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SUMMARY

A 1 kW (1.33 hp) single cylinder free-piston Stirling engine was installed in the test facilities at the Lewis laboratory. The engine was designed specifically for research of the dynamics of its operation. A more complete description of the engine and its instrumentation is provided in a prior NASA paper TM-82999 by J. G. Schreiber.

Initial tests at Lewis showed the power level and efficiency of the engine to be below design level. Tests were performed to help determine the specific problems in the engine causing the below-design-level performance. Modifications to engine hardware and to the facility where performed in an effort to to bring the power output and efficiency to their design values. As finally configured the engine generated more than 1250 watts of output power at an engine efficiency greater than 32 percent.

This report presents the tests performed to help determine the specific problems, the result if the problem was eliminated, the fix performed to the hardware, and the test results after the engine was tested. In cases where the fix did not cause the anticipated effects, a possible explanation is given.

INTRODUCTION

A 1 kW (1.33 hp) single cylinder free-piston Stirling engine was purchased and installed in the test facilities at the Lewis Research Center. The RE-1000 engine test program was designed to provide detailed engine operation data for use in the validation of free-piston Stirling engine computer codes and later to evaluate alternative load devices on the engine. This work is funded under a joint cooperative interagency agreement, number DE-AI05-82OR21005, between the National Aeronautics and Space Administration Lewis Research Center (NASA LeRC) and the Department of Energy Oak Ridge National Laboratory (DOE/ORNL).

Although the engine was found to be reliable, easy to operate and extremely quiet, the power output and efficiency calculations were much lower than the design goals. A major effort over the past 8 months has been made to detect and correct the problems in the RE-1000 and to restore the engine performance to the levels reached during acceptance tests at the contractor's site. Restoring the performance to the design level will allow data to be obtained for computer code validation.

The process of finding and correcting problems in the free piston Stirling engine has been time consuming, but has also provided an excellent tool for learning about the interrelationships of different design parameters and the operating dynamics of the engine.

This report covers some of the problems found and the methods by which they were detected. It also gives a description of corrective action taken and the results achieved by subsequent engine tests.

ENGINE DESCRIPTION

The RE-1000 engine tested at Lewis is shown in Figure 1; a cutaway drawing is shown in Figure 2. The engine was designed and fabricated at Sunpower Inc., Athens, Ohio. The RE-1000 was built with the intent to use it as a general free-piston Stirling test bed engine for future research, load device characterization, and data aquisition for computer code validation. To facilitate the installation of the desired instrumentation, the engine was built with many extra penetrations into selected areas of the heat exchangers.

The engine was optimized for maximum efficiency with helium as the working fluid at 7.0 MPa mean operating pressure; heater tube metal temperature set at 600° C; operating frequency of 30.0 Hz and a piston stroke of 2.54 cm. The engine design optimization was done using a previously designed heater head and cooler and does not represent the best overall optimized free piston Stirling engine possible.

The RE-1000 engine was designed with a posted displacer, an annular regenerator and cooler, and an electric resistance heater head. The power piston is shown in Figure 3 and the displacer in Figure 4. Since the engine was intended for research of the dynamics, a dashpot was used to absorb the power generated. Sliding surfaces were made of either hardened stainless steel or had chrome oxide surface coating for wear resistance. Effective sealing of different volumes was provided by minimal clearance of mating parts.

The basic measured dynamic parameters on the engine include piston and displacer position, piston velocity, compression space pressure, force exerted on the piston by the load, and heat exchanger pressure drops. The steady state measurements include mean pressure of the working space and bounce space along with many metal, gas, and coolant temperatures. Other parameters are calculated from these basic measurements by either local analog circuits or by the LeRC IBM 370 data system. These parameters include the brake power, indicated power, piston stroke, displacer stroke, power input to the heater head and heat rejected from the cooler. More information on the engine and test facilities can be obtained in reference 1.

OPERATION DEFICIENCY CORRECTION

The characteristics of the engine operation before the repair process was initiated can be summarized as follows: the displacer had a high phase angle with respect to the piston and a low stroke; the pressure phase angle was small in magnitude; the power output and efficiency were low; the heat rejected by the cooler was high; and the displacer ran centered toward the compression space. Several defects in the engine were known but it was decided that only one corrective action should be taken at a time. It was felt that this would provide more meaningful information and experience.

From past experience in the assembly of the RE-1000, it was known that at some point in the stack up of the heat exchangers, some amount of misalignment existed. Careful inspection and measurements taken showed that the problem originated at the mounting flange of the heater head. At this point the centerline of the heater head deflected at a slight angle from the centerline of the cooler assembly. The flange at which this misalignment originated is indicated in Figure 5.

The first step taken to refurbish the engine was to remachine the mounting flange of the heater head to insure a truer stack up of the heat exchanger assembly. Prior to this remachining process, the displacer would very lightly rub on the cylinder wall. The rub was light enough that the engine would still run smoothly. The effect of the rub, however, was to cause some amount of damping to the displacer motion.

The RE-1000 is designed such that the natural frequency of the displacer is approximately 28 Hz. Since the engine operates at 30 Hz due to the dominant mass of the power piston, the ratio of the driving frequency to the natural frequency of the displacer is 30/28 = 1.07. It can be seen in Figure 6 that for the case of forced vibration with viscous damping, at a frequency ratio of 1.07, the phase angle between the driving force and the driven response would decrease as the damping increased (Ref. 2) on the RE-1000. However, the phase angle between the power piston and displacer is not the same as the phase angle being shown in Figure 6. This is explained more fully in the following.

Figure 7 is a phasor diagram of a possible operating point, similar to the design point, for the RE-1000. The phasor diagram is used to show the amplitudes and phase relationships of the different sinusoidal engine parameters. The displacer motion is shown to lead the power piston motion by 45° and the pressure in the compression space lags the piston motion by 25°. Since there is no kinematic linkage in the free piston Stirling engine to drive the displacer, the only forces exerted on the displacer body come from the internal gas spring and the pressure in the working space. To preserve proper sign convention, the driving force on the displacer caused by pressure variations in the working space is along the negative pressure vector. The phase angle between the negative pressure vector and the displacer motion vector is the phase angle represented in Figure 6. It can be shown that this phase angle is 110°.

With the frequency ratio at 1.07, Figure 6 shows that the phase angle between the driving force and driven response will decrease as the damping increases. This is the same as saying that the phase angle between the power piston and displacer will increase as the damping increases. The remachining of the heat exchanger stack up was intended to reduce the damping by eliminating the slight rub and therefore should reduce the phase angle between the power piston and displacer.

Subsequent engine test runs showed almost no difference in power output, efficiency, or phase relationship in the engine. The only noticeable difference was that the engine was easier to start. The reason that this elimination of the rub had so little effect on the engine's operation is that the rub would generally occur when the displacer was at the extreme end of its cylinder toward the expansion space. During operation, the displacer spent little, if any, time at the end of the cylinder since during normal operation the displacer does not travel to the end of the cylinder.

