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RE–1000 Free-Piston Stirling Engine Hydraulic Output System Description

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SUMMARY

The NASA Lewis Research Center has been involved in free-piston Stirling engine research since 1976. Most of the work performed in-house was related to the characterization of the RE-1000 engine. The data collected from the RE-1000 tests was intended to provide a data base for the validation of Stirling cycle simulations. The RE-1000 was originally built with a dashpot load system which did not convert the output of the engine into useful power, but was merely used as a load for the engine to work against during testing.

As part of the interagency program between NASA Lewis and the Oak Ridge National Laboratory, (ORNL), the RE-1000 has been converted into a configuration that produces useable hydraulic power. A goal of the hydraulic output conversion effort was to retain the same thermodynamic cycle that existed with the dashpot loaded engine. It was required that the design must provide a hermetic seal between the hydraulic fluid and the working gas of the engine. The design was completed and the hardware has been fabricated. The RE-1000 was modified in 1985 to the hydraulic output configuration. The early part of the RE-1000 hydraulic output program consisted of modifying hardware and software to allow the engine to run at steady-state conditions.

This report presents a complete description of the engine, in sufficient detail so that the device can be simulated on a computer. Tables are presented showing the masses of the oscillating components and key dimensions needed for modeling purposes. Graphs are used to indicate the spring rate of the diaphragms used to separate the helium of the working and bounce spaces from the hydraulic fluid.

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INTRODUCTION

A free-piston Stirling engine designed for research purposes has been under test since 1979 at the NASA Lewis Research Center. The engine used in the research program was the RE-1000 engine, designed and fabricated by Sunpower Inc., of Athens, Ohio. The purpose of the test program was to accumulate a large data base with which to validate computer simulations of the Stirling cycle. These tests are discussed in references 1 to 3.

After completion of these tests, which were intended to focus on the thermodynamic processes occurring inside of the engine, the RE-1000 was converted into an engine producing hydraulic power. Several research goals were established for the hydraulic engine program. One goal was the investigation and development of a viable hydraulic output system. Another goal was the investigation of the dynamic interaction between the free-piston engine and the load device. It was known from previous research involving free-piston engines and compressor loads that there exists a strong interaction between the characteristics of the two systems. There must be more consideration given when matching an engine to a load than simply matching the power levels. A third goal was the investigation of the effects of compression-space cooling on the cylinder wall thermodynamic conditions and the overall engine performance.

The ability to change the load characteristics of the hydraulic output device was incorporated in the hydraulic load. By changing some key components of the pump on the hydraulic device, its spring content can be altered, thus altering its effect on the engine. For example, a compressor used as a load for an engine has much more spring content than a pure hydraulic pump. The term spring content refers to the spring component of the total force exerted by the load device. The total force exerted by the load device on the power diaphragm is the vector sum of the spring component and the damping component.

Requirements for the hydraulic output unit performance were outlined by NASA and the design of the unit was performed by Foster-Miller Associates, of Waltham, Massachusetts, with assistance from Sunpower Inc., of Athens, Ohio. The design and fabrication effort involved not only the hydraulic output unit but also the computerized control system along with the instrumentation used for the control system. The preliminary and detail design efforts were documented and published in reference 4.

DESCRIPTION

A cutaway view of the hydraulic engine is shown in figure 1. The thermodynamic section of the engine is generally the same as the dashpot loaded engine described in references 1 to 3. It was determined that the performance of the engine could be improved with a regenerator of higher porosity than was used with the dashpot engine, and a displacer tuned to slightly different dynamics. The displacer is shown in figure 2 along with some of the critical dimensions. With the addition of the hydraulic output unit to the engine, the dead volume of the compression space was increased. However, most of the dead volumes in the engine are the same as those values given in earlier reports. Dead volumes for this engine are given in table I. The compression space is now divided into two sections. Figure 3 shows a cross section of the compression space portion of the engine. The section of the compression space adjacent to the displacer is connected to the compression space adjacent to the power diaphragm by 18 passages of 0.414 cm (0.163 in.) diameter. Each passage is 4.57 cm (1.80 in.) long. In addition there are two holes of 0.516 cm (0.203 in.) diameter in parallel to the 18 holes.

The power diaphragm and the bounce diaphragm are both 0.51 mm (0.020 in.) thick and made of a 300 series stainless steel. The stiffness of both diaphragms were measured and are given in figure 4 in the form of force as a function of displacement. Supports were incorporated on either side of the diaphragms to protect the hardware in the event of complete pressure loss on one side of a diaphragm. Both diaphragms were designed to be annular and therefore each one has a hub mounted at the center. The diaphragms were mounted to the housing of the engine between O-rings. Similarly, the center hubs were mounted to the diaphragms with O-rings. The dimensions of the diaphragms are given in table II. The supports for the power diaphragm can be seen in figure 3. Photographs of the power diaphragm and the bounce diaphragm hardware are shown in figures 5 and 6.

Hydraulic fluid is contained in the volume above the power diaphragm. Power generated in the working space of the engine is transferred to the hydraulic fluid through the power diaphragm. As the power diaphragm moves, the hydraulic fluid forces the power transmission piston to move. The fluid above the power transmission piston forces the bounce diaphragm to move in a similar fashion. The total oscillating mass of the engine is the summation of the individual oscillating masses.

Attached to the power transmission piston is a rod that transmits the power from the power transmission piston to the hydraulic pump. The pump was designed with a center null band in which there is no load on the engine and no pumping takes place. The pumping work is performed only at the ends of the stroke. The reason for incorporating this feature in the design was to enhance the stability of the engine/load system. The hydraulic pump can be assembled in either one of two configurations. The pump can be assembled in either the four pulse-per-cycle or the two pulse-per-cycle configuration. Figure 7 shows the pump housing center null band cross section in both configurations. The absence or presence of a center null band accumulator determines how many pulses per cycle the pump will provide. In figure 7(a), a solid flat metal plate is shown covering the center null band ports. This configuration will provide four output pulses per cycle as shown schematically in figure 8. In figure 7(b), a center null band accumulater is shown connected to the center null band ports. This configuration will provide two output pulses per cycle as shown schematically in figure 9. Testing is currently being done with the pump in the four pulse-per-cycle configuration. A theoretical graph of the load and power output verses stroke is shown in figure 10. The theoretical load on the engine as a function of time is shown in figure 11.

Other significant features of the pump include small diaphragm accumulators on the inlet and outlet sides of the pump block as shown in figure 12. The purpose of the diaphragm accumulators is to smooth the oil flow to and from the pumping chambers. The inlet accumulator is charged with ~100 psi air while the outlet accumulator is charged with ~1000 psi nitrogen.

Between the power transfer piston and the hydraulic load, the pump rod passes through a cylindrical section for which a balance system was designed. The balance system consists of an annular piston that moves in the opposite direction as the power transmission piston. The annular piston is driven by a small piston which attaches to the center rod. The balance system can be seen in figure 1. If the balance system is not used, a single mass is substituted for the annular piston and its driving piston. This single mass is attached to the center rod to bring the total oscillating mass up to the design specification. Figure 13 shows the pump rod assembly. To the left of the pump rod is the annular piston used with the balance system. The small piston near the mid-point of the rod is used to drive the balance piston. If the engine is to be operated without the balance system, the large piston on the right side of the photograph is attached to the pump rod. The oscillating mass of the system is equivalent with either of the two modes. A schematic of the hydraulic output device is shown in figure 14.

