Status of the Development of Low Cost Radiator for Surface Fission Power - II

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Abstract. NASA Glenn Research Center (GRC) is developing fission power system technology for future Lunar and Martian surface power applications. The systems are envisioned in the 10 to 100kWe range and have an anticipated design life of 8 to 15 years with no maintenance. NASA GRC is currently setting up a 55 kWe non-nuclear system ground test in thermal-vacuum to validate technologies required to transfer reactor heat, convert the heat into electricity, reject waste heat, process the electrical output, and demonstrate overall system performance. The paper reports on the development of the heat pipe radiator to reject the waste heat from the Stirling convertors. Reducing the radiator mass, size, and cost is essential to the success of the program. To meet these goals, Advanced Cooling Technologies, Inc. (ACT) and Vanguard Space Technologies, Inc. (VST) are developing a single facesheet radiator with heat pipes directly bonded to the facesheet. The facesheet material is a graphite fiber reinforced composite (GFRC) and the heat pipes are titanium/water Variable Conductance Heat Pipes (VCHPs). By directly bonding a single facesheet to the heat pipes, several heavy and expensive components can be eliminated from the traditional radiator design such as, POCO™ foam saddles, aluminum honeycomb, and a second facesheet. As mentioned in previous papers by the authors, the final design of the waste heat radiator is described as being modular with independent GFRC panels for each heat pipe. The present paper reports on test results for a single radiator module as well as a radiator cluster consisting of eight integral modules. These tests were carried out in both ambient and vacuum conditions. While the vacuum testing of the single radiator module was performed in the ACT's vacuum chamber, the vacuum testing of the eight heat pipe radiator cluster took place in NASA GRC's vacuum chamber to accommodate the larger size of the cluster. The results for both articles show good agreement with the predictions and are presented in the paper.

Keywords: Heat Pipe Radiator, Surface Fission Power, Low Cost Radiator, Single Face Sheet Radiator

I. INTRODUCTION

NASA Glenn Research Center (GRC) is developing fission power system technology for future Lunar surface power applications. The systems are envisioned in the 10 to 100kW$_e$ range and have an anticipated design life of 8 to 15 years with no maintenance. A nominal lunar fission surface power design has been developed and is shown in FIGURE 1 [2]. The nuclear reactor supplies thermal energy to Brayton (or Stirling) convertors to produce electricity, and uses a heat pipe radiator to reject the waste heat generated by the convertors. The radiator panels must reject heat from both sides to achieve the highest efficiency; therefore, the optimum mounting position is vertical. The radiator panels contain embedded heat pipes to improve thermal transfer efficiency. Since the heat pipe evaporator is on the bottom, the heat pipes are gravity aided and can work as a thermosyphon. This is advantageous because the heat pipe is not required to pump the working fluid back to the evaporator against gravity. Heat is supplied to the heat pipes through a titanium/water heat exchanger that is coupled with the coolant loop in the radiator.

Currently, NASA GRC is putting together a Fission Power System Technology Demonstration Unit (TDU) [1, 3]. The TDU is a non-nuclear demonstration unit that will be tested in thermal-vacuum to demonstrate integrated system performance. The primary goals for the early systems are low cost, high reliability and long life. To help achieve these goals, ACT, NASA GRC, and VST are developing a single facesheet direct-bond radiator; see FIGURE 2. The radiator will have VCHPs made from titanium and will use water as the working fluid.

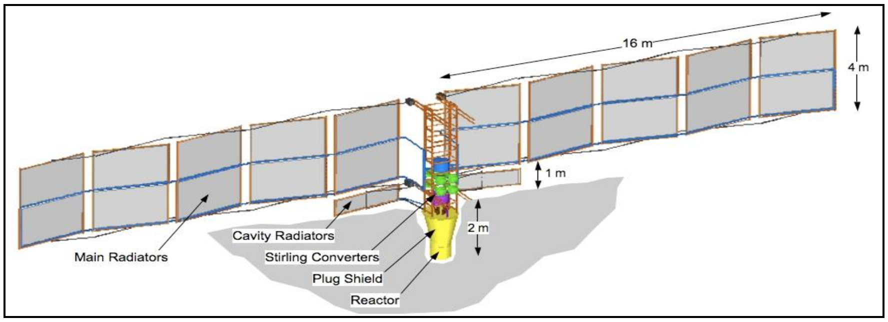


FIGURE 1. Fission surface power system concept [5]