Now that it was known that the complete engine assembly was true and that no misalignment existed in the engine, it was decided that the next task would be to get the displacer to operate more centered in the working space. The displacer had been running toward the compression space, always risking collision with the physical limitation of its motion and also leaving a great amount of dead volume in the expansion space.

The centering port system for the displacer consists of two ports in the displacer rod and two ports in the bore of the displacer. The two ports in the bore of the displacer have a circumferential groove in the surface of the bore connecting them to one another. When the ports in the bore of the displacer line up with the ports in the displacer rod, the displacer gas spring is able to communicate with the main bounce space or buffer space in the pressure vessel atop the engine. The volume in the pressure vessel is large enough that it has almost no pressure variation throughout the engine cycle. The displacer centering port system is therefore able to reference the displacer gas spring to a steady pressure, the engine mean operating pressure, when the centering ports open. This system is needed to make up for any leakage of gas into or out of the gas spring from the compression space. The system is shown in Figure 8.

It was found that the centering ports in the displacer were rotated 60° from the centering ports on the displacer rod. Although the circumferential groove in the displacer bore allows for this condition, it was felt that if the ports were aligned so as to not need the circumferential groove, a stronger centering system would exist. Because of provisions in the engine for some future instrumentation, it was possible to assemble the engine in a configuration which would permit the centering ports to be aligned.

Tests were run on the engine in this configuration. The displacer phase angle was lowered significantly and the displacer was operating slightly closer to being centered. The tests showed that the displacer phase angle relative to the power piston was reduced from about 62° to approximately 53°. The large change in phase angle was somewhat unexpected, although the explanation is rather simple.

The stiffness of the gas spring inside of the displacer is determined by the gas being used, the mean pressure of the gas, and the mean volume of the gas spring. The gas for these tests is always helium and the mean pressure is always 7.0 MPa. When the displacer was forced to operate more centered by the improved centering port system, the mean volume of the gas spring was increased slightly. This had the effect of making a softer gas spring for the displacer to rebound against, and therefore lowered the natural frequency of the displacer. Lowering the natural frequency of the displacer produces a higher ratio of the driving frequency to the driven frequency for the displacer. Referring to Figure 6, it can be shown that for a fixed amount of damping, the higher frequency ratio will yield a higher phase angle between the driving force and the driven response. This higher phase angle produces a lower phase angle between the piston and displacer.

Even though the test results showed that some of the problems in the engine were partially corrected, many others still existed. The power and efficiency were still low, the pressure phase angle was low, and the heat being rejected by the cooler was high.

In order to determine if any extra damping on the displacer motion was being caused by unwanted flow resistance in the heat exchangers, tests were run with a nitrogen gas flow test system. Pressure drops were measured with Validyne differential pressure transducers. The tests were run with the flow from compression space to expansion space, and then reversing from expansion space to compression space. Inlet pressure levels of 2.10 MPa and 1.40 MPa were used and the flow rate was varied to get data in the laminar, transition, and turbulent ranges.

The flow test data indicated that there may be a leak in the heat exchangers such that the same amount of gas was not flowing through all components of the heat exchanger assembly. This conclusion was reached by comparing the flow test results for opposite directions of flow, but for similar conditions of pressure and flow rate. It was impossible to determine if the differing results were due to a leak path or if it was merely caused by the different entrance and exit loss paths being reversed.

The mating junction between the cooler housing and the regenerator was suspected as being a potential leak path. A leak here would allow gas to go directly from the expansion space into the cooler or from the cooler directly into the expansion space. This would explain the high heat rejection rate at the cooler.

A guide ring is installed in the engine during assembly to insure proper alignment of the cooler to the regenerator. A seal made of high temperature RTV was formed in the gap by the alignment ring during assembly. Flow tests were run to check if the RTV seal would change the test results and to insure the integrity of the seal to withstand the pressure forces it would be subjected to during engine operation. The flow tests showed very little change in pressure drop measurements with the RTV seal in place. The engine was assembled and tested but no significant difference in performance was found. It was concluded that if a leak did exist it was not a major factor in the engine performance.

The next area of the engine to be repaired was the power piston cylinder. During past tests, rub marks would be found on the power piston and inside of the power piston cylinder. The rub marks inside of the power piston cylinder would generally be around the centering ports drilled through the cylinder wall. The power piston centering port system is very similar to the centering port system used on the displacer. When the piston reaches a predetermined point, ports in the cylinder wall line up with ports on the piston surface. These ports when aligned allow the working space to communicate with the main pressure vessel gas which remains at the engine's mean operating pressure.

Engine tests were run after the cylinder was honed to eliminate any high spots. The friction on the power piston in its cylinder was reduced significantly. This had the effect of raising the brake power output by reducing the friction, but did not change the indicated power.

At this point the engine produced 900W indicated power, had a displacer phase angle of 53° and showed a pressure phase angle of -12°. The computer simulation of the RE-1000 at Sunpower Inc., with some of the measured test parameters used as inputs, predicted the indicated power generated should have been slightly over 1250W. It was noted that although both the displacer phase angle and the pressure phase angle relative to the power piston were off from the design values, the phase relationship of the displacer with respect to the pressure wave was nearly at the design value. This indicates that the dynamics of the displacer are not too far from that desired. If the phase angle of the pressure relative to the piston position could be corrected to a value of -20° to -22°, the phase angle of the displacer relative to the piston position would follow and come down to 44° to 42°.

A pressure decay half-life pressure test was then performed to check the level of leakage past the power piston. This test is performed by clamping the power piston at some desired position. The pressure vessel of the engine is removed so as to expose the back side of the power piston to atmosphere. The working space is then pressurized with helium to a pressure of 1200kPa. The amount of time for the presure in the working space to decay to 600kPa is measured. For the engine to be able to run as designed, the half life of the pressure should be about 15 seconds or more. The half-life tests showed the measured time to be approximately 2 seconds. At some time during earlier tests the half life was measured to be 15 seconds. Since the gas leaking out of the working space travels out the annular gap between the piston and cylinder to the centering ports and then out through the centering ports, it was theorized that honing away the high spots that surrounded the centering ports allowed the leak rate to increase. Figure 9 showed the leak path being investigated during the half life tests.

To see if the leak rate could be slowed without rechroming the cylinder wall to minimize the gap between the piston and cylinder, 2 of the 4 centering ports drilled through the cylinder wall were plugged with epoxy. Half-life pressure tests were run to see if this lessened the leak rate, but the half life was still about 2 seconds. This indicated that even with 2 of the 4 centering ports blocked, the major restriction to leakage was the piston to cylinder wall clearance, and therefore the piston to cylinder clearance must be reduced to control the leak rate.

