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TESTING OF RECIPROCATING SEALS FOR APPLICATION IN A STIRLING CYCLE ENGINE

J. F. Curulla and T. L. Beck
Boeing Commercial Airplane Company

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Prepared for
NATIONAL AERONAUTICS AND SPACE ADMINISTRATION
Lewis Research Center
Under Contract NAS 3-20612

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FIGURES (Concluded)

1.0 SUMMARY

It is necessary in the development of the Stirling cycle engine to provide an effective sealing system that will maintain internal gas pressure, minimize heater contamination due to excessive oil leakage, reduce internal friction and have long life capabilities. The primary objective of this program was to evaluate various single-stage reciprocating seal designs in the adverse Stirling engine environment. This was accomplished by: (1) definition of design/test requirements; (2) selection of seal designs that gave promise of meeting the engine requirements; (3) design and fabrication of a cyclic test rig to simulate the engine environment; and (4) performance of cyclic endurance testing.

Six seal configurations were selected for evaluation. These included the Boeing Footseal, the NASA Chevron polyimide seal, the Bell three-piece footseal, the Quad seal, the Tetraseal and the Dynabak seal. The Stirling engine seal requirements were obtained from the operating characteristics of Ford Motor Company's Stirling cycle engine. These included a maximum helium gas leakage of .002 cc/sec per seal for 1500 hours of testing at gas pressure of 1.22×10^7 PA (1750 psig), 7.19 m/sec peak rod speed (3000 rpm) and 408 K (275°F) seal temperature. A 2.22 cm (7/8 inch) diameter rod size was used in this evaluation.

Testing was accomplished on all six seal configurations. None of the seal configurations survived the adverse requirements mentioned above. A number of high temperature failures occurred in all seal configurations. Failures were also observed when the requirements of rod speed, gas pressure and seal temperature were relaxed. The longest seal test was 63.6 hours on the Bell seal at lowered limits of 2.16 m/sec peak (900 rpm) and 1.22×10^7 PA (1750 psig) and a starting seal temperature of 294 K (70°F). However, this seal had leakage rates up to .13 cc/sec during this test. Breakaway friction rates varied from 45.5 N (10 lb) per Boeing Footseal, 89 N (20 lb) per Bell seal, and up to 455 N (100 lb) per NASA Chevron seal. None of the other seal configurations were tested for friction.

It can be concluded from the results of this evaluation that none of the six reciprocating single-stage seals is acceptable for use on the Stirling cycle engine. It is recommended that any future single-stage seal investigation evaluate configurations that lengthen seal leakage path, minimize leakage path clearances with the reciprocating shaft and provide intermediate venting of the leakage path. In addition, the use of Teflon-filled material, such as fiberglass, bronze, graphite, could improve endurance life of seal configurations. It is further recommended that various cooling schemes be provided such as forced air or oil mist cooling of the rod.

2.0 INTRODUCTION

The concept of the Stirling cycle engine offers a significant opportunity to improve engine performance and efficiency over conventional automotive internal combustion engines. The Stirling cycle produces power by compressing a working gas (hydrogen) at low temperature and expanding it at high temperature. It is a closed cycle using the same working gas. Combustion takes place external to the enclosed volume, and the working gas is heated or cooled through heat exchangers. The ideal thermal efficiency of this cycle is significantly better than that of the current internal combustion engine. However, the development of this more complex engine requires design features that are an extension of the state-of-the-art, one of which is fluid sealing. The successful identification of reciprocating single-stage seal designs improving seal leakage, power loss, and life over elastomeric O-rings in the adverse environment of the Stirling cycle engine is a necessary step in making this engine a viable competitor for highway vehicle applications. To ensure the selection of a satisfactory seal, it is necessary to screen the various possible seal designs and to validate the choice by demonstrating seal performance using a laboratory simulation of the Stirling engine fluid system cyclic operation. This is currently the most reliable and least expensive method to develop seals for new applications.

The purpose of this report is to summarize the work performed by the Boeing Commercial Airplane Company, under contract to NASA/LRC, in evaluating various single-stage seal configurations for possible application in the Stirling cycle engine. The scope of this effort changed during the program. Initially, the seal design requirements were based on engine conditions of the Ford/N. V. Philips experimental four cylinder Stirling cycle engine (reference 1). In this engine the reciprocating piston rods are driven against a cam swash plate to provide engine rotational output. The engine seal under investigation is located on the rod and separates the pressurized hydrogen working gas from the crankcase lubricating oil. The engine efficiency is highly dependent on minimum leakage of the working gas from the working volume.

When the Ford Stirling engine program was terminated, the engine requirements were modified to reflect the prototype engine developed by United Stirling of Sweden. This engine is an industrial type engine modified for automotive applications. As described in reference 2, the engine has four parallel cylinders, dual crankshafts and generates approximately 29.84 kw (40 hp).

The seal development program was divided into a number of activities:

(1) Define the seal design/test requirements

(2) Select and/or design the most promising single-stage seal designs

(3) Design and fabricate a laboratory simulation of the Stirling cycle engine

(4) Conduct an evaluation test of various seals to determine performance and endurance capability.

3.0 SEAL DESIGN ANALYSIS

This task evaluated a number of single-stage reciprocating seal designs based on the requirements of the Stirling cycle engine. The majority of seal designs investigated were configurations in wide use in aircraft and/or industrial applications. The NASA Chevron seal was designed specifically for this evaluation.

3.1 DESIGN REQUIREMENTS

To evaluate various single-stage rod seals for the Stirling engine application, the seal design criteria and test objectives of this program were established to reflect the proposed configuration of the Ford/Philips Stirling cycle engine. These requirements were later modified to reflect the engine conditions of the prototype engine from United Stirling of Sweden. The design requirements common throughout this evaluation are the use of a 2.22 cm (7/8 inch) diameter rod and a reciprocating stroke of 5.08 cm (2 inches). Helium was used as the pressurized gaseous medium instead of hydrogen due to the safety procedures required. This was mutually agreed upon by NASA/LRC and Boeing. A high temperature synthetic hydrocarbon-based fluid (MIL-H-83282) was used as the lubricant.

3.1.1 FORD ENGINE REQUIREMENTS

The Ford/Philips Stirling cycle engine is rated at 126.82 kw (170 hp) at 7.19 m/sec peak (3000 rpm) with an engine displacement of 3523 cc (215 cubic inches). The seal under investigation is located on the reciprocating rod and separates the hydrogen gas working medium from the enginecase lubricating fluid.

The requirements for the Ford Stirling engine included:

(1) rod reciprocating speed of 7.19 m/sec (3000 rpm maximum)

(2) helium gas pressure (upstream side of the test seal) of 1.2069×10^7 PA (1750 psig)

(3) lubricating oil (downstream of the test seal) maintained at atmospheric conditions

(4) temperature environment at the test seal of 408 K (275°F)

(5) desired endurance life of 1500 hours at 50 cps (2.7×10^8 cycles).

3.1.2 UNITED STIRLING ENGINE REQUIREMENTS

As the seal evaluation progressed, the seal test requirements changed significantly when Ford terminated all activity on the 4-215 Stirling engine. An alternative engine under investigation is being developed by United Stirling of Sweden and modified for automotive applications. Although the leakage criteria, endurance requirements and shaft diameter are the same, the following seal design parameters did change:

(1) rod reciprocating mean speed of 4.79 m/sec (2000 rpm)

(2) helium mean gas pressure of 5.28×10^6 PA (750 psig)

(3) ambient temperature at the test seal of 332 K (120°F).

3.1.3 LEAKAGE CRITERIA

The leakage criteria for this seal evaluation was based on the Ford/Philips engine requirements. The maximum allowable leakage rate for the test seals is based on one hydrogen recharge per year for a typical Stirling cycle engine. This amounts to a leakage rate of six grams of hydrogen from each seal for 565 hours of engine "on" time and 8195 hours of engine "off" time. For an engine speed of 7.19 m/sec peak (3000 rpm) with a pressure differential of 1.22×10^7 PA (1750 psid) and 408 K (275°F) seal temperature, the maximum allowable hydrogen volumetric leakage rate is .004 cc/sec. Helium, the test gas used in this evaluation, has an equivalent maximum allowable mass leakage rate of .002 cc/sec.
No account was made for the variation in the rate of diffusion of helium compared to hydrogen or solubility into lubricating oil. The diffusion of the gaseous medium is a parameter that is not controlled by the seal configuration and thus was not investigated. The solubility of the gaseous medium into the oil would be indicated by a constant leakage rate dependent on the temperature and pressure at the interface.

The maximum allowable hydrocarbon leakage (weepage) into the helium gas is 9.0×10^{-5} cc/minute. Again this requirement is based on the Ford engine requirement.

3.2 SEAL CONFIGURATIONS

Six seal configurations were evaluated during this rod seal investigation. They were the Boeing Footseal, NASA Chevron Seal, Bell Footseal, Quad Seal, Tetraseal, and the Dynabak seal. All seals were selected from aerospace and/or vendor standards, except the NASA Chevron seal configuration which was designed specially for this evaluation.

3.2.1 BOEING FOOTSEAL

The Boeing Footseal design (figure 1) was the first seal to be evaluated. This seal was developed by Boeing for commercial aircraft flight control actuators with variable stroke and dither operations conveniently placing the O-ring into a static seal configuration. In aircraft applications, typical surface velocities range up to .25 m/sec (50 ft/min) with differential oil pressures of 2.06×10^7 PA (3000 psid). The seal consists of a Teflon material shaped like a foot and in contact with the rod. It is loaded by an elastomer O-ring made of Viton material which is compatible with petroleum-based fluids. This seal is pressure-actuated by the upstream angle of the foot. Saw-tooth grooves are cut into the foot circumferentially to provide lubricant to the seal/rod interface. A special seal retainer is required to fill the triangular section not filled by the seal at the downstream end of a standard groove.

3.2.2 NASA CHEVRON SEAL

The NASA Chevron seal (figure 2) was designed for the 2.22 cm (7/8 inch) rod size by similarity to the successfully-tested 2.54 cm (1 inch) rod size (ref. 3). For fabrication, Dupont SP-21 polyimide material was used because of previously successful experience. In this

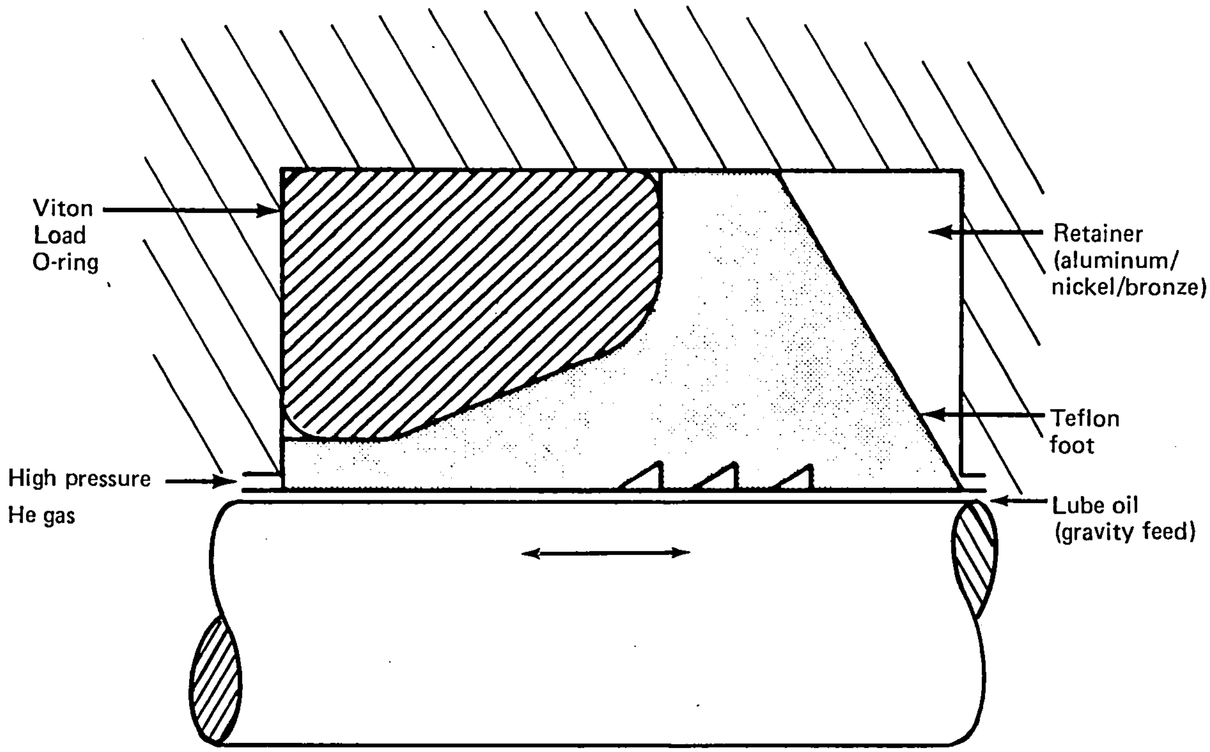


Figure 1. — Boeing Footseal Configuration

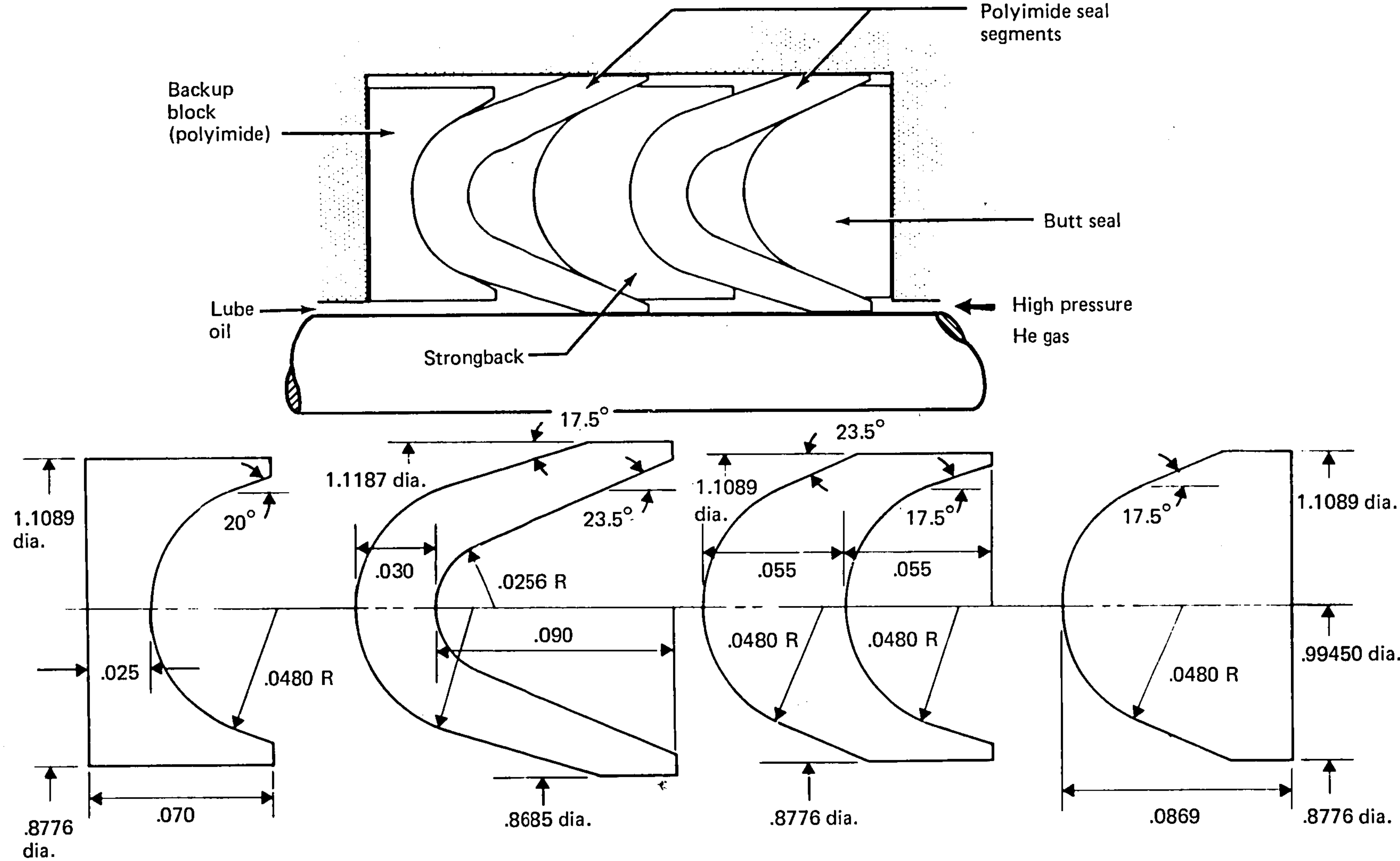


Figure 2. — NASA Chevron Seal Configuration

design, the Chevron sealing-elements' cross section acts as two cantilevered structural beams in flexure about the apex of the geometric shape, the two elements providing redundant sealing capability. This configuration was designed to maintain sealing pressure with the rod surface over a temperature range of 244 K (-20°F) to 422 K (300°F) in both the pressurized and unpressurized conditions. The other parts of the five-piece assembly complete the seal for installation in a standard MIL-G-5514 O-ring groove.

3.2.3 BELL SEAL (THREE-PIECE FOOTSEAL)

The Bell seal three-piece footseal design, as shown in figure 3, consists of a seal and backup ring made from Turcon (a Teflon-filled material). A hexagonal cross section lathe-cut seal ring made from nitrile material is used to load the seal. The dimensions of the seal gland are similar to MIL-P-5541F. This seal was supplied by Robert Brent of Bell Helicopter Company.

3.2.4 QUAD SEAL

The Quad Seal is a four-lobed elastomer seal, as shown in figure 4, made of nitrile rubber with a 70 Durometer Shore hardness. This configuration had been used with reasonable success by GM Research under helium pressures to 1.03×10^7 PA (1500 psig) and 3950 rpm. This seal is manufactured by Minnesota Rubber Company.

3.2.5 TETRASEAL

The Tetraseal, as shown in figure 5, is a seal with a lathe-cut square cross section. The Tetraseal inside diameter, which is less than a corresponding O-ring inside diameter, reduces seal squeeze, which in turn reduces seal wear and extends the life. This seal is made of nitrile rubber with a 70 Durometer Shore hardness. This seal displayed some success during GM Research testing in the 1960's. The Tetraseal is fabricated by Goshen Rubber Company.

3.2.6 DYNABAK SEAL

The Dynabak Seal, as shown in figure 6, has a triangular-shaped cross section backup ring and is loaded by a standard O-ring. The backup ring is made of Teflon and the seal of nitrile or viton. It was originally designed for aerospace applications in the early 1960's. The Dynabak is designed to prevent noticeable seal extrusion at the downstream side of the seal gland.

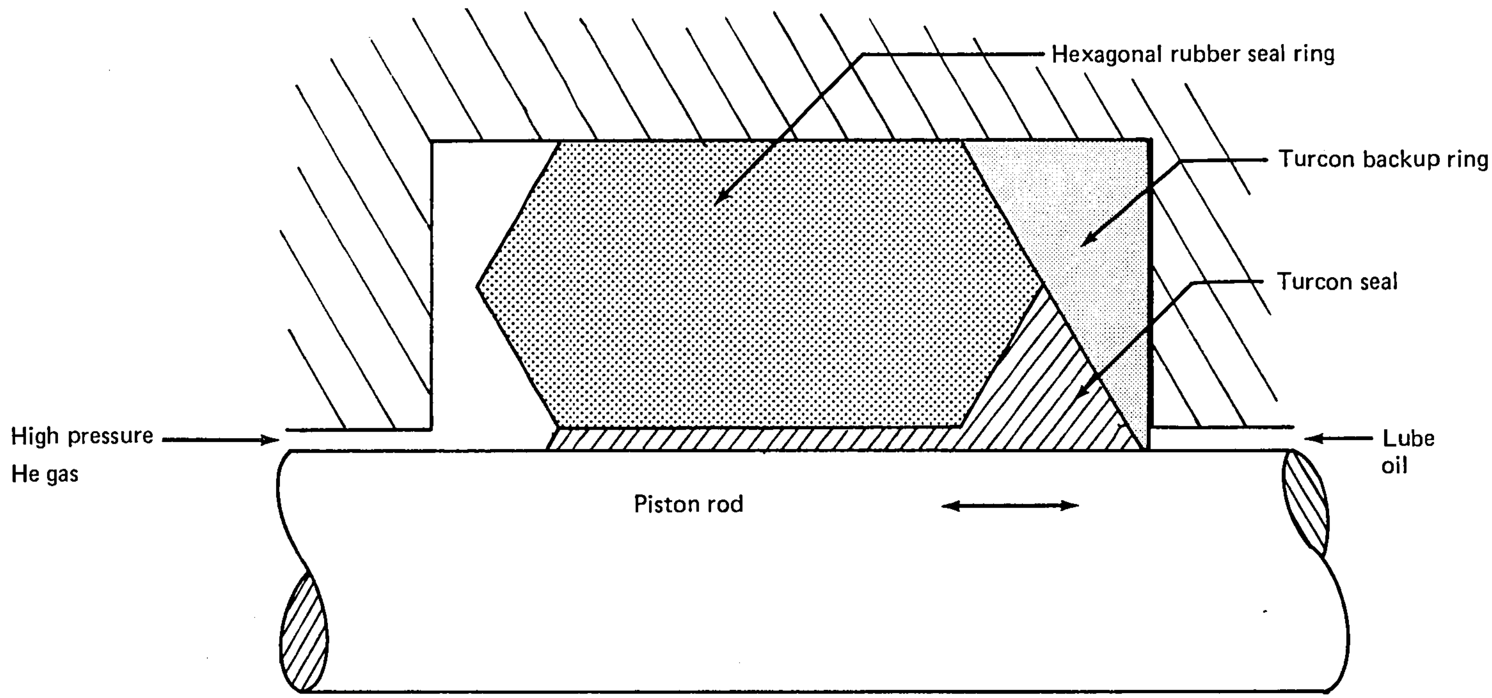


Figure 3. — Bell Seal Configuration—Three-Piece Foot Seal

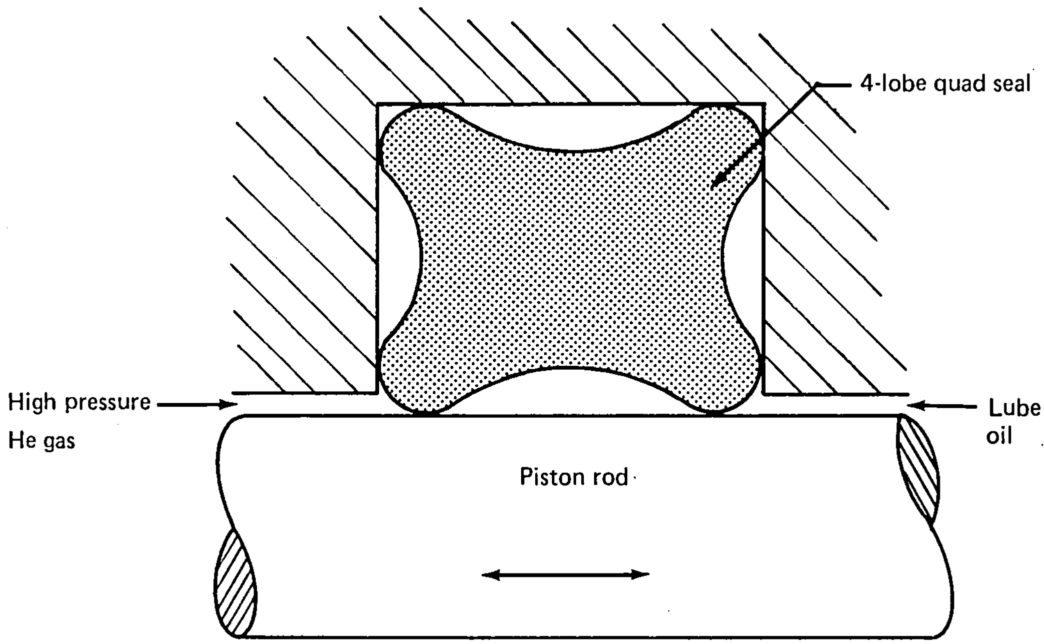


Figure 4. — Quad Seal Configuration

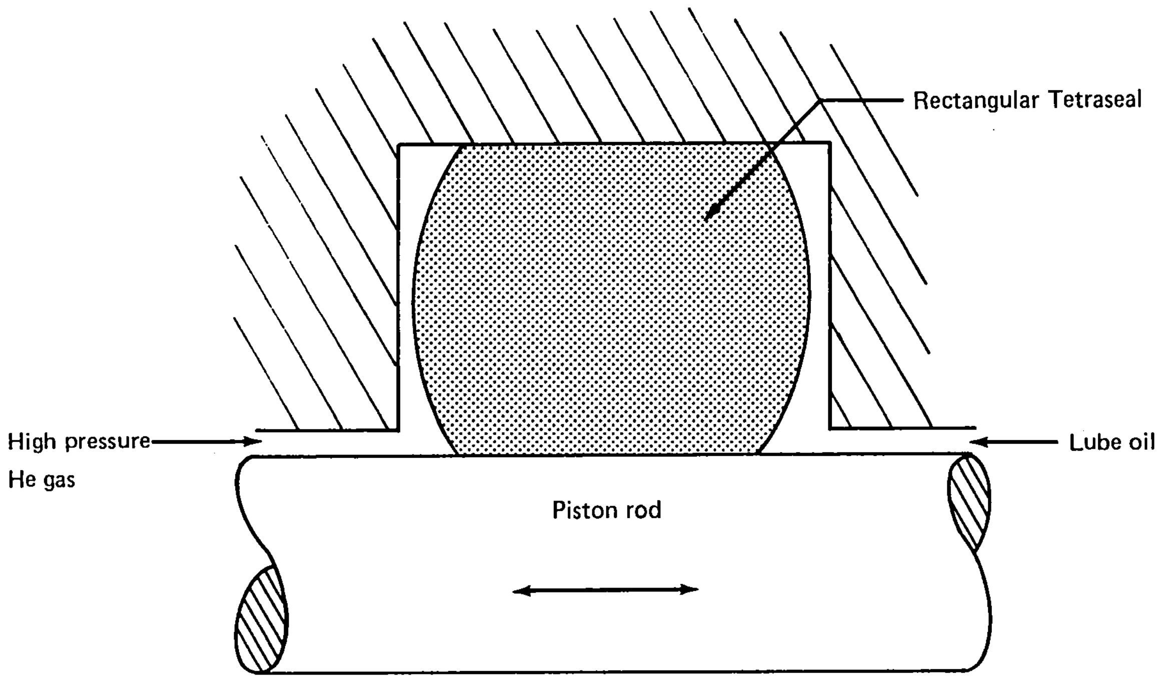


Figure 5. — Tetraseal Configuration

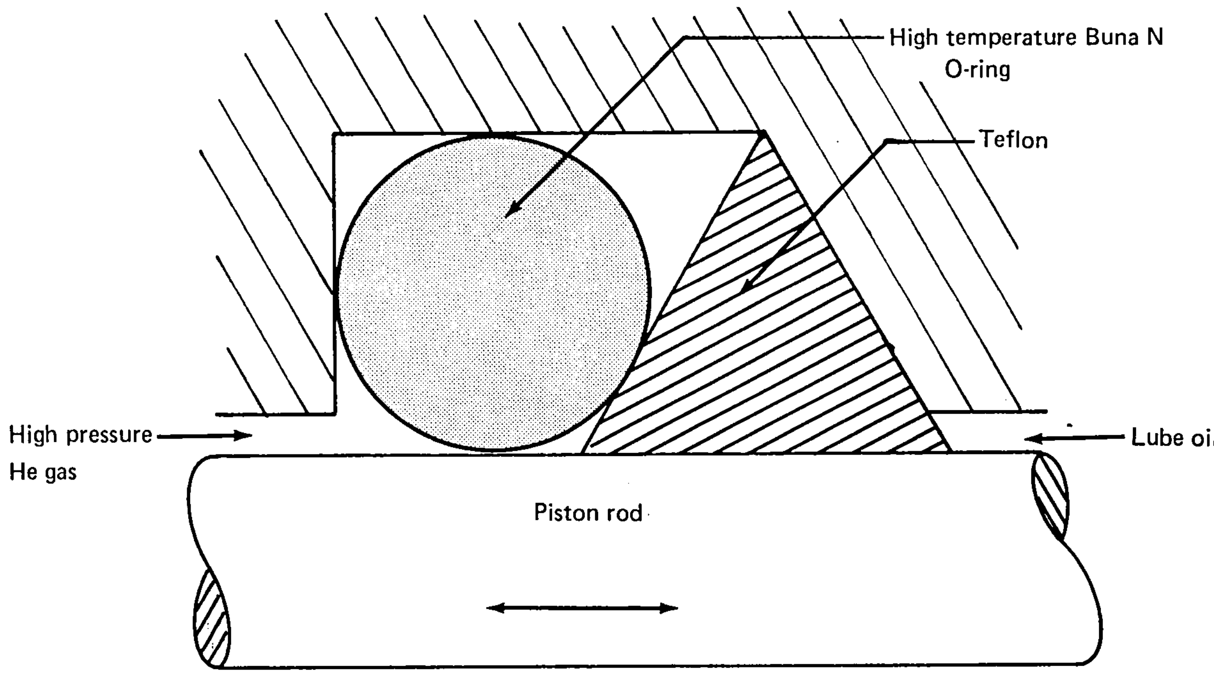


Figure 6. – Dynabak Seal Configuration

4.0 TEST SYSTEM DESCRIPTION

The testing apparatus used to simulate the reciprocating motion and environmental conditions of the Stirling cycle engine is shown schematically in figure 7. This test system consists of a 11.19 kw (15 hp) variable speed drive connected to a 2.54 cm (1 inch) eccentric. This provides a 5.08 cm (2 inch) reciprocating rod motion through each of two test seals held within the test seal retention fixture. The installation of the link-rod end-bearing lubrication and cooling system is required to prevent failure of the bearing. A ball bushing (pillow block) is used to react against side loads caused by the eccentric or other misalignments. An oil cavity reservoir supplies lubrication oil to the low pressure side of the two test seals. Pressurized helium is supplied to the upstream side of each test seal. Helium leakage across the test seals is collected in oil-filled helium accumulation reservoirs. The oil in these reservoirs rises into a leakage manometer for each seal. An oil sight gage is used to maintain a fully lubricated rod throughout testing. A gas chromatograph sampling port is provided to obtain helium gas samples for gas chromatograph analysis to determine hydrocarbon concentration. An environmental chamber allows test operation to 408 K (275°F). Automatic safety shutdowns are provided for excessive seal temperatures, high leakage rates and shaft operation. Modifications to the test system are described in Appendix A.