DESCRIPTION OF THE FACILITY

The RE-1000 free-piston Stirling engine hydraulic output test facility was built to supply all of the support systems needed to carry out the research program. The facility had a high pressure, 15.5 MPa, (2250 psi) gas system to supply the working gas to the engine, a water system to cool the engine, a separate water system to cool the compression space and the power diaphragm area of the engine, a dc electric power supply system to heat the engine, and a hydraulic system to supply and receive hydraulic fluid to and from the hydraulic pump as well as to maintain the correct relative positions of power and bounce diaphragms with the hydraulic pump rod. Along with these systems, a high speed data system was built to help gather data. Each of these systems will be described in detail. A schematic of the hydraulic system is shown in figure 15.

The gas system was designed with the ability to charge the engine with helium, hydrogen, nitrogen, or argon. The supply pressure to the engine was set with a remote control pressure regulator. The flow of gas into or out of the engine was controlled by motorized needle valves in the supply and vent lines respectively. To start the engine, a pair of solenoid valves alternatively connected the high pressure supply line or the low pressure vent line directly to the working space. The gas pulses would start the displacer and power diaphragm in motion. If the motion and the heat exchanger heat transfer were great enough, the engine cycle would continue to operate.

The water cooling system consisted of two separate systems. A closed loop water system was used for the engine cooler. This system was filled with distilled water. A heat exchanger with a feedback system controlled the temperature of this water loop. An open loop system was used to cool the compression space and the power diaphragm area of the engine. This system had no temperature control. Temperatures and flow rates were measured in both systems to allow the calculation of heat rejection rates of the engine.

The engine heater power supply system consisted of two Sorensen electric power supplies connected in parallel. Each power supply unit had the capability of delivering 1000 A of direct current to the engine. The two power supplies were regulated by an automatic controller which used a thermocouple on one of the engine heater tubes as feedback.

The hydraulic system consisted of two sub-systems: (1) a hydraulic pumping loop section and (2) a hydraulic control section. The hydraulic pumping loop began at the low pressure sump. The hydraulic fluid traveled through a valve, a filter, a check valve, a flowmeter, and an accumulator before it reached the inlet side of the hydraulic pump. Figure 15 shows that the check valve was connected in parallel with an air driven pump. The air driven pump was used as a primer pump to circulate oil through the pumping loop and was never used when the hydraulic output unit was running. The purpose of the accumulator was to smooth the upstream flow so that the flow rate could be measured more accurately. After the fluid exited the outlet side of the hydraulic pump, the fluid either entered the high pressure accumulator or passed through the variable resistance relief valve. The variable resistance relief valve acted as the load in the hydraulic pumping loop. The pressure drop across this valve can be varied to simulate a wide range of engine loads. A Fema control unit was used to control the variable resistance relief valve. The fluid that passed through the variable relief valve passed through a heat exchanger before it returned to the low pressure sump for recirculation. The hydraulic control system consisted of computer controlled solenoid valves which could adjust the oil inventory as well as the relative mean positions of the diaphragms and power transfer piston. The control system will be described in detail in the following section. The control room for these systems is shown in figure 16.

CONTROL SYSTEM

A schematic of the control system is shown in figure 15. The solenoid valves of the control system are operated by a computer system based on a Kaypro II personal computer. The computer prepares the hydraulic output unit for running and also monitors the operation of the engine to see if the center of each diaphragm stroke or the center of the power transmission piston stroke is too far from its desired center position. If the measured center position is outside of a predetermined tolerance, the computer will open the appropriate valve or valves to correct the position. The computer also maintains the oil inventory. As an example, if the mean position of the power diaphragm is too far from the working space and the mean position of the bounce diaphragm is too far from the bounce space, this means that there is too little hydraulic oil between the two diaphragms. When the control program recognizes this, it opens a valve that bleeds hydraulic oil from the high pressure accumulator of the facility to the engine. When the inventory of oil reaches the correct value, the control system closes the valve. Although much of the control system could have been incorporated into the hardware in the form of passive systems, the desire to be able to alter the control system without altering the hardware required an external control system.

The control program basically has two modes of operation: (1) the cold-start mode and (2) the run mode. The cold-start mode is used to place the power and bounce space diaphragms and the power transmission piston in their proper relative positions prior to the starting of the engine. The oil inventory and the relative positions are checked and adjusted only once. The run mode keeps the diaphragms and the power transmission piston in their proper relative positions as the engine is running. The oil inventory and the relative positions are constantly being monitored and adjusted as necessary. The run mode maintains the mean center positions within the specified tolerances. The functions of each solenoid valve as well as the cold-start and run modes of operation are described below.

Valve Functions

Hydraulic Solenoid Valves

VBO - This valve, under normal operating conditions, remains closed. Its purpose is to bleed oil from the hydraulic output unit if the oil inventory is too large. The oil is bled from the oil cavity above the power transmission piston to the low pressure sump.

VCC - This valve is in the closed position in the cold-start mode when the power transmission piston is being raised to its proper position. If the power transmission piston is raised too far above its proper position, VCC will open in combination with VEQ, VUP, and VDP to allow the piston to lower. When both diaphragms and the power transmission piston are in their proper relative positions, all of the solenoid valves will close except for VCC. This is an indication that the engine is ready to be started. The reason that this valve must be open is that the oil in the cavity above the pump rod must have somewhere to go as the pump rod oscillates. When VCC is open, the oil in the cavity above the pump rod is able to flow to and from the cavity just below the power transmission piston.

VDP - The purpose of this valve when in the run mode is to lower the mean position of the power transmission piston. This valve allows oil to migrate from the cavity below the power transmission piston to the cavity above the power transmission piston.

VEQ - This valve is used only in the cold-start mode of the control program. VEQ allows the oil to quickly migrate between the cavities above and below the power transmission piston. This valve was added at NASA Lewis. VEQ's function was previously handled by VDP and VUP. Since needle valves are required in series with these two valves (VDP and VUP), too much time was required for the oil migration and the control system was unable to converge on the start-up conditions. The addition of VEQ reduced the time required for the cold-start mode of the control program to complete its task.

VIO - This valve is used to inject oil into the hydraulic output unit when the oil inventory is low. VIO allows oil from the high pressure accumulator to be injected into the oil cavity above the power transmission piston.

VOC - This valve is used only in the cold-start mode of the control program. VOC opens in combination with VEQ, VUP, and VDP to raise the power transmission piston to its proper position. When VOC opens, the pressure of the oil in the cavity above the pump rod decreases allowing the pump rod and power transmission piston to rise. VOC allows the oil in the cavity above the pump rod to drain to the low pressure sump.

VUP - This valve is used in the run mode and also in the cold-start mode. The purpose of this valve when in the run mode is to raise the mean position of the power transmission piston. This valve allows oil to migrate from the cavity above the power transmission piston to the cavity below the power transmission piston.

Hydraulic Hand Valves

Hand Valve - This valve must be closed during the cold-start mode of the control program and must be opened immediately after the cold-start procedure has ended. This valve was added at NASA Lewis and has a similar purpose to VCC. It was found that the flow losses in the line controlled by VCC were too great to allow the engine to start. As a temporary solution to the problem, stainless steel tubing of a greater diameter and of a shorter length was connected in parallel with VCC. A hand valve was placed in this new line so that the cold-start procedure could still function properly. Current plans are to replace this hand valve with a solenoid valve so that it can be controlled by the Kaypro.

