E-7681

NASA Technical Memorandum 105883

Test Results of a Stirling Engine Utilizing Heat Exchanger Modules With an Integral Heat Pipe

Robert C. Skupinski, Leonard K. Tower, and Frank J. Madi
Sverdrup Technology, Inc.
Lewis Research Center Group
Brook Park, Ohio

and

Kevin D. Brusk
National Aeronautics and Space Administration
Lewis Research Center
Cleveland, Ohio

April 1993

NASA

Test Results of a Stirling Engine Utilizing Heat Exchanger Modules with an Integral Heat Pipe

Robert C. Skupinski, Leonard K. Tower, Frank J. Madi
Sverdrup Technology, Inc.
Lewis Research Center Group
Brookpark, Ohio 44142

Kevin D. Brusk
National Aeronautics and Space Administration
Lewis Research Center
Cleveland, Ohio 44142

ABSTRACT

The Heat Pipe Stirling Engine (HP-1000), a free-piston Stirling engine incorporating three heat exchanger modules, each having a sodium filled heat pipe, has been tested at the NASA-Lewis Research Center as part of the Civil Space Technology Initiative (CSTI).

The heat exchanger modules were designed to reduce the number of potential flow leak paths in the heat exchanger assembly and incorporate a heat pipe as the link between the heat source and the engine. An existing RE-1000 free-piston Stirling engine was modified to operate using the heat exchanger modules.

This paper describes heat exchanger module and engine performance during baseline testing. Condenser temperature profiles, brake power, and efficiency are presented and discussed.

INTRODUCTION

The Stirling space power project is part of the NASA High Capacity Power Program funded under the Civil Space Technology Initiative (CSTI). The objective of the CSTI High Capacity Power Program is to develop the technology base needed to meet the long duration, high capacity power requirements for future NASA space initiatives. The primary goal of the Stirling space power project, described in [1], is the development of a 12.5 kilowatt per cylinder Stirling converter with a lifetime of 60,000 hours.

The shell-and-tube heat exchangers used in many Stirling engines, including the first-generation space power converter, the Space Power Demonstrator Engine (SPDE), have a large number of joints. Reducing the number of joints reduces the number of potential leak paths in the converter which aids in meeting the lifetime requirements. A conceptual design of a converter optimized for space power was performed for NASA Lewis by Sunpower, Inc. and is described in [2]. This design included a novel heat exchanger module which allowed a significant reduction in the number of joints.

This Sunpower 25-kilowatt single-cylinder design used 40 heat exchanger modules. Each heat exchanger module incorporated a sodium heat pipe as the link between the heat source and the engine. The outer surface of each condenser was slotted; the working fluid was heated as it flowed through these slots. Each module also contained a regenerator and cooler. The cooler had slotted working fluid passages similar to the heater.

The fabrication of the next-generation Stirling space power converter is now being completed by Mechanical Technology, Inc. (MTI) for NASA Lewis. This 12.5 kilowatt converter is known as the Component Test Power Converter (CTPC). The heat exchanger module design concept was not selected for the CTPC. However, a variation of these slotted heat exchanger modules is being used for the CTPC cooler, shown in figure 1. The CTPC also incorporates a sodium filled heat pipe, although of a different design from those used in the heat exchanger module concept.

To demonstrate the feasibility of the heat exchanger module concept, an existing NASA Lewis RE-1000 free-piston Stirling engine was modified to operate with three of the modules. This modular heat exchanger was designed and fabricated for NASA Lewis by Sunpower; Thermacore, Inc. built the heat pipes for the modules. The engine test stand was designed so that the entire engine could be inverted, allowing testing of the heat pipes operating against gravity or in the gravity-assisted mode. This enabled examining the effect of the location of excess liquid sodium in the heat pipes.

This report briefly describes the design of the heat exchanger modules and performance of the engine during baseline testing. Condenser temperature profiles, brake power, and efficiency are presented for the heat pipes operating against gravity and in the gravity-assisted mode. References [3] and [4] provide a further description of the heat exchanger modules and early test results on the modified RE-1000 engine.