The engine was assembled for testing with the only change being in the centering port system. During this test run it was found that for the same operating pressure, temperatures and piston stroke, the heat rejected from the cooler was lower by 200 W and the power input to the heater was lowered by 200W. The mean gas temperature in the compression space was lowered by approximately 10° C and the gas temperature between the cooler and regenerator was lowered by approximately 8° C. The gas temperature between the regenerator and heater and the expansion space gas temperature was not changed. Because the piston had a much weaker centering port system than before, the piston tended to operate slightly farther in the cylinder toward the compression space.

It is interesting to note that the design of the centering ports for the displacer is less critical than the design of the centering ports for the power piston. In the case of the displacer gas spring, the centering ports will flow enough gas to make up for any leakage into or out of the gas spring. If the gas spring pressure had no hysteresis there would be very little flow through the displacer centering port system and therefore very little power loss. The power piston centering port system, however, will yield large losses if too much gas is allowed to flow. Since there is a much larger P-V diagram in the compression space, the working gas will try to flow to the buffer space or bounce space when the piston is on the expansion part of the cycle. Similarly, the gas from the bounce space will try to flow into the working space on the compression part of the cycle. P-V diagrams are shown in Figure 10.

Figures 11 to 15 show the changes to the engine gas temperatures and the brake power output resulting from these modifications. Shown are the results after correcting the alignment of the heat exchangers and rotating the displacer rod, the results after sealing the regenerator with RTV, and the results after honing the cylinder and eliminating 2 of the 4 power piston centering ports.

It appeared that the major problem left was to reduce the leakage of gas past the power piston. A fixture was made that replaced the power piston in the cylinder and sealed the working space with a static "O" ring. With this fixture in place the working space could be pressurized with helium to check for any leakage from the working space to the bounce space through leak paths other than the radial gap between the piston and the cylinder. The tests showed that there was no leakage out of the working space and into the bounce space through any static "O" rings or through any valves.

Based on the results of these leak tests it was determined that the radial clearance between the power piston and the cylinder should be reduced. Since the cylinder had recently been honed and was therefore known to be round and true, it was thought that the best solution would be to recoat the power piston and grind it to provide 0.020 mm diametral clearance instead of the original value of 0.036 mm. The displacer bore was honed and the displacer rod was recoated with chrome oxide at this same time to bring all of the close tolerance parts back to the original level of quality. The power piston, however, was coated with a fluorocarbon-based, low-friction substance. This coating was applied and ground to size such that it provided the 0.020 mm diametral clearance desired.

The engine was assembled for further test runs. The first test run showed an immediate power piston position change from previous test runs. On earlier test runs the power piston had been centering itself in toward the working space from the centering ports. Now the piston stroke was centered extremely far out, away from the working space. The power and efficiency had not changed greatly but the fact that the piston was centering itself quite far away from the working space was a significant clue to problems with the engine. Leakage through the radial gap around the piston will generally cause the power piston to drift toward the working space the same way that leakage by the displacer rod would cause the displacer to run further onto the displacer rod. If the seal along the power piston or along the displacer rod does not change much as the position of the piston or displacer changes, then the volumetric leakage past the seal in one direction will be roughly equal to the volumetric leakage past the seal in the opposite direction. In the case of the power piston, the leak path is between the working space and the bounce space, with most of the gas traveling along a path utilizing the power piston cylinder centering port system. In the RE-1000 engine there is almost no pressure variation in the bounce space due to its large volume; therefore, the bounce space pressure is approximately equal to the mean operating pressure of the engine. There is, however, a rather large pressure variation in the working space. When the working space pressure is lower than the bounce space pressure, some amount of gas will leak from the bounce space to the working space. When the working space pressure is higher than the bounce space pressure, the leakage will be from the working space to the bounce space. If the sealing ability of the piston is equivalent at both of these points in the cycle, then the volumetric leak rate out of the working space will equal the volumetric leak rate into the working space. However, because the density of the gas is greater during the high pressure part of the cycle, the mass flow rate out of the working space will be greater than the mass flow rate into the working space. This phenomena will cause the power piston to drift toward the working space. A similar argument can be used to show that the displacer in the RE-1000 will have a tendency to drift farther onto the posted displace rod.

The important point is that if the leak path or seal system is equivalent for both directions of leakage, the volumetric leak rates may be equivalent but the mass leak rates will differ resulting in a biased piston position. With the cylinder recently honed by the engine manufacturer to have a diametral variation of less than 0.0025 mm and the piston recently coated and ground to have a diametral variation over its length of less than 0.0025 mm, the piston should have operated with the mid-point of its stroke very near the centering ports or perhaps slightly in toward the working space. The fact, however, was that the piston was operating so far out from the working space that the centering ports were opening at the far inward end of the stroke. This indicated that there was some leak path that existed during engine operation that permitted a large flow of gas from the bounce space to the working space but not from the working space to the bounce space.

In an effort to isolate such a leak path all unnecessary instrumentation and facility plumbing was removed in an attempt to operate the engine in the same configuration as at the engine manufacturer's site. At that time, the engine did operate as per the design goals. In order to place the engine in the contractor's configuration, the following was done. The Validyne differential pressure transducers were removed. A valve known as the short circuit valve was removed. The short circuit valve existed to connect the working space to the bounce space when charging the engine or venting the engine. Check valves to allow gas from a supply system to be introduced to the working space and to allow gas to be vented from the working space were removed. With these two check valves removed, the engine would be brought up to full operating pressure after operation has started by slowly bleeding gas into the bounce space and letting the piston centering ports equalize the mean pressure across the power piston. These valves are shown schematically in Figure 16.

Subsequent tests showed the power level to be able to go well above 1200W brake power and efficiency levels above 32 percent as calculated by the following equation:

$$\text{Efficiency} = \frac{\text{Brake Power}}{(\text{Heat Input to the Head})}$$

Curves of brake power output and efficiency are shown in Figures 17 and 18. These tests were run with the cooling water inlet temperature set at 30° C and with helium at 7.0 MPa as the working gas.

Subsequent tests showed that the Validyne differential pressure transducer and the short circuit valve had no effect on engine performance. It was found that the working space venting check valve was not sealing properly although it is not known exactly when this valve began to fail. The method used at LeRC to check piston leakage would not show a leak problem in that check valve.

With the engine operation back to the original design goals, a complete mapping of the performance can be initiated. The engine mapping will be done over a range of conditions. These conditions include 4 heater head temperatures, 3 cooler temperatures, 3 mean pressure levels, and 6 different power piston stroke settings. The map may be expanded in several limited areas. The detailed data will be published for use in the validation of free-piston Stirling engine computer codes.

CONCLUDING REMARKS

The RE-1000 developed more than 1[illegible]60 watts of output power and was found to be a desirable engine for general free piston Stirling engine tests. The design of the engine made it reliable and allowed nearly silent operation. The overall layout made it very easy to work on.

Although many small problems were found in the engine, all of these problems are actually rather simple and easily diagnosed with a proper understanding of the dynamics involved. None of the problems encountered indicate that the free-piston Stirling engine is not suited to mass production.

