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ABSTRACT

The development and application of Stirling engines for space power production requires concomitant development of an advanced heat rejection system. We are currently involved in the design, development, and testing of advanced ceramic fabric (ACF) water heat pipes for optimal heat rejection from the Stirling cycle without the use of hazardous working fluids such as mercury. Our testing to-date has been with a 200-μm thick titanium heat pipe utilizing Nextel™ fabric as both the outer structural component and as a wick. This heat pipe has been successfully started up from a frozen condition against a negative 4 degree tilt (i.e., fluid return to evaporator was against gravity), with 75 W heat input, in ambient air. In a horizontal orientation, up to 100 W heat input was tolerated without experiencing dryout.

Introduction

Some devices such as Stirling engines are most effective in a space application with heat rejection temperatures in the 400 - 600 K range. The number of working fluids in this temperature regime is limited and they have many drawbacks; mercury is perhaps best, with some organics possible candidates. Water has excellent properties, especially in the 400 - 500 K range, but is usually not considered because of high vapor pressure, necessitating a massive radiator structure if constructed in metals. ACF properties permit use of water in a heat pipe space radiator without exacting a severe weight penalty.

The purpose of the fabric heat pipe work discussed here is twofold: (1) to construct and test several non-optimized heat pipes with a water working fluid, and (2) to design more advanced versions of fabric heat pipes by considering the entire radiator subsystem as it might integrate with a Stirling engine. The first item aims at a proof-of-concept, and will demonstrate our current knowledge of fabric radiator components. The water heat pipes being built at Pacific Northwest Laboratory (PNL) employ ceramic fabrics in the structural and wicking components. The second item will incorporate the knowledge gained from fabrication and testing to generate realistic radiator designs that will be constructed and tested during later phases of this project.

The approach taken in this study is to design and test simple, easily-constructed and modified devices. The intent of these activities is to proceed with design, construction and testing of ACF radiator components, and to begin planning and scheduling of activities for a sub-assembly demonstration test. This study utilizes data obtained from component-level tests that have been performed and are described in earlier publications,[1,2,3] and is a continuation of earlier testing of two prototype water heat pipes[1,2].

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Startup Tests from Frozen Condition

Before beginning the startup tests, the existing titanium heat pipe, which has specifications identical to those used in our earlier tests[1] (see Table 1 and Figure 1), was fully inspected. A borescope examination of the internals showed no visible deterioration, even though this heat pipe had been operated for a total of about 200 hours. The wicking behavior of the various readily-available Nextel™ fabrics was then determined. This was accomplished in the standard manner[4,5] by placing one end of each sample in contact with a container of deionized water and recording the vertical, capillary rise of the water in the material as a function of time.

Table 1. Test Heat Pipe Parameters

A non-optimized ACF heat pipe was built using currently-available materials. It was constructed as follows:

• Outer liner	430-μm (17-mil) thick biaxially-braided Nextel fabric
• Metal liner	200-μm (8-mil) titanium alloy, 16-mm (5/8-in) OD x 0.9-m (36-in) long
• Wick	ca. 250-μm (10-mil) Nextel tape, held in place with 100-mesh stainless steel screen
• End fittings	standard Swagelok™
• Working fluid	water

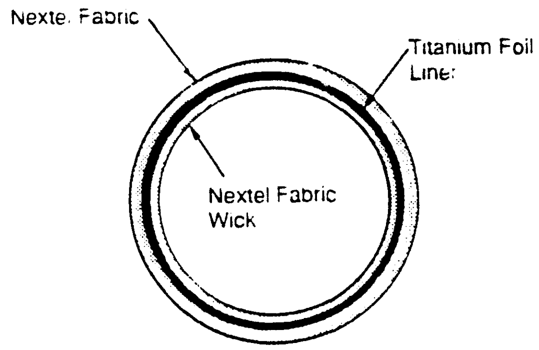

Figure 1. ACF Heat Pipe Cross-Section.

Based on these wicking tests, it was concluded that the variation in geometry among the four samples (two braided tubes; one braided artery; one woven tape) had a fairly minor effect on wicking. The permeability and effective pumping radius correspond approximately to a 200 - 250 mesh screen.[4] The 3-mm tube, prototypical of an arterial configuration, wicked nearly as well as the other samples while pumping much more liquid. This suggests that our next test/prototype heat pipes should incorporate arteries, and take advantage of liner corrugation (from internal pressure) for distributive wicking.[5]

Heat pipe startup from a cold condition is of great interest for space applications. The heat pipe working fluid could be in a frozen state prior to startup. Such a situation could occur after initial spacecraft launch, or following planned or unplanned variations in the heat rejection rate. Typical liquid metal heat pipes are started from frozen even at room temperature; however, the behavior of a liquid metal working fluid during startup differs significantly from water,[6] which is usually capillary flow limited.[7] Few startup tests of water heat pipes have been reported in the literature.[6,7] Previous tests showed that successful startup of a frozen heat pipe is greatly dependent on the boundary conditions at the outer surface of the evaporator and the condenser[6]. Heat input rate was generally limited to prevent evaporator dryout. Heat rejection was likewise restricted by various means so that the condenser temperature could rise.