4.1 TEST ROD DRIVE SYSTEM

The test rod drive system consists of a Sterling Speedtrol variable speed drive capable of speeds to 3000 rpm and bolted to a flat steel surface (figure 8). The flat surface is 1.91 cm (3/4 inch) steel plate mounted on top of I-beams of 25.4 cm (10 inch) width. The output shaft of the varidrive unit has an endcap assembly with a stud offset 2.54 cm (1 inch) from the shafts' rotational axis to provide a reciprocating motion of ±2.54 cm (1 inch). A high speed self-aligning ball bearing, connected to the stud, is located at the varidrive end of a connecting rod. This bearing is designed to withstand the frictional wear from the high rotational velocity and dynamic loading effects from the varidrive (inertia forces at the inside and outside bearing races). An aluminum/nickel/bronze bushing assembly, threaded into the opposite end of the connecting rod, provides connection to the dynamic seal test rod. The plane of the connecting rod and dynamic seal test rod is aligned perpendicular to the rotational axis of the varidrive output shaft and any misalignment is compensated for by the self-aligning characteristics of the ball bearing assembly.

One end of the dynamic seal test rod is supported by a pillow block unit. The other end is supported by the bearing surfaces in the seal retention test fixture thus permitting the test rod to sustain pure translational motion. The test rod is also aligned in a horizontal plane.

The test fixture is mounted inside a special environmental chamber capable of maintaining a test seal temperature of 408 K (275°F). Both the test seal retention fixture and the pillow block are mounted on a common steel channel that extends through the environmental chamber. This allows proper alignment of the rod, pillow block, and test seal retention fixture at a perpendicular to the varidrive output shaft centerline. In the environmental chamber, the channel is mounted on a 1.27 cm (1/2 inch) steel plate that is supported by bolts extended through the chamber floor to an I-beam mounting substructure. The environmental chamber is thus isolated from the test drive mechanism. The top of the

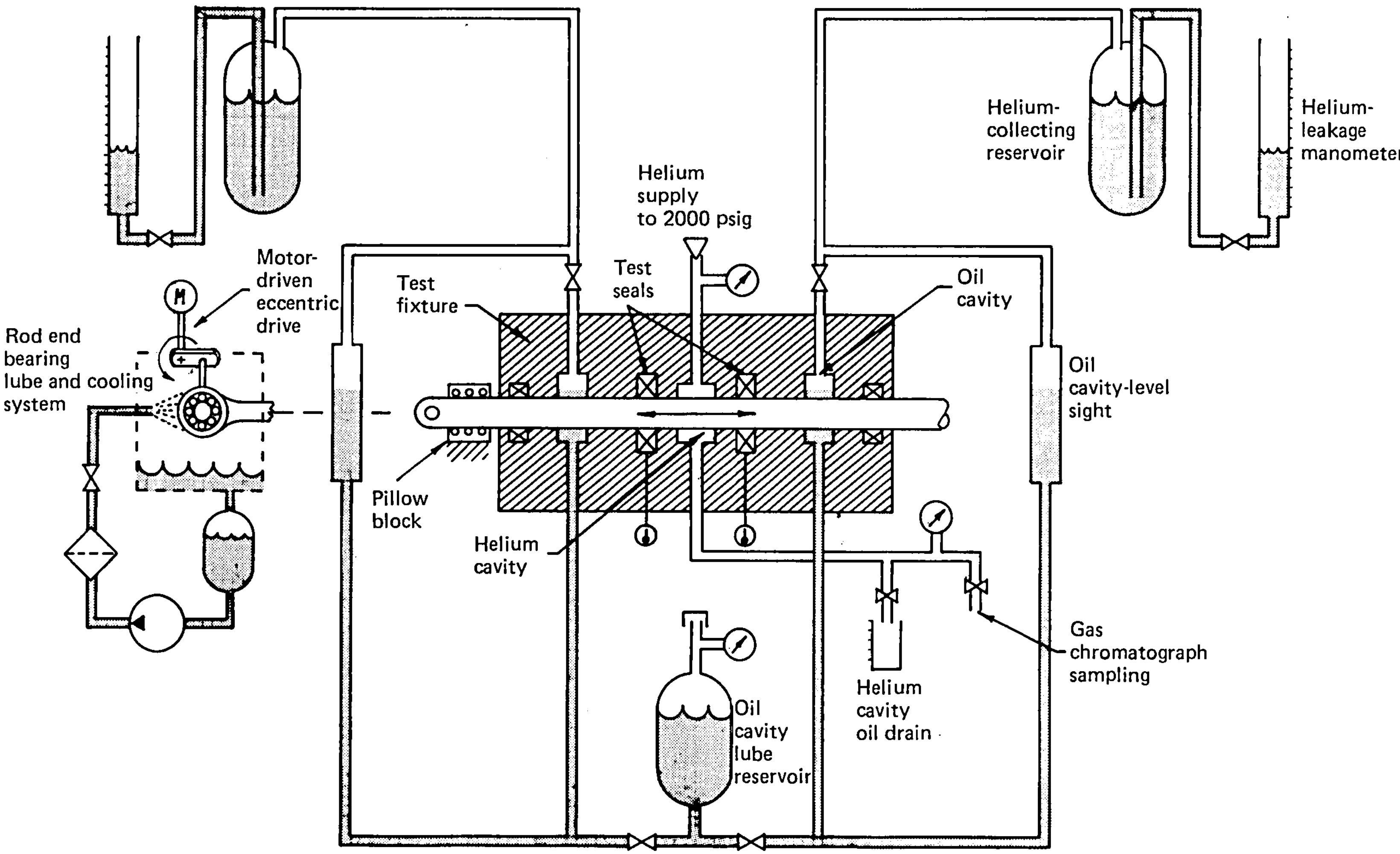


Figure 7. — Test System Schematic

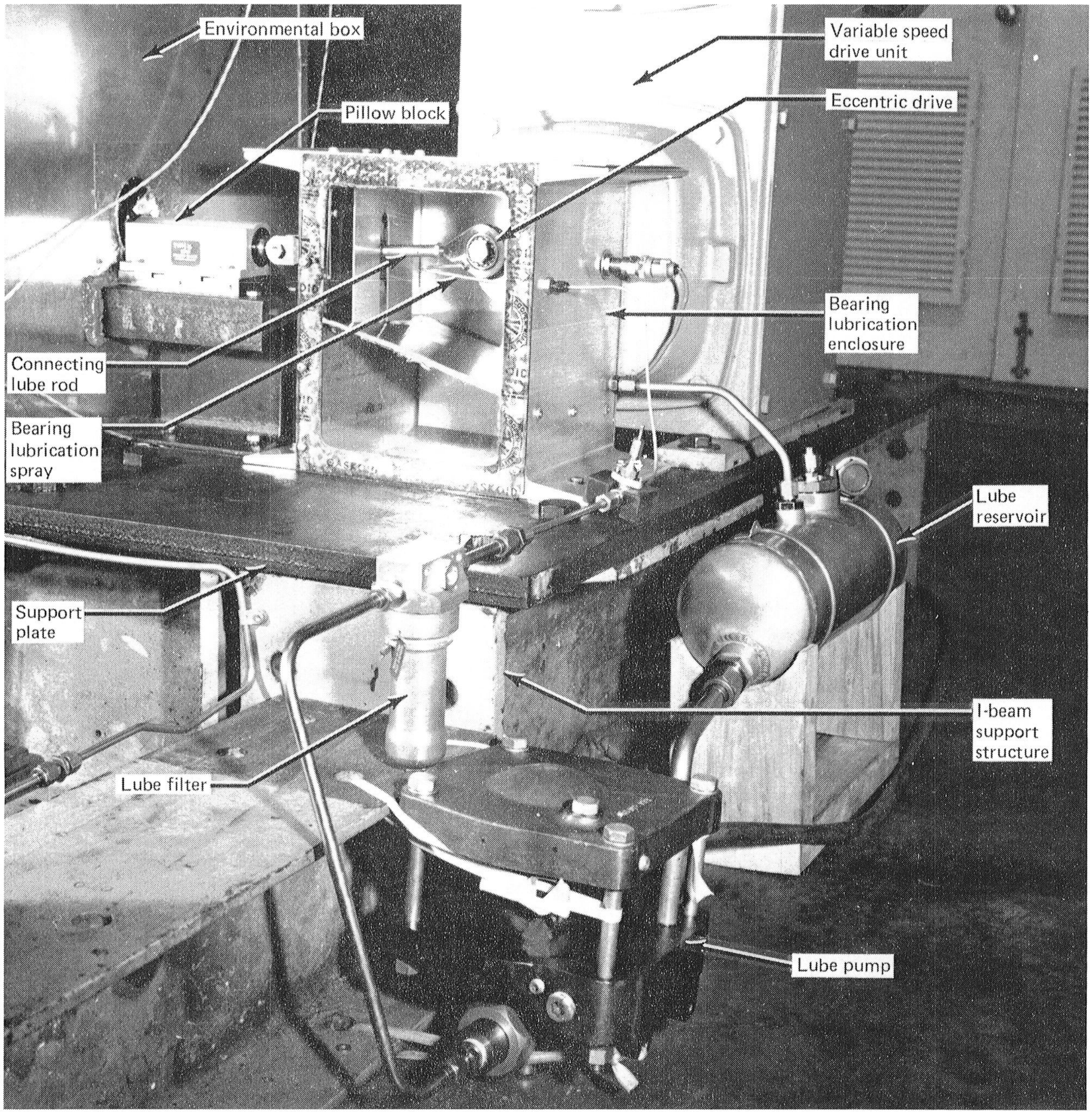


Figure 8. — Drive Linkage and Bearing Lubrication System

environmental chamber is enclosed during testing.

The reciprocating seal test rod was manufactured of nitriding steel ground to a 4-6 RMS surface finish. The rod was machined to a 2.22 cm (7/8 inch) diameter. The pillow block end of the seal test rod is 2.54 cm (1 inch) diameter to fit the pillow block of the nearest standard size inside diameter. The test rod is hollow to allow forced air convection cooling and to minimize inertia effects of the drive. Refurbishment of the two rods required chrome plating followed by grinding.

4.2 TEST SEAL RETENTION FIXTURE

The test seal retention fixture was manufactured from aluminum/nickel/bronze which is an excellent bearing material. The fixture shown in figure 9 is capable of simultaneously testing two seals of the same configuration. Because it is necessary to change gland sizes for each of the seal designs, the seal retention fixture is divided into three sections. The center section provides supply pressure to the upstream sides of the test seal gland walls, while the two outer sections act as supports for seal retainer rings. One seal retainer ring is installed for each seal acting as the downstream gland wall.

The downstream side of the test seal contains ambient pressure oil (MIL-H-83282) acting as the test rod lubrication supply. Capseals are used as rod end seals to prevent oil leakage from each end of the test fixture. The test seal retention fixture also has temperature probes for each seal, helium gas chromatograph sampling and helium cavity drainage ports.

4.3 HELIUM SUPPLY SYSTEM

The helium supply system is composed of a high pressure helium bottle with two pressure regulators and a hydraulic line from the bottle to the test fixture pressure supply port. The regulators can control pressure to a maximum of 1.39×10^7 PA (2000 psig) for proof pressure testing, and to a minimum of 1.05×10^6 PA (150 psig) for gas chromatograph sampling pressure control. The helium gas supply schematic (figure 10) has a pair of solenoid-operated control valves that act as a shutdown if failure occurs. A pressure sensing switch is used in this network to act as part of the systems' failsafe mechanism, monitoring helium supply pressure with shutdown occurring with supply over pressure.

4.4 OIL CAVITY LEVEL CONTROL SYSTEM

During initial testing, significant leakage past the rod end seals of the test fixture was discovered. As a result, a test block fluid level indication system was installed to maintain complete fluid exposure to the rod sealing surface. The fluid reservoir level system is partially shown in figure 11. The separate sight tubes are located outside the environmental chamber to allow easy reservoir level inspection and control. The oil sight tubes and oil level control unit outside of the environmental chamber are shown in figure 12. Hydraulic fluid supply to the test block oil reservoirs is provided from a pressurized lubrication reservoir mounted above the test block on the environmental chamber wall. Fluid levels are controlled by periodically interrupting the flow from the lubrication reservoir to the test block with a solenoid shutoff valve upstream of the test fixture.