MODULAR HEAT EXCHANGER DESCRIPTION

Each module consists of a heat-pipe heater, regenerator, and cooler as shown in figure 2. The engine heater is actually the condenser end of a heat pipe. Each heat pipe was fabricated from a 300 series stainless steel, contains 25 to 30 grams of sodium, and has a sintered powder metal wick with two arteries. Slots are milled in the outer surface of the condenser to form fins. The condenser end is then inserted and brazed into a cylinder, forming the gas paths. Three holes were drilled for thermocouples through this outer cylinder and into a condenser fin to the same depth as the milled slots. The thermocouples are located at roughly the beginning, middle, and end of each condenser to measure its axial temperature distribution. The average of all nine condenser temperatures (three heaters with three thermocouples each) is defined as the heater temperature.

The regenerator resembles a metal spool around which a knitted stainless steel mesh is wrapped. This regenerator is inserted into the outer cylinder against the condenser end of the heat pipe.

The cooler is similar in construction to the heater, a finned cylinder pressed into an outer cylinder. Water circulates around the outer cylinder to remove heat.

The assembled heat exchanger modules are shown connected to the displacer cylinder in figure 3.

HEAT-PIPE STIRLING ENGINE DESCRIPTION

An existing RE-1000 free-piston Stirling engine was modified to operate with three of the heat exchanger modules. This RE-1000 engine was built by Sunpower, Inc. for NASA Lewis. The modified RE-1000 engine was re-named the HP-1000 and is shown in figure 4. Both engines were designed to produce approximately 1 kilowatt brake power at a mean working space pressure of 7 MPa and a heater and cooler temperature of 873 K and 298 K, respectively.

The heat-pipe evaporators are heated with individual, close fitting electrical resistance heating elements. Evaporator temperatures are measured by thermocouples located at roughly the beginning, middle, and end of each evaporator. Waste heat is removed from the engine by circulating water through the cooler. The engine's power is absorbed using a dashpot with a remote controlled adjustable orifice.

The engine test stand was designed so that the entire engine could be inverted, allowing for testing of the heat pipes operating against gravity or in the gravity-assisted mode. This enabled examining the effect of the location of excess sodium in the heat pipes. A passive vibration absorber, also built by Sunpower, Inc. and described in [5], was used to reduce vibration of the engine and test stand. The absorber was tuned to minimize vibrations at a frequency of 30 Hz. The engine, test stand, and vibration absorber are shown in figure 5; the engine is shown in the inverted position with the heat pipes operating in the gravity-assisted mode.

TEST OBJECTIVES

The two basic objectives for testing the HP-1000 were (1) to demonstrate the feasibility of the heat pipe heat exchanger modules as an alternative to typical shell-and-tube pumped loop heat exchangers and (2) operate the engine in both orientations and determine the effect of the location of excess liquid sodium in the heat pipes. With the heat pipes operating against gravity, the condensers are below the evaporators and excess sodium would collect at the end of the condenser. This is expected to reduce heat transfer at the end of the condenser which would be indicated by a lower temperature at this location.

TEST PROCEDURE

The engine was pressurized with helium to approximately 700 kPa absolute. The cooling water flow rate was set at about 4 liters-per-minute and maintained between 293 to 298 K. Next, the power to the heating elements was turned on. When the heater temperature reached approximately 473 K the engine was started. At this pressure, the engine frequency was approximately 10 Hz. The stroke was purposely set as small as possible (around 1 cm) to reduce vibration.

As the engine heated up, the pressure was gradually increased to raise the engine frequency to 30 Hz, still keeping the stroke as small as possible. This was done to minimize vibration until reaching the optimum 30 Hz frequency of the vibration absorber.