The most difficult handicap in the investigation into the performance discrepancies of the RE-1000 was the fact that if more than one problem exists, the symptoms other than low power or low efficiency may hide one another. An example is the situation that existed with leakage through the piston to cylinder clearance gap along with the leaky check valve. The leak around the piston would cause low power output, low efficiency, and the power piston to drift in toward the working space. The leaky check valve would cause low power, low efficiency, and the power piston to drift out away from the working space. When both problems exist, the piston may not show a strong tendency to drift in either direction.

The program has shown that the RE-1000 will make an ideal test bed for general mapping, sensitivity testing, and potential endurance testing. The only reliability problems encountered were with delicate instrumention systems or with facility systems. Efficiency values of over 32 percent were achieved which is considered excellent for an engine which was not optimized for efficiency as a complete unit.

References

1. Schreiber, Jeff: Testing and Performance Characteristics of a kW Free-Piston Stirling Engine. NASA TM-82999, 1983.
2. Timoshenko, S; Young, D. H.; and Xieaver, LW., Jr.: Vibration Problems in Engineering. 4th Edition, John Wiley and Sons, 1974.

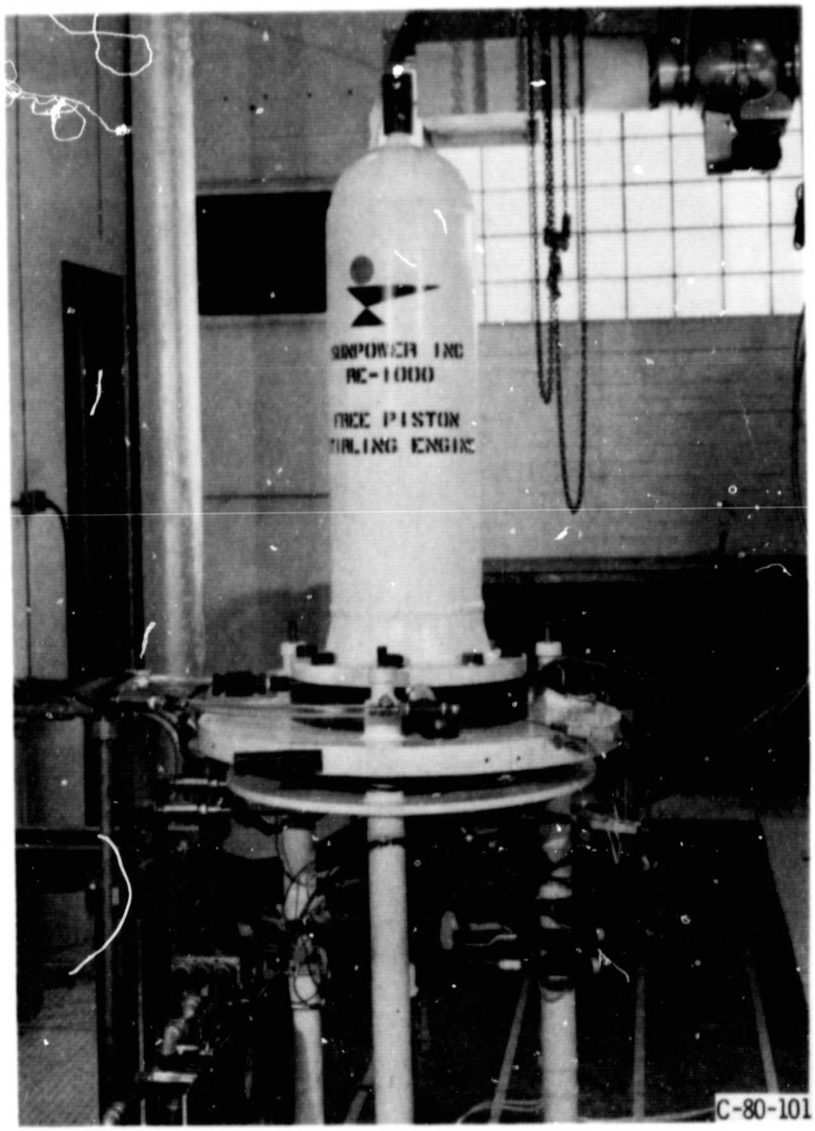

Figure 1. - RE-1000 free-piston Stirling Engine in test cell.

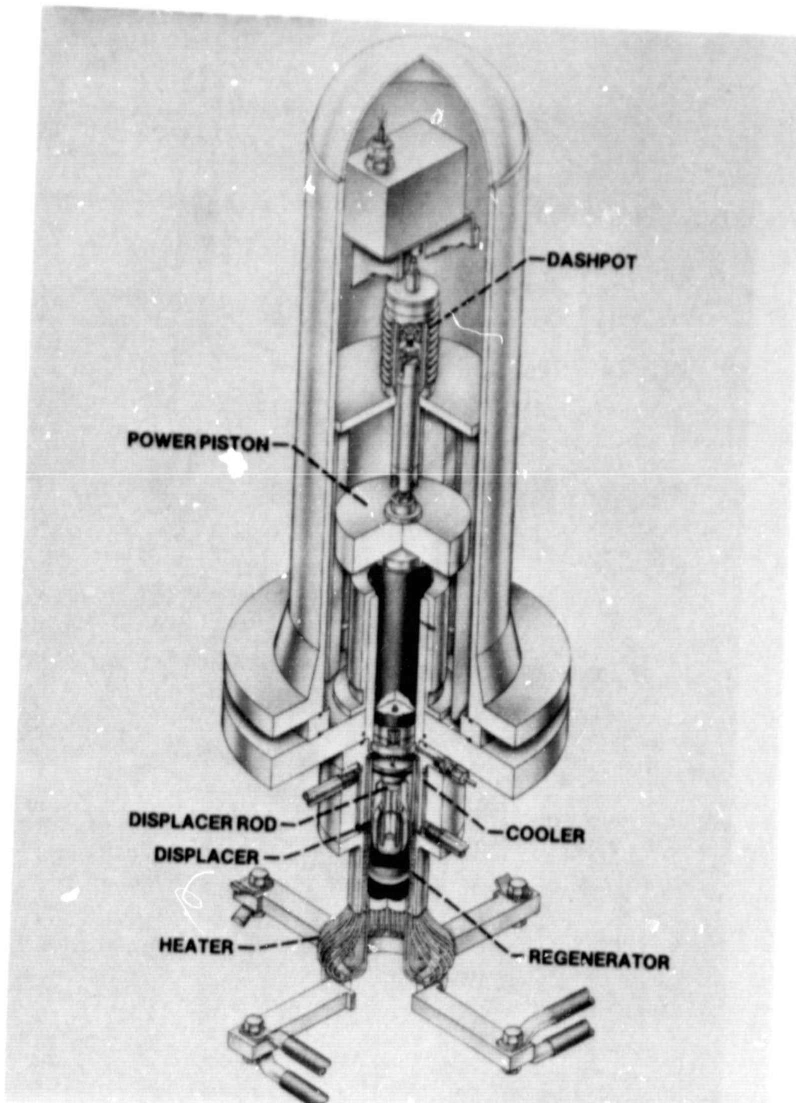

Figure 2. - Cutaway view of RE-1000 free-piston, free-displacer Stirling Engine.