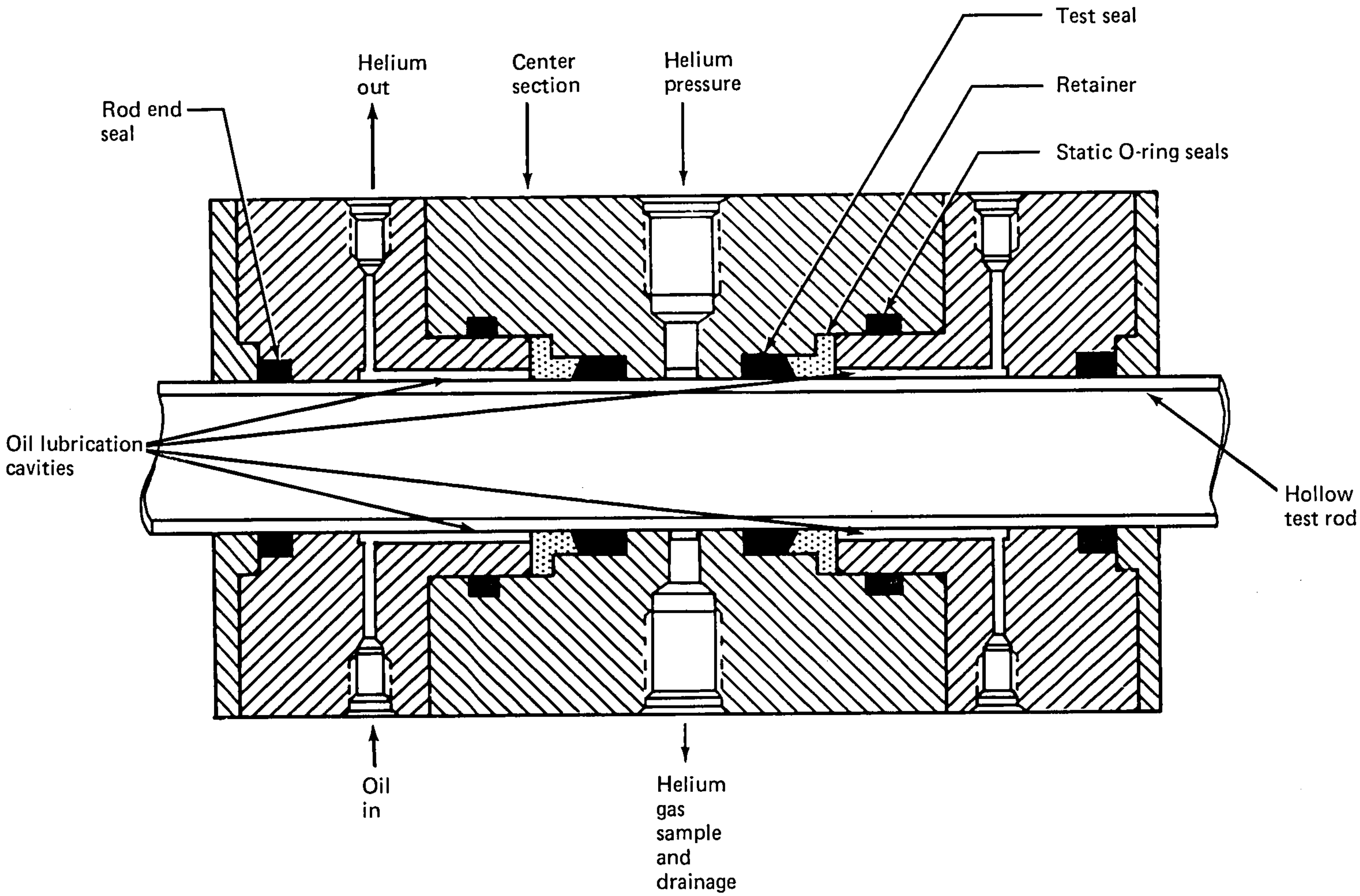


Figure 9. – Seal Retention Fixture

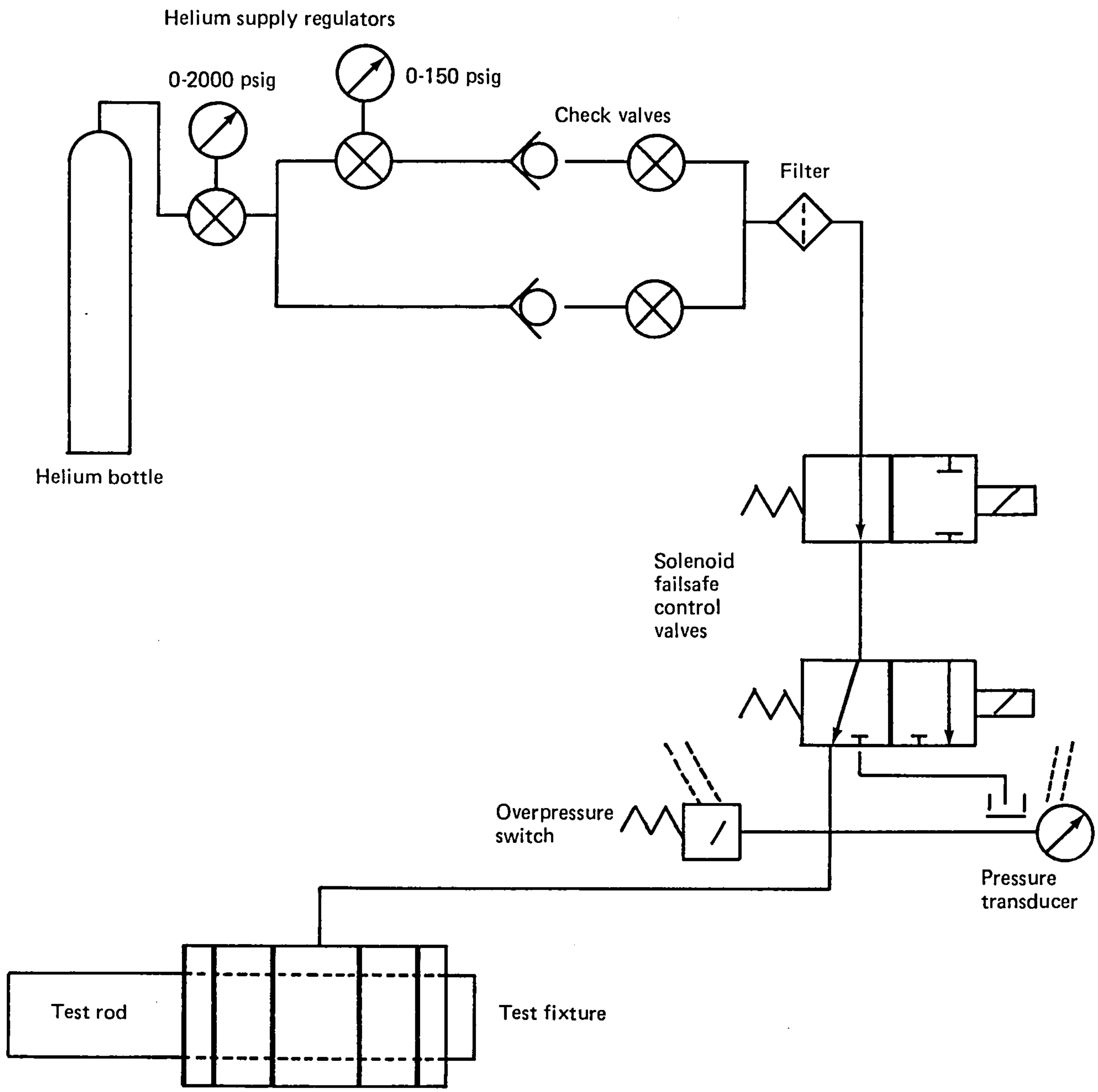


Figure 10. — Helium Supply System Schematic

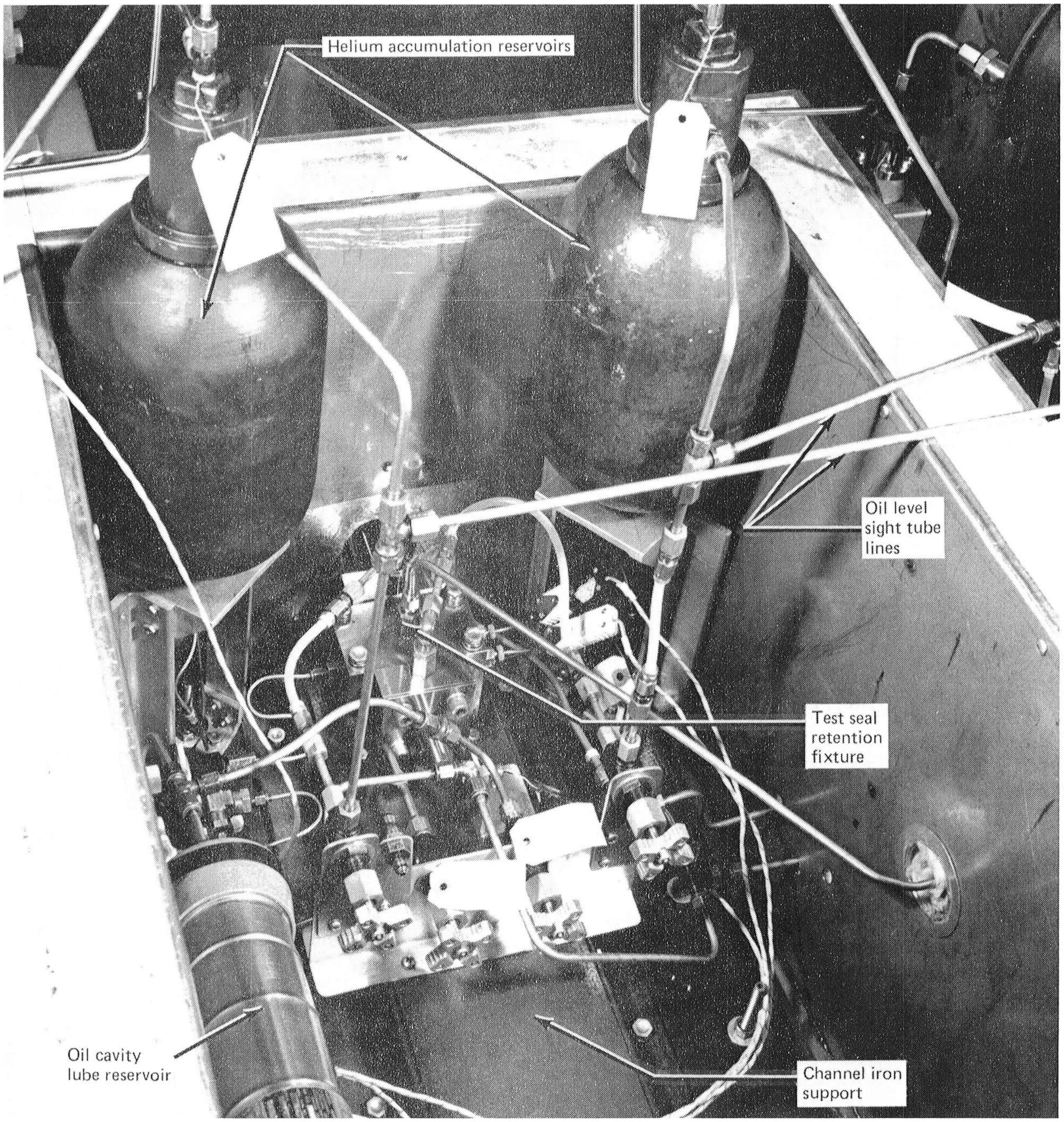


Figure 11. — Oil Lubrication System

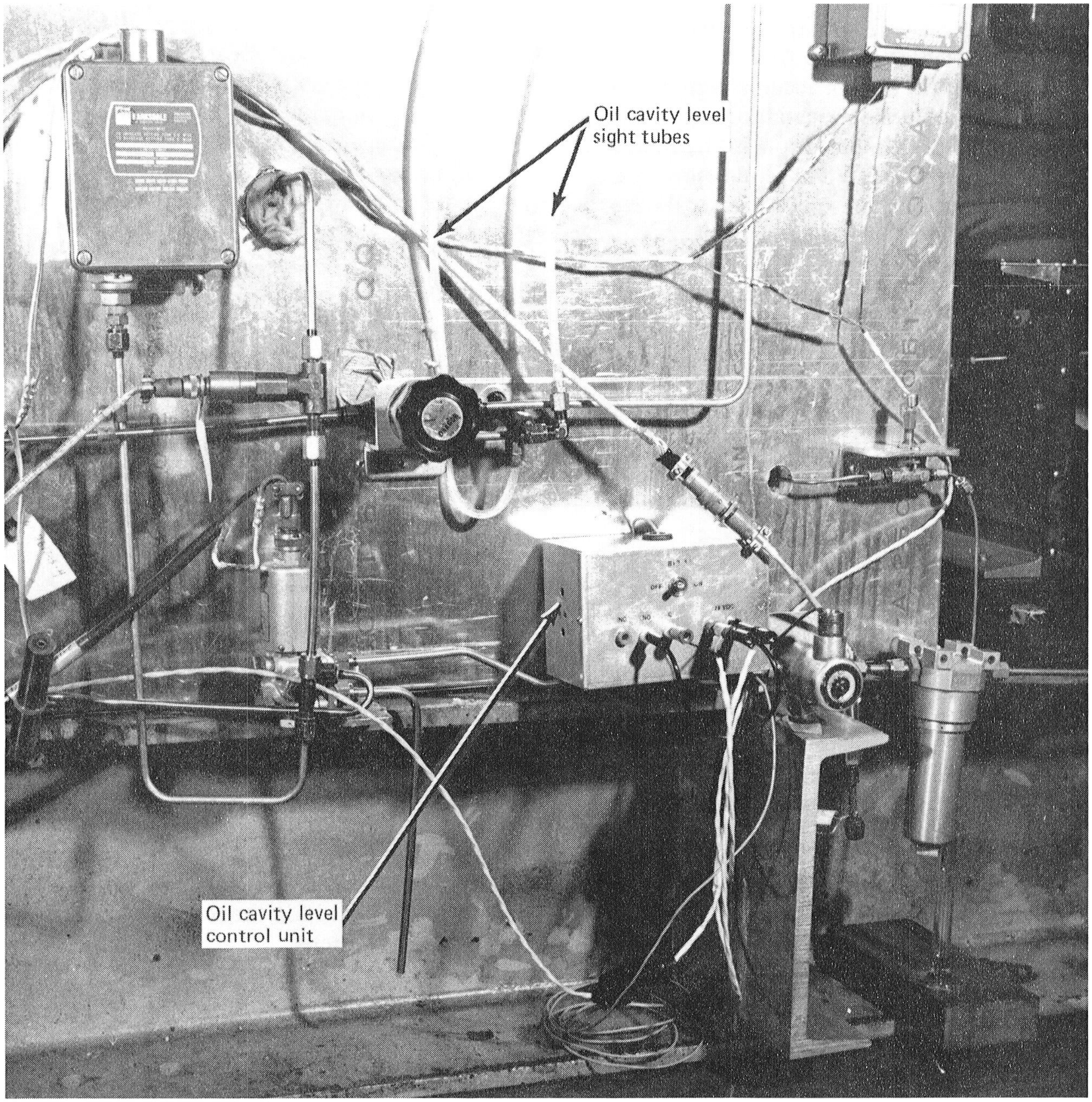


Figure 12. — Oil Cavity Level Control System

4.5 HELIUM LEAKAGE MEASUREMENT SYSTEM

The helium leakage measurement system consists of a large 2.86 cm (1-1/8 inch) inside diameter and small 1.27 (1/2 inch) diameter glass manometer pair and helium accumulation reservoir for each test seal. Each glass manometer stands eight feet high, as shown in figure 13. Helium leakage from the test seal retention fixture travels through hydraulic lines to the top of the helium accumulation reservoirs. The helium gas pressure buildup displaces hydraulic fluid up a stand tube from the bottom to the top of the helium accumulation reservoir and then out through a hydraulic line to the glass leakage manometers. The manometers are calibrated in centimeters, and hydraulic fluid levels in the large and/or small manometers are monitored versus seal test time. Interval points taken from this data are then used to determine helium leakage rates in cc/sec.

4.6 DYNAMIC BEARING LUBRICATION SYSTEM

To maintain high life requirements for the self-aligning bearing mounted on the eccentric drive, it is necessary to provide a method of lubricating the bearing running surfaces as well as to remove heat generated by contact friction. The wet lubrication system directs a stream of hydraulic fluid into the bearings' path. The lubrication system, shown in figure 8, consists of a reservoir, air driven hydraulic pump, hydraulic fluid filter, flow adjustment valve, nozzle, and an oil accumulation canister surrounding the bearing acting as a spash guard.

4.7 SEAL FRICTION MEASUREMENT SYSTEM

The measurement of both static and dynamic friction forces from the test seal to test rod interface is accomplished by utilizing a hydraulic actuator adapted to the end of the test rod. The actuator displaces the test rod in both travel directions. The values of the actuator rod and head pressures are recorded and the frictional forces opposing the test rod motion are then calculated. A schematic of the frictional measuring system is shown in figure 14.

4.8 TEST INSTRUMENTATION SYSTEM

The instrumentation setup for the Stirling engine dynamic rod seal test program records test parameters and provides the control of specific failsafe circuits within the test fixture. The test instrumentation is capable of reading the following parameters:

(1) temperatures

a) seal #1

b) seal #2

c) helium accumulation reservoir #1

d) helium accumulation reservoir #2

e) leakage manometer set #1

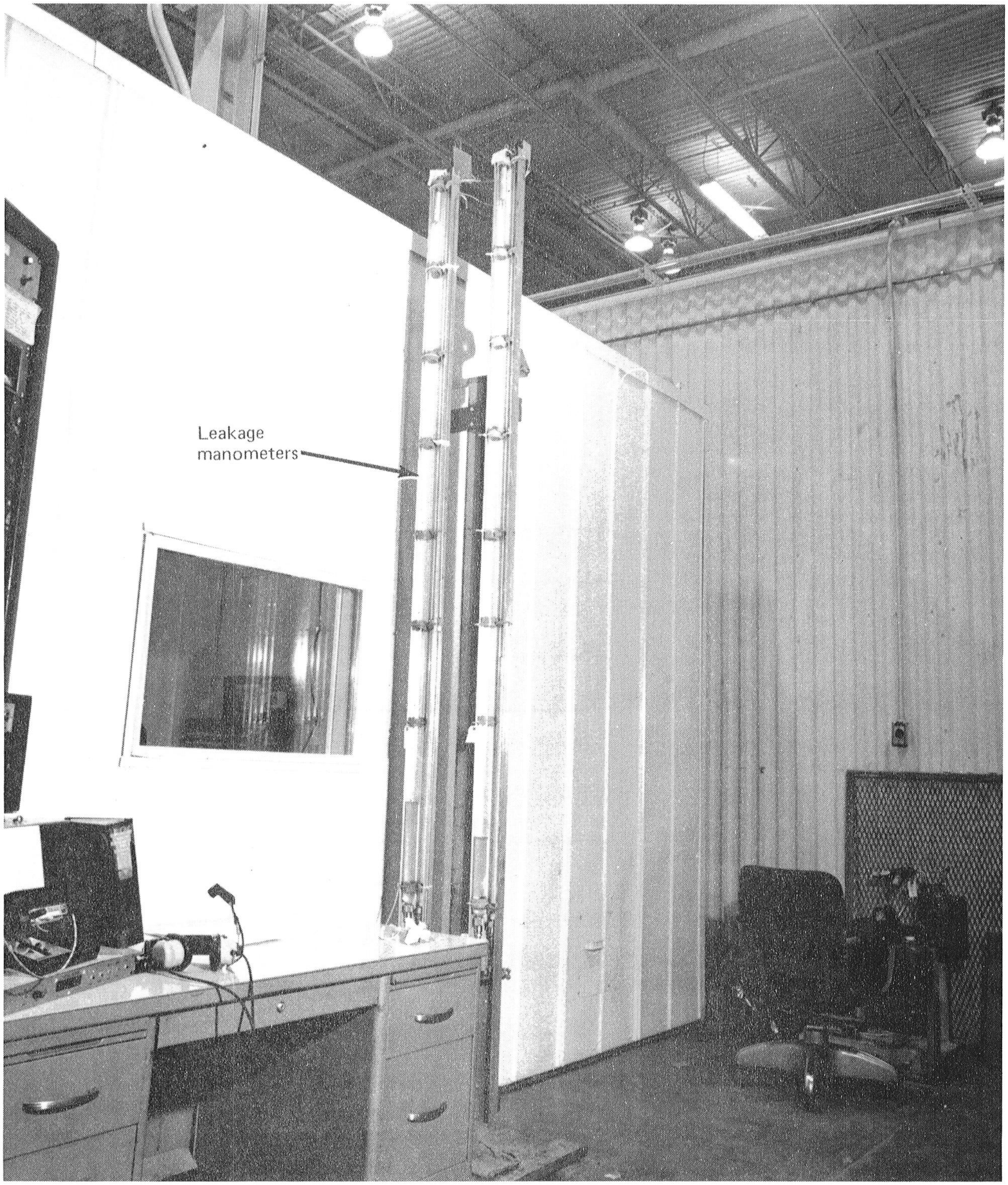


Figure 13. — Leakage Manometers

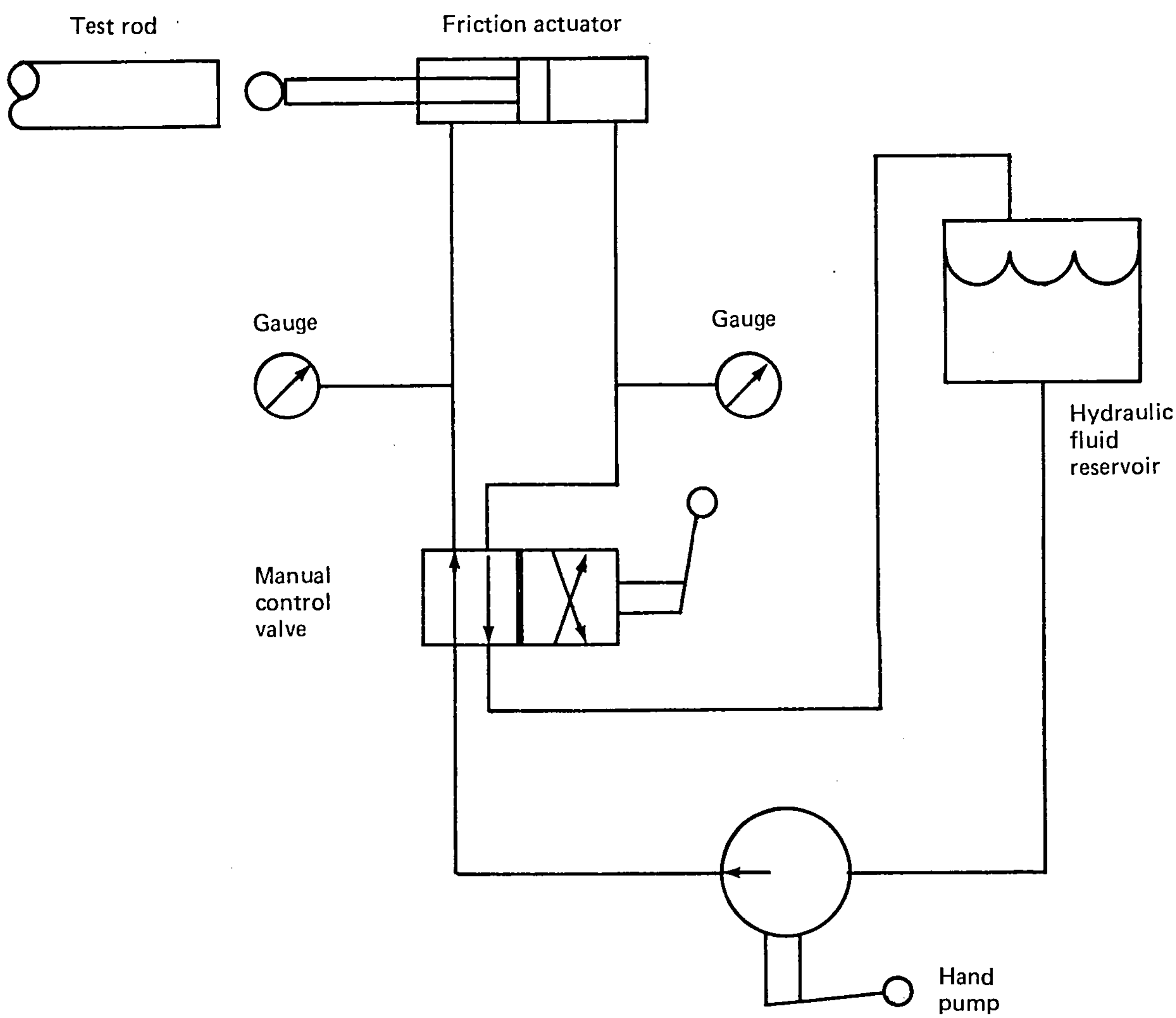


Figure 14. — Frictional Measurement Schematic

f) leakage manometer set #2

g) environmental chamber

h) ambient

(2) pressures

a) helium supply

b) helium accumulation reservoir #1

c) helium accumulation reservoir #2

(3) test rod frequency

(4) cycle counts

(5) test time

It was necessary to locate the test inside a noiseproof room with limited height clearance. The leakage manometers were located outside the room due to their height, and an intercom system was provided to allow communication between the test operator and data taker. A photo of the test instrumentation rack is shown in figure 15.

4.9 FAILSAFE CIRCUITS

The Stirling engine dynamic rod seal evaluation test facility is designed for continuous use without the aid of an operator. Because of this feature, numerous failsafe mechanisms are incorporated within the system that shutdown the test operation if a failure occurs. Failsafe circuits are incorporated for:

(1) helium supply overpressure limit of 2000 psig

(2) helium accumulation reservoir overpressure of 7.0 x 10^5 PA (100 psig)

(3) manometer level overflow limit

(4) test seal overtemperature limit of 422 K (300°F)

(5) environmental chamber overtemperature limit of 413 K (285°F).

4.10 ENVIRONMENTAL CHAMBER

The environmental chamber is designed to maintain a temperature of 408 K (275°F) at the location of the test seal. Both a strip element and blower heater are used for heating, while liquid nitrogen is used for cooling. The chamber is constructed with four-inch composite walls consisting of fiberglass insulation sandwiched between two aluminum sheets of .318 cm (.125 inch) thickness.

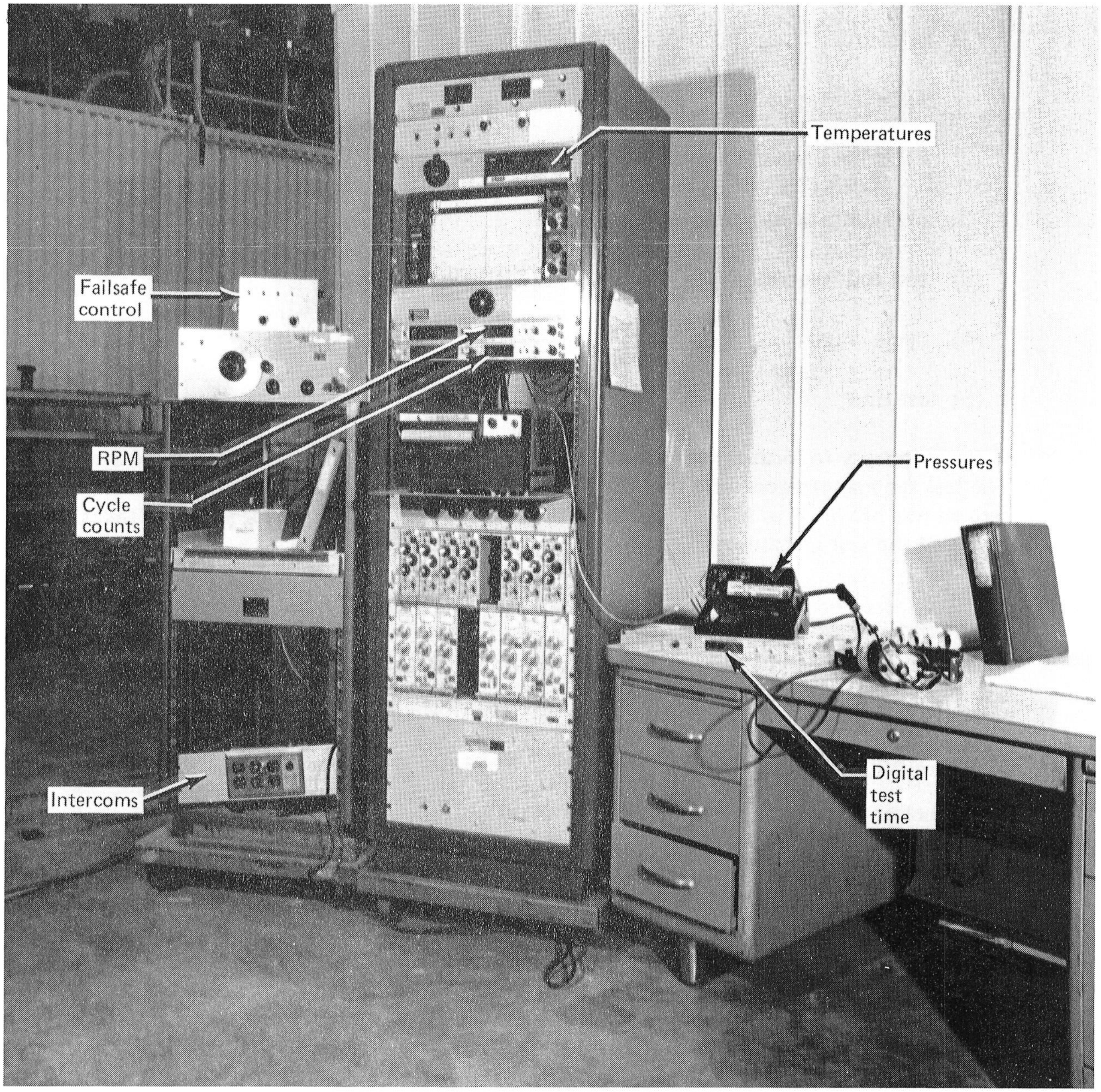


Figure 15. — Test Instrumentation

4.11 GAS CHROMATOGRAPH AND OIL SEEPAGE DRAINAGE PORTS

To monitor the amount of MIL-H-83282 hydraulic fluid leakage from the low to high pressure side of the test seals, it is necessary to provide a sampling port at the high pressure side of the test fixture for gas chromatograph measurement techniques. The gas chromatograph sampling provides the percentage, by volume, of hydraulic fluid in the helium gas. Because large leakage rates were obtained from each of the seal configurations tested, gas chromatograph samples were taken only for the first Boeing Footseal test run, and it was necessary to drain oil seepage out of the helium cavity to expose the total seal surface to the helium gas.

4.12 PROOF PRESSURE

Prior to each test seal endurance test, a static proof pressure test is conducted in which helium pressure of 13.74×10^6 PA (2000 psig) is applied to the seal for five minutes at two seal temperatures of 274 K and 408 K (70°F and 275°F).

5.0 RESULTS AND DISCUSSION

Testing was conducted on the six seal configurations described in section 3.0 using the testing apparatus of section 4.0. A number of catastrophic structural (deformation) failures occurred with all configurations, particularly the Boeing Footseal and the NASA Chevron seal. Failures were due to excessive temperatures at the rod seal interface. Rod resurfacing was required after three of the failures because of rough rod surfaces. None of the seals evaluated survived the test conditions of 1.22×10^7 PA (1750 psig), 7.19 m/sec peak (3000 rpm), and 408 K (275°F). At the lowest test requirements, leakage rates were still beyond the desired test goals (table 1). Endurance testing (table 2) was minimal on all seals since leakage rates were well beyond the leakage criteria. One set of Bell seals was tested for 54.5 hours before leakage became excessive. Of the three seal configurations (Boeing Footseal, NASA Chevron seal, and Bell seal) that underwent friction testing as shown in table 3, the NASA Chevron seal displayed frictional forces that were an order of magnitude larger than the Boeing Footseal. The Bell seal had twice the friction levels of the Boeing Footseal.

5.1 LEAKAGE RATE RESULTS

Testing of the various seals revealed that none of the configurations could meet the .002 cc/sec leakage criteria under the harsh conditions of 7.19 m/sec peak (3000 rpm) and 1.22×10^7 PA (1750 psig). Further testing of a Quad seal, Tetraseal, and Dynabak backup ring (in combination with a standard O-ring seal) provided similar unsatisfactory leakage results at the required test system operating conditions of 4.72 m/sec peak (2000 rpm) and 5.28×10^6 PA (750 psig) helium pressure. Appendix B summarizes graphs of leakage rates (cc/sec) versus test time (minutes) from selected test runs for each type of seal design. Selected final and maximum leakage rates, along with test parameters, are also displayed in table 1.

5.1.1 BOEING FOOTSEAL

Eight sets of Boeing Footseals were evaluated with four of the sets failing due to high temperature deformation without any leakage information being obtained. The seals were tested at the conditions of 1.22×10^7 PA (1750 psig) and 7.19 m/sec peak (3000 rpm), and gave leakage rates of 0.0154 and 0.0104 cc/sec for set #2 seals #1 and #2, respectively, which was five to seven times the leakage criteria goal of 0.002 cc/sec. All other leakage data obtained from the Boeing Footseal was at a helium pressure of 5.28×10^6 PA (750 psig) and a test rod velocity of 1.03 m/sec peak (430 rpm). Three runs were performed at these conditions on sets #1, #3, and #4 with set #3 providing the best leakage information. The lowest leakage rates obtained during this run were 0.0043 and 0.002 cc/sec for seal #1 at 9 and 9.5 minutes into the test. However, shortly after this time, the leakage rate jumped to .049 cc/sec on seal #2 and the test was terminated. Testing of sets #1 and #4 revealed similar leakage values ranging from 0.0104 to 0.68 cc/sec.

5.1.2 NASA CHEVRON SEAL

Three sets of NASA Chevron seals were evaluated, with two failures resulting in breakdown of the seal. The NASA Chevron polyimide five-piece leg seal showed improved leakage

Table 1. — Seal Leakage Results

Type	Set No.	Max Press MPA	Max Rod Vel. (peak) m/sec	Temperature Min-max K(°F)	Leakage rates		Type of Failure
					Seal #1 Final/max cc/sec	Seal #2 Final/max. cc/sec	
Boeing Footseal	1	12.2	1.03	390/422 (243/300)	.0119/.0119	.0126/.0126	Overtemp
	2	12.8	7.19	380/429 (224/313)	.0154/.0154	.0104/.0104	Overtemp
	3	5.28	1.03	292/305 (67/89)	.0020/.0273	.4900/.4900	Catastrophic leakage
	4	5.28	1.03	298/315 (77/108)	.2847/.4440	.4273/.6700	Catastrophic leakage
NASA Chevron Seal	1	.101	1.03	289/322 (61/121)	.0373/.0373	.0765/.0765	Drive link
	2	12.2	2.16	308/355 (95/179)	.2345/.2345	.2629/.2629	Catastrophic leakage
	3	12.2	2.16	273/409 (32/277)	No data/.0210	No data	Overtemp
Bell Seal	1	12.2	2.16	294/436 (70/325)	.0012/.1304	.0064/.1312	Overtemp
	2	12.2	7.19	292/301* (67/83)	.0281/.5866	.0158/.7732	Drive link
	3	5.28	4.19	295/346 (71/163)	No data/.0164	.0019/.0090	Helium cavity
	4	5.28	4.79	294/385 (70/233)	.0117/.0304	.0105/.0698	Catastrophic leakage
Quad Seal	1	12.2	7.19	295/408 (71/275)	.6315/.6315	.5990/.5990	Drive link
	2	5.28	4.19	294/314 (69/105)	.0017/.1078	No data/.0227	High leakage no failure
	3	5.28	4.79	No data	.0018/.0337	.0011/.0170	Catastrophic leakage
	4	5.28	4.79	311/353 (100/126)	.0042/.0461	.0025/.0567	Overtemp

* - Peak temp. = 371 K (208°F)
** - Peak temp. = 355 K (199°F)

Table 1. — Seal Leakage Results (Concluded)

Type	Set No.	Max Press MPA	Max Rod Vel. (peak) m/sec	Temp-erature Mini-max K(°F)	Leakage rates Seal #1 Final/max. cc/sec	Leakage rates Seal #2 Final/max. cc/sec	Type of Failure
Tetraseal	1	5.28	4.79	292/376 (67/218)	.0004/.0041	.0118/.0118	Overtemp
	2	5.28	4.19	334/380 (141/224)	No data/.0351	No data	Overtemp
	3	5.28	4.79	299/380 (78/148)	.0015/.0093	.0024/.0042	High leakage no failure
	4	5.28	4.79	311/402 (101/265)	.0100/.0380	.0177/.0178	Overtemp
	5	5.28	4.79	305/367 (89/202)	.2731/.2731	.0109/.0210	Catastrophic leakage
Dynabak Seal	1	5.28	4.79	294/362 (70/192)	.0032/.0168	.0047/.1110	Catastrophic leakage
	2	5.28	4.79	294/297** (70/75)	.0042/.0785	.0052/.0725	High leakage no failure

* - Peak temp. = 371 K (208°F)

** - Peak temp. = 355 K (199°F)

Table 2. — Endurance Test Results

	Maximum Accumulated Test time, Min. / Set No.	Maximum Continuous Run time, Min. / Set No.	Total Accumulated Test time, Min.*	Conditions: Helium Press Pa (psig)	Conditions: Test Rod Velocity RPM	Conditions: Test Rod Velocity m/sec (ft/min) peak
Boeing Footseal	475/1	78/2	590	1.28×10^7 (1840)	3000	7.19 (1414)
NASA Chevron Seal	330/1	137/2	549	1.22×10^7 (1750)	900	2.16 (424)
Bell Seal	3269/1	548/1	3818	1.22×10^7 (1750)	900	2.16 (424)
Quad Seal	555/4	215/4	629	5.28×10^6 (750)	2000	4.79 (943)
Tetraseal	172/4	49/4	476	5.28×10^6 (750)	2000	4.79 (943)
Dynabak Seal	126/2	50/2	198	5.28×10^6 (750)	2000	4.79 (943)

* For all seals tested of one configuration

Table 3. — Seal Friction Force

	Breakaway Friction Force per Seal	Creep Friction Force per Seal	Increase in Breakaway Friction force Per seal due to Increase in Temperature	Increase in Breakaway Friction force Per Seal Due to Increase in Pressure	
Conditions	340 K (67°F) 1.01×10^5 PA (0 psig)	340 K (67°F) 1.01×10^5 PA (0 psig)	ΔT = 117 K (210°F) 1.01×10^5 PA (0 psig)	338 K (65°F) P = 1.22×10^7 PA (1750 psig)	548 K (275°F) P = 1.21×10^7 PA (1750 psig)
Boeing Footseal	58.7 (13.2 lb)	52.5 N (11.8 lb)	14.2 N (3.2 lb)	36.9 N (8.3 lb)	17.4 N (3.9 lb)
NASA Chevron Seal	Max 497.7 N (112.0 lb) Avg 409.3 N (92.1 lb)	—	393.8 N (85.5 lb)	77.4 N (17.4 lb)	112.6 N (25.3 lb)
Bell Seal	103.2 N (23.2 lb)	—	—	—	—

results over those of the Boeing Footseal. However, because of the high frictional forces produced between the seals and test rod interface, excessive vibrations were produced that prevented varidrive operation above 2.16 m/sec peak (900 rpm) with 1.22×10^7 PA (1750 psig) helium gas pressure. Realignment of the test setup did not alleviate this condition. Leakage rates ranged from 0.0036 to 0.0146 cc/sec for seal set #2, with a failure leakage rate of 0.2629 cc/sec. Testing on this seal was discontinued in favor of the Bell seal.

