEST

### RECIRCULATING HYDRAULIC SYSTEM DEVELOPMENT

1. LOW TEMPERATURE SERVICE
2. HIGH TEMPERATURE SERVICE

### ADVANCED FLUID AND SEAL DEVELOPMENT

1. IMPROVED VISCOSITY FLUIDS
2. ELASTOMER WEAR RESISTANCE
3. HARD VACUUM SEALING
4. NON-ELASTOMERIC SEALS

### ACTUATION SYSTEM DESIGN CRITERIA FOR SPACE APPLICATIONS