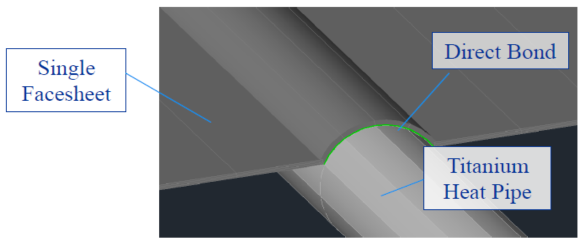


FIGURE 2. Single-facesheet radiator with direct bonding of the facesheet to the heat pipes [4]

A. Background

The single, direct-bond facesheet radiator has the advantages of reducing mass and cost of the system by eliminating the graphite foam saddles, aluminum honeycomb, and one of the graphite fiber reinforced composite (GFRC) facesheets, which are present in the previous ACT/VST heat pipe radiators [6].

TABLE 1. Summary of final radiator design.

Geometry	**0.75" Cond. OD Design**
Evaporator Length (cm)	13
Adiabatic Section Length (cm)	7.62
Condenser Length (cm)	170
NCG Reservoir Length (cm)	7.62
Fin Width Overhang (cm)	12
Total GFRC Area (m^2)	42.36
Total Number of Heat Pipe Modules	96
Total Number of Heat Pipe Clusters	12
Heat Pipe Redundancy Compared to Nominal Radiator (i.e. 36kW, 175K Sink, 400K inlet)	23
% Margin by Area Compared to Nominal Radiator	24
Thermal Performance & Mass	
Total Power Output (kW)	40
Specific Power (W/kg)	609.0
Dry Mass of Single Heat Pipe/Fin Module (kg)	0.685
Total Dry Mass of Radiator System (kg)	65.74

ACT and VST have previously demonstrated the feasibility of the single facesheet radiator by fabricating and testing a small-scale, two heat pipe radiator panel [5]. In a second paper with respect to this topic [4], the status of the Low Cost Radiator Development was presented where the final design was shown. It was decided that a 0.75 in. (1.91 cm) heat pipe condenser O.D. was more suitable for TDU integration, since it lowers the risk of composite fiber breakage during facesheet bonding.The larger diameter pipes also allow for more heat transfer area between the pipes and facesheet, thus lowering the thermal resistance and reducing the necessary number of heat pipes (radiator modules). TABLE 1 summarizes the mass and thermal performance of the final radiator design.

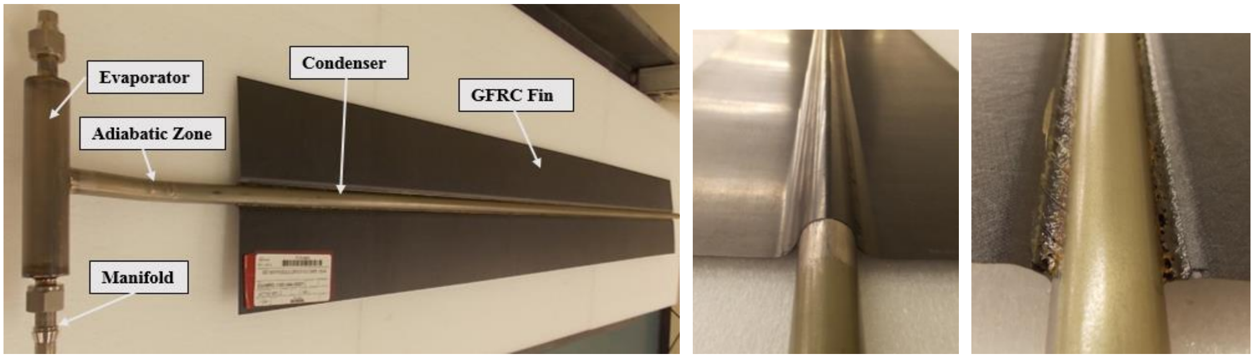
In addition, the face-sheet/heat pipe interface was also optimized. The wrapping angle of the GFRC around the 0.75 in. (1.91 cm) heat pipe condenser was investigated systematically. Based on thermal performance, a wrapping angle of 151° was chosen. This wrapping angle was the largest among the three wrapping angles that were investigated. The current paper presents a continuation of this development by showing testing results of both heat pipe radiator module and first heat pipe radiator cluster.