5.1.3 BELL SEAL

Four sets of Bell seals were evaluated during this program. The majority of leakage rates obtained for seal set #1 had values between 0.01 and 0.04 cc/sec. A total of five test readings were taken of rates below 0.005 cc/sec. Of course, three were at or below the 0.002 cc/sec leakage goal. A minimum rate of 0.0010 cc/sec (fig. 31, Appendix B) was obtained from seal #1 after 84 minutes of testing at conditions of 1.22×10^7 PA (1750 psig) and 2.16 m/sec peak (900 rpm). This rate and the post test value 0.0012 cc/sec at 54.48 hours into the test were the only values obtained below 0.002 cc/sec for the entire seal set. The second set of Bell seals indicated leakage rates below 0.03 cc/sec. This test was initiated at 2.16 m/sec peak (900 rpm) and after 4.62 hours was accelerated to 7.19 m/sec peak (3000 rpm). As the peak rod speed was obtained, the leakage rate increased dramatically to 0.78 cc/sec. This was the largest rate obtained by this seal set. Bell seal set #3 provided leakage rates from .0033 to .0100 cc/sec at test conditions of 5.28×10^6 PA (750 psig) and 2.09 m/sec peak (1750 rpm). The Bell seal set #4 was the final set run with pressure at 5.28×10^6 PA (750 psig), and the rod speed was step-incremented between 1.03 m/sec peak and 4.79 m/sec peak (430 and 2000 rpm) to allow seal temperatures to remain near 408 K (275°F). The most promising leakage results were obtained from this seal set with the majority of leakage data for both seals below 0.02 cc/sec. Also, a large set of data was obtained below the 0.002 cc/sec leakage goal with some values as low as 0.001 cc/sec over 5-minute time increments.

5.1.4 QUAD SEAL

Four sets of Quad seals were evaluated. The first set was run at 1.22×10^7 PA (1750 psig) and 7.19 m/sec peak (3000 rpm). The final leakage rates were 0.60 and 0.63 cc/sec for seals #1 and #2 and failure occurred after two minutes of testing. The second set was run at 5.28×10^6 PA (750 psig) and rod speed was increased from 1.03 m/sec peak (430 rpm) to 2.09 m/sec peak (1750 rpm). Leakage rates obtained from this set were significantly better than those of the previous run. Most of leakage data obtained were below 0.05 cc/sec, with one value for seal #2 as low as 0.0016 cc/sec for one minute. Seal set #3 was run with a helium supply pressure of 5.28×10^6 PA (750 psig) and an initial test rod velocity of 1.03 m/sec peak (430 rpm) with a velocity buildup to 4.79 m/sec peak (2000 rpm). Leakage rates for this seal set were as high as 0.01 cc/sec. For seal set #2, some leakage rates obtained were as low as 0.0011 cc/sec. Seal set #4 was exposed to a series of test runs alternating in steps between 1.03 m/sec peak (430 rpm) and 4.79 m/sec peak (2000 rpm). Most of the leakage results for this seal set were below 0.03 cc/sec.

5.1.5 TETRASEAL

Five sets of Tetraseals were evaluated with all testing conducted at an initial seal temperature of 294 K (70°F), helium gas pressure of 5.16 x 10^6 PA (750 psig), and motor drive speeds stepped between 1.03 m/sec peak (430 rpm) to 4.79 m/sec peak (2000 rpm). The data from the first two seal tests were unuseable due to surface irregularities in the test rod. Set #3 revealed a significant amount of data below the leakage goal with values at the beginning of the test run from 0.0005 cc/sec to 0.0032 cc/sec. The majority of leakage rate values for this set were below 0.005 cc/sec indicating good accuracy of test data. For the fourth set, considerable data were obtained below 0.0023 cc/sec for both test seals. This run provided the largest amount of leakage information at or near the leakage criteria limit. Similar leakage results were acquired from set #5 which was also run at either 1.03 m/sec peak (430 rpm) or 4.79 m/sec peak (2000 rpm). A gradual increase in leakage rates was observed from this run beginning below the seal leakage goal and extending to above 0.01 cc/sec before failure.

5.1.6 DYNABAK BACKUP RING

The majority of leakage results obtained for the first set of Dynabak backup ring and Buna-N (Nitrile) O-ring combination were below 0.01 cc/sec. Rates gradually increased during the test run until failure of seal #2. The second set also failed under operating conditions of 1.03 m/sec peak (430 rpm) to 4.79 m/sec peak (2000 rpm), 5.28 x 10^6 PA (750 psig) helium pressure, and ambient temperatures. Most of the leakage rates obtained for this seal set were between 0.01 cc/sec and 0.02 cc/sec.

5.2 ENDURANCE RESULTS

Endurance results for each of the reciprocating seals have been arranged in table 2, showing maximum continuous run time for one seal set, maximum accumulated time for one seal set, and total accumulated test time for all seals of one configuration. The Bell seal had the highest total accumulated test time of 3818 minutes (63.6 hours, 2.94 million cycles). Maximum continuous run time for this seal at 1.22 x 10^7 PA (1750 psig) and 2.16 m/sec peak (900 rpm) was 548 minutes (9.13 hours, 0.493 million cycles).

Testing of other seal designs resulted in less satisfactory endurance limits. Because leakage rates were much higher than anticipated, endurance testing of individual seal configurations was discontinued to allow leakage testing of alternate seal designs.

5.3 SEAL FRICTION RESULTS

Friction testing of the Boeing Footseal, NASA Chevron polyimide seal, and Bell seal was performed prior to seal endurance runs with results shown in table 3. The Boeing Footseal design showed the least amounts of frictional loading with a breakaway value of 58.7 N/seal (13.2 lb/seal), and a creep .635 cm/sec (0.25 in/sec) frictional load of 52.5 N/seal (11.8 lb/seal). These frictional forces were obtained at conditions of 291 K (65°F) and 1.01 x 10^5 PA (0 psig). The increase in friction due to a temperature from 291 K (65°F) to 408 K (275°F) was approximately 14.2 N/seal (3.2 lb/seal). With an increase of helium gas pressure from 1.01 x 10^5 PA

(0 psig) to 1.22 x 10^7 PA (1750 psig) the increase in friction was 36.9 N/seal (8.3 lb/seal) at 291 K (65°) and 17.4 N/seal (3.9 lb/seal) at 408 K (275°F). The NASA Chevron polyimide seal gave the highest breakaway frictional forces per seal with a maximum value at ambient conditions of 497.7 N/seal (112 lb/seal) and an average value of 409.3 N/seal (92.1 lb/seal). Frictional forces under creep conditions were not obtainable due to unstable conditions (rod stick/slip effects) just after breakaway.

5.4 FAILURE MODE DISCUSSION

During the course of dynamic leakage testing of the six reciprocating seal configurations, various types of failure modes were observed. Failure descriptions are also listed from each seal set in table 1. It was possible to categorize all failure types encountered as listed:

1. overtemperature failure - Smoke generation from seal and burning seal odor caused by high seal friction.

2. catastrophic leakage failure - Leakage manometer overflow due to pressure in helium accumulation reservoirs forcing oil out into the manometers.

3. drive linkage failure

   a) characterized by high vibration

   b) test rod buckling under frictional loads

   c) failure of bearing surfaces in retention fixture

   d) test rod bending

   e) varidrive bearing failures

4. high leakage rates - no failure

5. gas hydrocarbon weepage failure - helium cavity filled with hydraulic fluid

Of these five failure types, specific seal configurations were of one type of failure mode more than another. The NASA Chevron seal, because of its high frictional characteristics, tended to cause more drive linkage failures than the other seal designs. The Bell seal also had one failure of this type. All seals displayed high leakage and overtemperature failures. Catastrophic leakage failures were observed with the Boeing Footseal, Bell seal, and Quad seal all showing rapid manometer level rises. Figures 16 through 19 are photos of the types of failure modes for the Boeing Footseal, NASA Chevron Polyimide seal, Bell seal, and Quad seal. These photos show the most dramatic failure occurrences of each seal. Figure 20 shows the high frictional abrasion failure mode of the dynamic test rod.

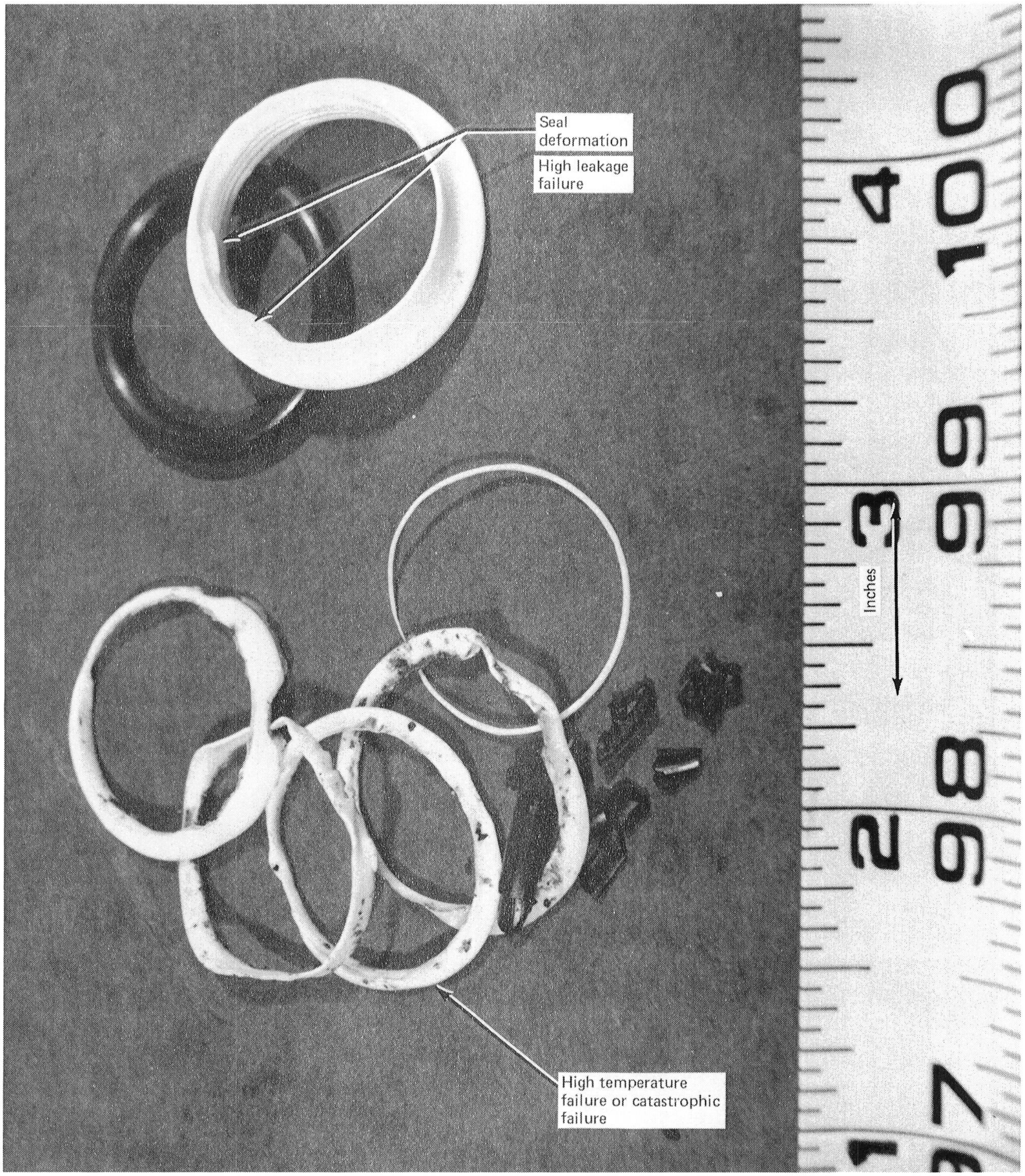


Figure 16. — Boeing Footseal Failures

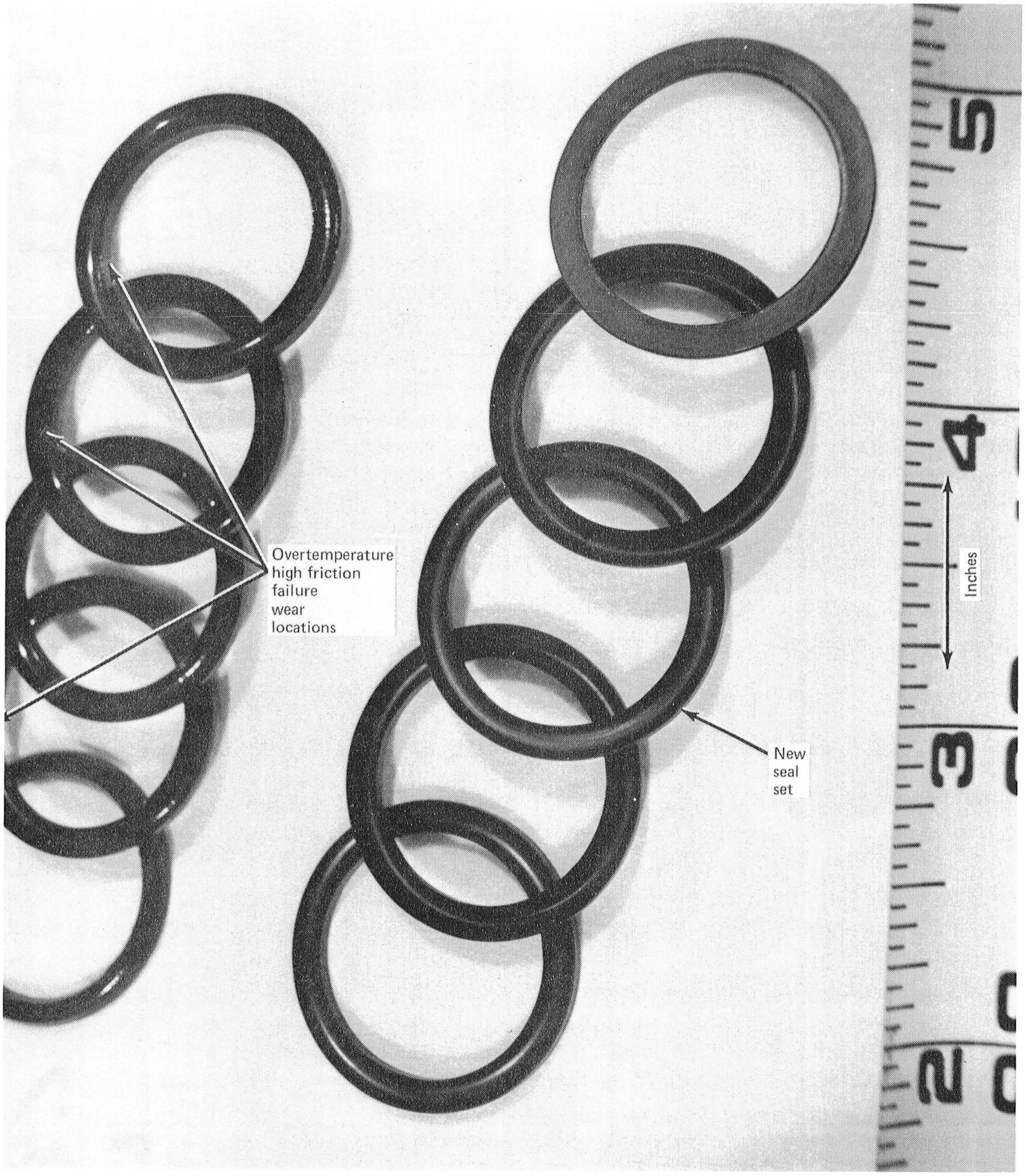


Figure 17. – NASA Chevron Seal Failure

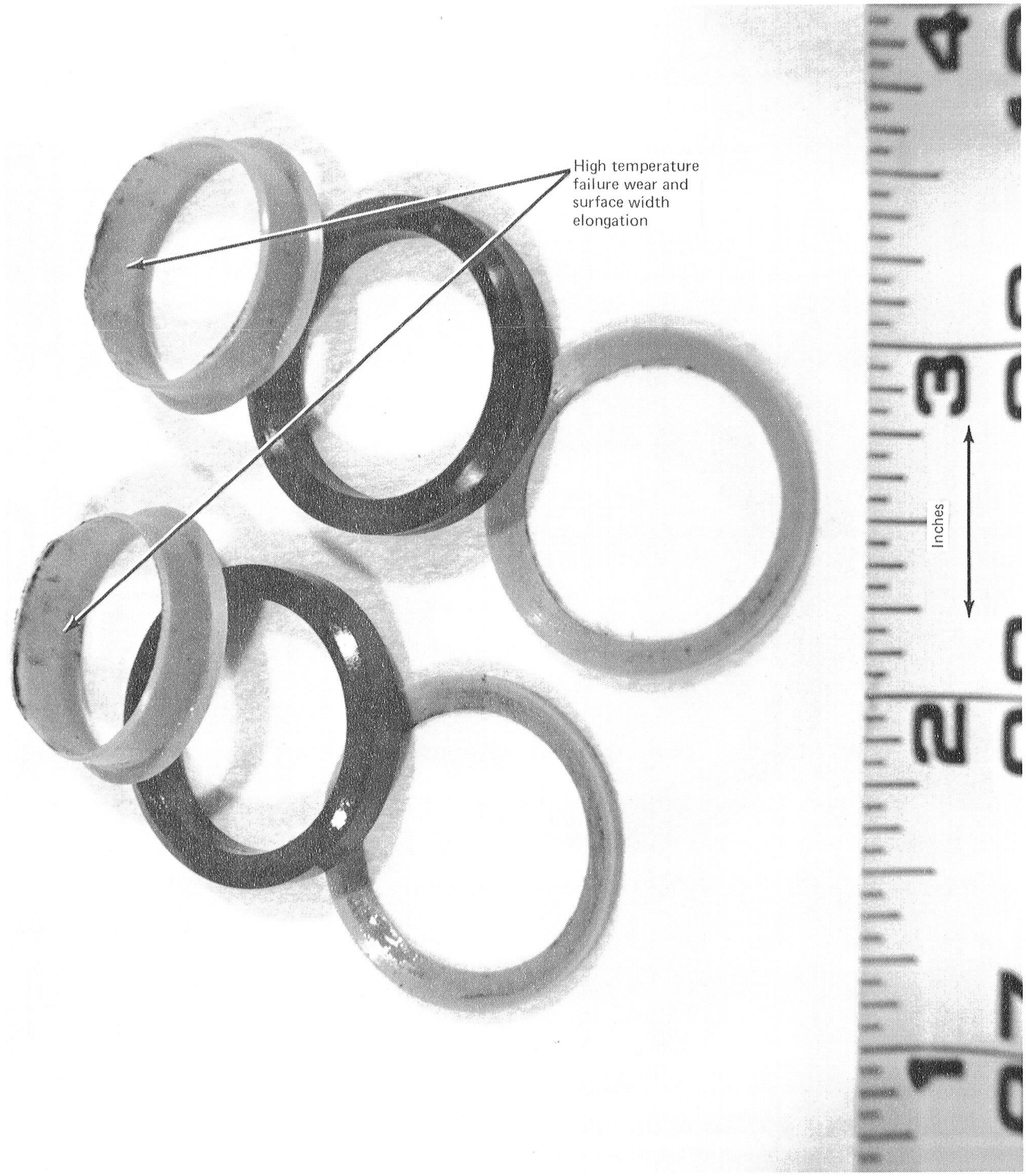


Figure 18. — Bell Seal Failure

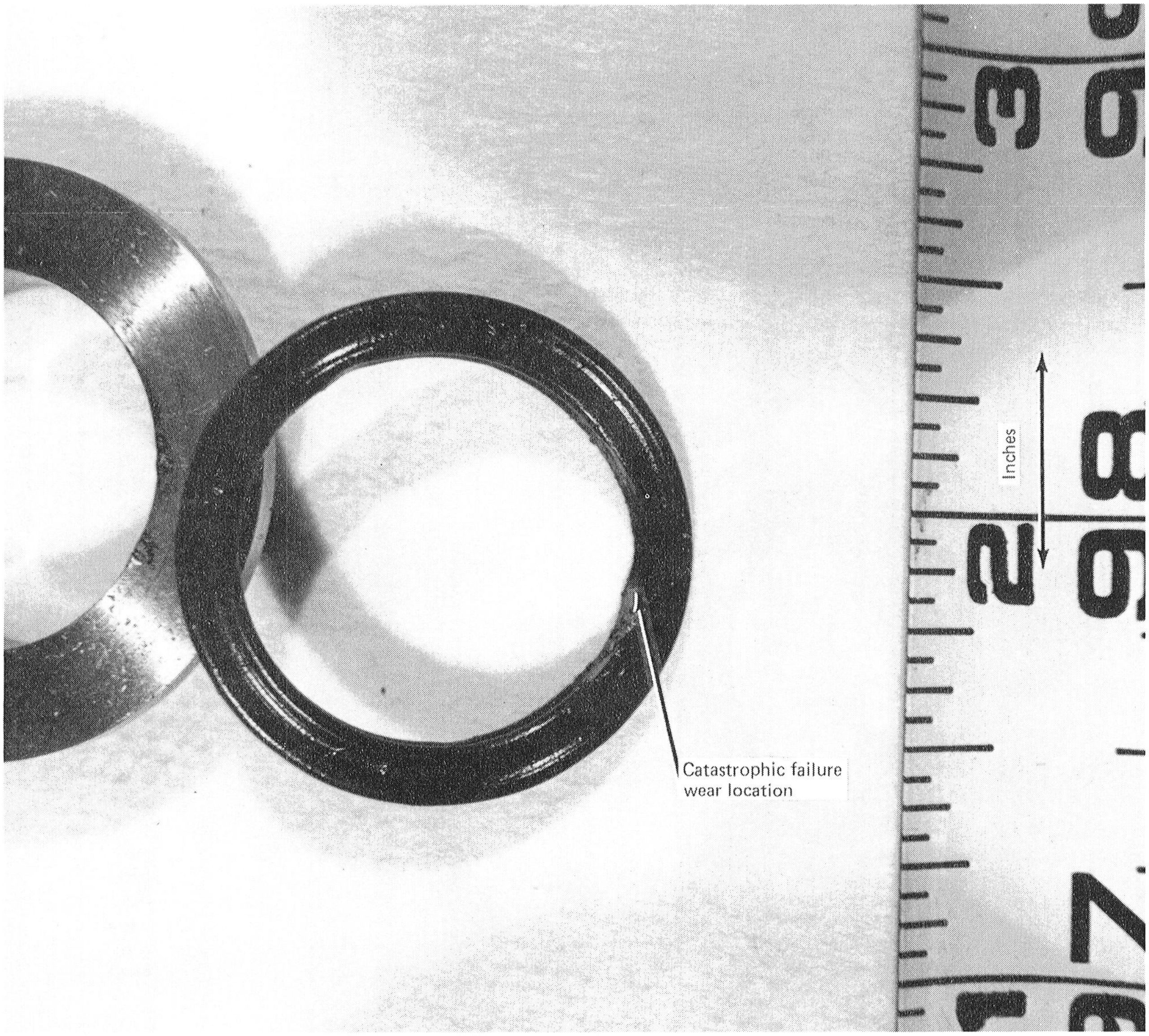


Figure 19. — Quad Seal Failure

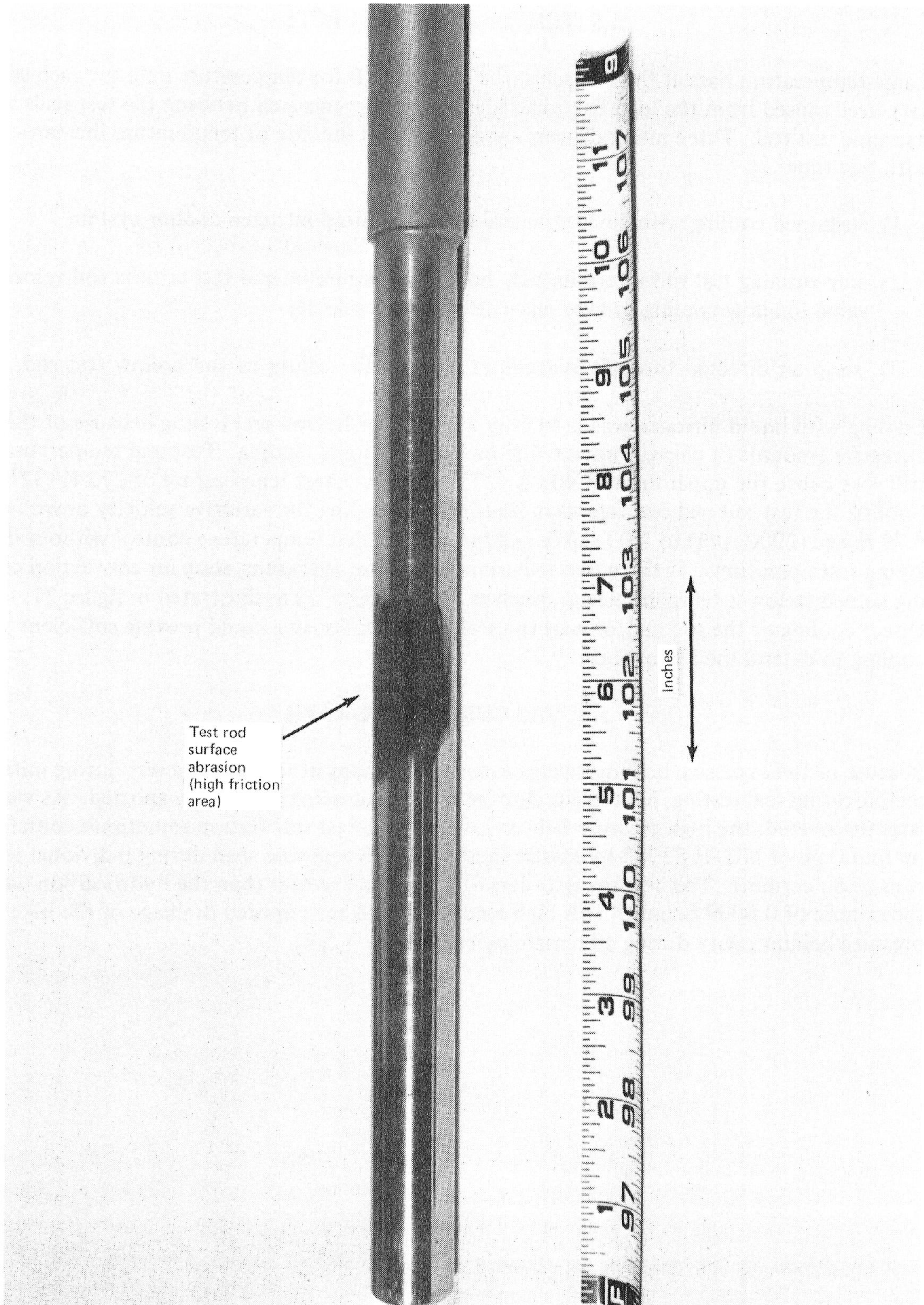


Figure 20. – Test Rod Surface Failure

5.5 TEMPERATURE EFFECTS

Large temperature rises at the test seals (see Appendix B for temperature data for each seal set) were caused from the high frictional loading forces generated between the test seals and dynamic test rod. Three methods were used to control the rate of temperature increase with test time:

1) sustained cooling with environmental chamber liquid nitrogen cooling system

2) step-running test rod intermittently between a minimum and test criteria rod velocity value to allow cooling to take place at the lower velocity

3) shop air directed through hydraulic tubing to the center of the hollow test rod.