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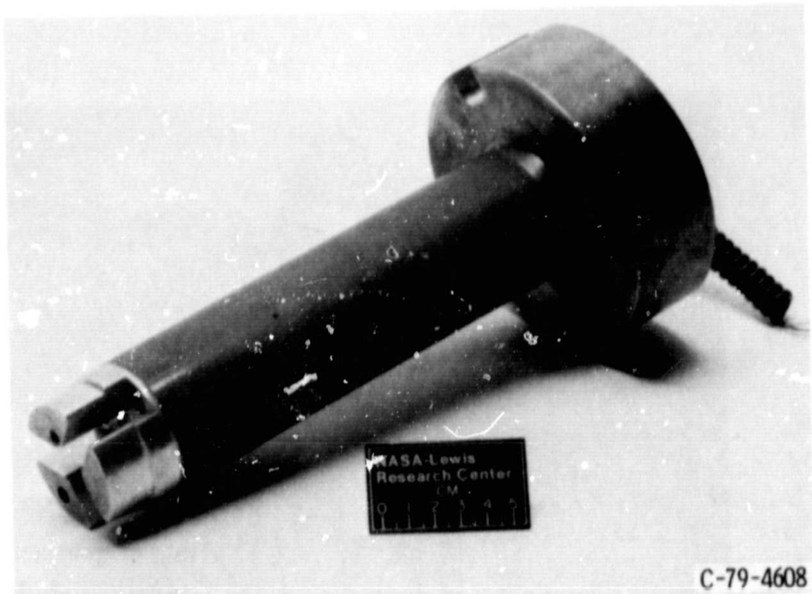

Figure 3. - RE-1000 free-piston Stirling Engine power piston.

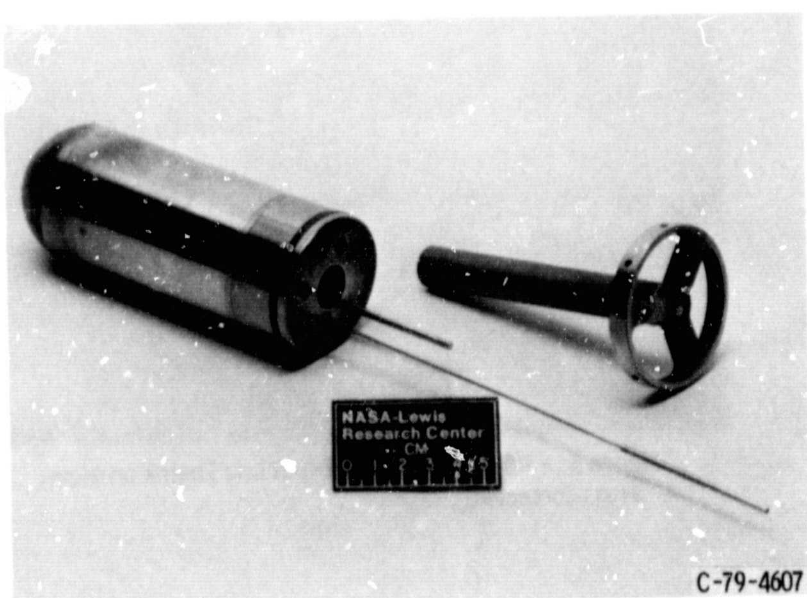

Figure 4. - RE-1000 free-piston Stirling Engine displacer and displacer rod.

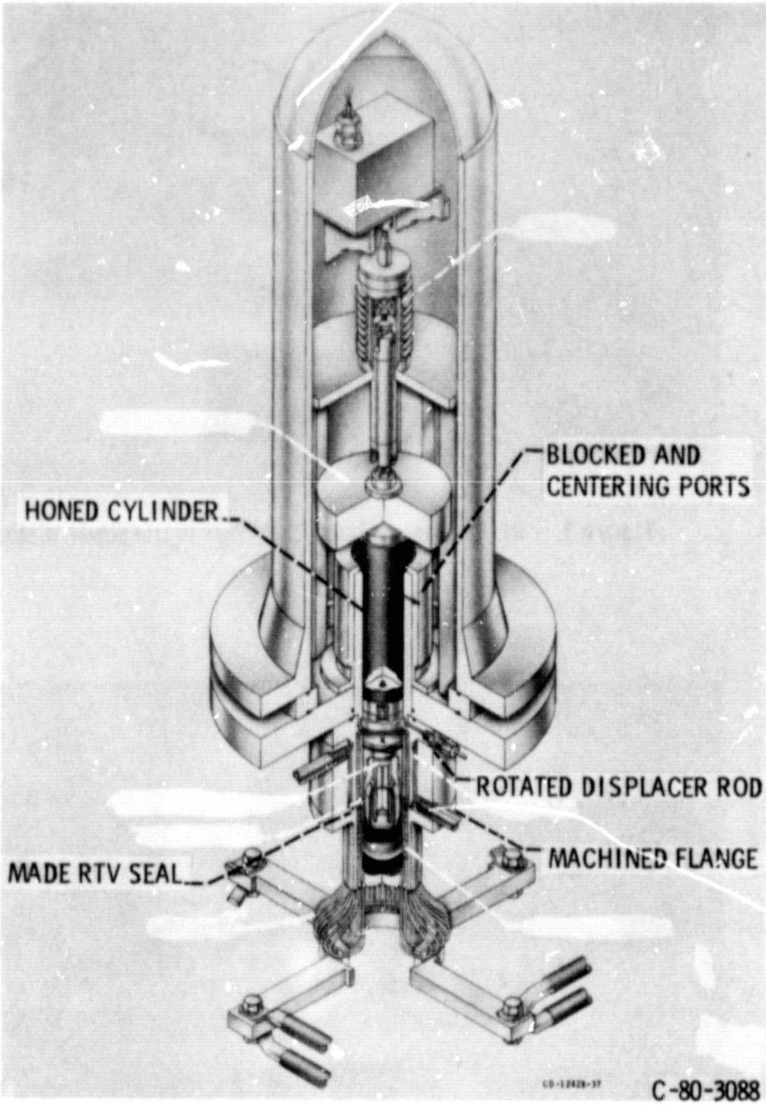

Figure 5. - RE-1000 free-piston Stirling Engine problem areas corrected.