II. HEAT PIPE RADIATOR MODULE TESTING

The development of the Heat Pipe Radiator Module consisted of the following: designing the module and associated heat pipes, developing of the direct bond region characterized by the wrapping angle, testing of the heat pipe, developing the testing setup (cold wall) and testing the module. While most of the items mentioned above were discussed and presented in previous papers [4,5], module testing setup and results are presented in this paper.

A. Heat Pipe Radiator Module

The purpose of the full-scale module development is to validate the heat transport and heat rejection capability of the entire radiator and the ability to start-up from a frozen state. The heat pipe outer surface was primed with BR-127 prior to direct bonding to the GFRC fin. Vanguard Space Technologies, Inc. (VST) fabricated the tool required to make the 0.2 in. (0.508 cm) depth radiator (corresponding to the 151° wrapping angle previously mentioned and described in the previous paper). The fabricated heat pipe radiator module is shown in FIGURE 3a and direct bonding details for both sides of the module are shown in FIGURE 3b and c.



a) b) c)

FIGURE 3. a) Fabricated radiator module ready to be tested. Direct bond of GFRC to titanium heat pipe b) Top side and c) Bottom side.

B. Test Setup

FIGURE 4a shows the testing setup with the module installed inside. The entire assembly is placed in ACT's thermal vacuum chamber for testing as seen in FIGURE 4b. The water line is connected to the manifold and the evaporator is heated by circulating hot water through the manifold, similar to the system will operate in NASA'S TDU.. Thermocouples are placed along the entire length of the heat pipe to measure thermal conductivity of the adhesive bond. RTDs are installed before and after the manifold for water temperature measurements. A digital flow meter is also installed to provide the necessary information for calorimetry calculations to evaluate the power rejected by the radiator module for a given water inlet temperature.

a)

b)

FIGURE 4. Fixture for thermal vacuum testing of the radiator module. The evaporator section of the sandwiched module can be seen near the bottom and the reservoir and the fill tube can be seen at the top of the condenser section b) Testing setup and the radiator module assembly as installed in ACT's vacuum chamber for testing.

C. Testing of the Heat Pipe Radiator Module

VST fabricated the single radiator module with an emphasis on bond development and improving the contact between the GFRC and titanium heat pipes. Thermal lap shear tests were conducted on coupons alongside module fabrication to evaluate bond strength. ACT performed thermal performance testing in both vacuum and ambient conditionsto evaluate the heat rejection capability and integrity of the adhesive bond. Thermal cycle testing was also conducted to determine if the direct bond experiences any degradation caused by CTE induced stresses. Thermal performance testing was repeated to detect and evaluate potential damages or changes to the direct bond joints. After extensive testing, the original evaporator (5.5in (14 cm) long) was cut and replaced with a 7in. (17.8 cm) one to determine the effect of evaporator length on overall thermal performance.

1. Heat Pipe Radiator Module Testing in Vacuum

The radiator module was tested in vacuum in a gravity aided orientation. However, because of the limited size of the vacuum chamber at ACT (92 cm in diameter), the module was inclined at a ~ 45º angle. Three parameters were varied during testing: sink (cold wall) temperature, water inlet temperature and water flow rate.

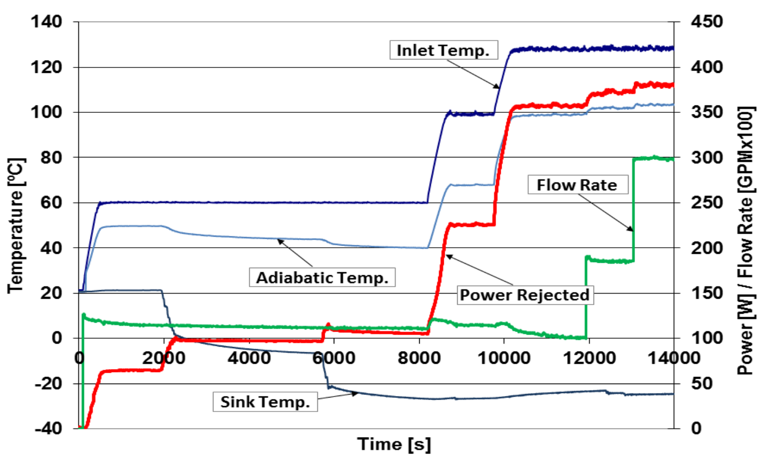


FIGURE 5. Power test before thermo-cycling test.