Cooling with liquid nitrogen was used only during the Chevron seal testing because of the excessive amounts of energy generated from high frictional loading. Test seal temperatures still rose above the upper limit of 408 K (275°F) from a start temperature of 273 K (32°F). Cooling the test rod and seal retention housing by stepping the varidrive velocity down from 4.79 m/sec (2000 rpm) to 1.03 m/sec (430 rpm) provided temperature control without delaying test operations. If these two techniques were not sufficient, shop air convection cooling helped to lower temperatures as much as 18.6 K (33.5°F) as illustrated in figure 21. Direct cooling of the rod and/or near the seal retention housing could provide sufficient cooling to extend the life of seals.

5.6 GAS CHROMATOGRAPH

Because of the excessive helium leakage rates encountered at the manometers during initial reciprocating seal testing, helium gas chromatograph sampling plans were aborted. As was later discovered, the high pressure helium cavity of the test seal fixture sometimes contained up to 4.0 ml of MIL-H-83282 hydraulic fluid after a five-minute span during individual test runs (0.80 cc/min). This was many orders of magnitude greater than the hydrocarbon weepage criteria of 0.0009 cc/min. This high amount of leakage required drainage of the high pressure helium cavity during all remaining test runs.

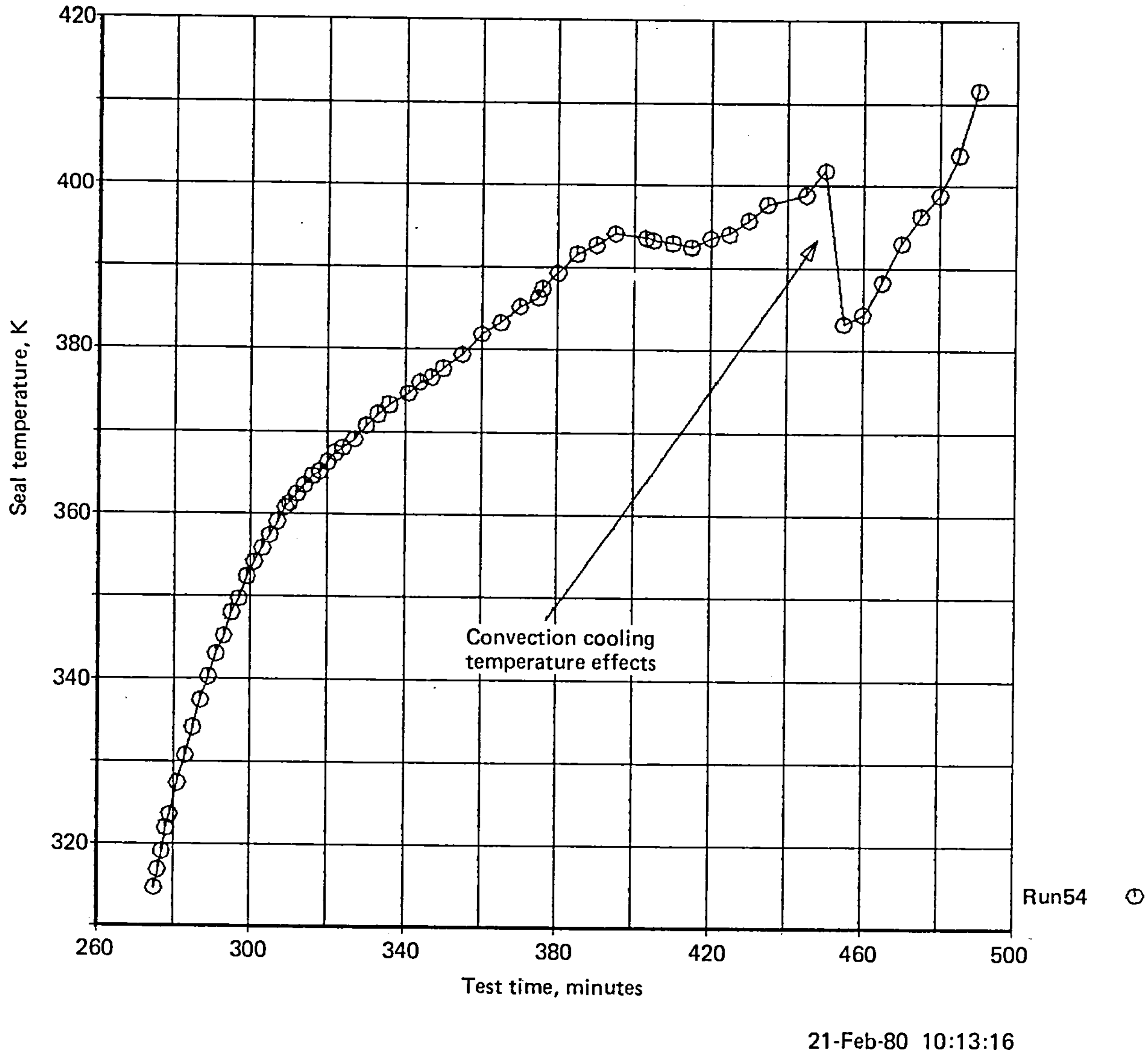


Figure 21. — Convection Cooling Temperature Effects, Quad Seal Configuration

6.0 CONCLUSIONS AND RECOMMENDATIONS

Completion of dynamic leakage testing of six different single-stage reciprocating piston rod seals for Stirling cycle engine applications has revealed poor helium leakage and endurance results for all seals evaluated; Boeing Footseal, NASA Chevron polyimide seal, Bell seal, Tetraseal, Quad seal, and Dynabak seal. It is concluded that none of the six seals evaluated is acceptable for the Stirling cycle engine reciprocating seal application in their present form. The Tetraseal showed the lowest leakage rates with an average minimum rate of .014 cc/sec (still above the leakage goal of 0.002 cc/sec). The Bell seal displayed the most promising endurance results with total run time on one seal set of 63.6 hours, while the endurance criteria was 1500 hours. The Bell seal had an average minimum leakage rate of .026 cc/sec. In general, the elastomeric seals (Quad, Tetraseal, and Dynabak) in contact with the test rod did perform slightly better with respect to leakage than the Bell seal or Footseal. Frictional forces of the NASA Chevron polyimide seal were particularly high. High frictional forces produced at the seal and test rod contact surface generated heat levels that elevated seal temperatures above the test criteria value of 408 K (275°F), and eventually caused test seal failure.

In any future investigation of seals in a Stirling cycle environment, it is concluded that some method of shaft and/or seal cooling is required. A recommended method of cooling the seal environment is to direct a pressurized air or oil mist at the test rod at the seal interface. Cooling fluid jackets should be provided in the seal retention housing as close to the test seal as possible to carry away additional heat. Additional cooling can be provided by using forced convection on the inside diameter of the test rod. These cooling methods will improve the endurance life of the seal but will not necessarily decrease the high leakage rates of the various seals.

To improve leakage characteristics it is recommended that seal configurations be investigated that: 1) lengthen the leakage path; 2) minimize leakage path clearance with the rod; and 3) provide intermediate venting of leakage path. In addition, the use of Teflon-filled materials (bronze, fiberglass and/or graphite) will improve the seal rigidity (endurance characteristics) over the virgin Teflon of the Footseal or the Bell seal. To improve the testing system, it is recommended that: 1) a strain-gaged link be used between test rod and varidrive to obtain both static and dynamic friction; 2) a larger tubing (3/8 inch) be used from test block to manometer to minimize resistance to helium leakage flow which would help reduce the random nature of the leakage measurements; and 3) an improvement be made in the strength and/or surface hardness of test rod.

APPENDIX A
TEST SYSTEM MODIFICATIONS

During the schedule of reciprocating seals testing, it was discovered that several system modifications were necessary to properly continue testing. This Appendix discusses all modifications that were implemented during the program.

The initial test setup, as shown schematically in figure 22, had a 1/4-inch O.D. hydraulic line from the top of the test fixture to the base of the helium accumulation reservoir. The line from the bottom of the test fixture was attached to the helium leakage manometers. This bottom line was of 1/8-inch O.D., and the smaller diameter acted as a flow restriction of oil flow to the leakage manometers, but was also designed to reduce helium gas passage.

Because of the high helium leakage rates encountered up to this time, helium leakage was not confined exclusively to the helium accumulation reservoirs, but was discovered as helium gas bubbles rising directly in the leakage manometers. As pressure was built-up in the helium accumulation reservoirs, the alternate path to the leakage manometers provided less resistance to helium flow. Therefore, the first plumbing modification was made by rerouting the line from the top of the test fixture to a stand tube in the base of the helium accumulation reservoir. From there, another line from the base of the helium accumulation reservoir was connected to the leakage manometers (figure 23). The stand tube assured helium accumulation in the top volume of the accumulation reservoir with remaining oil being displaced out of the reservoir base to the leakage manometers.

As significant wear was discovered on the test rod at the dynamic seal contact region, it was determined that a more efficient method of seal lubrication was necessary. In the second system plumbing change, the line out of the bottom of the test fixture was connected to a separate oil lubrication reservoir. This reservoir was pressurized with 1.36×10^5 PA (5 psig) air and used to maintain lubrication oil supply to the test rod assembly.

A major change to the test system resulted from a consistent failure of the connecting rod end bearing between the varidrive and test rod assembly. The high dynamic loading forces and rotational cycling velocities revealed that it was mandatory to redesign the bearing to provide high loading and long life properties. The bushing bearing was then replaced with a self-aligning ball bearing assembly designed specifically for this application.

With the replacement of the connecting rod end bearing assembly, a much greater amount of run time was achieved. However, due to the rapid rise in temperature at the bearing contact surfaces, a wet lubrication system was developed to provide a method of heat transfer away from the rod bearing assembly. This modification is described in detail in section 4.5.

Additional test system modifications occurred in September and October of 1979. The first modification in this set was to use air convection directed inside the hollow test rod to provide an alternate method of removing heat from rod/seal during rapid cycling.

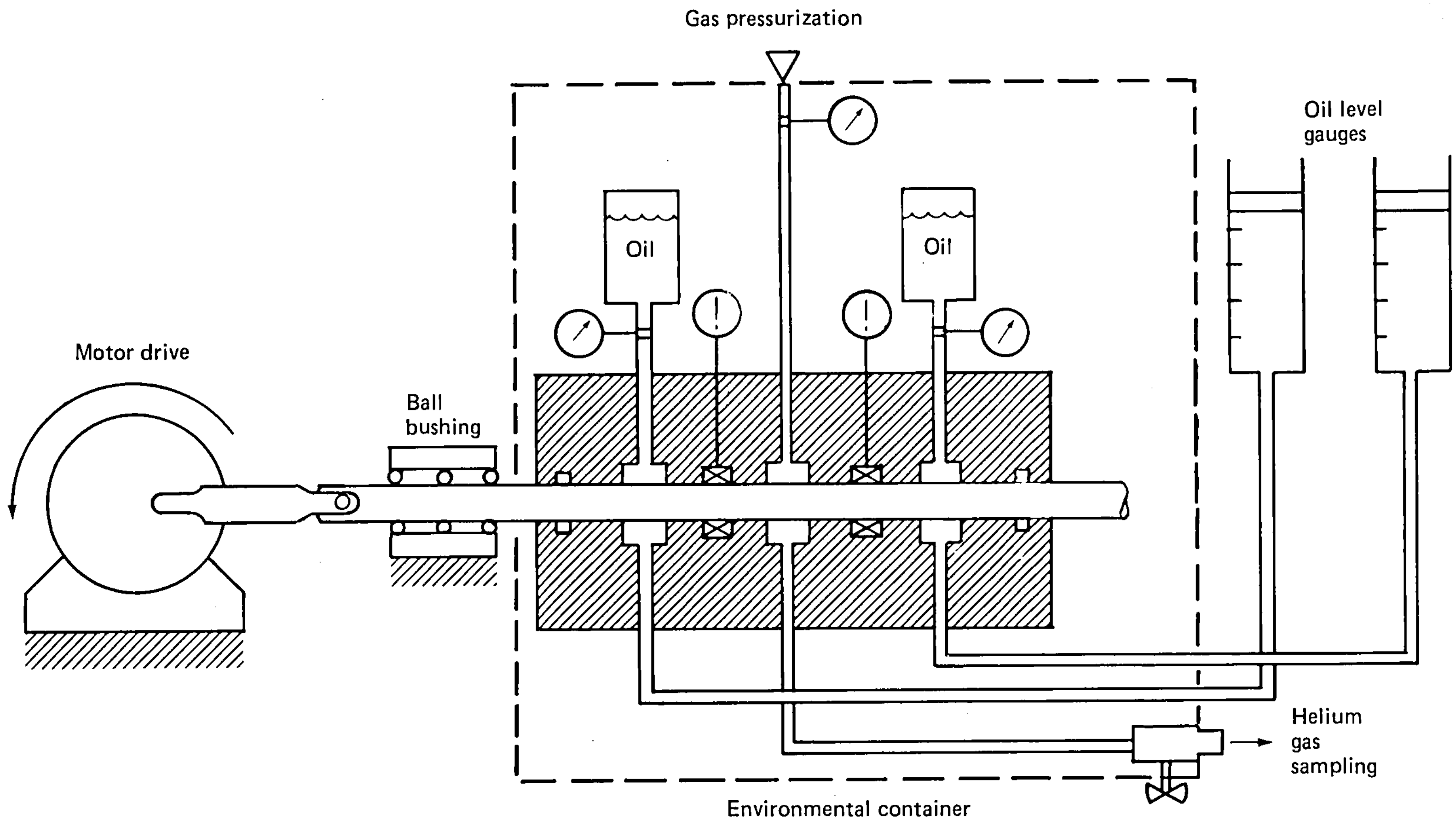


Figure 22. – Test System Schematic

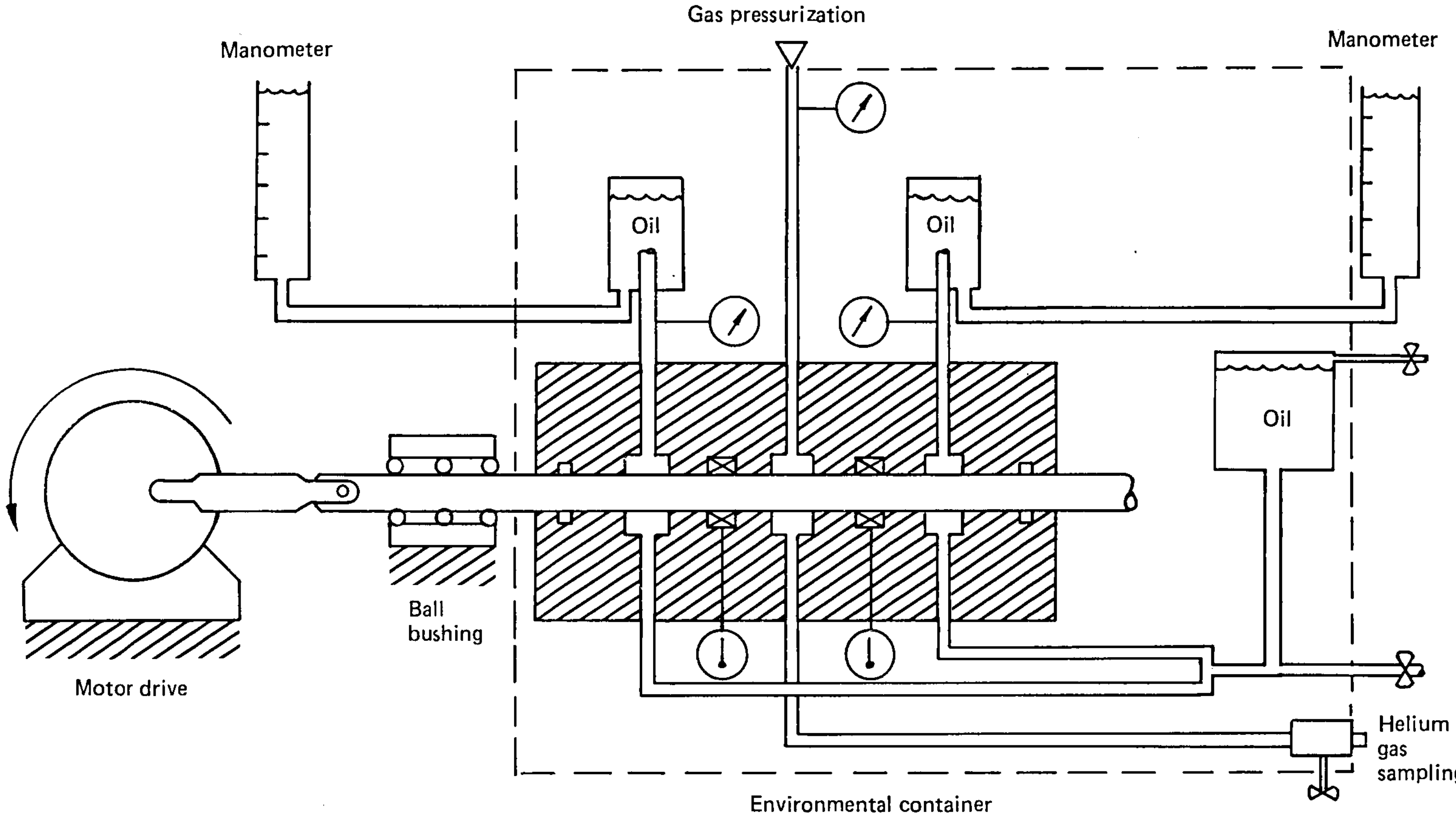


Figure 23. — Test System Schematic, Modification 1

The third plumbing change was made at the helium accumulation reservoirs so that both the inlet and outlet (with stand pipe) were at the top of the reservoir (figure 7). This assured helium pressure against the hydraulic fluid in the reservoir and only one leakage path for oil out of the reservoir through the stand tube. This helped to eliminate negative leakage rates from manometer oil draining back into the helium accumulation reservoirs during testing. The line from the helium accumulation reservoirs to the leakage manometers was also plumbed from the top of the reservoir so all connections at the helium accumulation bottle could easily be inspected for leakage. Previous connections at the base of the helium accumulation reservoirs were blocked.

During a posttest inspection of a seal test in September 1979 it was discovered that the oil lubrication cavities of the test fixture were dry. Oil cavity-level sight tube assemblies were installed as shown in figure 12. The sight tubes were adjusted to bring the oil levels in the test fixture up to the same elevation as the top of the test seals. In this way, the total contact surface of the test seals were exposed to the lubricating fluid and there was still enough open volume at the top of the oil cavity to provide an unobstructed leakage path for helium flow to the accumulation reservoirs. The sight tubes were installed as a parallel loop between the top and bottom of the test fixture oil lubrication cavities. The oil cavity-level control was obtained by installing an on/off solenoid valve inline from the oil lubrication reservoir to the oil lubrication cavities at the test fixture.

The final requirement for modification of the reciprocating seals evaluation system was discovered during a posttest inspection of the seal retention fixture. Lubrication oil from the oil lubrication cavities (as much as 2 cc's) on the low pressure side of the test seals had bled across the seals into the helium cavity of the test fixture. Because this would allow a much smaller helium exposure area to the test seal, and thus affect leakage results, the test fixture was again modified to provide oil drainage from the helium cavity.

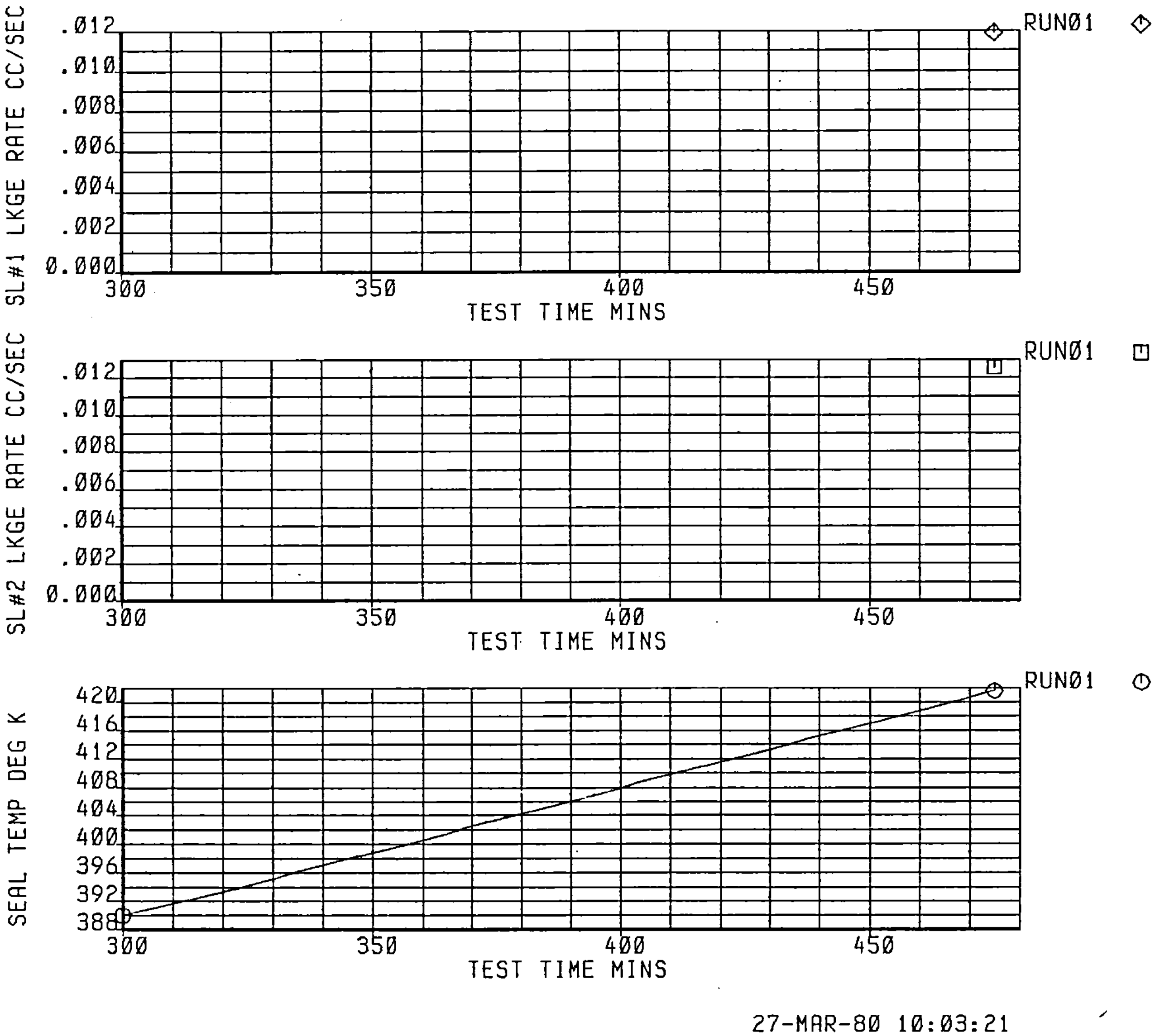
APPENDIX B
SEAL LEAKAGE AND TEMPERATURE DATA

Seal Test Runs

Run	Date	Seal Description	Pressure	RPM	Run Sequence
01	071878	Footseal set #1	1750	430	1
02	080178	Footseal set #2	1840	3000	1
03	100678	Chevron set #1	0	430	1
04	110178	Chevron set #2	1750	900	1
05	120178	Chevron set #3	1750	900	1
06	120478	Bell seal set #1	1750	900	1
07	120578	Bell seal set #1	1750	900	2
08	120678	Bell seal set #1	1750	900	3
09	121378	Bell seal set #1	1525	900	4
10	122078	Bell seal set #1	1775	900	5
11	122178	Bell seal set #1	1750	900	6
12	010279	Bell seal set #1	1750	900	7
13	010379	Bell seal set #1	1750	900	8
14	010579	Bell seal set #2	1750	900	1
15	010579	Bell seal set #2	1750	3000	2
16	011079	Bell seal set #2	1750	3000	3
17	071179	Quad seal set #1	1750	3000	1
18	091079	Tetraseal set #1	750	400-2000	1
19	091179	Tetraseal set #2	750	1750	1
20	092479	Quad seal set #2	750	430	1
21	092479	Quad seal set #2	750	1000	2
22	092479	Quad seal set #2	750	1750	3
23	092479	Quad seal set #2	0	430	4
24	092479	Quad seal set #2	0	1750	5
25	092579	Bell seal set #3	0	430	1
26	092579	Bell seal set #3	750	430	2
27	092579	Bell seal set #3	750	1750	3
28	092579	Bell seal set #3	0	1750	4
29	092679	Tetraseal set #3	750	430	1
30	092679	Tetraseal set #3	750	1750	2
31	092879	Quad seal set #3	750	430	1
32	092879	Quad seal set #3	750	2000	2
33	100279	Footseal set #3	750	430	1
34	100279	Bell seal set #4	750	430	1
35	100279	Bell seal set #4	750	2000	2
36	100479	Bell seal set #4	750	430	3
37	100479	Bell seal set #4	750	2000	4
38	101079	Tetraseal set #4	750	430	1
39	101079	Tetraseal set #4	750	2000	2
40	101079	Tetraseal set #4	750	430	3
41	101079	Tetraseal set #4	750	2000	4
42	101079	Tetraseal set #4	750	430	5
43	101079	Tetraseal set #4	750	2000	6
44	101079	Tetraseal set #4	750	430	7
45	101079	Tetraseal set #4	750	430	8

Sea Test Runs

Run	Date	Seal Description	Pressure	RPM	Run Sequence
46	101079	Tetraseal set #4	750	2000	9
47	101079	Tetraseal set #4	750	430	10
48	101679	Quad seal set #4	750	430	1
49	101679	Quad seal set #4	750	2000	2
50	101679	Quad seal set #4	750	430	3
51	101679	Quad seal set #4	750	2000	4
52	101679	Quad seal set #4	750	430	5
53	101779	Quad seal set #4	750	430	1
54	101779	Quad seal set #4	750	2000	2
55	101779	Quad seal set #4	750	430	3
56	101979	Footseal set #4	750	430	1
57	102279	Tetraseal set #5	750	430	1
58	102279	Tetraseal set #5	750	430	2
59	102279	Tetraseal set #5	750	2000	3
60	110779	Dynabak set #1	750	430	1
61	110779	Dynabak set #1	750	430	2
62	110779	Dynabak set #1	750	2000	3
63	111579	Dynabak set #2	750	430	1
64	111579	Dynabak set #2	750	2000	2
65	111679	Dynabak set #2	750	430	1
66	111679	Dynabak set #2	750	2000	2
67	111679	Dynabak set #2	750	430	3