Working Space Gas Solenoid Valves

VBG - This valve is used only when the engine is in the run mode. If too much gas is in the working space and not enough gas is in the bounce space, VBG opens to shuttle gas from the working space to the bounce space. The condition that causes the opening of VBG to be necessary exists when the mean position of the power diaphragm is too far away from the working space and the mean position of the bounce space diaphragm is too close to the bounce space.

VBS - This valve is used to either pressurize or vent the bounce space. It is opened in combination with VFG or VVG.

VEG - This valve is used only when the engine is in the run mode. If too much gas is in the bounce space and not enough gas is in the working space, VEG opens to shuttle gas from the bounce space to the working space. The condition that causes the opening of VEG to be necessary exists when the mean position of the power diaphragm is too close to the working space and the mean position of the bounce space diaphragm is too far away from the bounce space.

VES - This valve is used to either pressurize or vent the working space. It is opened in combination with VFG or VVG.

VFG - This valve is used to pressurize the engine with gas.

VVG - This valve is used to vent gas from the engine.

Cold-Start Mode

When executed the program checks the oil inventory in the hydraulic output unit by checking the positions of the power and bounce space diaphragms. For example, if the static position of the power diaphragm is toward the working space end of its stroke while the static position of the bounce space diaphragm is toward the bounce space end of its stroke, then there is too much hydraulic oil between the two diaphragms. The control program will then open VBO, VEQ, VDP, and VUP until the oil inventory reaches the correct value. Once the oil inventory is correct, the power transmission piston is ready to be moved to its proper position relative to the power diaphragm. The piston is raised to its proper position by opening VOC, VEQ, VDP, and VUP. VOC connects the small volume of oil above the pump rod to the low pressure sump which causes the net force acting on the pump rod to be in the upward direction. If the power transmission piston is raised past its proper relative position, VOC will close and VCC will open making the total net force in the downward direction due to gravity. Once the power transmission piston is in its proper position, all of the solenoid valves will close then VCC will open. This is a signal that the diaphragms and the power transmission piston are in their proper positions and that the engine is ready to be started. The hand valve connected in parallel with VCC must then be opened before the engine can be started.

Run Mode

The RE-1000 free-piston Stirling engine hydraulic output unit is started using the same starting procedure that was used for the RE-1000 dashpot load configuration. That is by giving the engine pressure pulses from the starter system. Once the engine begins running, the control program enters its run mode of operation. In this mode under steady-state conditions, only VCC and the hand valve connected in parallel with VCC are open. All of the other solenoid valves are closed.

The oil inventory in the engine is controlled by VIO and VBO, as described previously. The mean position of the power transmission piston is controlled by either VDP or VUP depending on direction of the needed correction. Similarly, the mean diaphragm positions are controlled by valves VBG and VEG.

INSTRUMENTATION

Due to the design goal of having a general purpose test bed for a free-piston Stirling engine with hydraulic output, the unit was designed to incorporate a great amount of instrumentation. The instrumentation used to measure the performance of the Stirling cycle basically remained the same as was used during the tests outlined in references 1 to 3. Position transducers were used to measure the displacer position, the power diaphragm position, the pump rod position, and the bounce diaphragm position. The linear voltage differential transformers (LVDT), are used for both the control system and for research data system. The LVDT used to measure the pump rod position is mounted at the top of the pump rod, above the pump housing. This volume is filled with hydraulic oil when the engine is running. Oil in this area must be permitted to flow through the LVDT housing to allow the pump rod to oscillate. In the configuration used to date, this equalization has been accomplished through valve VCC and the hand valve shown in figure 15.

The LVDT used to measure the bounce diaphragm position was mounted in the gas filled bounce space. The LVDTs used to measure the power diaphragm position and the displacer position were mounted in the oil filled space above the power diaphragm as shown in figure 3. Ideally, the LVDT windings used to measure the displacer position should be located in the compression space. This would eliminate the thin wall (0.0065 in.) tube required to seperate the oil from the working space gas. The LVDT core mounted on the displacer moves axially inside the thin wall tube. This tube, which is mounted on the power diaphragm hub, moves axially inside the LVDT windings. The displacer LVDT core moves at a phase angle with respect to this tube equivalent to the phase angle between the power diaphragm and the displacer. The relative motions between the LVDT core, the thin wall tube, and the LVDT windings have caused an appreciable amount of wear to the tube. It is inevitable that this tube would fail if testing continued. Therefore, it was necessary to redesign the displacer position measurement system. It was impractical to mount the LVDT windings in the compression space. Mounting the LVDT windings in the compression space would require a large compression space dead volume which would lower the engine performance. The final position measurement design chosen to replace the old system was based on a proximity probe measurement system. All four LVDTs currently being used will be replaced with the new system in order to preserve the integrity of the measurement system. In the new system, sensors and targets are used to measure the absolute positions of the moving parts. Tapered targets will be mounted to the displacer, power diaphragm hub, bounce diaphragm hub, and the pump rod. Since the sensors and targets are quite small in size, the entire displacer measurement system can now be contained within the compression space without significantly increasing the compression space dead volume.

At some future time, fast response pressure transducers will be added in the pump housing to measure the pressure exerted on the pump rod throughout the cycle. A listing of all of the instrumentation currently being used in the tests is given in table III. A listing of the calculations performed in the data reduction process is given in table IV.

CONCLUDING REMARKS

This report is intended to provide the parameters needed to simulate the RE-1000 based hydraulic output engine on a computer. Included in the description of the hardware are experimental measurements of the stiffness of the two main diaphragms and measurements of the volumes of the gas filled spaces of the engine. The design of the hydraulic output unit was intended to make the engine a rugged test bed for research.

Data collected during the testing of the hydraulic engine will be published and used to validate the NASA Lewis computer code. Testing with variable cooling in the compression space and variations in the configuration of the load will aid in this purpose.

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TABLE I. - HYDRAULIC ENGINE DEAD VOLUMES

Bounce space, cc ($in.^3$)	2714 (165.6)
Total working space, cc ($in.^3$)	479.5 (29.26)
Compression space, cc ($in.^3$)	264.2 (16.12)
Regenerator (matrix), cc ($in.^3$)	65.8 (4.02)
Regenerator (total), cc ($in.^3$)	80.5 (4.91)
Cooler, cc ($in.^3$)	31.9 (1.95)
Heater, cc ($in.^3$)	36.2 (2.21)
Expansion space, cc ($in.^3$)	66.7 (4.07)

TABLE II. - HYDRAULIC ENGINE KEY DIMENSIONS

Hydraulic oil kinematic viscosity, cm^2/sec ($in.^2/sec$)	0.026 (4.1×10^{-3})
Hydraulic oil density, g/cm^3 ($lb/in.^3$)	0.860 (0.0311)
Displacer mass, g (lb)	414.0 (0.89)
Displacer gas spring mean volume, cm^3 ($in.^3$)	23.2 (1.42)
Power transmission piston diameter, cm (in.)	5.07 (1.998)
Pump rod diameter, cm (in.)	0.953 (0.375)
Hydraulic pump piston diameter, cm (in.)	1.62 (0.639)
Power transmission piston/rod mass, g (lb)	959 (2.114)
Substitute for balance piston, g (lb)	1719 (3.79)
Power diaphragm outer diameter, cm (in.)	20.31 (7.996)
Power diaphragm inner diameter, cm (in.)	6.058 (2.385)
Bounce diaphragm outer diameter, cm (in.)	20.31 (7.996)
Bounce diaphragm inner diameter, cm (in.)	1.588 (0.625)
Power diaphragm hub mass, g (lb)	338 (0.745)
Bounce diaphragm hub mass, g (lb)	181 (0.399)
Distance from power diaphragm to power transmission piston at mean position, cm (in.)	26.65 (10.49)
Power diaphragm to power transfer piston passage diameter, cm (in.)	7.62 (3.0)
Distance from bounce diaphragm to power transmission piston at mean position, cm (in.)	13.55 (5.33)
Bounce diaphragm to power transfer piston passage diameter, cm (in.)	7.62 (3.0)
VCC line length, cm (in.)	87.6 (34.5)
VCC line diameter, cm (in.)	1.09 (0.43)