Once 30 Hz was achieved at the desired heater temperature, the stroke was gradually increased. Data was taken beginning with a stroke of 1.4 cm and at increments of 0.2 cm. This also required increasing the pressure to keep the engine frequency at 30 Hz, and increasing the power to the heating elements to keep the heater temperature constant. When a desired stroke was reached, data was taken every 5 to 10 minutes for at least 20 minutes. Previous tests determined that system efficiency stabilized after approximately 20 minutes. The stabilization period was necessary primarily due to the large thermal mass of the insulation and structure surrounding the heating elements.

Data was taken at increasingly larger strokes until the evaporator temperature on any one heat pipe approached 1023 K. This was regarded as the highest temperature at which the heat pipes could be run without greatly reducing operational lifetime. After the power to the heating elements was turned off, the engine was allowed to run until it stopped in order to cool itself.

RESULTS

The tests at a heater temperature of 823 K with the heat pipes in the gravity-assisted orientation required 3 runs to complete testing over the possible range of strokes due to hardware and instrumentation problems. This created a discontinuity in the condenser temperature data because as the engine is disassembled and rebuilt, it is likely the position of the heating elements with respect to the evaporators change. This may cause slightly higher or lower evaporator temperatures with similar changes in the condenser temperatures for the same power throughput. The test with the heat pipes oriented against gravity was repeated to confirm the temperature differences between the two orientations.

In general, the condenser temperature distribution is 10 to 30 K larger when the heat pipes are operating against gravity. The temperatures at the end of the condensers account for most of this difference, decreasing as the stroke (or power) is increased. This is probably due to recession of the meniscus in the evaporator wick with the increased difference between the vapor and liquid pressure required for capillary pumping. This expels some liquid from the evaporator wick into the excess liquid pool in the end of the condenser, further blocking heat transfer.

The heat pipes will be referred to as heat pipes #1, #2, and #3 for identification purposes. Figure 6 shows the condenser temperature distribution for heat pipe #3. The thermocouples are located at approximately the beginning, middle, and end of the condenser and are labeled T1, T2, and T3, respectively. The temperature at the end of the condenser in this heat pipe was very dependent on its orientation, decreasing by nearly 20 K when operating against gravity. A probable explanation for this is a considerable overfill of this particular pipe, which causes a pooling of sodium in the condenser with the evaporator up, thereby reducing heat transfer.

The condenser temperature distribution for heat pipe #2 (not shown) did not show as large a temperature difference between the two orientations. This suggests a more optimal sodium fill. Three extra thermocouples installed on this condenser, 180° opposite those used to measure the temperature distribution, indicated a circumferential nonuniformity of up to 8 K. This is possibly due to variations in the wick thickness around the pipe.

With regards to engine performance, the results indicate brake power as a function of stroke was independent of the heat pipe's orientation as can be seen in figure 7. However, other factors change with engine orientation which make direct comparisions of power difficult. For example, when the engine is oriented such that the heat pipes operate against gravity, the mean position of the piston is further from the compression space due to gravity. This results in a larger volume of gas in the engine.

When the heat pipes were oriented in the gravity-assisted mode, the engine produced approximately 1030 Watts at a stroke of 2.9 cm before reaching the evaporator's temperature limit, while the engine produced approximately 800 Watts at a stroke of about 2.4 cm before reaching the evaporator's temperature limit when operating against gravity. This may have been due to a structural problem in heat pipe #1. Although the condenser temperatures between both positions were similar, suggesting no large pool of excess sodium, the evaporator temperatures were much higher when operating against gravity. This may be due to partial dryout of the wick which leaves less evaporation area. Wick separation or possible underfill of this heat pipe could be the cause. The heat pipe was not cut open for examination because of possible future engine tests.

Brake thermal efficiency based on rejected heat was approximately 19% to 21% over the range of piston strokes for both orientations as shown in figure 8. At a stroke of 1.4 cm, the efficiency was noticably lower with the heat pipes oriented in the gravity-assisted mode. However, this appears to be due to a data acquisition error.