RPM	430
PRESS MPA	12.09

Figure 24. — Boeing Footseal Set #1 Test Data

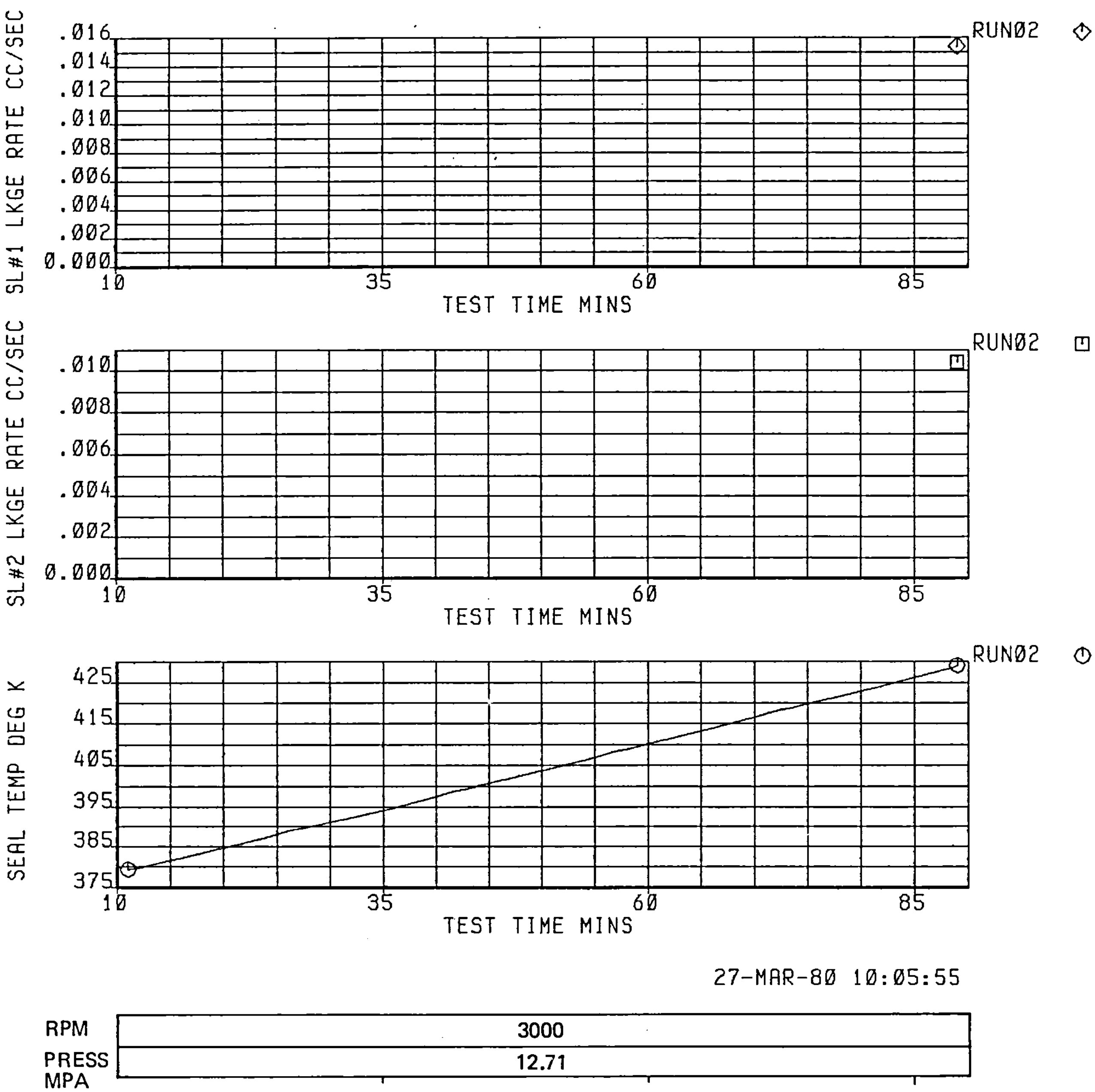


Figure 25. — Boeing Footseal Set #2 Test Data

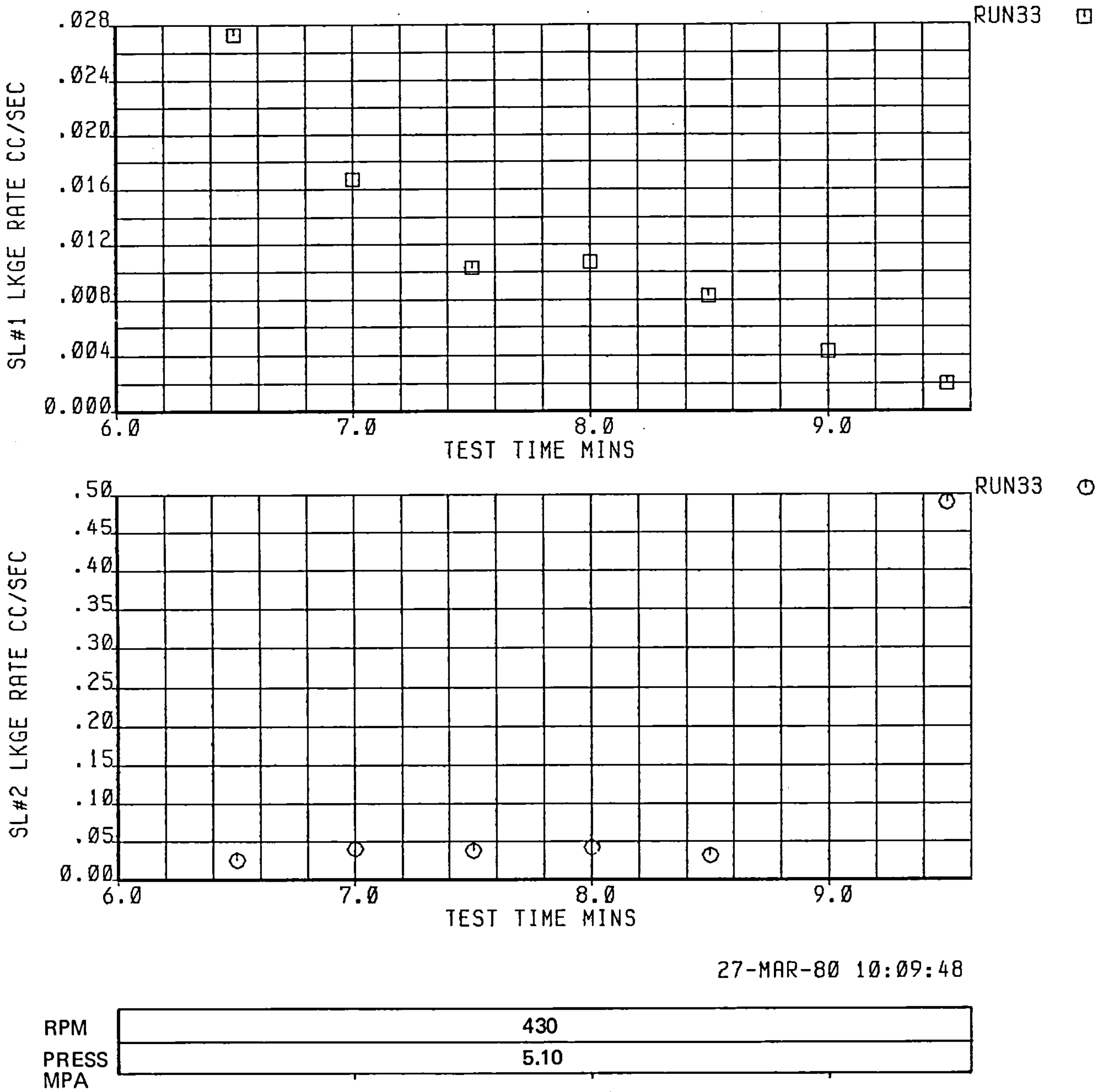


Figure 26. — Boeing Footseal Set #3 Test Data

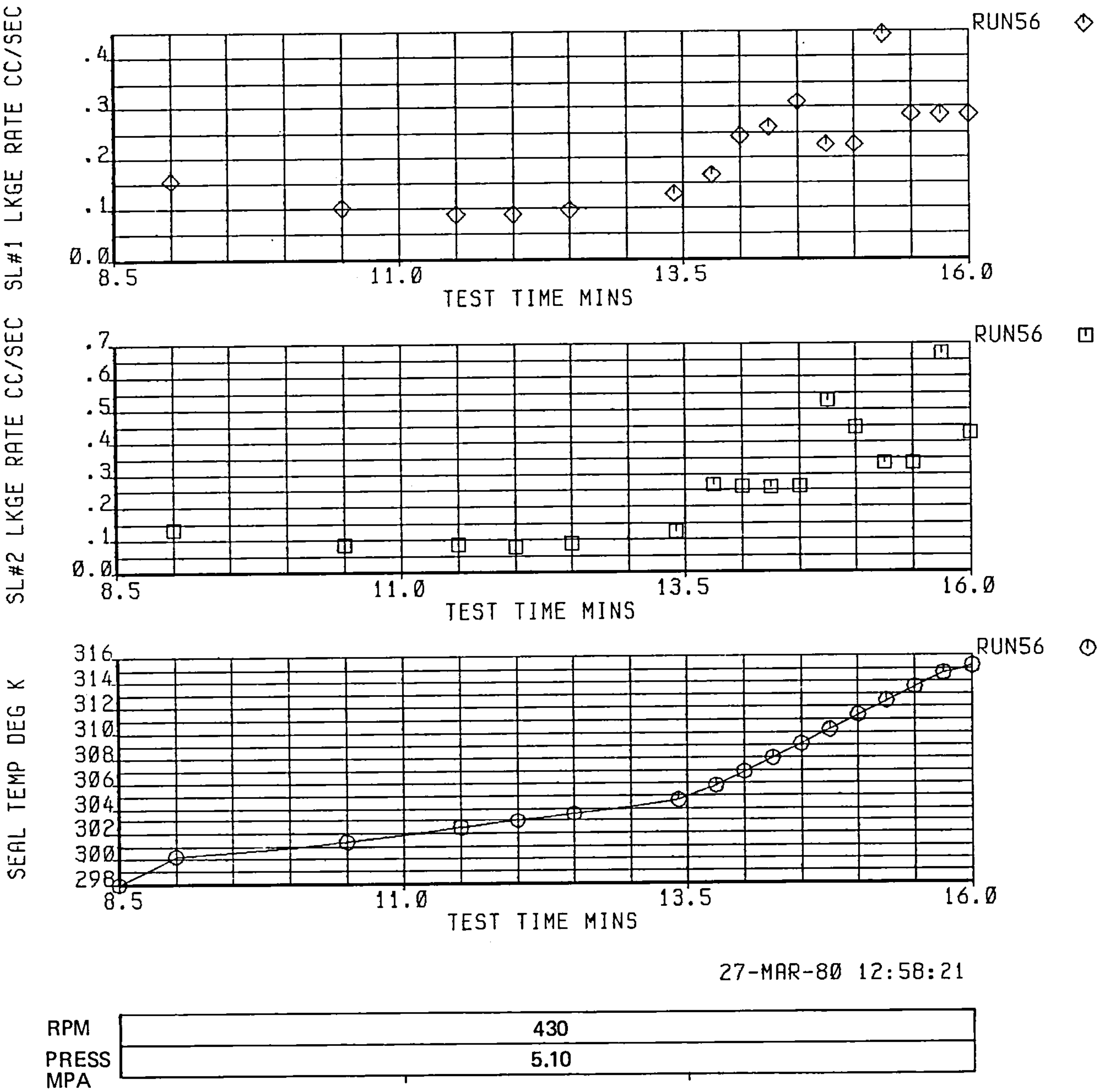
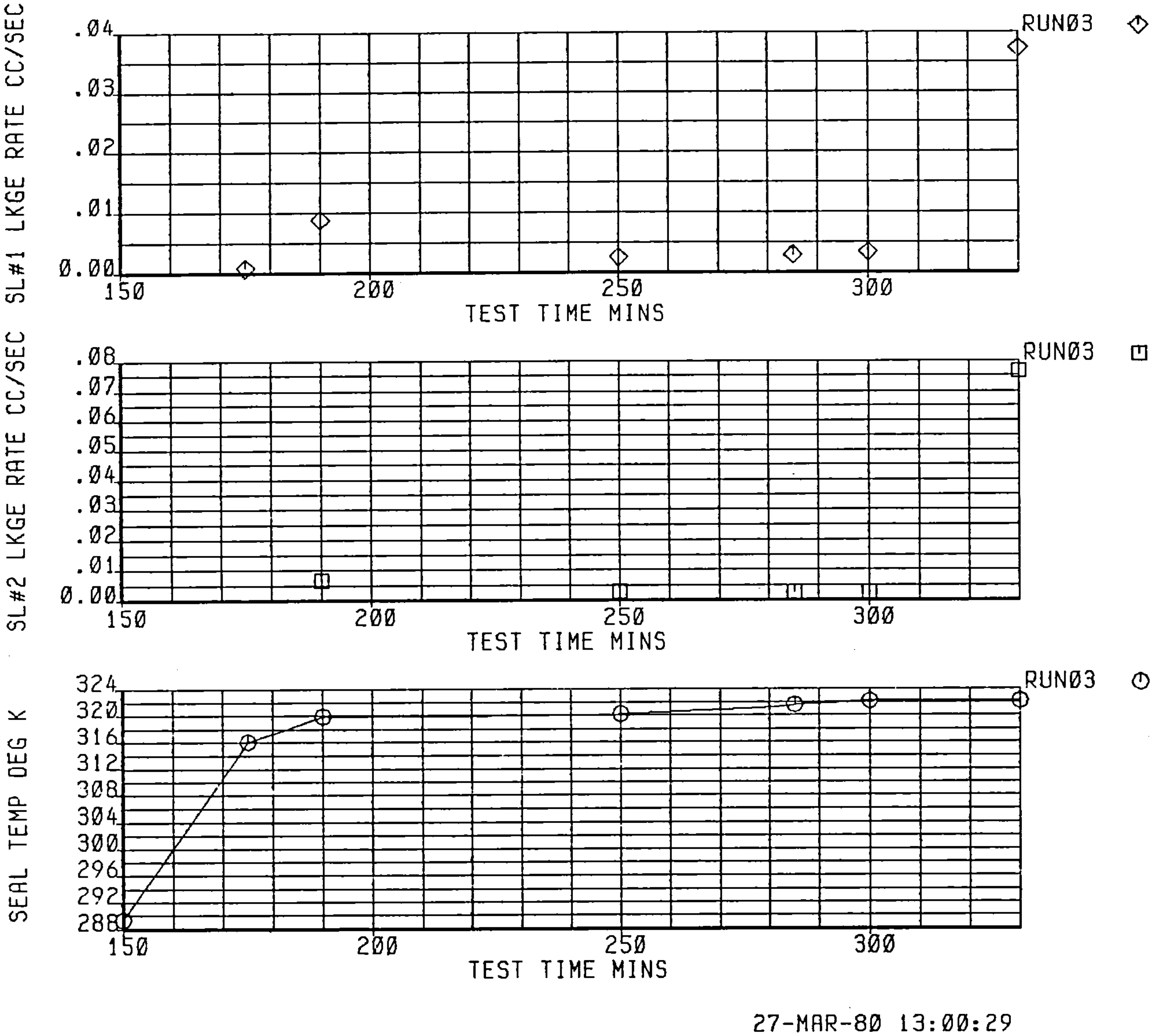
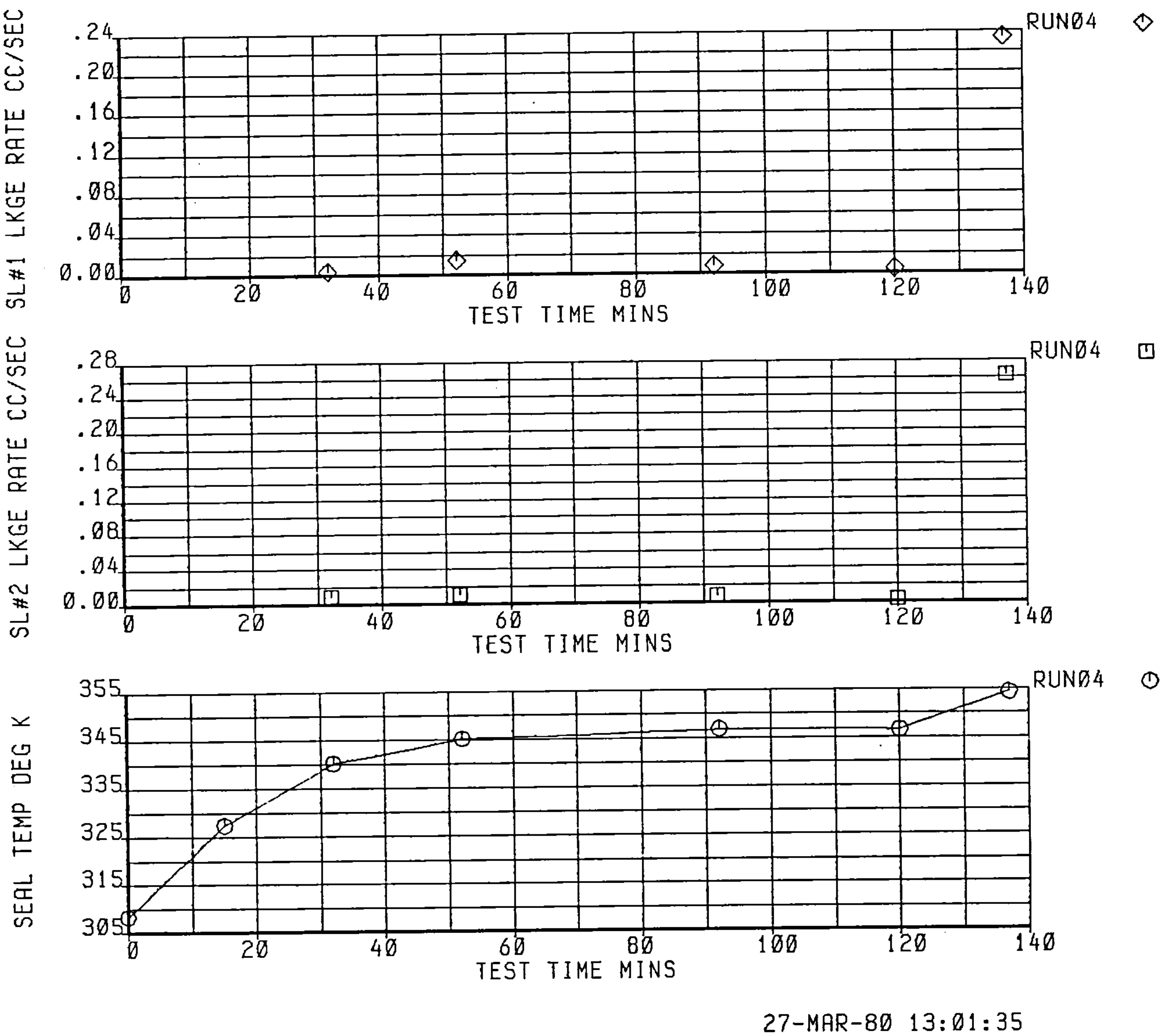


Figure 27. – Boeing Footseal Set #4 Test Data



RPM	430
PRESS MPA	.101

Figure 28. — NASA Chevron Seal Set #1 Test Data



RPM	900
PRESS MPA	12.09

Figure 29. – NASA Chevron Seal Set #2 Test Data

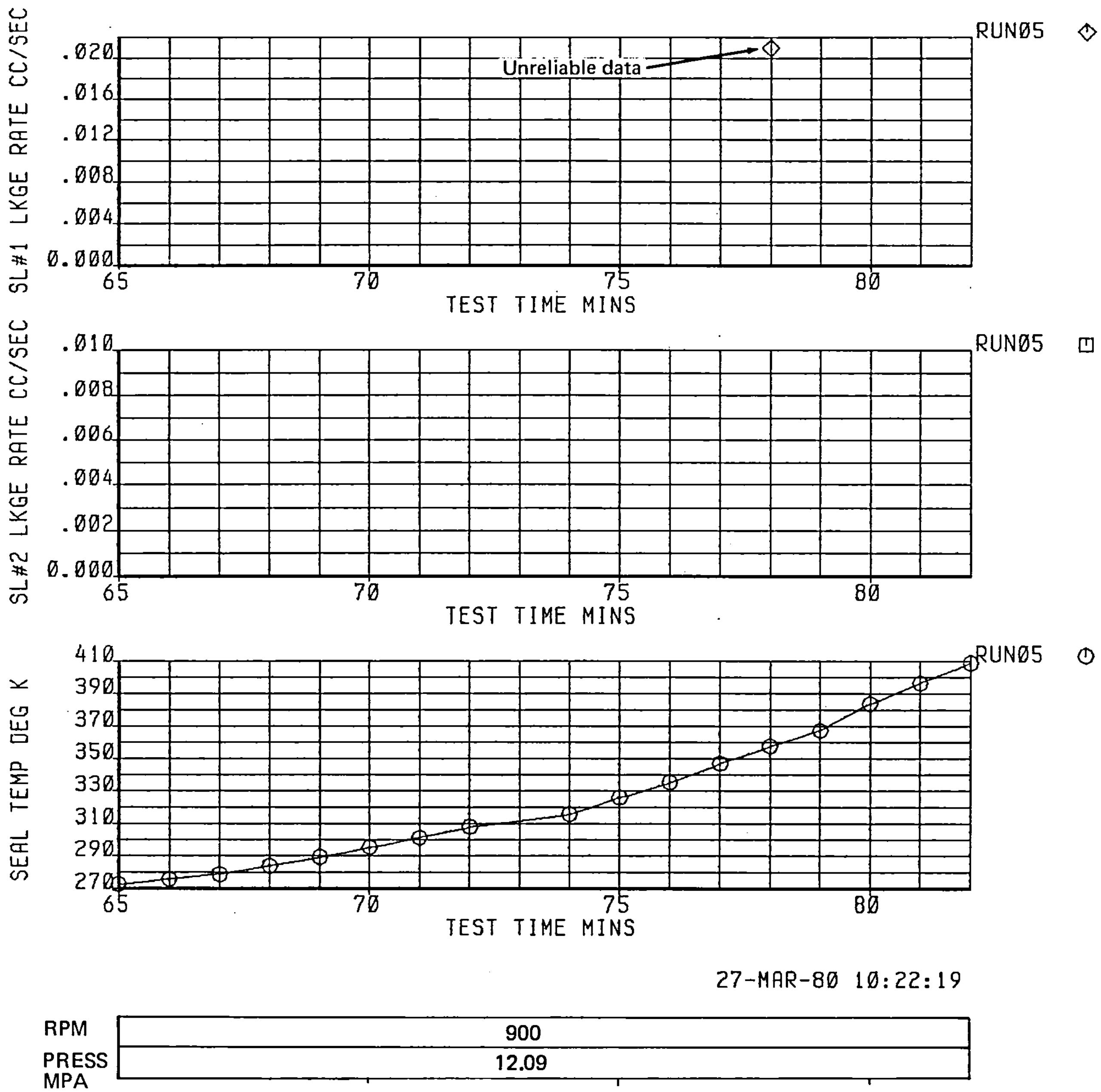
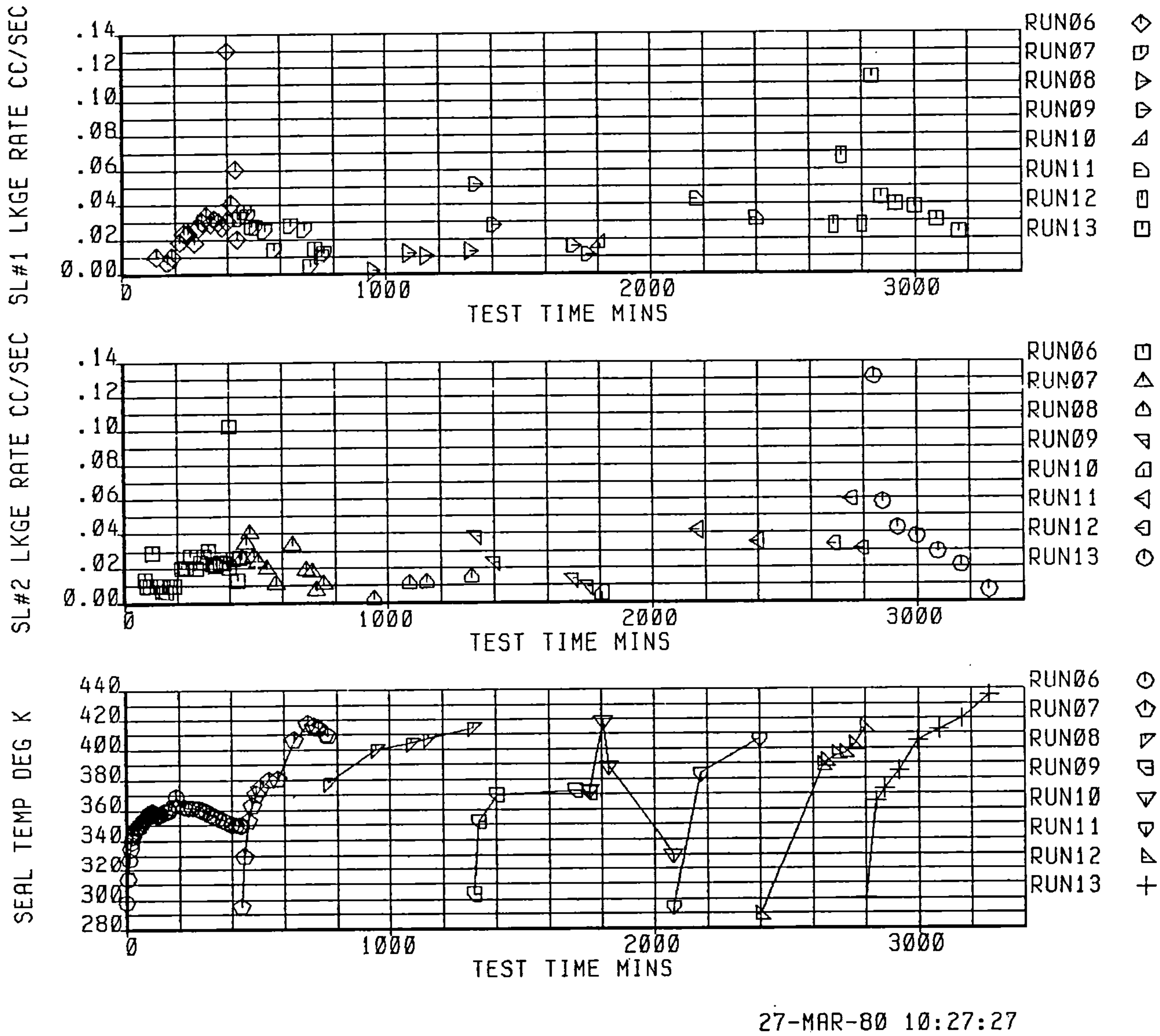