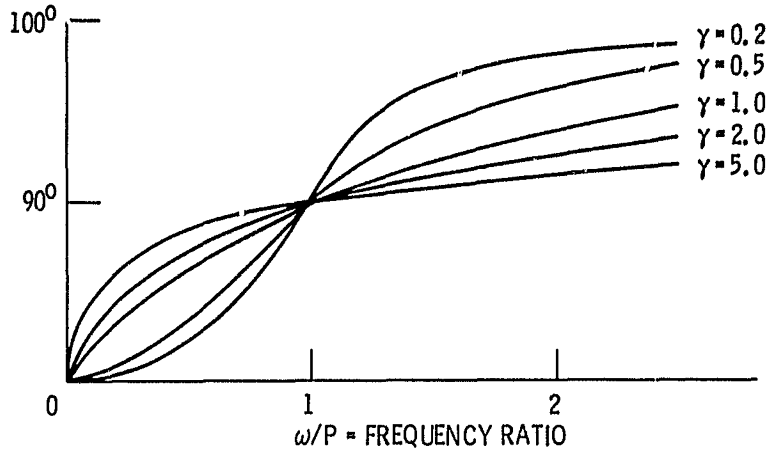

Figure 6. - Relationship of phase angle vs frequency ratio at several damping values.

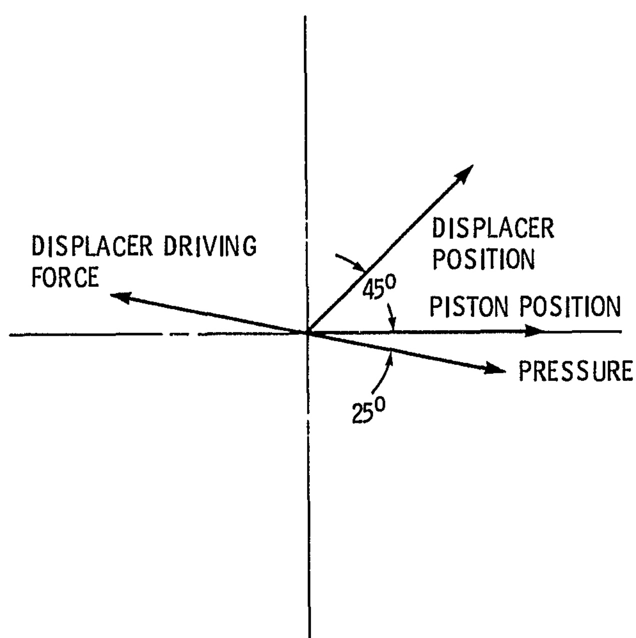

Figure 7. - Typical RE-1000 phasor diagram operating point.

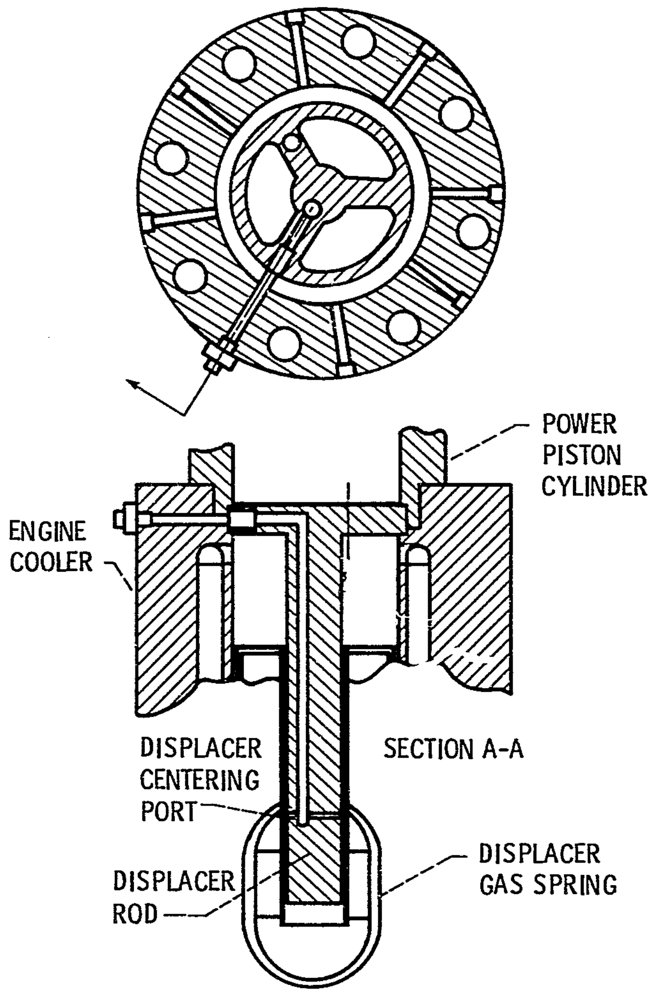

Figure 8. - Displacer rod mounting and communication ports for RE-1000 engine.

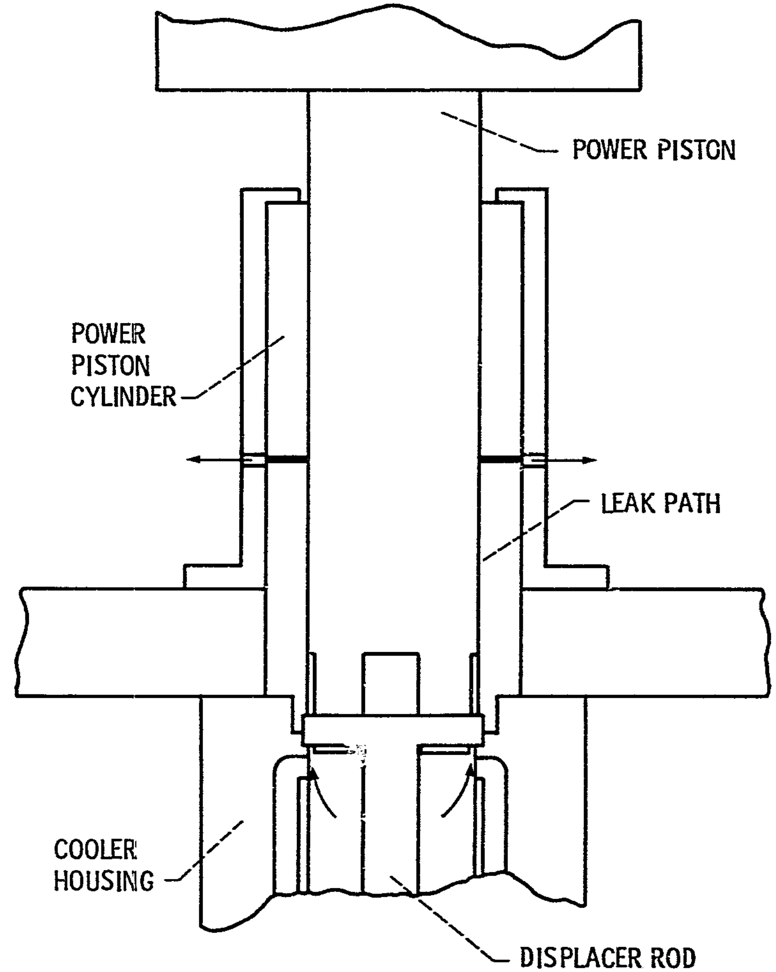

Figure 9. - Power piston leak path of RE-1000 engine.