During the *first testing sequence*, all parameters mentioned above were varied starting with unfavorable values. For instance, the test began with a higher than nominal sink temperature (ambient), a lower than nominal water inlet temperature (60°C), and a lower than nominal water flow rate (1.1GPM). The values and order of these changes was as follows: 1) Sink temperature: ambient, -5°C, -23°C, 2) Water inlet temperature: 60°C, 100°C, 127°C and 3) Water flow rate: 1.1GPM (4.2 L/min), 1.8GPM (6.8 L/min) and 3GPM (11.4 L/min). FIGURE 5 shows the initial vacuum testing of the module where the parameters were changed as previously described. In the end, when all the nominal values were reached, the power rejected by the module was 380W. Note that the predicted radiator module performance is 416W.

The *second testing sequence* was thermal cycling in vacuum. Three cycles were performed by simultaneously changing sink temperature (between ambient and -165°C), water flow rate (between ~ 1.254GPM and 0) and water inlet temperature (between 25°C and 122°C). All the cycles are shown in FIGURE 6 where it can also be noticed that the resulting power during the thermal cycling test was between 0 and 360W.

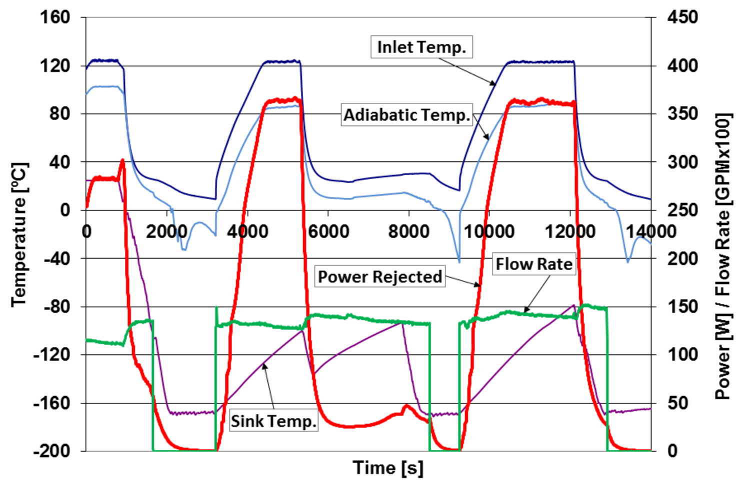


FIGURE 6. Thermo-cycling test.

The *third testing sequence* was a simple power test (also in vacuum) to check the integrity of the direct bond after the thermal cycling sequence. As seen in FIGURE 7, the power rejected by the radiator after the thermal cycling was 380W, which is equal to the power rejected before the thermal cycling in similar conditions (i.e. similar vacuum, sink temperature, water temperature and flow rate). The conclusion was that no degradation of thermal resistance was observed after thermal cycling.

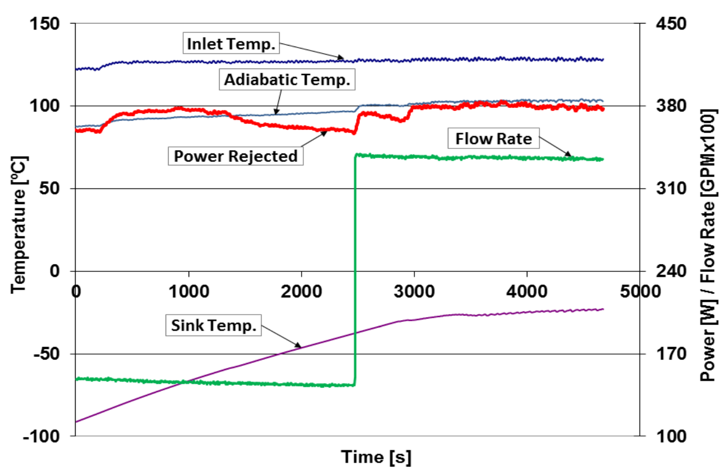
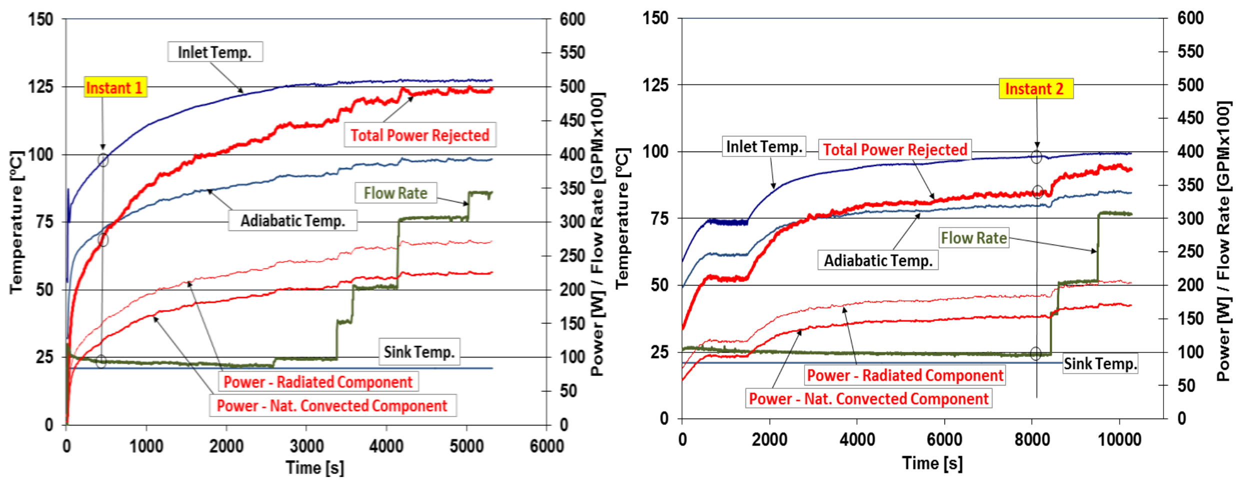