Figure 30. — NASA Chevron Seal Set #3 Test Data



RPM	900			
PRESS MPA	12.09	10.53	12.26	12.09

Figure 31. — Bell Seal Set #1 Test Data

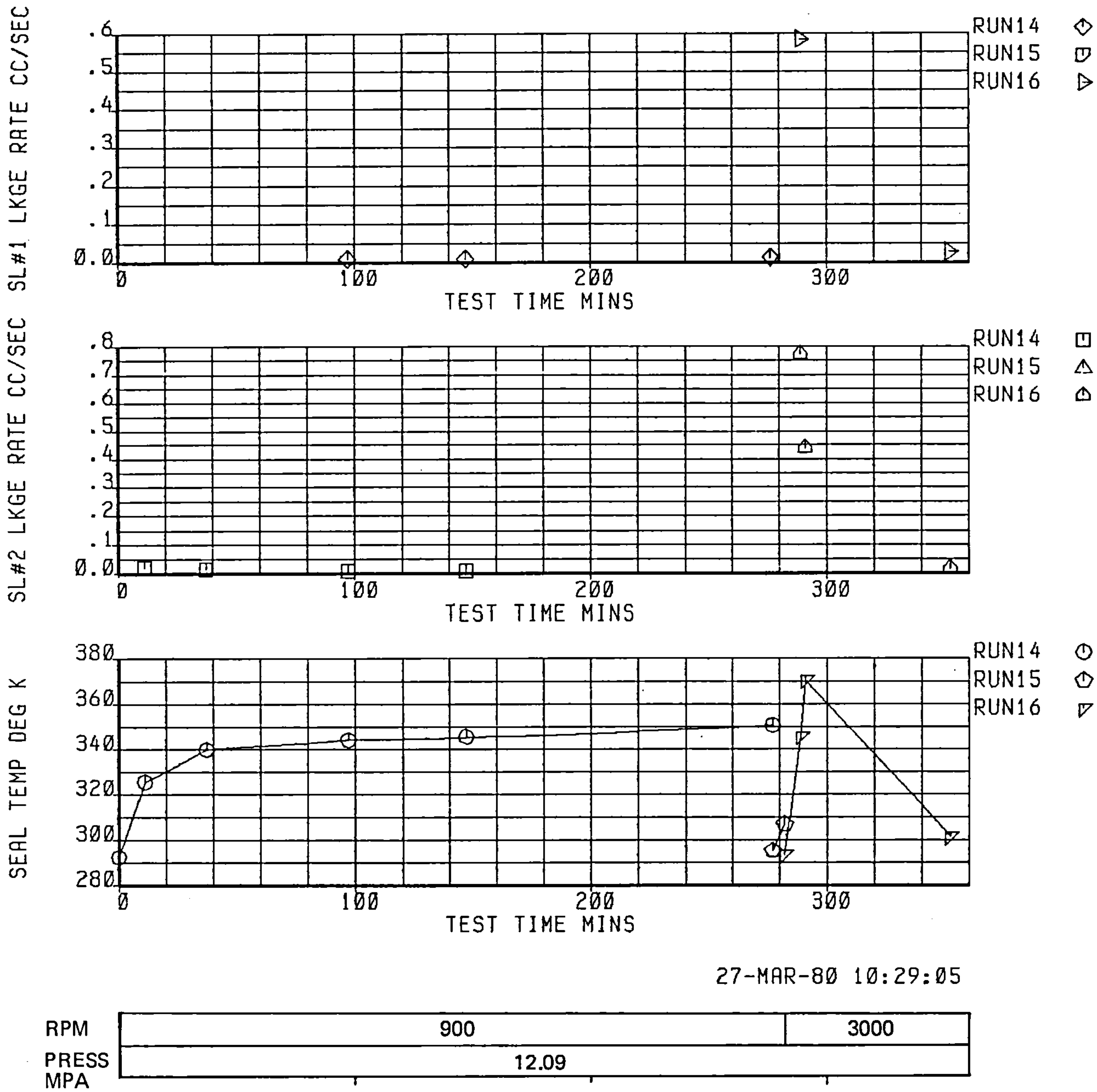


Figure 32. — Bell Seal Set #2 Test Data

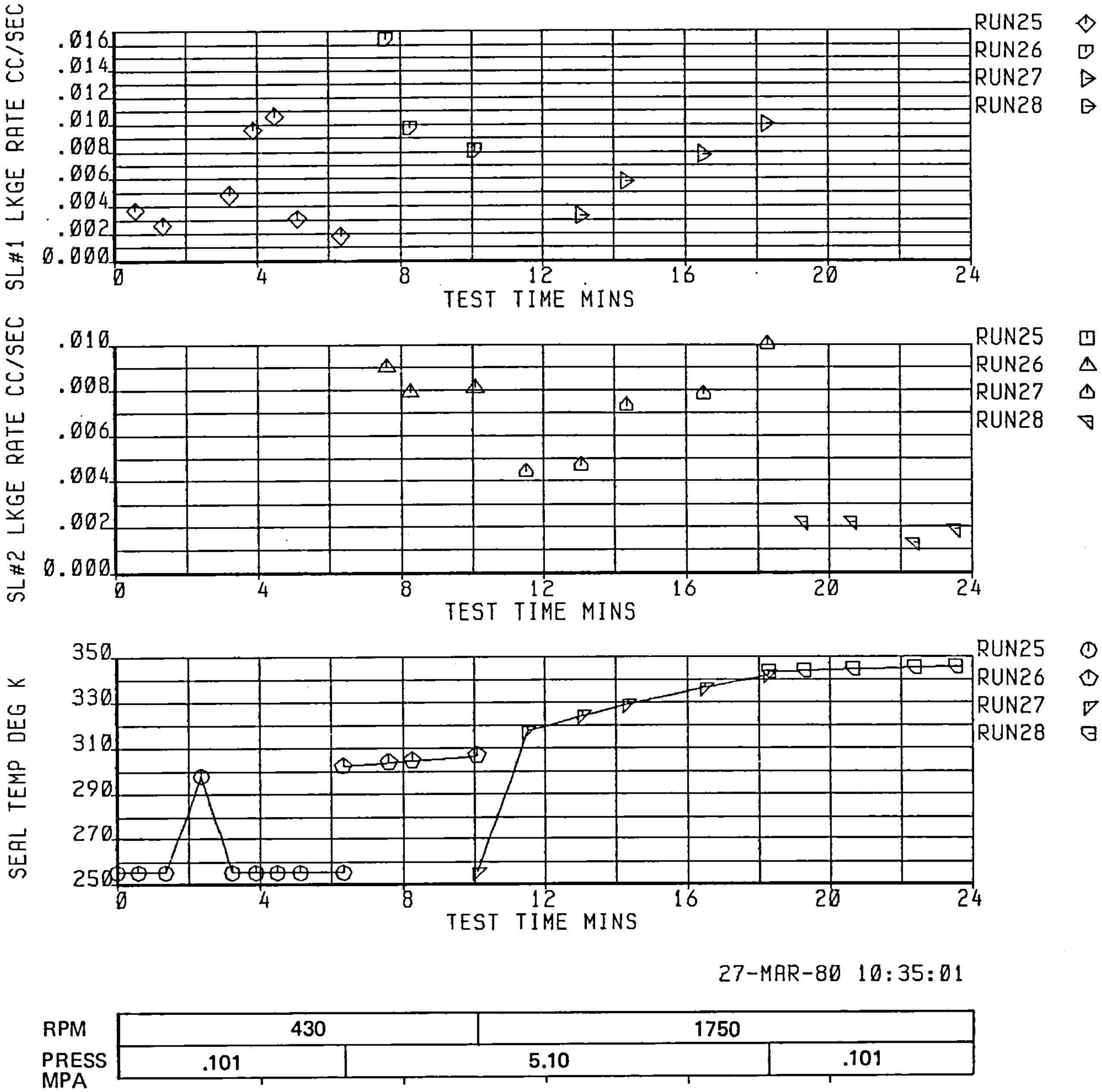


Figure 33. — Bell Seal Set #3 Test Data

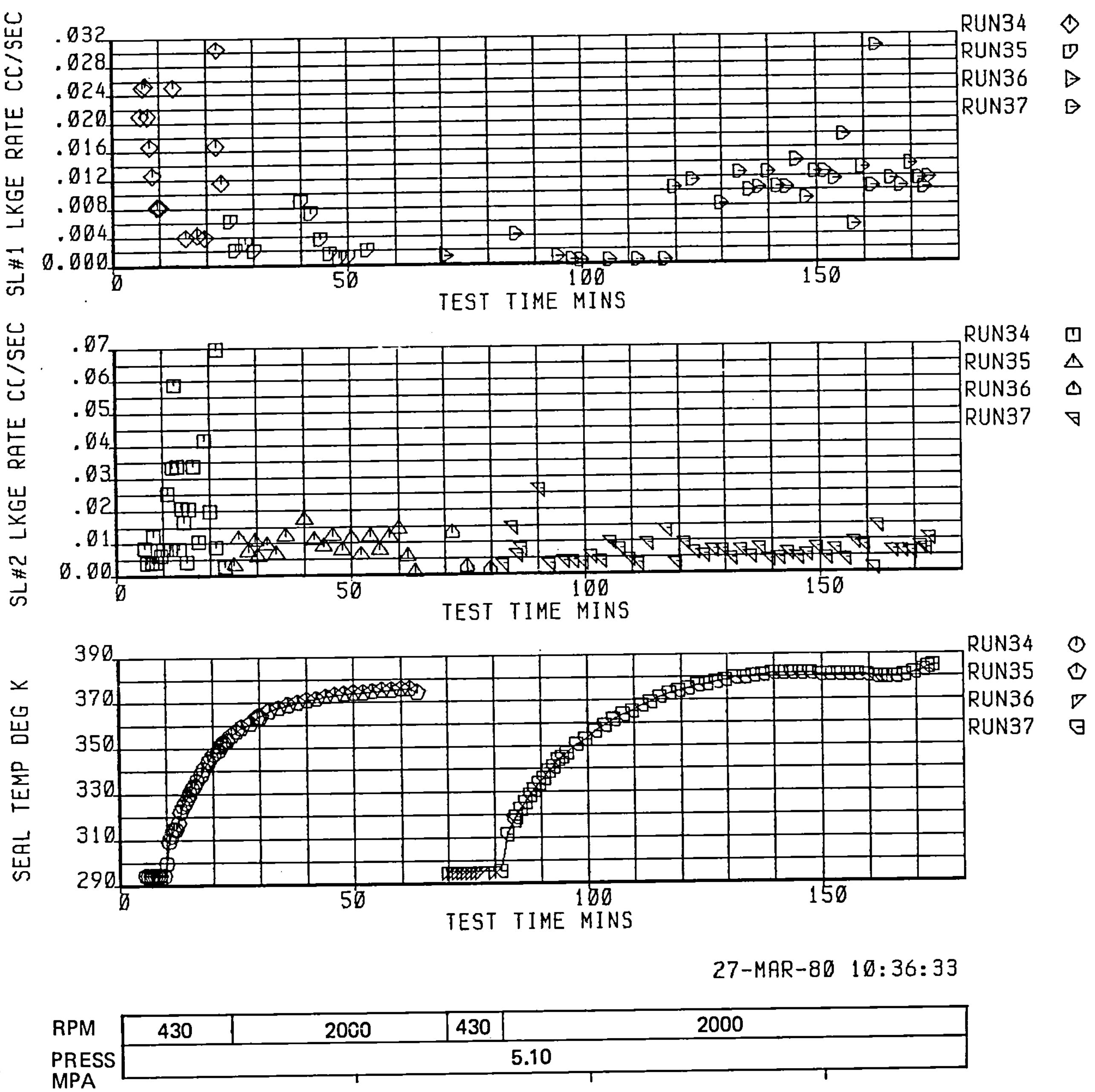


Figure 34. — Bell Seal Set #4 Test Data

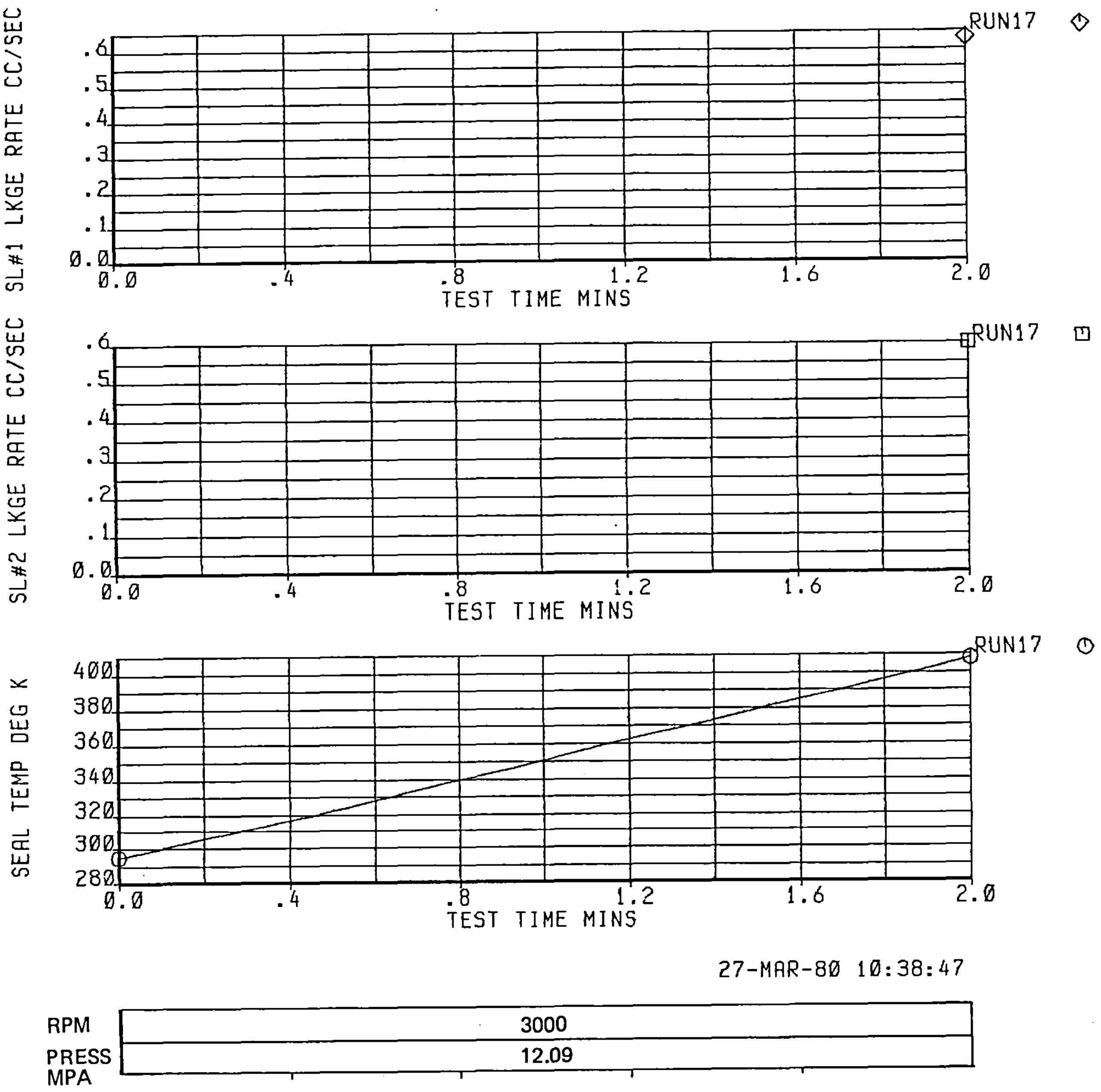
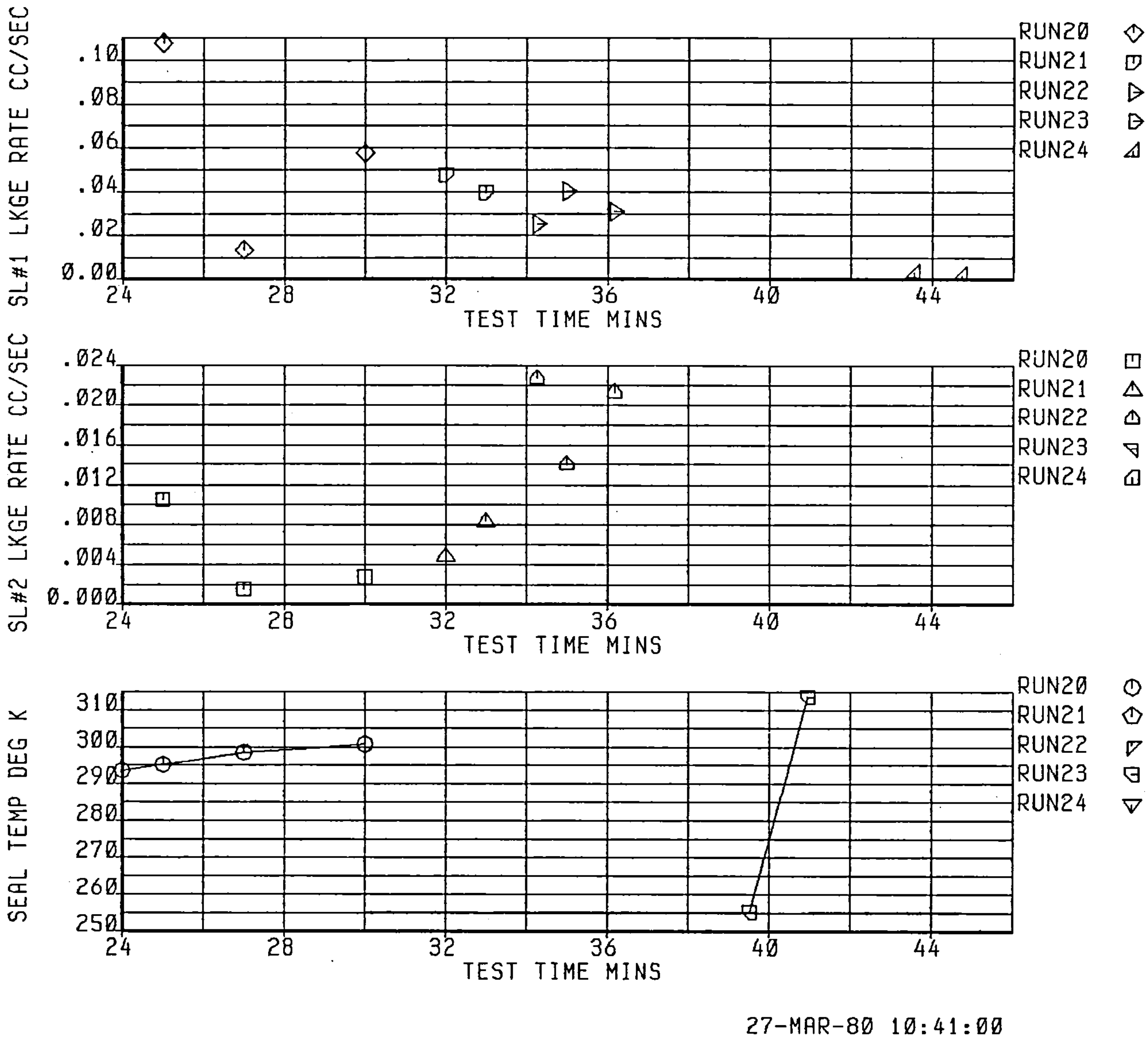