In our tests, we have not had to restrict startup conditions, and have performed the tests in essentially the same manner as earlier tests done at ambient with the water in a liquid form. Full power was always applied at the beginning of the startup and maintained throughout the test. The only difference between frozen and ambient startups was in restricting the heat influx from the ambient environment for a frozen startup condition. This was accomplished by insulating the entire heat pipe, and removal of the insulation over the condenser only when heat pipe temperatures approached ambient.

For maximum realism, the startup test matrix consisted of startup from frozen with a cold (adiabatic) boundary, in various orientations. Different orientations during freezing (-4°, 0°, +4°) and during thaw/startup (-4°, 0°)[1] were tested. The worst-case scenario was attained by storing the heat pipe at negative tilt, to allow maximal draining of the water into the condenser section, followed by a negative-tilt startup. Typical freeze conditions were attained by overnight storage of a fully-insulated heat pipe in a freezer held at 257 K. The testing involved removal of the heat pipe from the freezer and connection of power and instrumentation leads. This process required less than 5 minutes. The insulation on the condenser was kept in place until the condenser temperature began to approach ambient (about 15 minutes after turning on power to the heater tape). At that time this insulation was removed and the condenser was free to convect/radiate heat to ambient. The test matrix described in Table 2 was executed. It included variations in the freeze/startup orientations and the heat input rate.

Table 2. Test Matrix for Nextel/Ti/Nextel/100 mesh SS Heat Pipe Startup from Frozen.

Test No.	Orientation Degrees[1] Freeze	Orientation Degrees[1] Startup	Heat Input Rate, W	Results
1	0	0	50	Successful
2	0	-4	75	Successful
3	0	0	100	Successful
4	-4	-4	75	Successful
5	-4	-4	100	Dryout
6	+4	-4	100/75	Dryout/ successful

[1] 0 = horizontal; negative angles represent condenser lower than evaporator

Startup Test Results

The test heat pipe was instrumented with thermocouples mounted to the exterior fabric; a pressure transducer; and a power (heat input) wattmeter (see Figure 2). A microcomputer-based data acquisition system sampled all data every 20 seconds during a test.

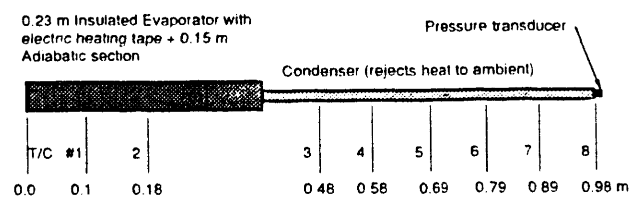

Figure 2. Instrumentation Arrangement on Test Water Heat Pipel

A typical test is represented by test #3, as illustrated on Figure 3. For clarity, only some of the thermocouples are shown on Figure 3. This test was at high power (100 W) with the heat pipe oriented horizontal during both freeze and startup. The transient temperature profiles are

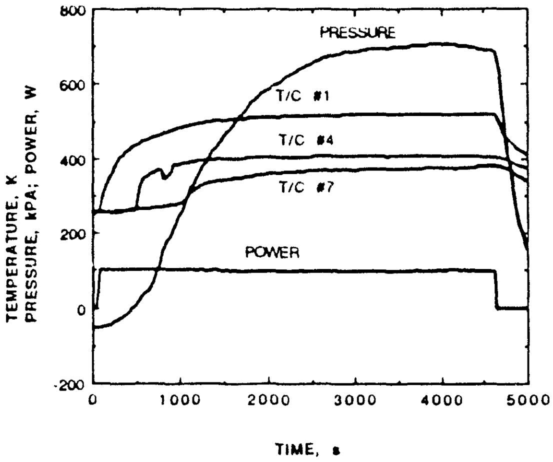

Figure 3. Temperature, Pressure, and Power Profiles During Test #3.

very similar to those measured during startups from ambient[2], although profile development was somewhat retarded by the additional phase change required from the frozen state. The condenser section is nearly isothermal over its working length, as evidenced by thermocouples #4 and #7 (thermocouple #8 was mounted on the pressure transducer at the very tip of the condenser, and its temperature was always significantly lower because of the fin effect of the transducer, plus liquid pooling during tests at negative tilt).