TABLE III. - RE-1000 INSTRUMENTATION

	Mnemonic	Parameter	Range	Type	SS	V
1	MEANCP	Mean comp space pressure, KPa	0-13 800	Strain gauge	X	-
2	MEANCP	Mean bounce pressure, KPa	0-13 800	Strain gauge		-
3	PRESUP	Gas supply pressure, KPa	0-13 800	Strain gauge		-
4	T01HTR	Heater tube metal temperature, °C	400-825	Thermocouple		-
5	T02HTR					-
6	T03HTR					-
7	T04HTR					-
8	T05HTR					-
9	T06HTR					-
10	T07HTR					-
11	T08HTR					-
12	T09HTR					-
13	T10HTR					-
14	T11HTR					-
15	T12HTR					-
16	T03HED	Head metal temperature, °C				-
17	T13REG	Regenerator vertical profile, °C				-
18	T14REG	Regenerator vertical profile, °C				-
19	T15REG	Regenerator circular profile, °C	250-825			-
20	T16REG					-
21	T17REG					-
22	T18REG					-
23	T19REG	Regenerator vertical profile, °C	20-250			-
24	TGBOUN	Regenerator vertical profile, °C	20-80			-
25	TGCOMP	Compression space gas temperature, °C	20-250			-
26	TGREGC	Regenerator-cooler gas temperature, °C	20-250			-
27	TGREGH	Regenerator-heater gas temperature, °C	400-825			-
28	TGEXP	Expansion space gas temperature, °C	400-825			-
29	TWINDI	Diaphragm water inlet temperature, °C	10-70			-
30	TDLDT1	Diaphragm H_2O delta temperature (top), °C	0-20			-
31	TDLDT2	Diaphragm H_2O delta temperature (bottom), °C	0-20			-
32	TWODI	Diaphragm water outlet temperature, °C	10-70			-
33	TWINCL	Cooler water inlet temperature, °C	10-70			-
34	TDLCL	Cooler water delta temperature, °C	0-20			-
35	TWOCL	Cooler outlet temperature, °C	10-70			-
36	TDLDB1	Diaphragm H_2O delta temperature (top), °C	0-10			-
37	TDLDB2	Diaphragm H_2O delta temperature (bottom), °C	0-10			-
38	THFIN	Hydraulic inlet temperature, °C	20-30			-
39	THFOUT	Hydraulic outlet temperature, °C	20-50			-
40	THFCLR	Hydraulic fluid cooler, °C	20-50			-
41	AMPS1	Heater amps, supply 1, A	0-1000	Ammeter		-
42	AMPS2	Heater amps, supply 2, A	0-1000	Ammeter		-
43	VOLTG	Heater voltage, V	0-20	Voltmeter		-
44	FLODI	Diaphragm water flow, l/min	0-10	Turbine meter		-
45	FLOCLR	Cooler water flow, l/min	0-10	Turbine meter		-
46	FLOHYD	Hydraulic oil flow, l/min	0-10	Turbine meter		-
47	VX1HOR	Horizontal vibration, cm/sec	0-3.8	Accelerometer		-
48	VY1VER	Vertical vibration, cm/sec	0-3.8	Accelerometer		-
49	PDIAST	Power diaphragm stroke, mm	0-7	Stroke meter		-
50	DISPST	Displacer stroke, cm	0-4	Stroke meter		-
51	INDPWR	Indicated power, W	0-3000	Analog circuit		-
52	PWROUT	Brake power, kW	0-3.0	Analog circuit		-
53	FPIST	Load force on piston, N	0-1600	F transducer	-	X
54	XDIAP	Power diaphragm position, N	±3.5	LVDT	-	
55	XDOTP	Piston velocity, m/sec	±8.0	LVT	-	
56	XDISP	Displacer position, mm	±2.0	LVDT	-	
57	XDIAB	Bounce diaphragm position, mm	±3.5	LVDT	-	
58	XPUMP	Pump position, cm	±30.0	LVDT	-	
59	PDYNC	Compression space pressure, kPa	±2000	Crystal	-	
60	PDLCLR	Cooler delta pressure, kPa	68.9	Differential	-	
61	PDLREG	Regenerator delta pressure, kPa	138	Pressure transducer	-	
62	PDLDIS	Displacer delta pressure, kPa	138		-	
63	PHFIN	Hydraulic pressure in, MPa	0-1		-	
64	PHFOUT	Hydraulic pressure out, MPa	0-10		-	
65	PDYNDB	Displacer gas spring pressure, MPa	10.0	Strain gauge	-	
66	PAPRES	Phase angle of pressure, deg	0-360	Phase meter	X	-
67	PADISP	Phase angle of displacer, deg	0-360	Phase meter	X	-
68	FREQ	Engine frequency, Hz	0-50	Frequency to dc	X	-
69	PAMPC	Compression pressure amplitude, kPa	0-2000	Crystal	-	X

TABLE IV. - RE-1000 CALCULATIONS

MNEMONIC	Description of the calculation
PWRIN	Electric power input to the heater head
QCOOLR	Heat rejected by the engine cooler
QDPRMT	Heat rejected from the top of the engine diaphragm
QDPRMB	Heat rejected from the bottom of the engine diaphragm
NIT	Engine indicated thermal efficiency based on the indicated power and the heater power input
TAVHTR	Average heater tube outside temperature
NST	System thermal efficiency based on the gross load power and heater power input
AMPS	Total amperage to the heater head
QDISPG	Heat conduction through the gas in the displacer
QDISP	Heat conduction through the displacer body
QREG1	Outer regenerator wall conduction
QREG2	Outer regenerator wall conduction
QREG3	Inner regenerator wall conduction
QHEAD	Power input to engine head
GRLPWR	Gross load power output from the hydraulic pump
INDPWR	Indicated power output, analog calculation
PDLHYD	Pressure rise across hydraulic pump
PDIAST	Power diaphragm stroke
DISPST	Displacer stroke

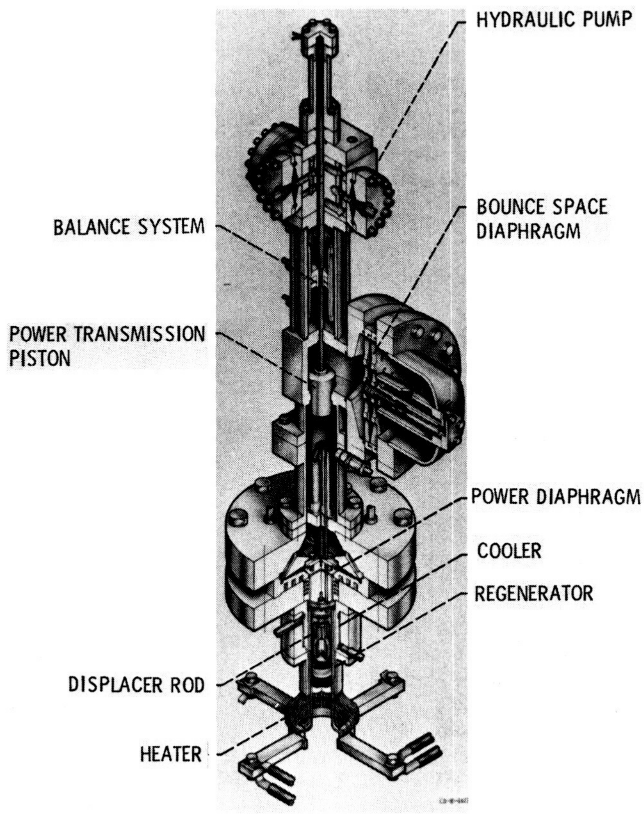


Figure 1. - Cutaway view of the RE-1000 Free-Piston Stirling engine with hydraulic load.