FIGURE 7. Power test after thermo-cycling test.

2. Module testing in ambient

Since all the radiator clusters tested at ACT must be in ambient conditions, the single radiator module was also tested in ambient. However, the results from the ambient testing are still helpful in determing how the radiator cluster will perform under vacuum conditions. The ambient testing results of the module are shown below in FIGURE 8a. The rejected power is 500W while the coolant temperature was 127ºC and the flow rate was 3.4 GPM (12.9 L/min). As shown in the next seciton, radiator module testing in ambient also served for comparison between the performance of the module with the original evaporator of 5.5in (14 cm) and the module with an extended evaporator (7in (17.8 cm) long). The module in the new configuration was tested again in ambient and the results are shown in FIGURE 8b.



a) b)

FIGURE 8. Module testing in ambient: a) initial evaporator length of 5.5in (14.cm) b) 7in (17.8 cm) evaporator length

3. Large evaporator module testing in ambient

As mentioned previously, the module was also tested with a longer evaporator secition to determine the effect on thermal performance. However, during testing of this module in ambient conditions the water heater had some technical problems, which prevented the water temperature from exceeding 100°C. To compare the heat rejection capability of the two configurations of the module (5.5 inch evaporator and 7 inch evaporator), instants where relevant parameters are similar (i.e.sink temperature, coolant flow rate and water inlet temperature) were chosen. As seen in FIGURE 8a and b parameters corresponding to Instants 1 and 2 in the two plots are: Coolant Flow rate = 1 GPM, Sink Temperature = 21ºC and Coolant Inlet Temperature = 96°C. In the above conditions it was observed that:

- Power rejected by the module with a 5.5in (14.7 cm) evaporator was 275 W
- Power rejected by the module with a 7in (17.8 cm) evaporator was 345 W

In conclusion, the power rejected by the module with a 7in (17.8 cm) evaporator is approximately 25% higher than the one with a 5.5in (14 cm) evaporator. This percentage is expected to increase when the inlet temperature becomes nominal (127°C) because of the radiation component of the total power in ambient. In vacuum, this percentage would increase even more than in ambient simply because the radiation is the only heat rejection mode. It was decided that, although the performance of the 5.5in (14 cm) evaporator was satisfactory, the rest of 11 clusters will use 7in (17.8 cm) evaporators to increase the margin of performance of the TDU ultimate radiator.

III. HEAT PIPE RADIATOR CLUSTER DEVELOPMENT

The plan in the beginning of the program was that only the first radiator cluster will be tested in vacuum at NASA GRC's vacuum chamber. Since this cluster was the first one, it has short (5.5in, 14.cm) evaporators, as shown in FIGURE 9. As soon as the cluster arrived at ACT valves were installed on the heat pipes and the heat pipes were charged with water. A flow meter was installed and RTDs were placed in the hot water supply stream before and after the cluster for calorimetric measurements.

FIGURE 9. First cluster a) Actual radiator cluster b) Detail showing the manifolds, the evaporators and the adiabatic sections.