System efficiency, shown in figure 9 and defined as the percentage of brake power output to heater power input, varied from approximately 10% to 14% over the range of strokes. The low system efficiency compared to the brake thermal efficiency is primarily due to heat loss from the heating system. In spite of efforts to insulate around the heating elements, calculations and tests indicated that from 1 to 3 kilowatts was lost to the ambient through the insulation while running the engine.

The system efficiency does seem to change with engine orientation. Regardless of the stroke, the efficiency was lower with the heat pipes operating against gravity. This may be due to reduced heat transfer because of the apparent poor operation of heat pipe #1 and excess sodium in heat pipe #3. However, other factors change with engine orientation which make direct comparisons of efficiency difficult; for example, the heat transfer from the heating element assembly to the surroundings, and the position of the heating elements with respect to the evaporators. The heating elements are not fixed in place and can shift to accommodate movement due to thermal expansion.

The results at a heater temperature of 873 K had the same trends as the 823 K results, except for the system efficiency. At a heater temperature of 873 K, the differences in system efficiency between engine orientations were less with neither orientation having a distinct advantage. This may be due to a smaller liquid inventory in each heat pipe. However, at a heater temperature of 873 K, the evaporator temperature limit was reached at a piston stroke of 1.8 cm for both orientations; therefore, the comparisons to the 823 K results are based on a limited stroke range.

COMPARISONS TO RE-1000 RESULTS

Comparing brake power and brake thermal efficiency (based on rejected heat) between the HP-1000 and the RE-1000 on the basis of identical mean pressure, stroke, heater and cooler temperatures, shows similar results. For example, at a heater temperature of 823 K and a piston stroke of 2.8 cm, the HP-1000 produced 986 Watts while the RE-1000 produced 981 Watts. The brake thermal efficiency for the HP-1000 and RE-1000 was 24.0% and 24.8%, respectively.

Although direct comparisons between both engines are difficult because of the differences in their heat exchanger assemblies, the heat exchanger module did perform as expected, delivering similar power and efficiency as the RE-1000.

The brake thermal efficiency was calculated based on rejected heat since the means used to supply heat to both engines differed substantially. In the RE-1000 heater, gas flowed through electrically heated tubes made of Inconel 718. RE-1000 test results are documented in [6].

CONCLUSIONS

The feasibility of a modular heat exchanger design with substantially fewer critical joints than a typical shell-and-tube heat exchanger has been demonstrated. The heat exchanger modules in the HP-1000 performed as expected, delivering similar power and efficiency as the RE-1000. The use of a heat pipe to transfer heat from the heat source to the engine was also sucessfully demonstrated.

It may be possible to improve the performance of the heat pipe heater by reducing the amount of excess sodium. This could be done by determining a method for a more optimal fill, or by providing a well for excess sodium which would be located so as not to interfere with heat transfer.

REFERENCES

1. J.E. Dudenhoefer and J.M. Winter, "Status of NASA's Stirling Space Power Converter Program," NASA TM-104512, 1991.

2. L.B. Penswick, "1050 K Stirling Space Engine Design," NASA CR-182149, 1988.

3. J.G. Schreiber, "The Design and Fabrication of a Stirling Engine Heat Exchanger Module With an Integral Heat Pipe," NASA TM-101296, 1988.

4. J.G. Schreiber, "Initial Characterization of a Modu lar Heat Exchanger With an Integral Heat Pipe," NASA TM-102097, 1989.

5. L.B. Penswick, W.T. Beale, and J.G. Wood, "Free-Pis ton Stirling Engine Conceptual Design and Technologies for Space Power, Phase 1-Final Report," NASA CR-182168, 1990.

6. J.G. Schreiber, S.M. Geng, and G.V. Lorenz, "RE-1000 Free-Piston Stirling Engine Sensitivity Test Results," NASA TM-88846, DOE/NASA/1005-11, 1986.