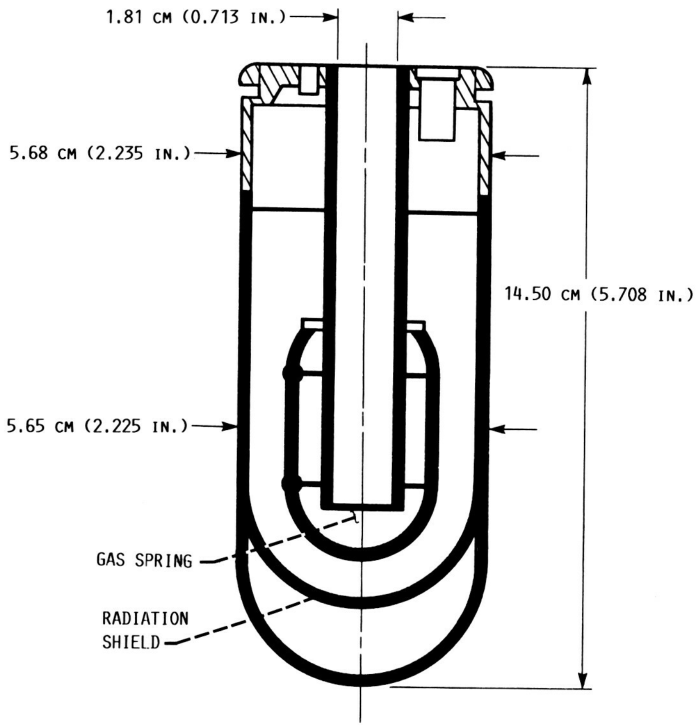


FIGURE 2. - DISPLACER 3 CROSS SECTION. DISPLACER WEIGHT 414.0 G (0.89 LB), GAS SPRING MEAN VOLUME, 23.2 CM^3 (1.42 $IN.^3$).

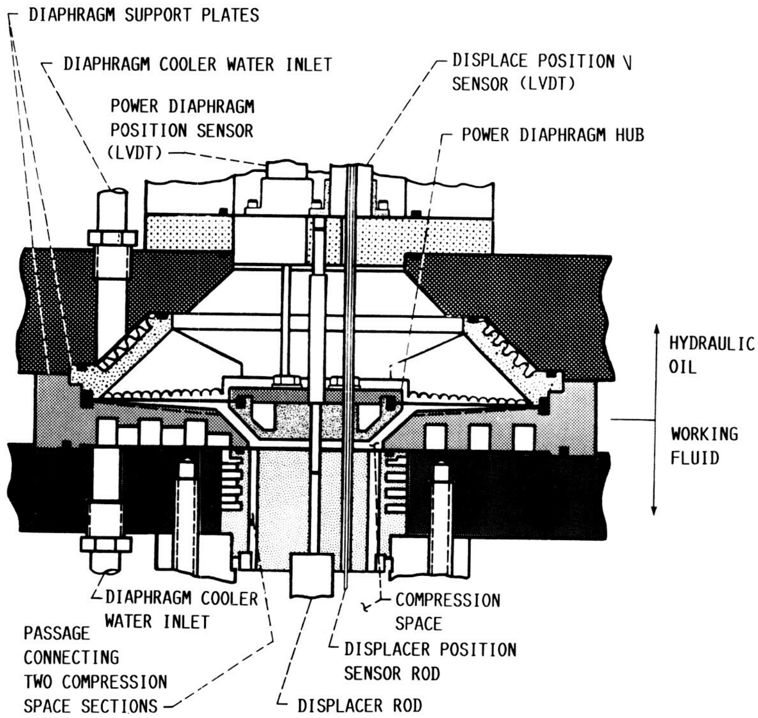
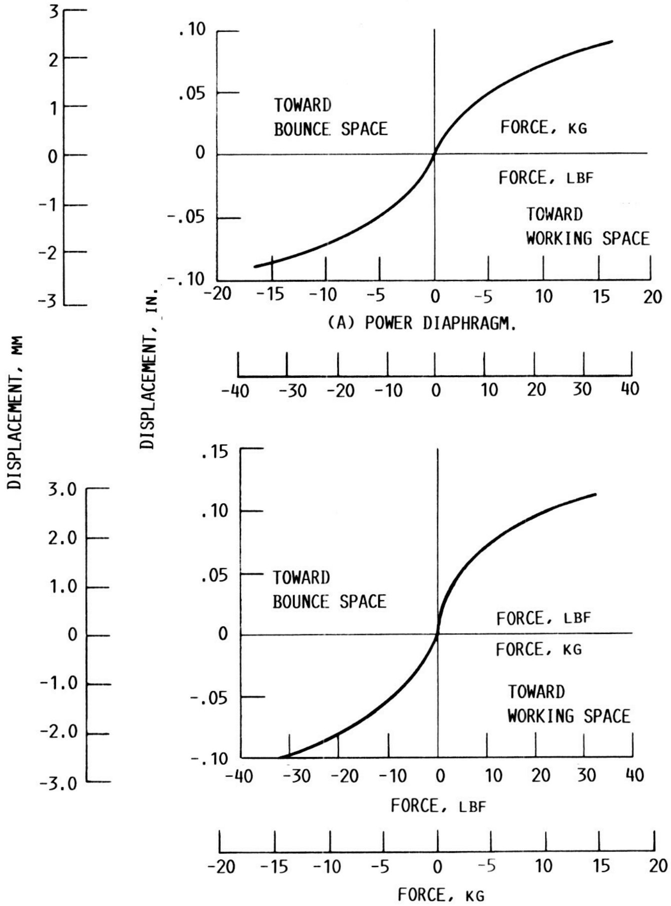


FIGURE 3. - COMPRESSION SPACE CROSS SECTION.



(B) BOUNCE DIAPHRAGM.

FIGURE 4. - RE-1000 HYDRAULIC ENGINE DIAPHRAGM SPRING RATES.