A. Full Testing in Ambient at ACT

After a preliminary testing the cluster was fully instrumented and also charged with NCG. FIGURE 10 shows the thermocouple map used for both full ambient testing at ACT and future vacuum testing at NASA GRC. As it can be seen, each condenser was provided with 6 TCs marked as P1C1, P1C2 …P1C6 …P8C1…P8C6. The condenser TCs were installed on GFRC and not on the titanium pipe. The reservoirs had one TC each marked as P1R …P8R. The adiabatic sections also have one TC each marked as P1A …P8A. In addition to the two fluid “in” and “out” RTDs, 6 other TCs were installed on the manifold surfaces between the evaporators. These thermocouples are marked as CC1 …CC6.

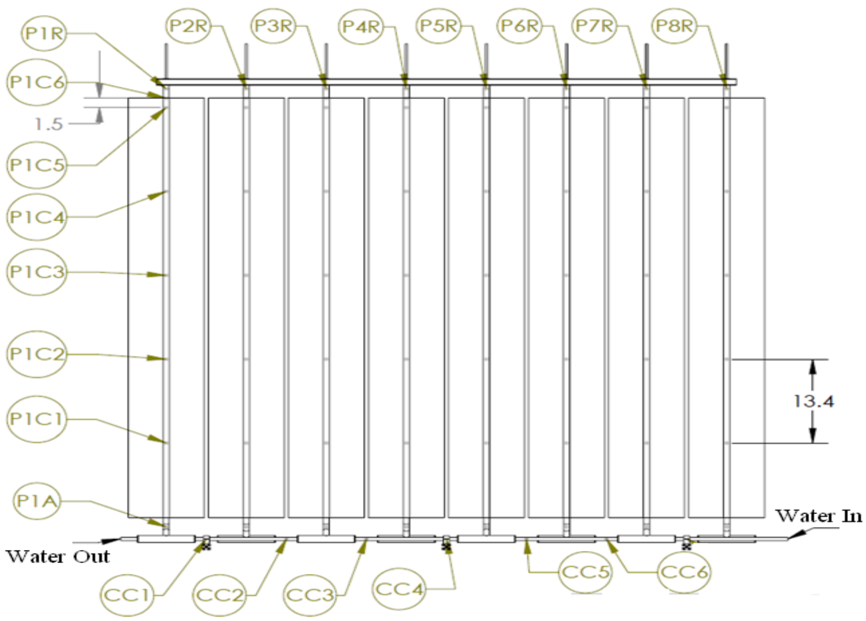


FIGURE 10. Thermocouple map on the First Cluster.

The actual testing consisted of a power test for various water inlet temperatures. The sink temperature was always ambient (21ºC) and the flow rate was always 6 GPM (22.7 L/min). Since the two manifolds are connected in parallel, it was assumed that the flow rate per manifold was approximately 3 GPM (11.35 L/min).

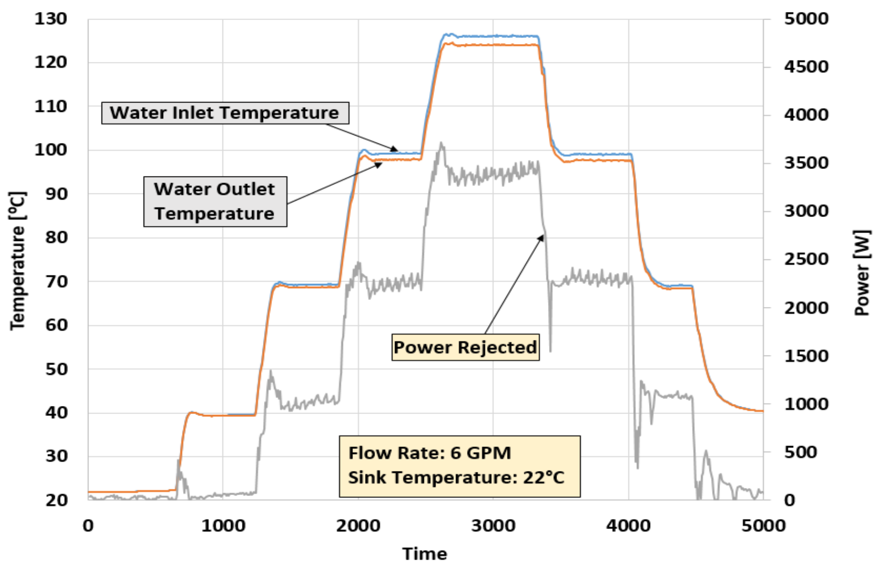


FIGURE 11. Power test: heat rejected for various water inlet temperatures

FIGURE 11 shows the power test results in ambient at ACT. As seen, the water inlet temperature was increased in steps from ambient all the way to the nominal value of 127°C. The intermediate steps were at 40°C, 70°C and 100°C. At each temperature step steady state was allowed to be reached. The maximum rejected power was again ~3.5 kW. However, there was conservative aspect of this test that is described in more detail below..