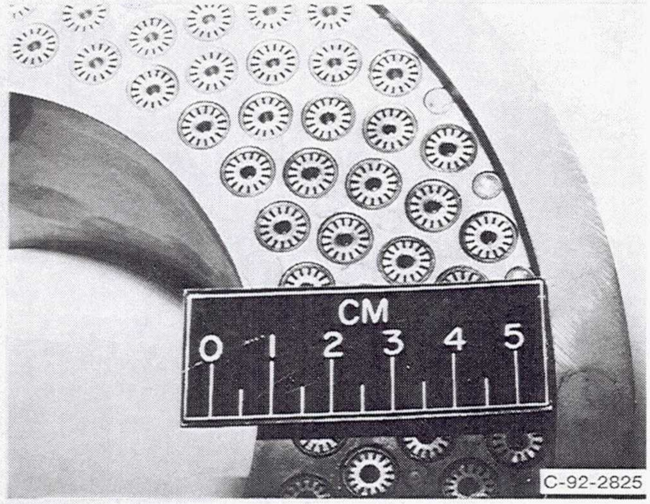

Figure 1.—Component test power converter cooler.

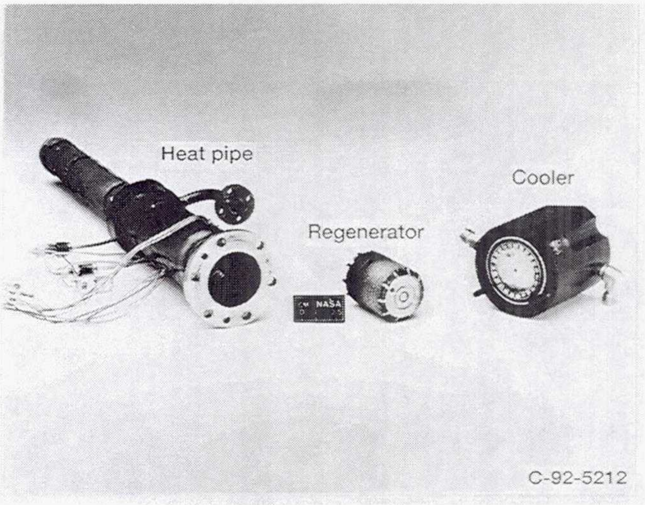

Figure 2.—Heat exchanger module components.

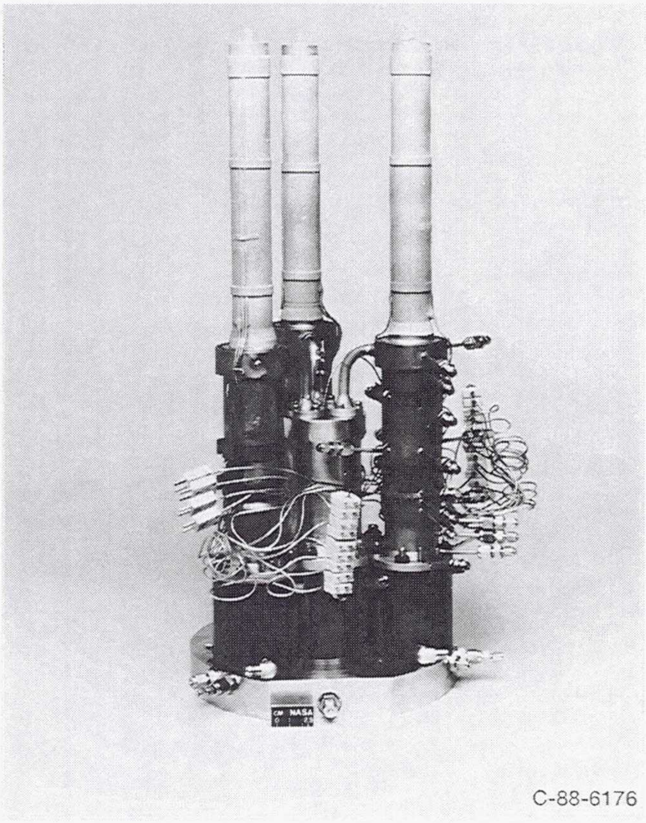

Figure 3.—Assembled heat exchanger modules and displacer cylinder.