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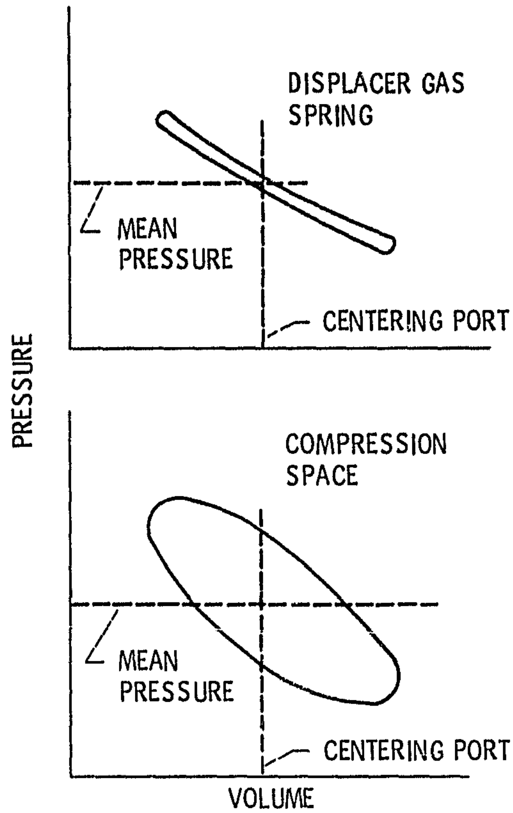

Figure 10. - RE-1000 engine centering port pressure drop for displacer gas spring and compression space.

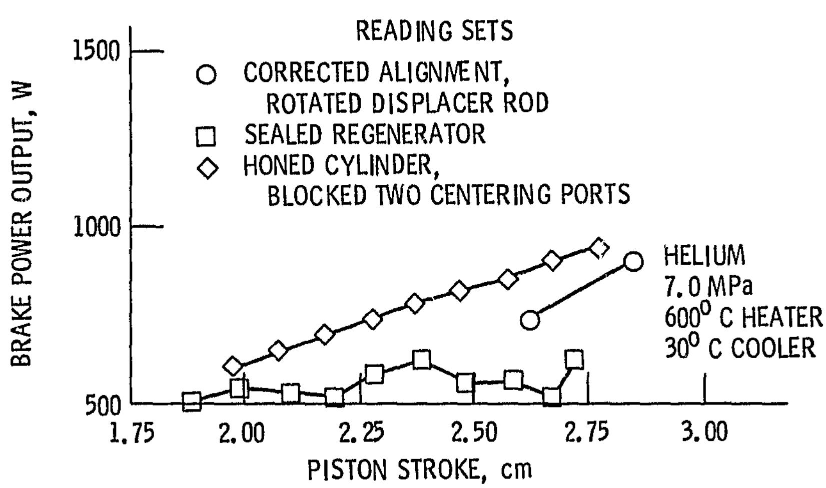

Figure 11. - Brake power vs piston stroke for RE-1000 engine.

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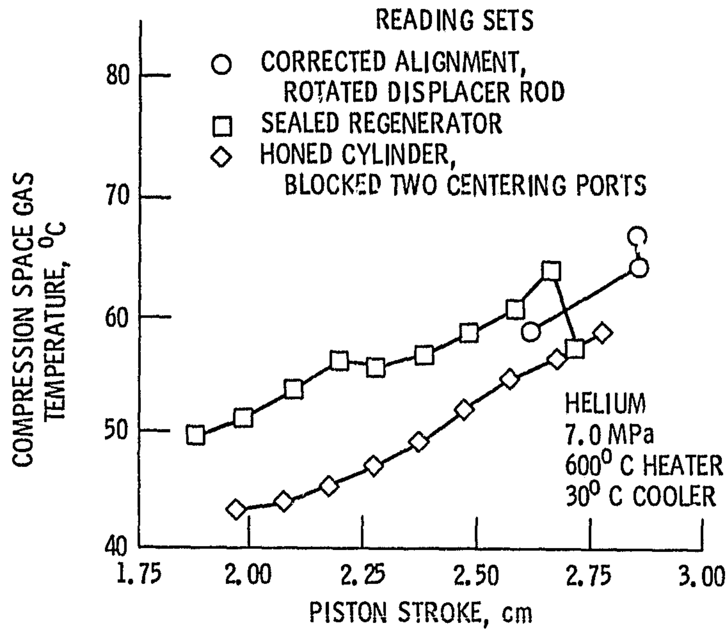

Figure 12. - Compression space gas temperature.

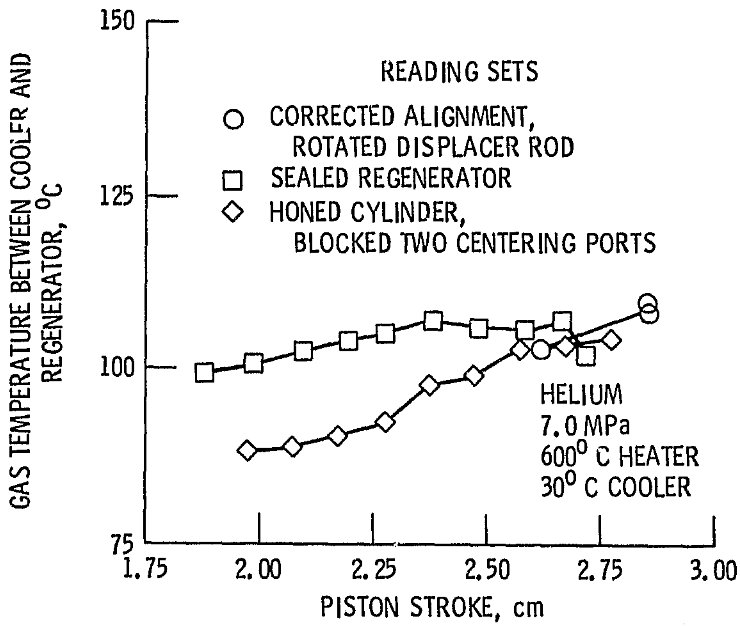

Figure 13. - Gas temperature between cooler and regenerator.

