Performance Results of Microplate Heat Exchangers

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ABSTRACT

High effectiveness heat exchangers are required for recuperative cycles such as Brayton and Joule-Thomson cryocoolers, as well as precooled hybrid coolers. System efficiency is greatly improved by higher exchanger effectiveness. An improvement of the effectiveness from 98% to 99% can reduce the input power by 24% for a typical Brayton cryocooler. The microplate heat exchangers tested use a parallel plate geometry. A new manufacturing method has been developed allowing higher performance in a parallel plate geometry while reducing the performance degradation caused by flow maldistribution.

We have manufactured and tested the performance of several heat exchangers. The heat exchanger effectiveness has been measured. Effectiveness values of 98.7% have been measured. These experimental results will be presented

While there is still much to be learned about reliably designing and building these exchangers, this work shows important improvements in performance. With the lessons learned here, we expect the next generation of heat exchangers to achieve effectiveness an above 99%.

INTRODUCTION

Recuperative cycle cryocoolers typically use some form of shell-in-tube heat exchanger.[1] Joule-Thomson (J-T) coolers typically use a tube-in-tube or multi-tube-in-tube exchanger which works well for the low mass flows of these systems but doesn't scale well to larger sizes. Brayton and Collins-cycle coolers often use a perforated plate[2] or Collins style[3] exchangers.

Precooled J-T coolers or remote cooling applications, as well as Brayton coolers, are much more efficient than simple J-T coolers but often require larger mass flows making the heat exchanger a dominate part of the overall system both in terms of mass and power. These types of coolers are well suited for both remote terrestrial and space applications where mass, power, and ease of integration are of primary importance.

Over the last several years we have developed a microplate heat exchanger with a diffusion bonded exchanger made of stacked thin foils creating a parallel plate exchanger, see Figure 1. Various materials have been used including titanium, Inconel, and stainless steel. The parallel plate geometry has the smallest theoretical size[4] but has a number of challenging aspects that have prevented its practical use. To give a sense of scale, the exchanger pictured in Figure 1 replaces a tube-in-tube exchanger that is about 8" in diameter and weighs twice as much while transferring 45 W of heat compared to 20 W for the tube-in-tube exchanger.

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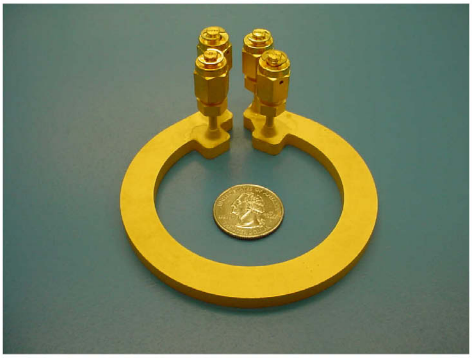

Figure 1. Microplate Heat Exchanger.

The most challenging problem is overcoming the mass flow imbalance between the various flow channels.[5] There are also a number of other practical manufacturing problems, including creating a reliable hermetic seal with diffusion bonding, the sealing the bridge bond, and sealing the inlet/outlet ports.

The bridge bond occurs when either the hot or cold side fluid enters the exchanger. There is an unsupported section at this location. Diffusion bonding requires heat and pressure to make a good joint, which is problematic at this unsupported location. Any leak in the bridge bond is internal to the exchanger, but it does reduce the overall system efficiency since not all the fluid reaches the expansion device. It could also lead to a long-term reliability issue as the leak rate could increase with time. The problem is magnified by different fluids in the hot and cold sides of the exchanger, in which case any leak is intolerable. We have tried various solutions to the bridge bond and have had some success although most solutions have not worked consistently from unit to unit. Our latest approach appears to have successfully solved the problem creating a hermetic bridge bond joint in 6 total units tested.

Another seemingly easy problem, which has proven more difficult than anticipated, is the bonding of the inlet/outlet ports. In the past, including the units tested here, we have used soldering or brazing to attach the tubes into the exchanger after welding some type of fitting to the tubes. Soldering has not created robust joints, and exchanger handling difficulties have led to leaks. While brazing creates a good robust joint, the braze process is difficult to control and reproduce, and has the potential to plug flow passages.