Figure 35. – Quad Seal Set #1 Test Data



RPM	430	1000	1750	430	1750
PRESS MPA	5.10			.101	

Figure 36. — Quad Seal Set #2 Test Data

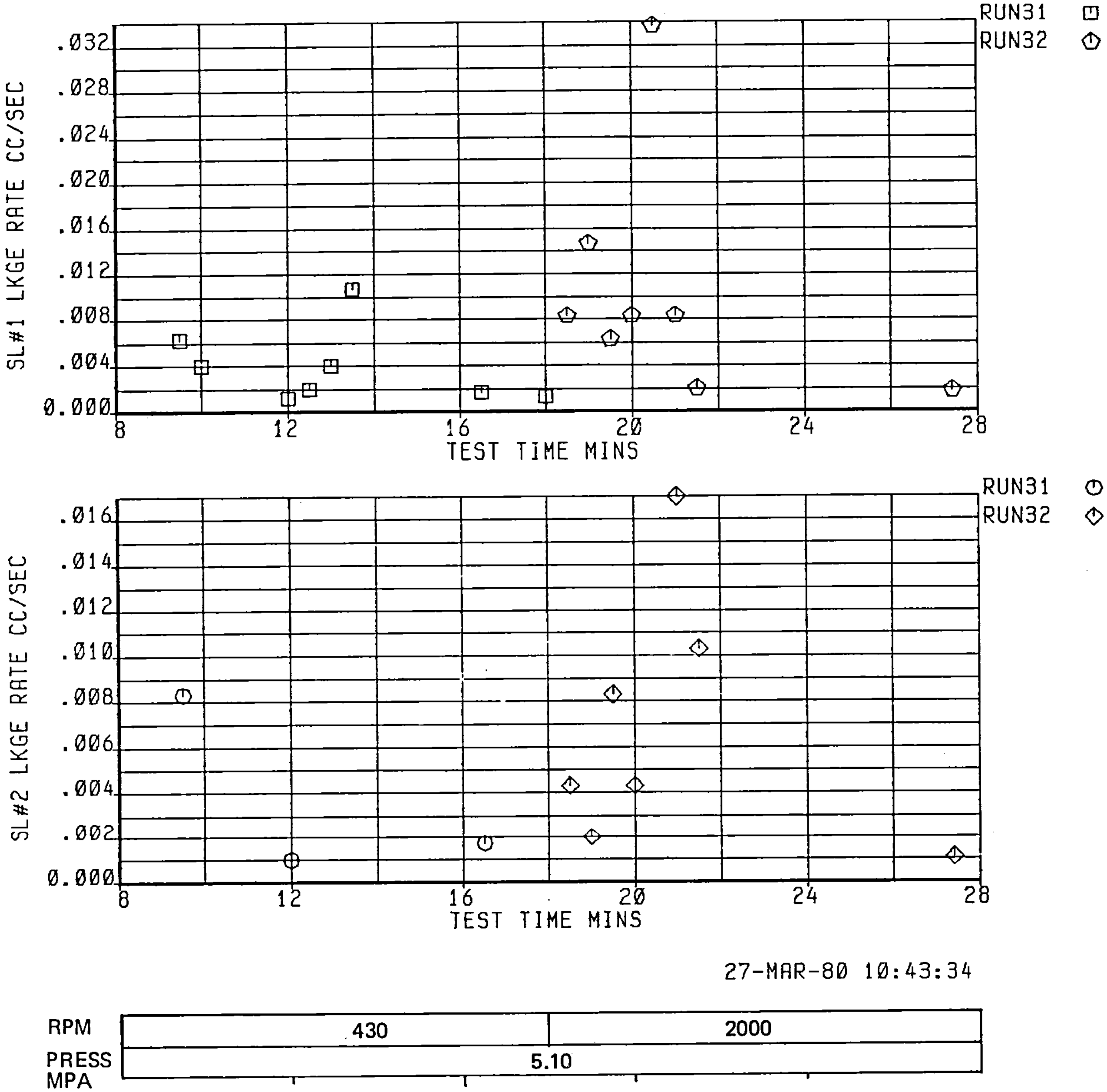


Figure 37. — Quad Seal Set #3 Test Data

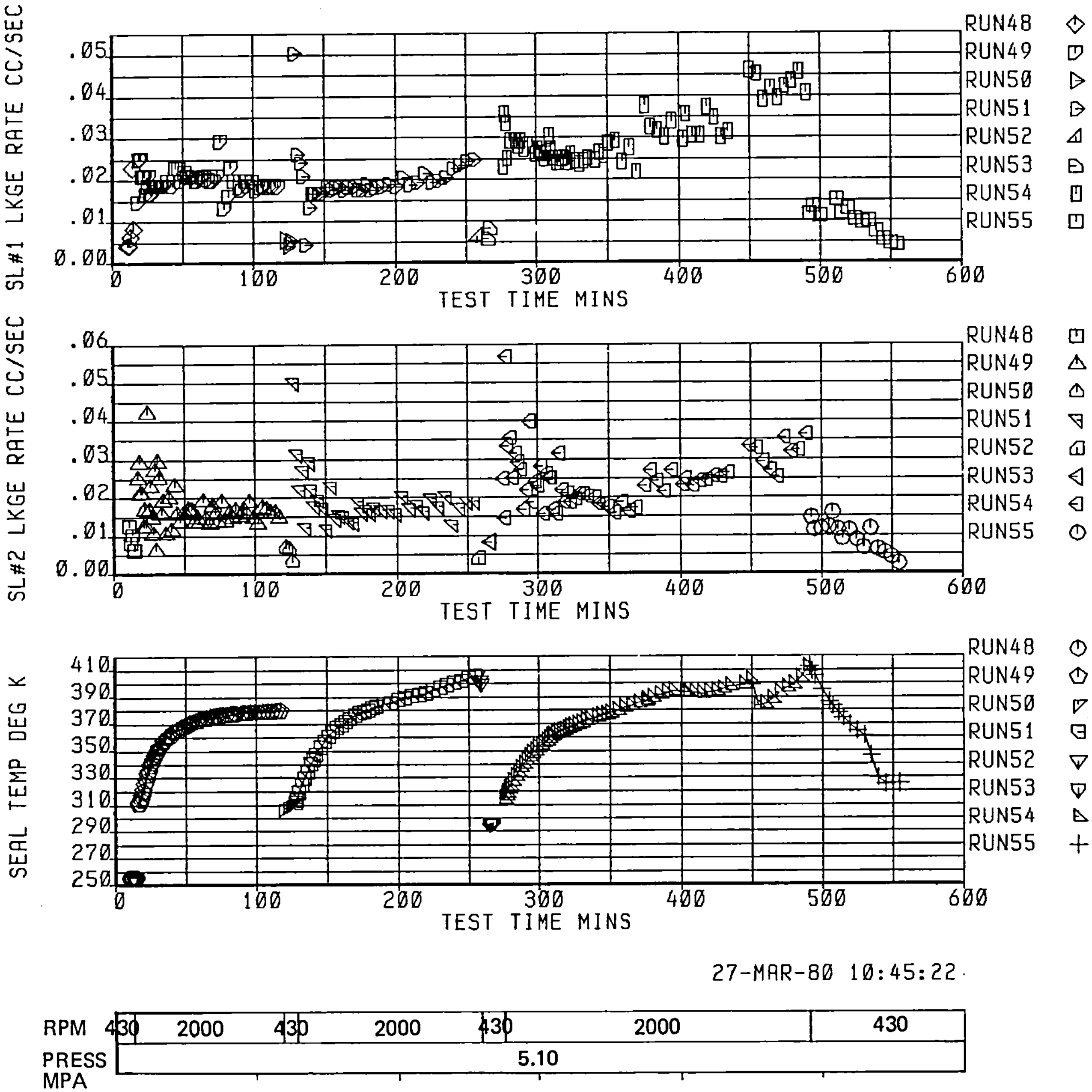


Figure 38. — Quad Seal Set #4 Test Data

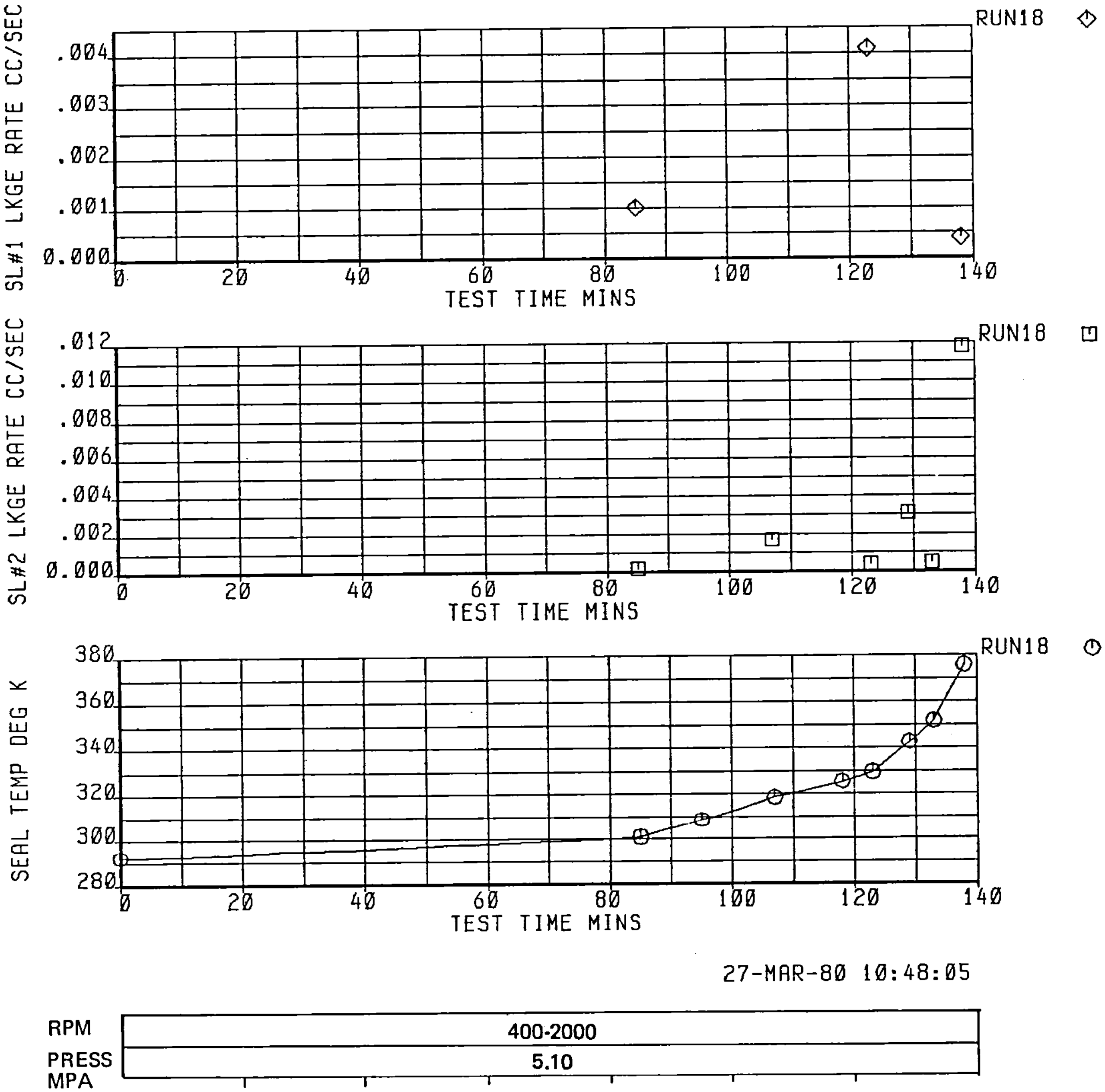


Figure 39. — Tetraseal Set #1 Test Data

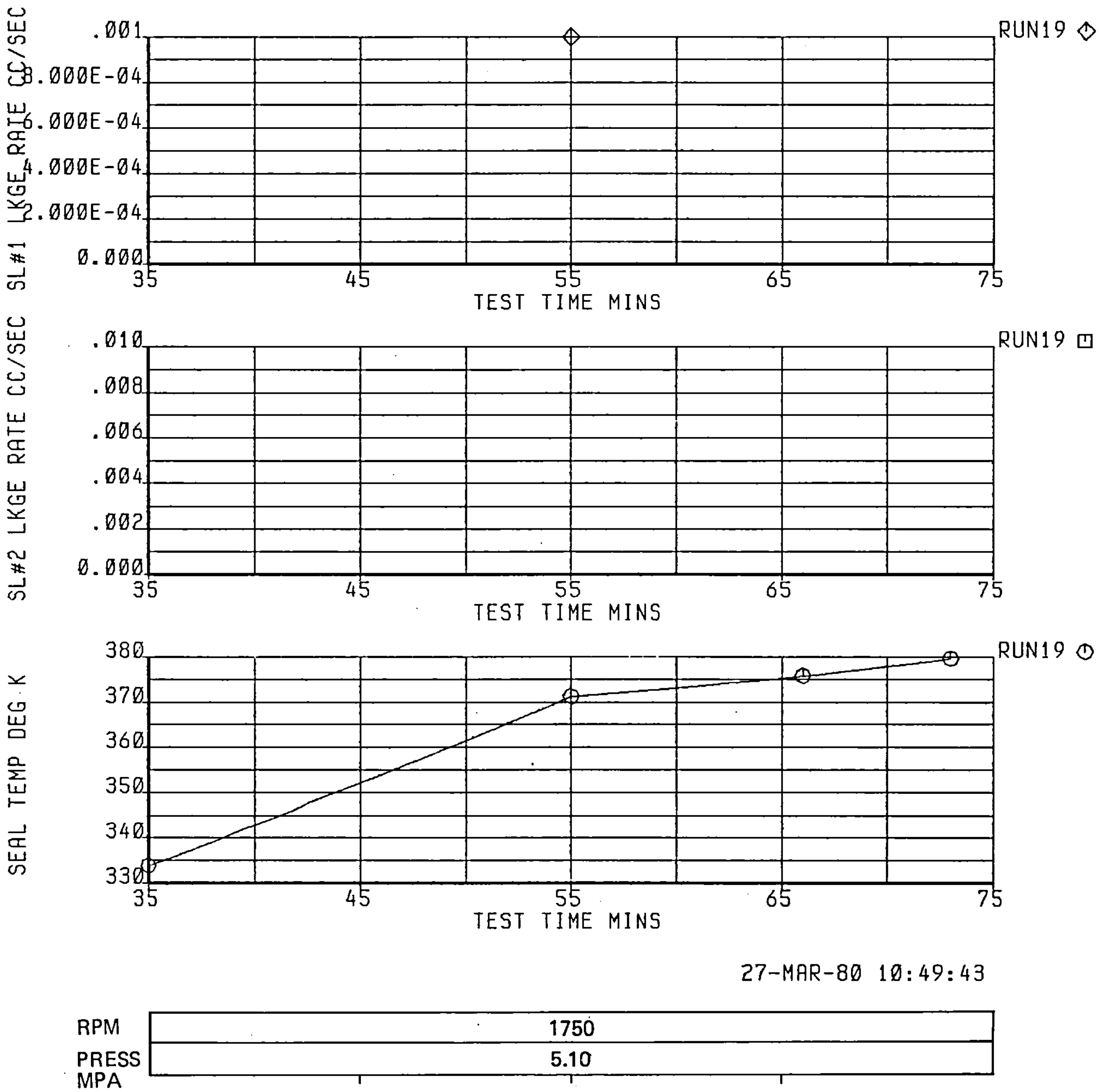


Figure 40. — Tetraseal Set #2 Test Data

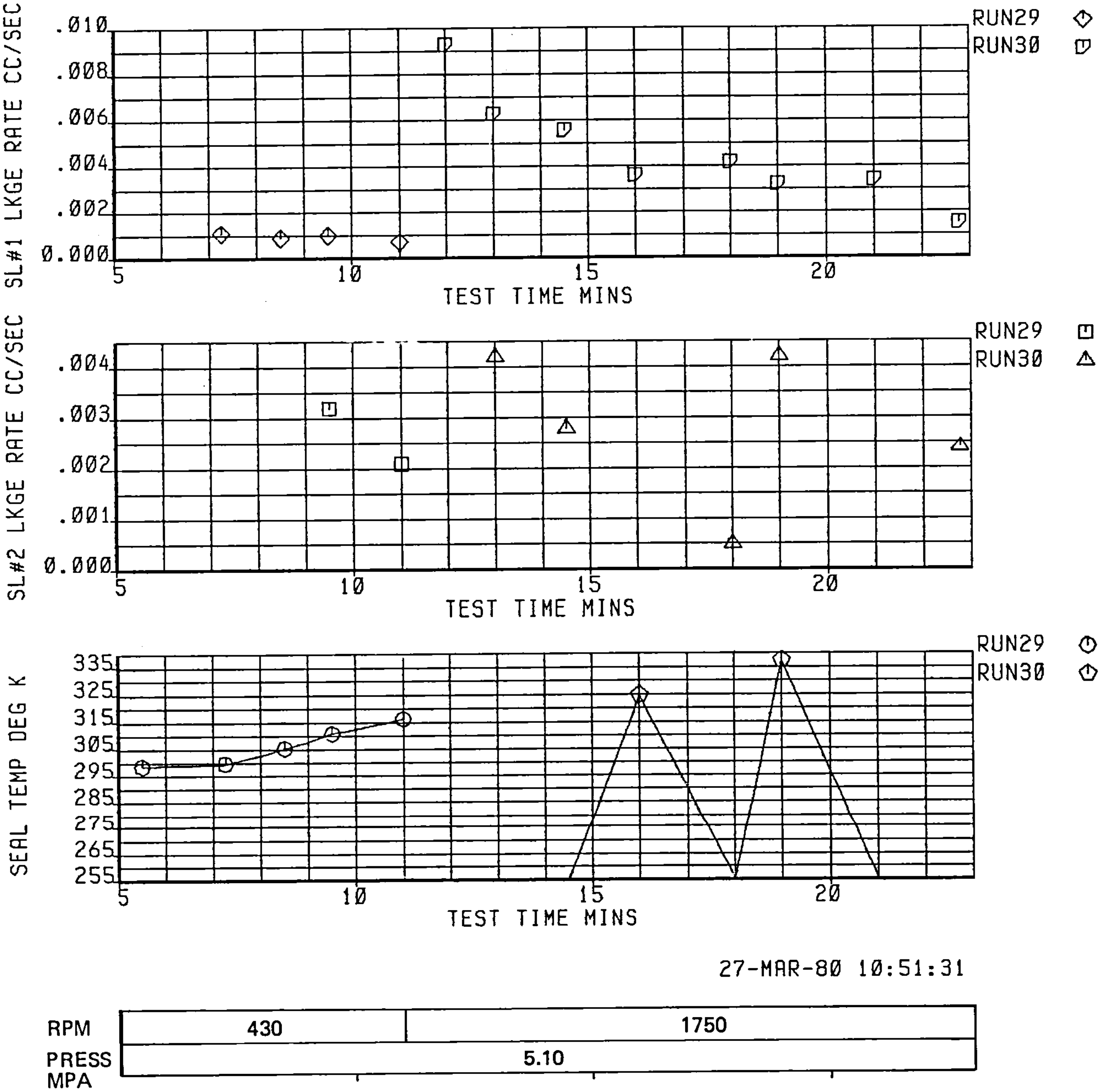
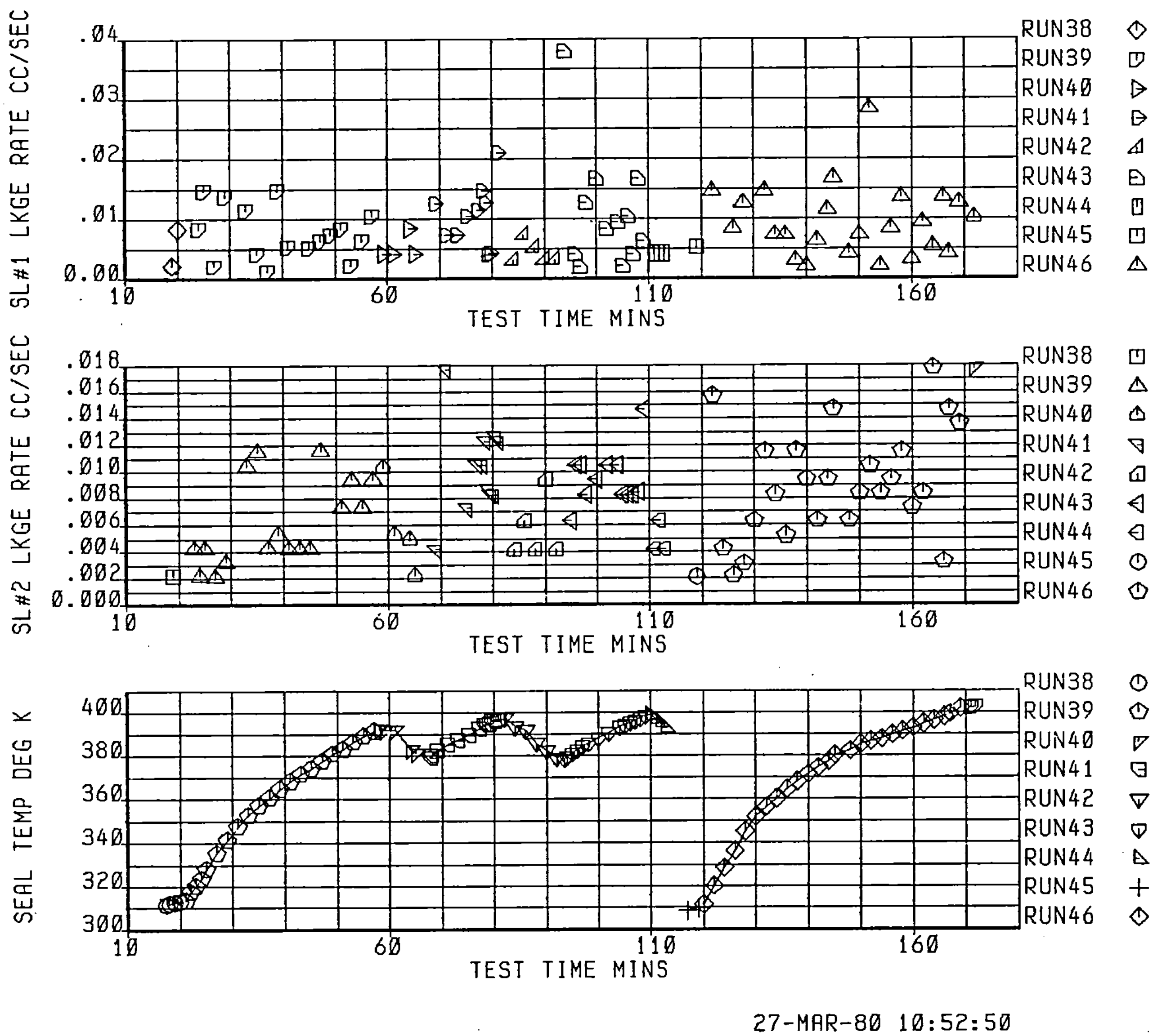


Figure 41. – Tetraseal Set #3 Test Data



RPM	430	2000	430	2000	430	2000	430	2000	430
PRESS MPA	5.10								

Figure 42. – Tetraseal Set #4 Test Data

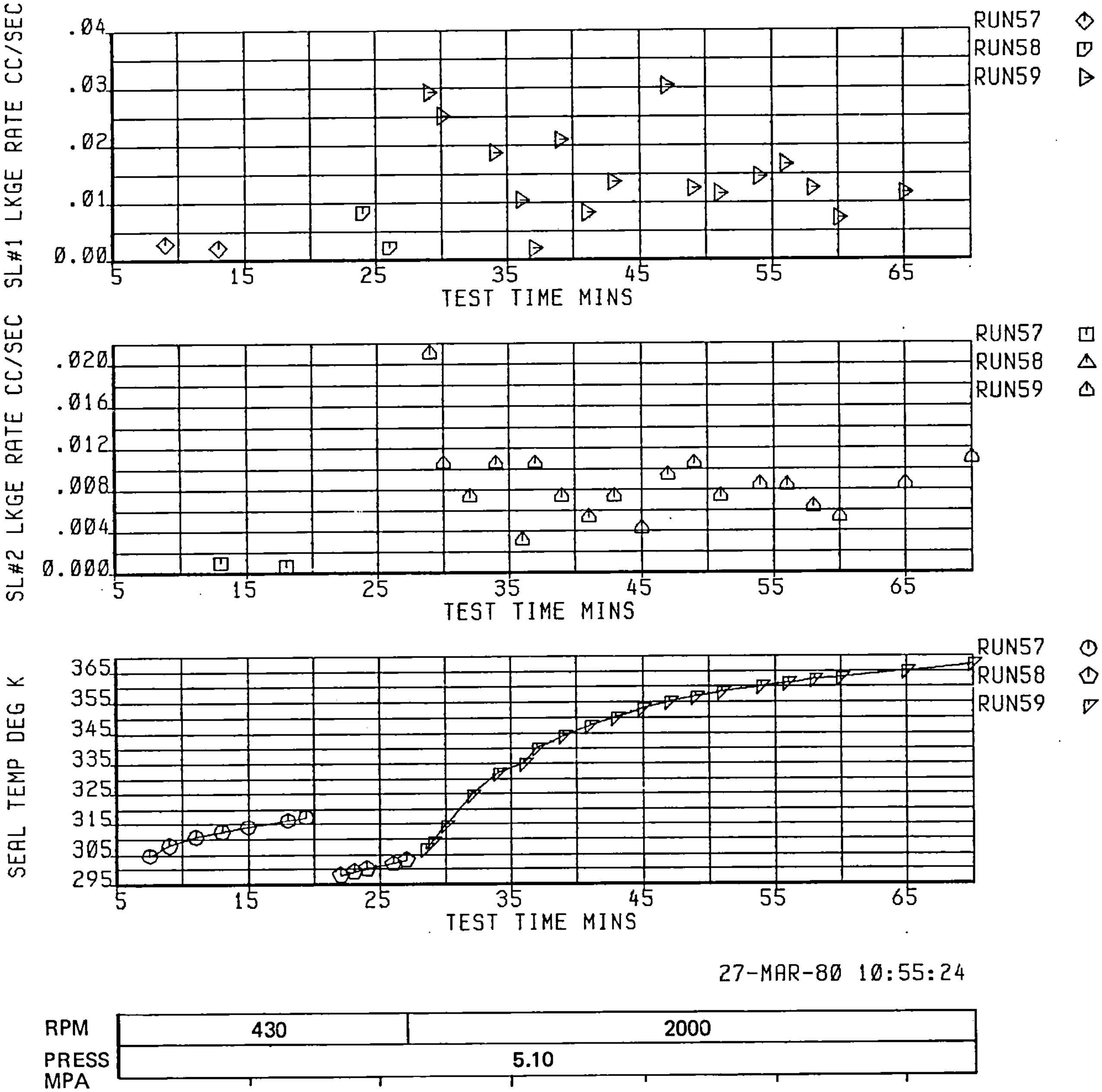


Figure 43. – Tetraseal Set #5 Test Data

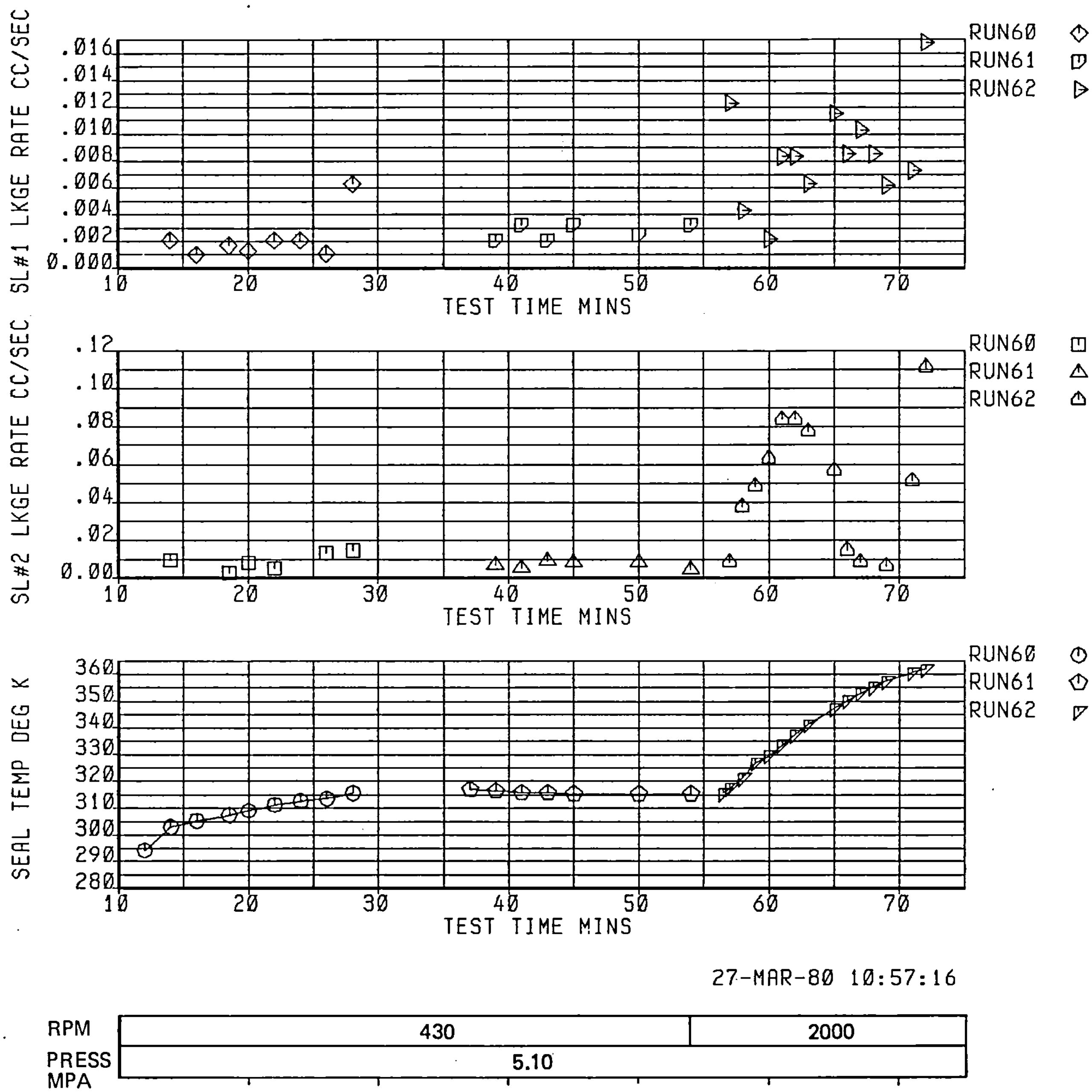


Figure 44. — Dynabak Seal Set #1 Test Data

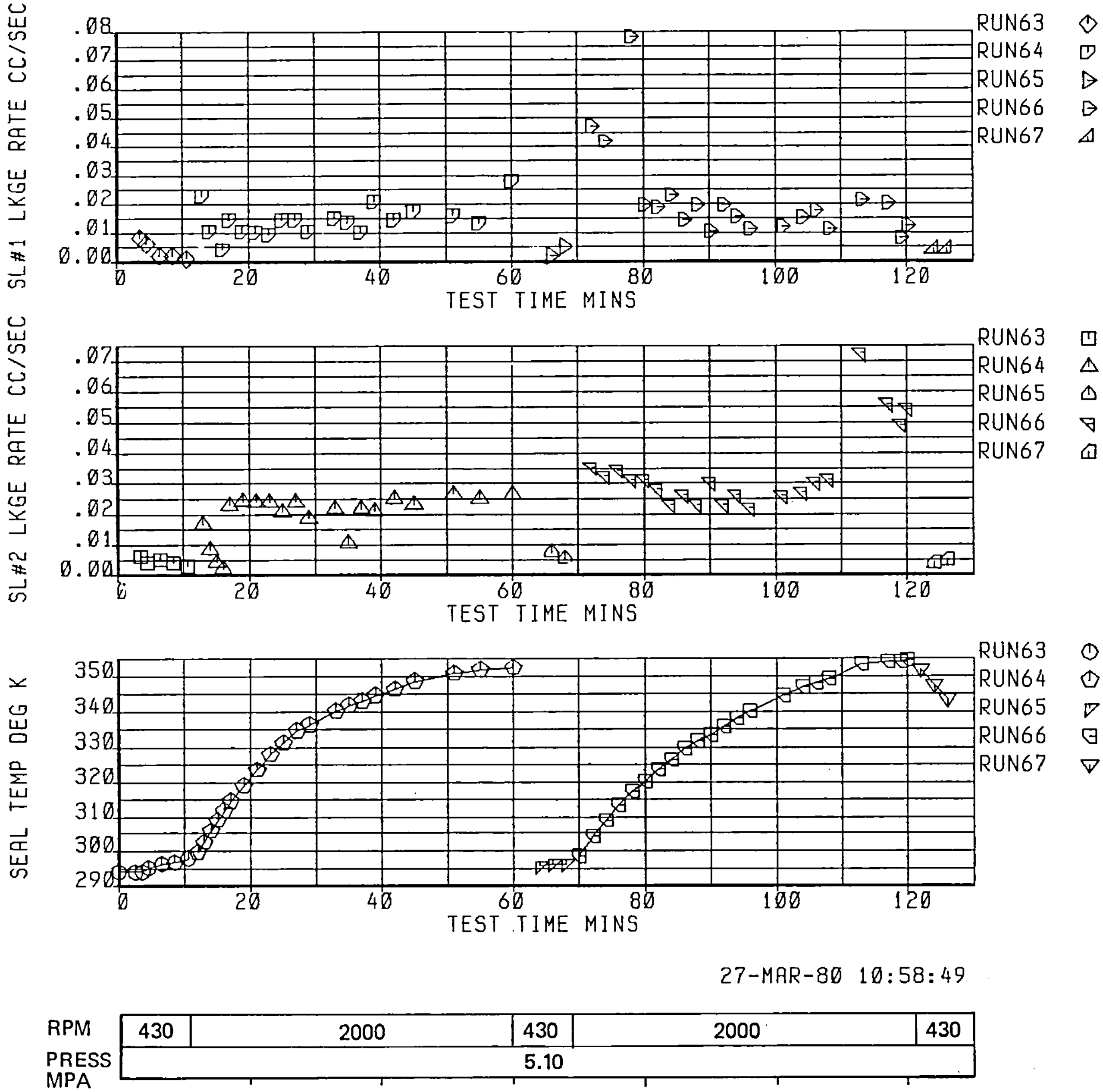


Figure 45. — Dynabak Seal Set #2 Test Data

REFERENCES

1. Automotive Stirling Engine Development Program, DOE/NASA/4396-79-5, NASA CR-159518, February 1980.

2. Assessment of State of Technology of Automotive Stirling Engine, DOE/NASA/0032-79-4; NASA CR-159631.

3. Testing of Polyimide Second Stage Rod Seals for Single State Applications in Advanced Aircraft Hydraulic Systems, NASA CR-135191, May 1977.

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15. Supplementary Notes

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16. Abstract

Evaluation of six single-stage reciprocating seal configurations to the requirements of the Stirling cycle engine was accomplished. The seals tested were: the Boeing Footseal, NASA Chevron polyimide seal, Bell seal, Quad seal, Tetraseal, and Dynabak seal. None of these seal configurations could meet the leakage goals of .002 cc/sec at helium gas pressure of 1.22×10^7 PA (1750 psig), rod speed of 7.19 m/sec peak (3000 rpm) and seal environmental temperature of 408 K (275°F) for 1500 hours. Most of the seals failed due to high temperatures. Catastrophic failures were also observed for a minimum number of test runs characterized by extremely high leakage rates and large temperature rises. The Bell seal did attain 63 hours of run time at significantly lowered test conditions.

17. Key Words (Suggested by Author(s))		18. Distribution Statement
Reciprocating Seals	Bell Seal	Unclassified-unlimited
Stirling Engine	Boeing Footseal	
Seal Friction	NASA Chevron Seal	
Gas Leakage	Quad Seal	
Dynabak	Tetraseal	

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Unclassified	Unclassified	75	

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