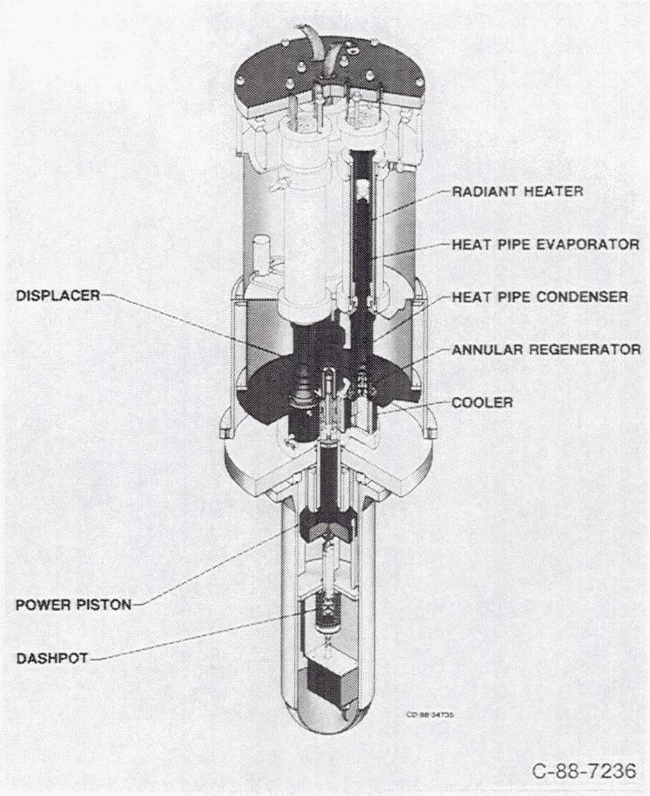

Figure 4.—Cut-away of HP-1000.

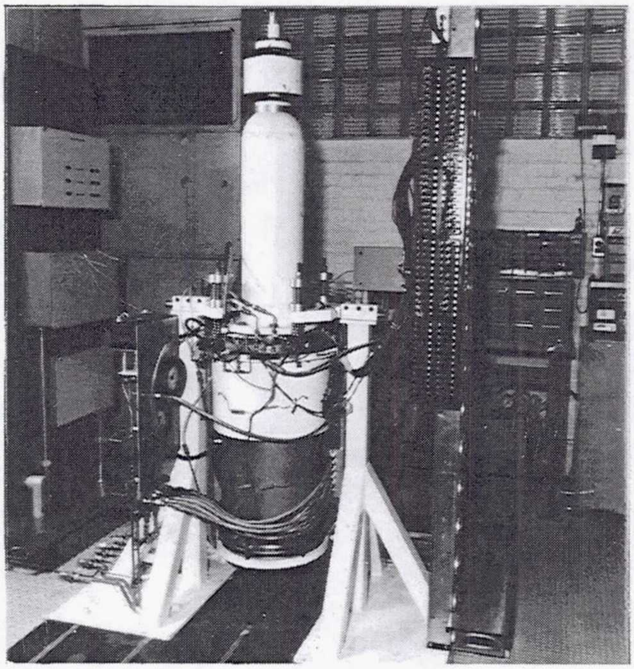

Figure 5.—HP-1000 orientated with heat pipes in gravity-assisted mode.