The actual working fluid temperature can be estimated indirectly. Pressure transducer readings can be used to determine water saturation temperature; this has been found to be on the order of 10 K higher than the temperature measured by thermocouple #4. Unfortunately, we experienced erratic behavior by the pressure transducer temperature compensation circuitry, so the pressure traces illustrated have a margin of error estimated to be about 50 kPa (equivalent to about 4 K in saturation temperature). During other heat pipe tests we have installed thin foil thermocouples under the fabric at the same locations as the existing, outer thermocouples. At the condenser, the temperature difference has typically been measured to be about 3 K, implying that the major temperature drop is across the titanium liner and wick. We have also inserted extremely fine (25-μm) thermocouples into the fabric, and these have indicated virtually identical temperatures as the surface thermocouples. This data, combined with a sensitive infrared camera, has been used to determine the fabric surface emittance, which is estimated to be about 0.93 at a 475 K surface temperature.

The negative tilt mode was investigated at lower power (75 W) in test #4, as illustrated on Figure 4. Both pressure and temperatures are lower than during test #3, and the difference in temperatures between thermocouples #4 and #7 is greater, indicating that the capillary flow limit is being approached.

High power at negative tilt was applied in test #6 (see Figure 5). The temperature at the evaporator (thermocouple #1) and internal pressure climbed rapidly to levels indicative of wick dryout, so power was reduced to 75 W. This caused the temperatures and pressure to moderate and ultimately hold at levels consistent with test #4, with one exception. Thermocouple #1 continued to indicate a high (about 550 K) temperature, about 50 K higher than measured in test #4. However, thermocouple #2 that is also on the evaporator but 75 mm closer to the condenser measured a temperature of about 500 K (not illustrated; usually thermocouple #1 and 2 readings were within a few degrees of each other). Thus it appears that the capillary limit was being exceeded in the evaporator, with local dryout at thermocouple #1. Future tests will encompass conditions with an additional Nextel wick layer in the evaporator to decrease the dryout potential.

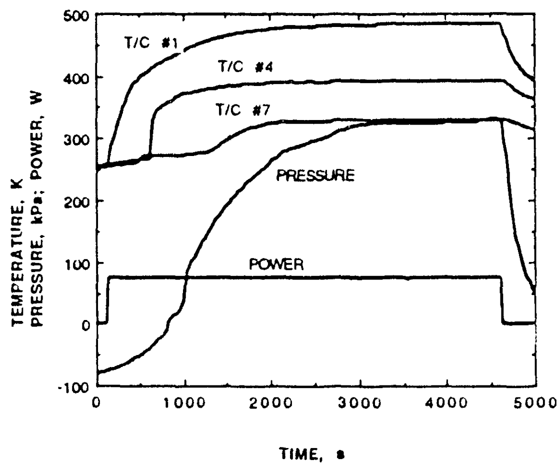

Figure 4. Temperature, Pressure, and Power Profiles During Test #4.

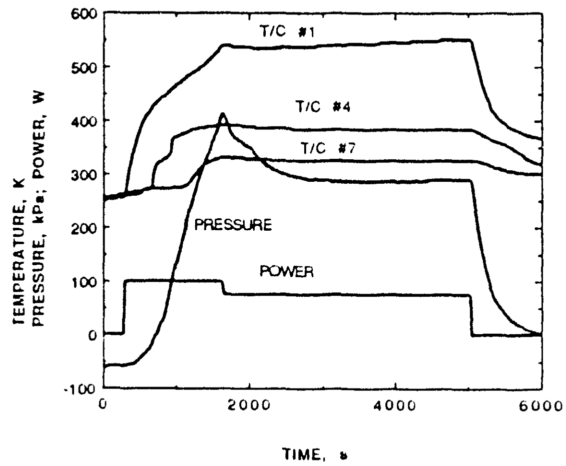

Figure 5. Temperature, Pressure, and Power Profiles During Test #6.

Conclusions

Good progress has been made in the development and testing of a fabric/foil water heat pipe. Preliminary heat pipe and wick tests indicate that the goal of a survivable, low-mass (2 to 3 kg/m^2) heat pipe space radiator is achievable with this technology. Issues raised in the past regarding startup problems of water heat pipes from a frozen condition have not surfaced during our test series. This test series was meant to explore the entire matrix of conditions that might be encountered in a frozen startup. Nevertheless, it would be well to perform one more frozen startup test, with a radiative thermal boundary condition truly representative of a space environment.

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