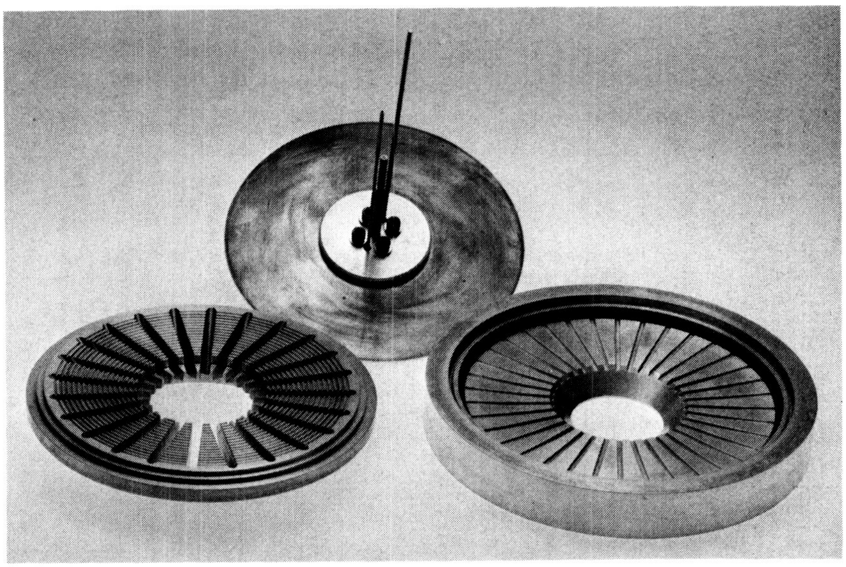


FIGURE 5. - RE-1000 POWER DIAPHRAGM WITH SUPPORT PLATES.

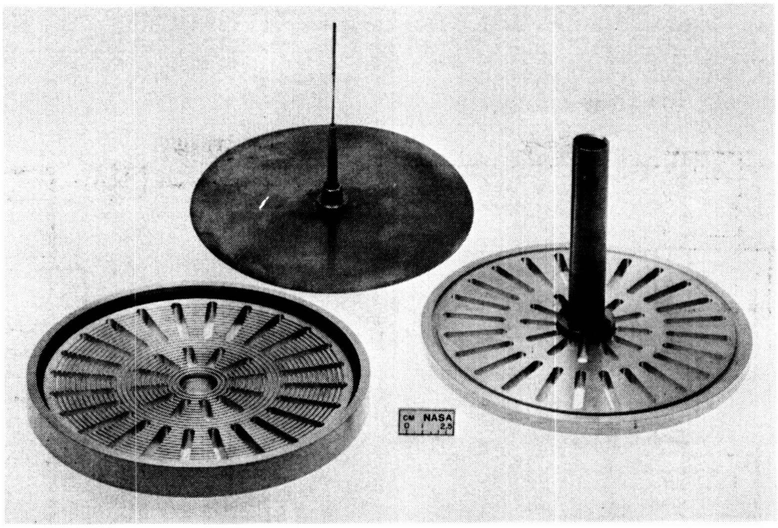
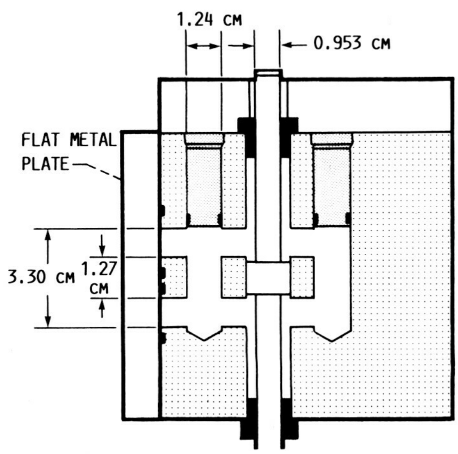
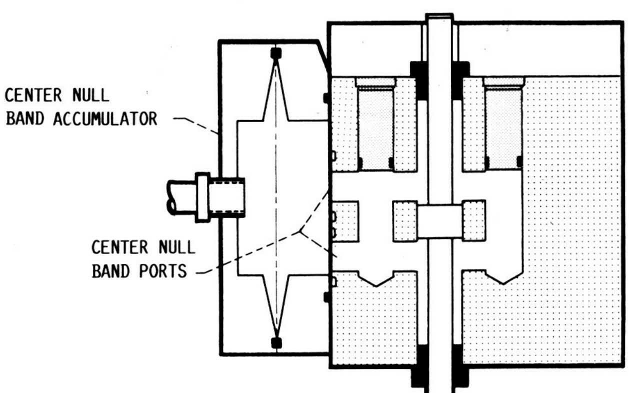


FIGURE 6. - RE-1000 BOUNCE DIPHRAGM WITH SUPPORT PLATES.

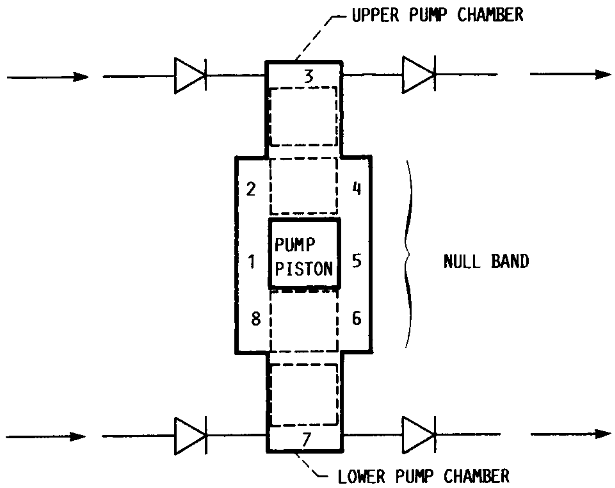


(A) FOUR PULSE-PER-CYCLE CONFIGURATION.



(B) TWO PULSE-PER-CYCLE CONFIGURATION.

FIGURE 7. - PUMP HOUSING CENTER NULL BAND CROSS SECTION.



1-2: NO OIL ENTERS OR LEAVES THE PUMP CHAMBERS. ALL OIL FLOW IS WITHIN NULL BAND

2-3: OIL IS PUMPED OUT OF THE UPPER PUMP CHAMBER AND IS DRAWN INTO THE LOWER PUMP CHAMBER

3: TOP DEAD CENTER OF PUMP PISTON STROKE

3-4: OIL IS PUMPED OUT OF THE LOWER PUMP CHAMBER AND IS DRAWN INTO THE UPPER PUMP CHAMBER

4-5-6: NO OIL ENTERS OR LEAVES THE PUMP CHAMBERS, ALL OIL FLOW IS WITHIN NULL BAND

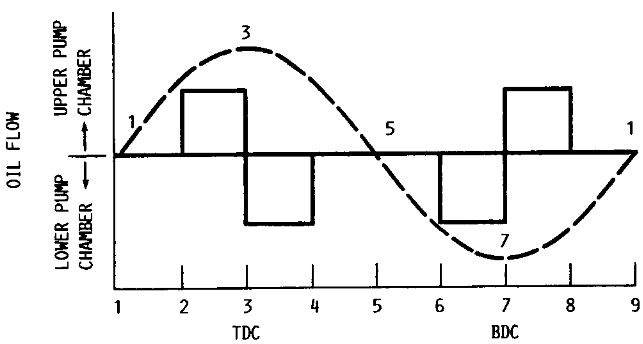
6-7: OIL IS PUMPED OUT OF THE LOWER PUMP CHAMBER AND IS DRAWN INTO THE UPPER PUMP CHAMBER

7: BOTTOM DEAD CENTER OF PUMP PISTON STROKE

7-8: OIL IS PUMPED OUT OF THE UPPER PUMP CHAMBER AND IS DRAWN INTO THE LOWER PUMP CHAMBER

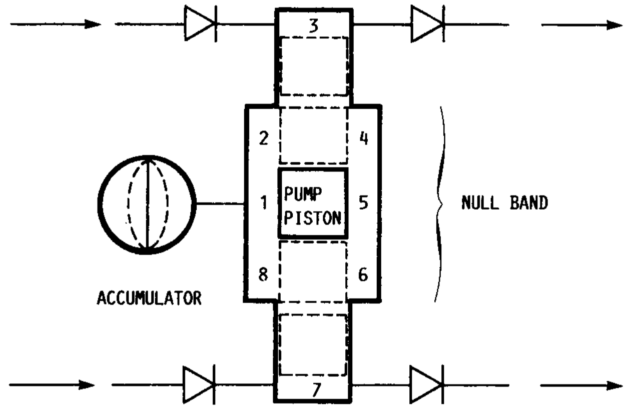
8-1: NO OIL ENTERS OR LEAVES THE PUMP CHAMBERS, ALL OIL FLOW IS WITHIN NULL BAND

(A) PUMP SCHEMATIC.