During this experiment the heat pipes worked as VCHPs and not as CCHPs. As the next plots will show, the NCG charge is slightly too large which preventedthe heat pipe condensers to be fully active at nominal water inlet temperature. As a consequence, it is expected that the panel would reject more than 3.5 kW in ambient conditions at nominal water inlet temperature and flow rate if the NCG amount is properly adjusted.

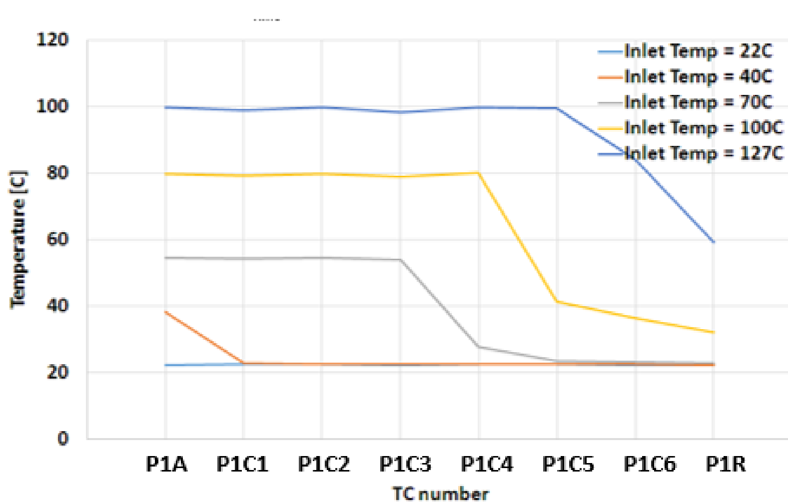


FIGURE 12. Steady state temperature profiles along Pipe No 1 corresponding to each water inlet temperature.

FIGURE 12 shows temperature profiles along pipe number 1 during the steady states at each water inlet temperature. This representation was necessary to evaluate the active length of the condenser. Indeed, it can be observed the NCG is slightly oversized. This is shown mainly by the temperature profile at the nominal water inlet temperature of 127°C where TCs P1C6 and P1R show lower temperatures when compared to the rest of them. More complete analysis (not shown here) revealed similar temperature distributions in each pipe. The slight NCG overcharge is due to the fact that the charge was calculated for a nominal sink temperature of -23°C rather than 22°C, resulting inthe reservoir being warmer than expected.

B. Cluster Testing in Vacuum at GRC

The first cluster was tested in vacuum in two rounds. During the first round the nominal parameters could not be used because of various reasons. In other words, sink temperature was 2ºC (compared to the nominal value of -23ºC), water inlet temperature and flow rate were 100ºC and 3.9 GPM (14.8 L/min) respectively compared to the nominal values of 127ºC and 6GPM (22.7 L/min), respectively. Therefore, a second round of testing where water inlet temperature was nominal was performed at a later time. Since the testing results obtained during the second round were very recent at the time of writing the paper, only the results of the first round of testing are presented in this paper.

The actual test consisted of a power test, followed by thermocycling and another power test to verify the status of the thermal resistance of the direct bond between GFRC and the titanium condenser. The water inlet temperatures during the two power tests were 40ºC, 70ºC and 100ºC. During the thermal cycling sequence, the water inlet temperature was varied between 100ºC and ambient. Water flow rate and sink temperature were always 3.88 GPM (14.7 L/min) and 2ºC, respectively. As seen in FIGURE 13, two rejected powers are represented: one power resulted from calorimetric calculations and is represented by a highly scattered succession of data points, and other power resulted from calculations of direct radiation from the panels. As it can be observed, the agreement between the two power representations is good. It can be concluded that the power rejected by the radiator at 100ºC water inlet temperature is 1.94 kW. Moreover, the rejected power after thermocycling did not change for all three water inlet temperatures. This fact confirms the integrity of the bond. Other conservative factors are discussed below in the conclusion section.