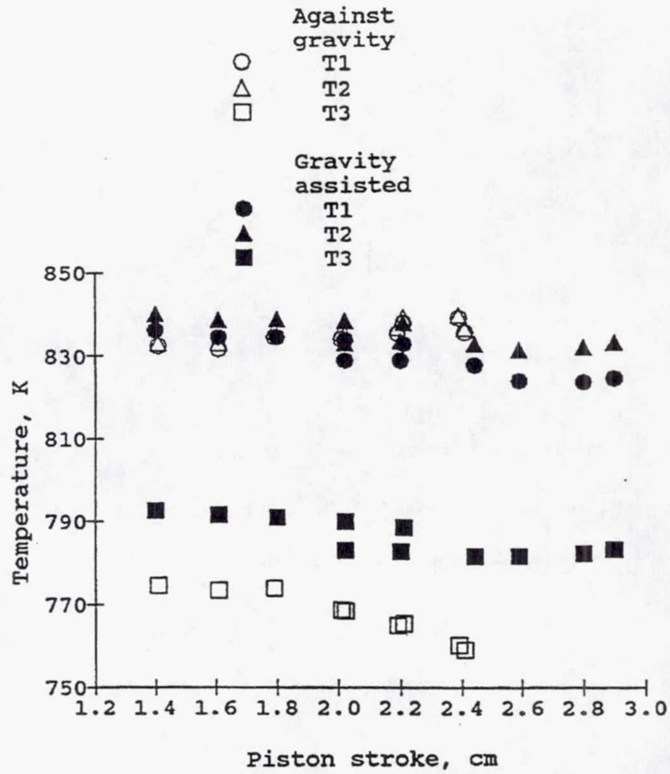

Figure 6.—Condenser temperature distribution versus piston stroke at heater temperature of 823 K.

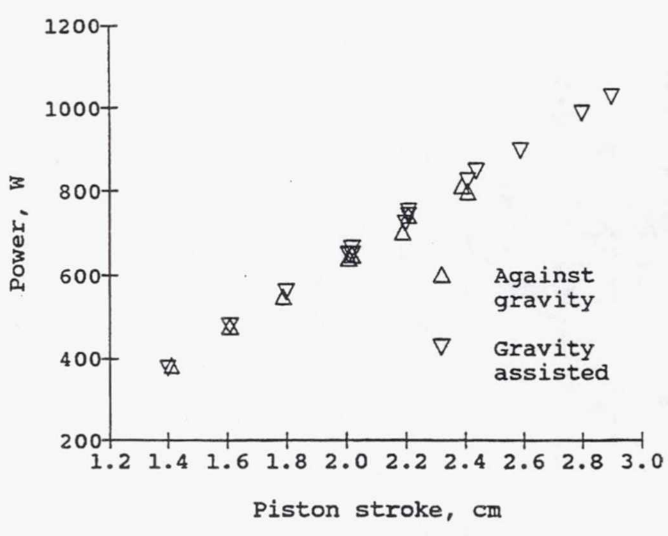

Figure 7.—Engine brake power versus piston stroke at heater temperature of 823 K.

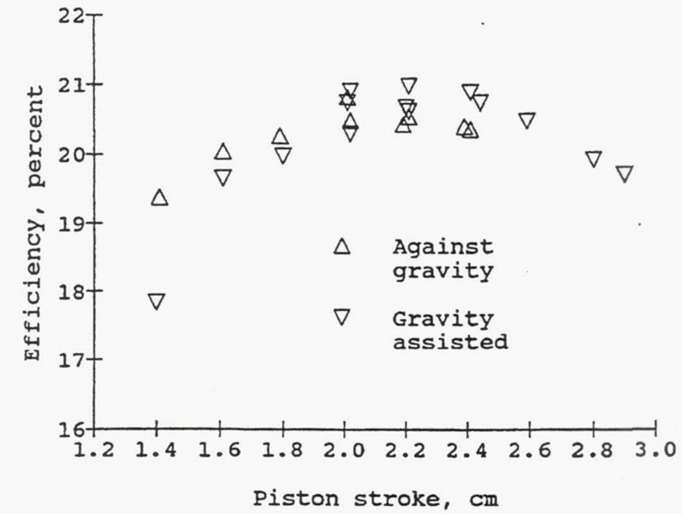

Figure 8.—Brake thermal efficiency versus piston stroke at heater temperature of 823 K.