(B) OIL FLOW OUT OF PUMP.

FIGURE 8. - DOUBLE-ACTING FOUR PULSE-PER-CYCLE PUMP.



1-2: NO OIL ENTERS OR LEAVES THE PUMP CHAMBERS, ALL OIL FLOW IS WITHIN NULL BAND

2-3: OIL IS PUMPED OUT OF THE UPPER PUMP CHAMBER AND IS DRAWN FROM THE ACCUMULATOR

3: TOP DEAD CENTER OF PUMP PISTON STROKE

3-4: OIL IS DRAWN INTO THE UPPER PUMP CHAMBER AND IS PUMPED INTO THE ACCUMULATOR

4-5-6: NO OIL ENTERS OR LEAVES THE PUMP CHAMBERS, ALL OIL FLOW IS WITHIN NULL BAND

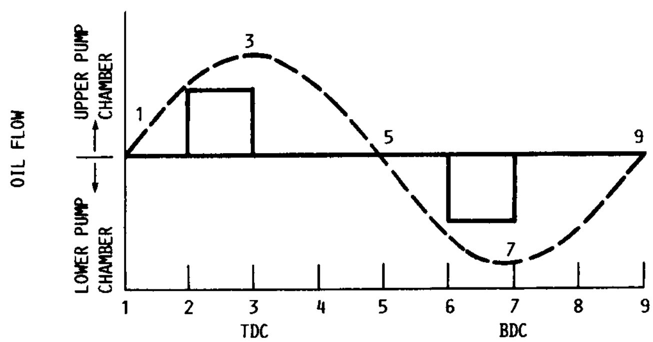
6-7: OIL IS PUMPED OUT OF THE LOWER PUMP CHAMBER AND IS DRAWN FROM THE ACCUMULATOR

7: BOTTOM DEAD CENTER OF PUMP PISTON STROKE

7-8: OIL IS DRAWN INTO THE LOWER PUMP CHAMBER AND IS PUMPED INTO THE ACCUMULATOR

8-1: NO OIL ENTERS OR LEAVES THE PUMP CHAMBERS. ALL OIL FLOW IS WITHIN NULL BAND

(A) PUMP SCHEMATIC.



(B) OIL FLOW OUT OF PUMP.

FIGURE 9. - DOUBLE-ACTING TWO PULSE-PER-CYCLE PUMP.

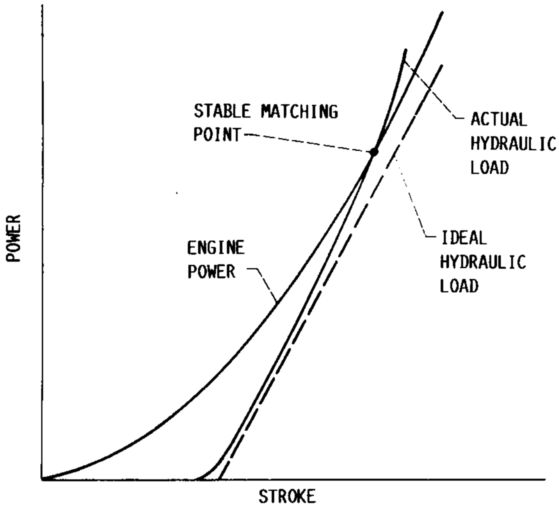


FIGURE 10. - HYDRAULIC LOAD AND POWER OUTPUT MATCHING.

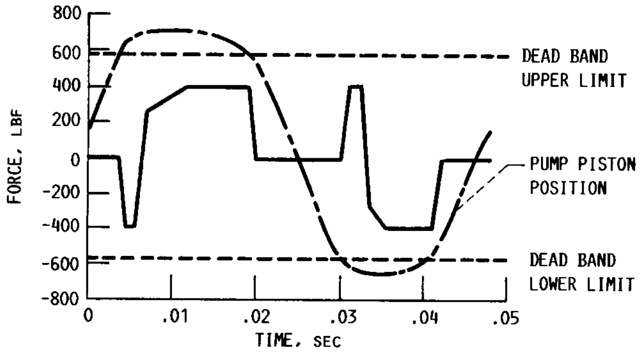


FIGURE 11. - HYDRAULIC LOAD VERSUS TIME. OIL FORCE ON PUMP PISTON VERSUS TIME FOR THE DOUBLE-ACTING, FOUR PULSE-PER-CYCLE PUMP CONFIGURATION.

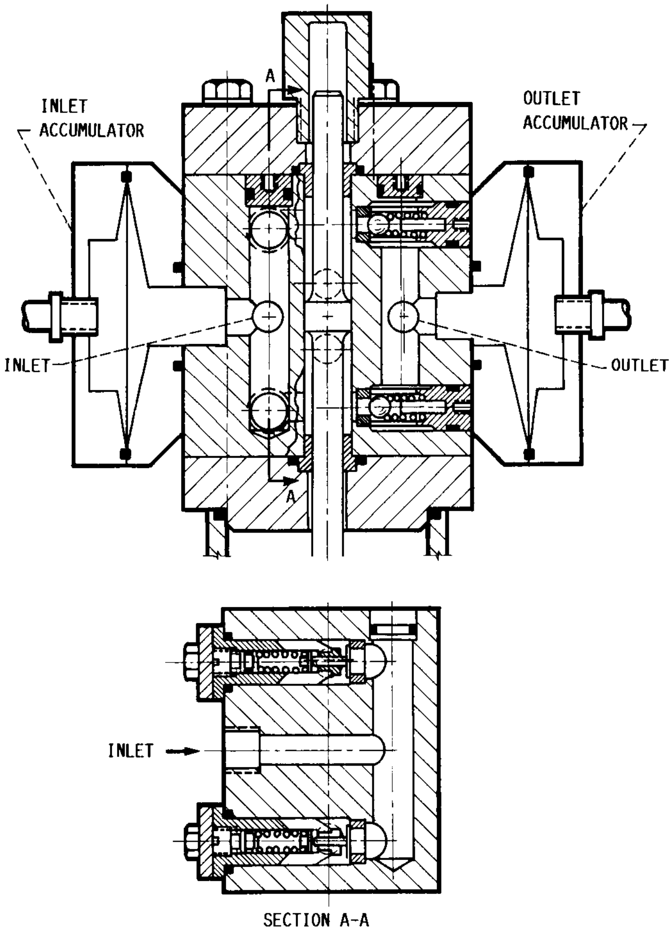


FIGURE 12. - PUMP HOUSING CROSS SECTION SHOWING THE INLET AND OUTLET PASSAGES.