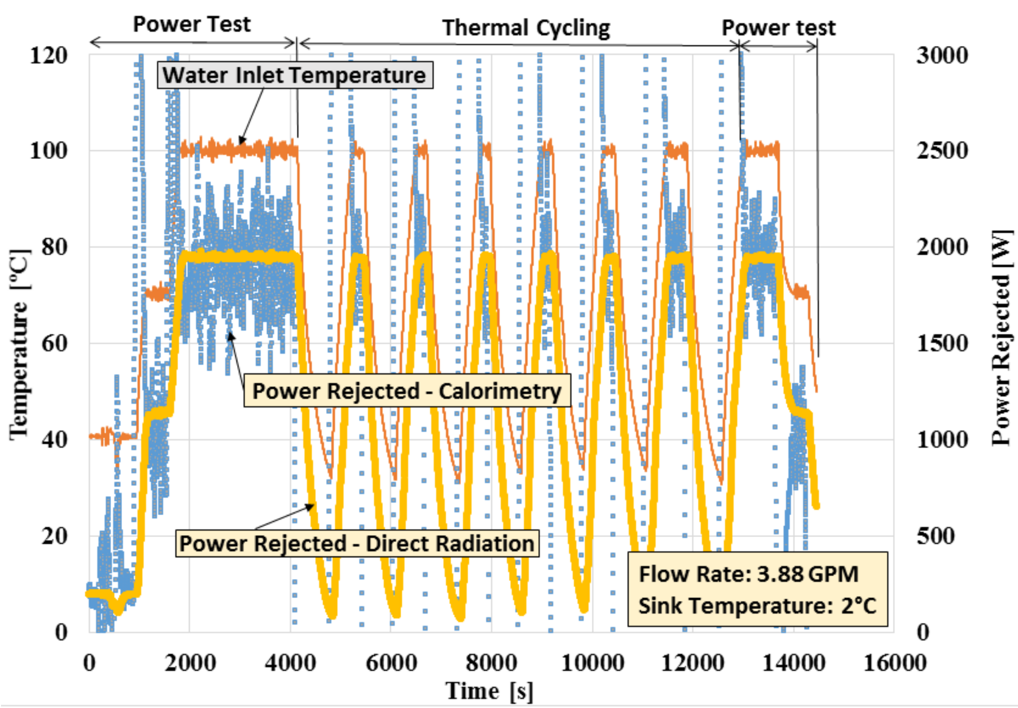


FIGURE 13. Radiator cluster testing in vacuum – first round. Testing was carried as power test, thermocycling and power test again.

IV. CONCLUSION

The paper mainly presented testing results for the heat pipe radiator module and for the radiator cluster both in ambient and vacuum. The radiator module was tested in vacuum at nominal parameters. The module rejected 380W, which is slightly less than the predicted 418W. However, the module initially had a short evaporator. After increasing the size of the evaporator, new ambient testing showed that the performance of the module increased by 25%. Also, thermal cycling in vacuum of the module showed that the direct bond was not affected by the repeated exposure to thermal stresses. The cluster, which has the shorter evaporator, was also tested in both ambient (at ACT) and vacuum (at GRC). Ambient testing showed a performance of 3.5 kW at nominal water temperature and flow rate. However, the sink temperature was ambient. In addition, an oversized NCG charge was observed in all 8 heat pipes of the cluster. In conclusion, several conservative factors influenced the performance of the radiator in ambient conditions. Vacuum testing of the cluster included an initial power test, thermocycling, and a secondpower test. None of the parameters were nominal. The power rejected by the radiator in vacuum was 1.94kW for the highest water inlet temperature. Again, the power test carried after thermocycling showed no degradation of the direct bond. In conclusion, it is expected that the performance of the cluster in vacuum would be significantly higher if all three parameters (water inlet temperature, flow rate and sink temperature) were at nominal conditions. In addition, other conservative factors during vacuum testing were: 1) the oversized amount of NCG, whicheven at nominal temperature would not allow a fully open condenser, 2) the fact that water inlet temperature was 27ºC less than nominal, which further amplified the effects of the oversized amount of NCG, and lastly, the short evaporator. A second round of cluster testing was performed in vacuum where the water inlet temperature was raised to nominal values. These results will be presented in a future paper. Finally, it is also important to mention that all subsequent cluster builds will be fabricated with long evaporators.

V. ACKNOWLEDGMENTS

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