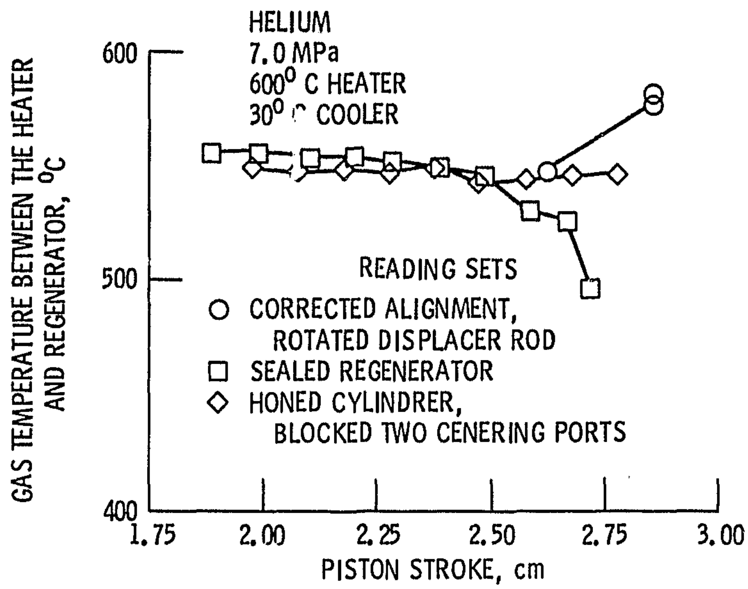

Figure 14. - Gas temperature between heater and regenerator.

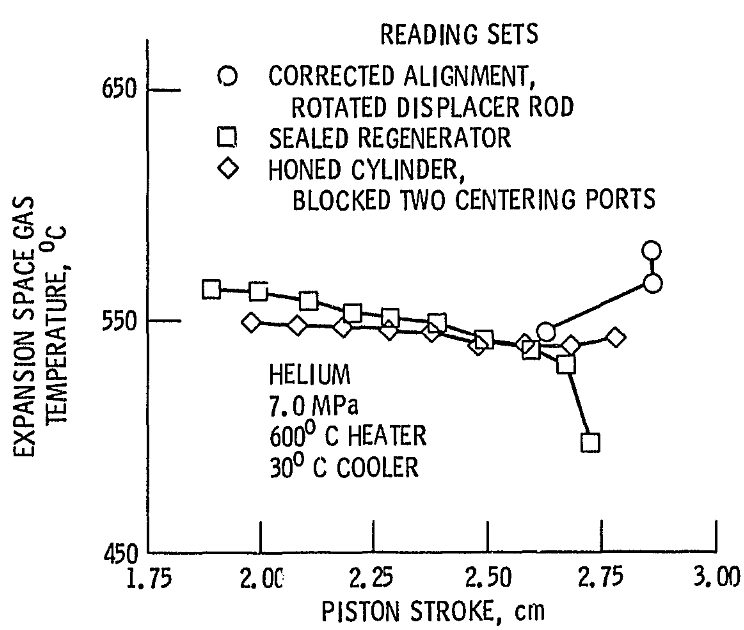

Figure 15. - Expansion space gas temperature.

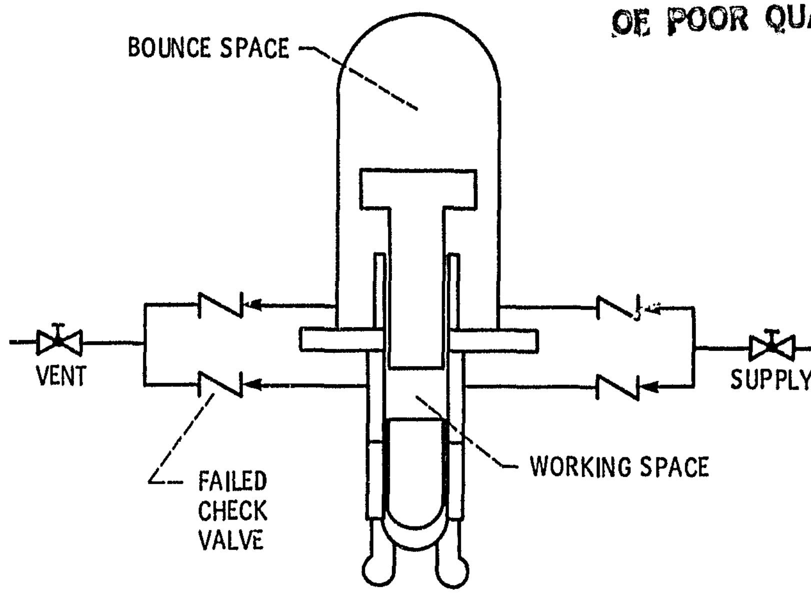

Figure 16. - Supply and vent check valve system.

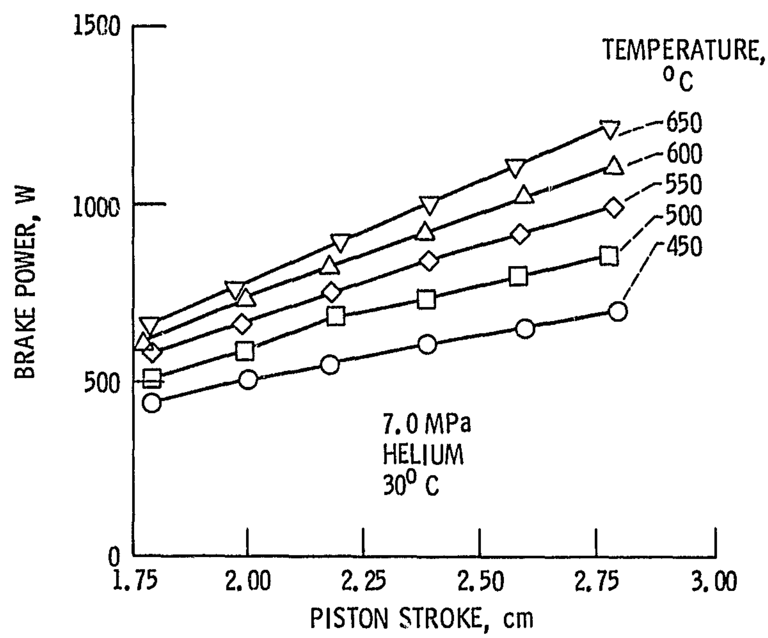

Figure 17. - Brake power output vs piston stroke.

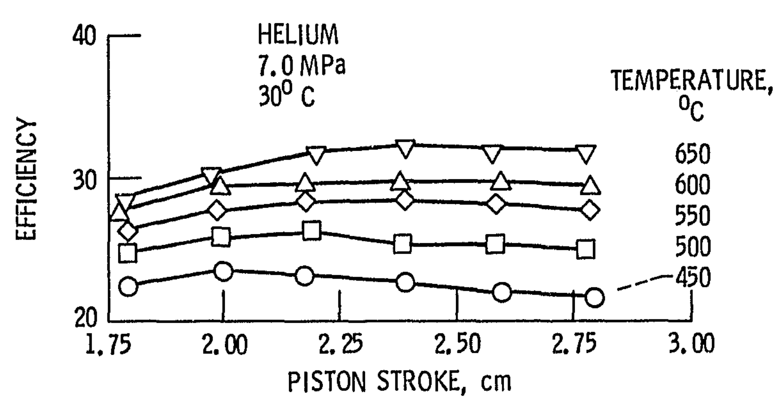

Figure 18. - Efficiency vs piston stroke.