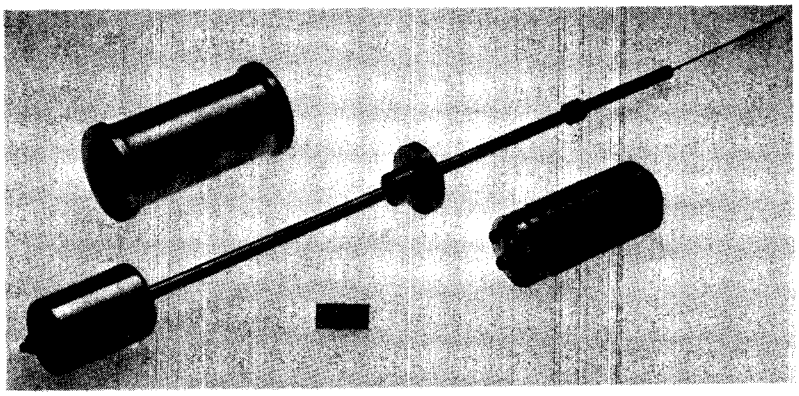


FIGURE 13. - HYDRAULIC LOAD PUMP ROD ASSEMBLY WITH BALANCE PISTON.

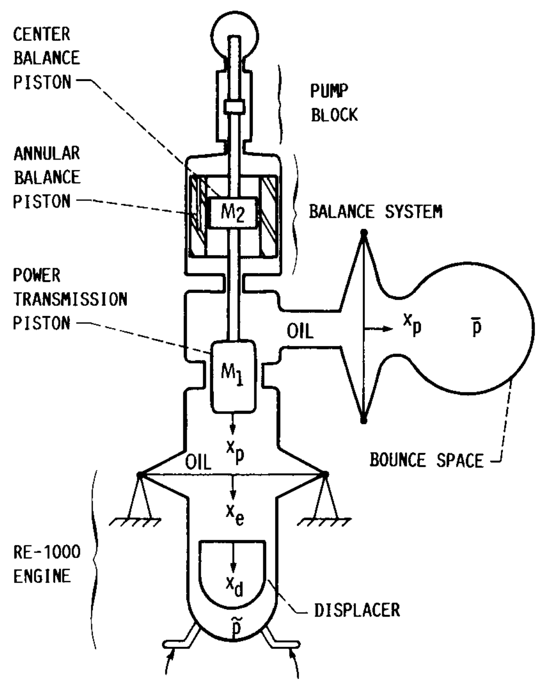


FIGURE 14. - SCHEMATIC OF HYDRAULIC OUTPUT DEVICE.

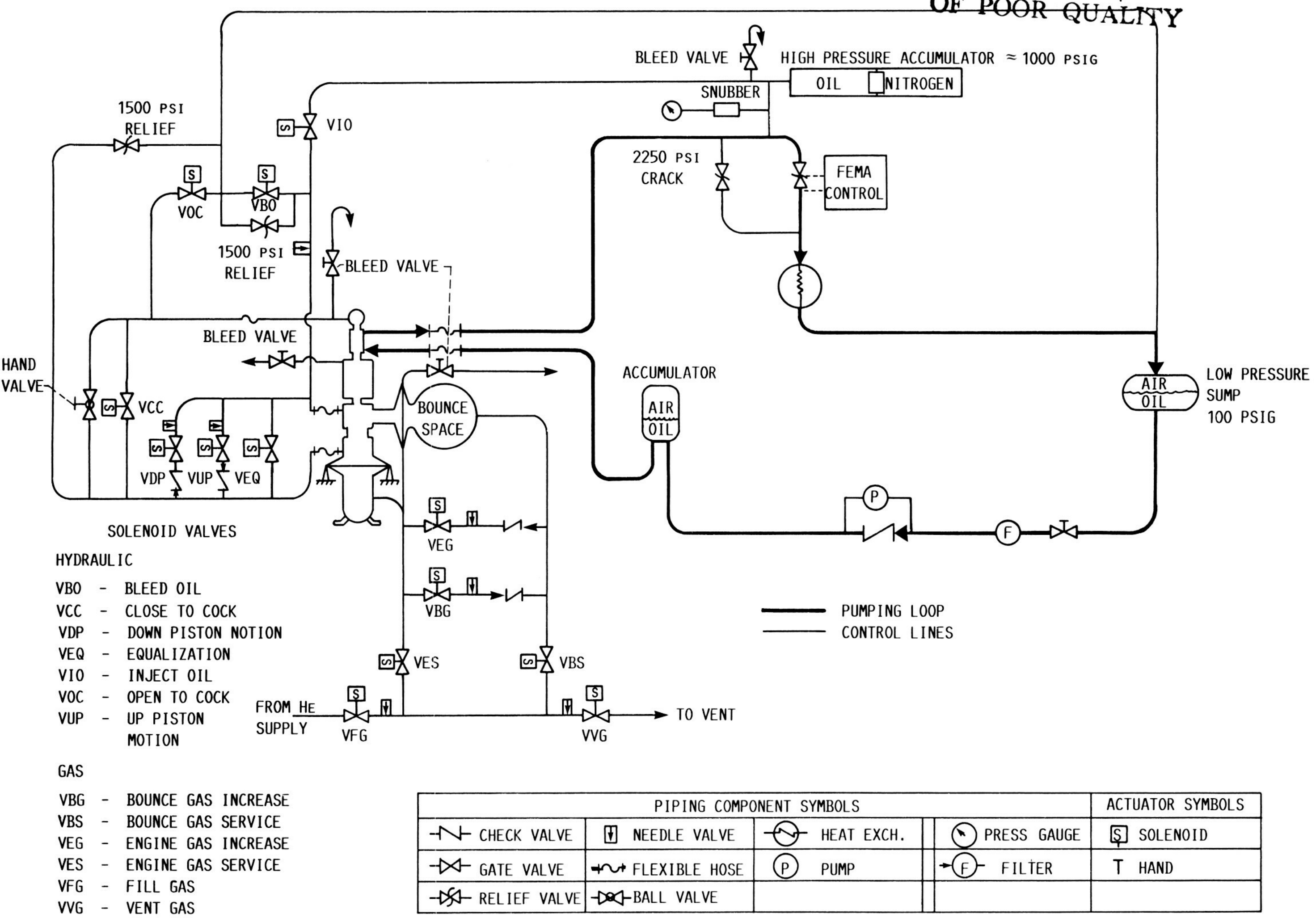


FIGURE 15. - SCHEMATIC OF HYDRAULIC SYSTEM.

FIGURE 16. - CONTROL ROOM.

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15. Supplementary Notes

16. Abstract

The NASA Lewis Research Center has been involved in free-piston Stirling engine research since 1976. Most of the work performed in-house was related to the characterization of the RE-1000 engine. The data collected from the RE-1000 tests was intended to provide a data base for the validation of Stirling cycle simulations. The RE-1000 was originally built with a dashpot load system which did not convert the output of the engine into useful power, but was merely used as a load for the engine to work against during testing. As part of the interagency program between NASA Lewis and the Oak Ridge National Laboratory, (ORNL), the RE-1000 has been converted into a configuration that produces useable hydraulic power. A goal of the hydraulic output conversion effort was to retain the same thermodynamic cycle that existed with the dashpot loaded engine. It was required that the design must provide a hermetic seal between the hydraulic fluid and the working gas of the engine. The design was completed and the hardware has been fabricated. The RE-1000 was modified in 1985 to the hydraulic output configuration. The early part of the RE-1000 hydraulic output program consisted of modifying hardware and software to allow the engine to run at steady-state conditions. This report presents a complete description of the engine, in sufficient detail so that the device can be simulated on a computer. Tables are presented showing the masses of the oscillating components and key dimensions needed for modeling purposes. Graphs are used to indicate the spring rate of the diaphragms used to separate the helium of the working and bounce spaces from the hydraulic fluid.

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