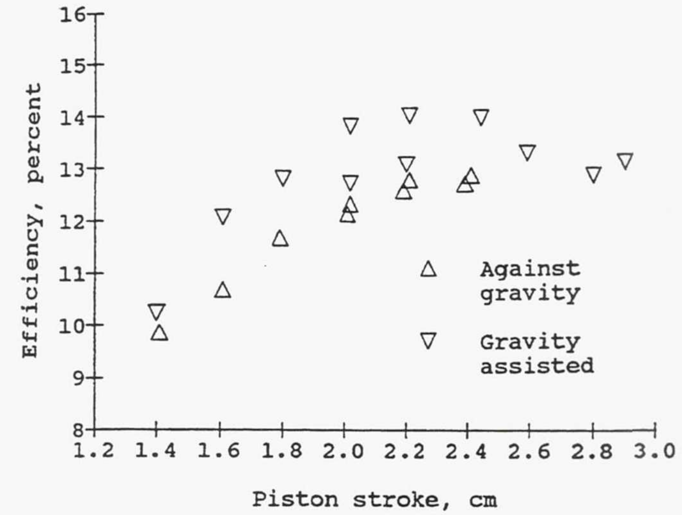

Figure 9.—System efficiency versus piston stroke at heater temperature of 823 K.

REPORT DOCUMENTATION PAGE

Form Approved
OMB No. 0704-0188

1. AGENCY USE ONLY *(Leave blank)*

2. REPORT DATE
April 1993

3. REPORT TYPE AND DATES COVERED
Technical Memorandum

4. TITLE AND SUBTITLE
Test Results of a Stirling Engine Utilizing Heat Exchanger Modules With an Integral Heat Pipe

5. FUNDING NUMBERS
WU–590–13–11

6. AUTHOR(S)
Robert C. Skupinski, Leonard K. Tower, Frank J. Madi, and Kevin D. Brusk

7. PERFORMING ORGANIZATION NAME(S) AND ADDRESS(ES)
National Aeronautics and Space Administration
Lewis Research Center
Cleveland, Ohio 44135–3191

8. PERFORMING ORGANIZATION REPORT NUMBER
E–7681

9. SPONSORING/MONITORING AGENCY NAMES(S) AND ADDRESS(ES)
National Aeronautics and Space Administration
Washington, D.C. 20546–0001

10. SPONSORING/MONITORING AGENCY REPORT NUMBER
NASA TM–105883

11. SUPPLEMENTARY NOTES
Robert C. Skupinski, Leonard K. Tower, Frank J. Madi, Sverdrup Technology, Inc., Lewis Research Center Group, 2001 Aerospace Parkway, Brook Park, Ohio 44142 and Kevin D. Brusk, Lewis Research Center. Responsible person, Robert C. Skupinski, (216) 891–2260.

12a. DISTRIBUTION/AVAILABILITY STATEMENT
Unclassified - Unlimited
Subject Category 44

12b. DISTRIBUTION CODE

13. ABSTRACT ***(Maximum 200 words)***

The Heat Pipe Stirling Engine (HP–1000), a free-piston Stirling engine incorporating three heat exchanger modules, each having a sodium filled heat pipe, has been tested at the NASA-Lewis Research Center as part of the Civil Space Technology Initiative (CSTI). The heat exchanger modules were designed to reduce the number of potential flow leak paths in the heat exchanger assembly and incorporate a heat pipe as the link between the heat source and the engine. An existing RE–1000 free-piston Stirling engine was modified to operate using the heat exchanger modules. This paper describes heat exchanger module and engine performance during baseline testing. Condenser temperature profiles, brake power, and efficiency are presented and discussed.

14. SUBJECT TERMS
Stirling engines; Free-piston engines; Heat pipes; Heat exchangers

15. NUMBER OF PAGES
12

16. PRICE CODE
A03

17. SECURITY CLASSIFICATION OF REPORT	18. SECURITY CLASSIFICATION OF THIS PAGE	19. SECURITY CLASSIFICATION OF ABSTRACT	20. LIMITATION OF ABSTRACT
Unclassified	Unclassified	Unclassified	