EXPERIMENTAL DESCRIPTION

The heat exchanger performance was characterized using an experimental apparatus which simulates the environment where the exchanger will be used. This allows heat exchanger performance characterization to be performed independently of the final cryocooler configuration. This test configuration measures the actual exchanger performance directly.

The apparatus directly measures heat imbalance between the two flow streams. A two-stage Gifford-McMahon (G-M) cryocooler is used to set each end of the exchanger to a fixed temperature. The gas flow is first routed to a precooling counterflow heat exchanger used to reduce the load on the G-M cooler. The gas flow is next routed to the G-M first-stage where it is isothermalized at the heat exchanger hot end temperature before it enters the exchanger. After flowing through the exchanger under test, the gas is passed through a second isothermal heat exchanger attached to a calibrated heat flow sensor attached to the G-M 2nd-stage. This is also the heat exchanger cold end temperature before it enters the cold side of the heat exchanger under test, see Figures 2 and 3. The heat measured by the heat flow sensor is a direct measurement of the heat loss of the exchanger including all the parasitic losses. Temperature sensors are also placed at all four exchanger inlets and outlets to double check the heat loss although the resolution of this measurement is not as high as the direct heat flow method.

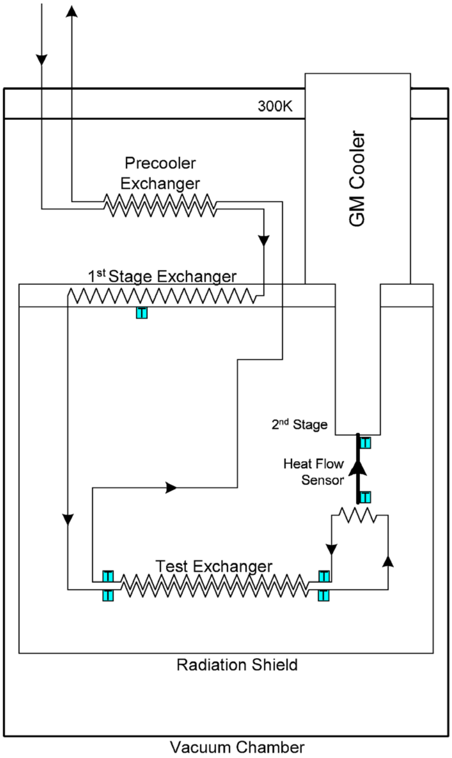

Figure 2. Experimental setup schematic showing how the fluid is routed though the system

Helium is flowed through the system at various mass flow rates to characterize the exchanger performance. The G-M cooler temperatures can be set at different temperatures to provide additional performance characterization. The fluid heat capacity rates, C_{hot} and C_{cold}, are calculated for both the hot and cold flow steams using the inlet temperatures and mass flow rate. The heat exchanger effectiveness is determined using

$$\varepsilon = 1\text{-}(Q_{loss}/C_{min}) \tag{1}$$

where ε is the effectiveness, Q_{loss} is the measured heat loss, and C_{min} is the minimum of C_{hot} and C_{cold}..

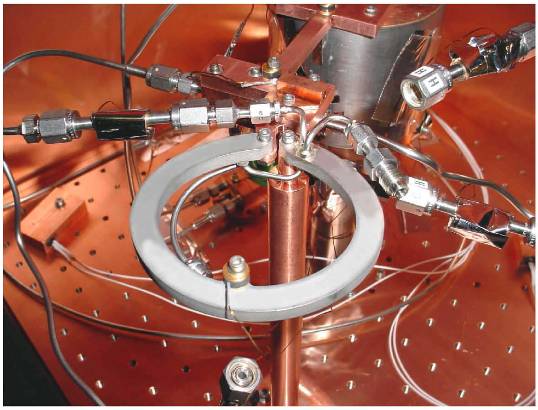

Figure 3. Experimental setup showing heat exchanger under test, heat flow sensor, and various temperature sensors.

RESULTS & DISCUSSION

These tests were used to correlate a model of heat exchanger performance, and this model was then used to predict exchanger performance at the cryocooler operating conditions. Figure 4 shows the measured performance of three heat exchangers. Heat exchanger (HX) 1 is designed to operate between room temperature and the first-stage of a Stirling precooler. HX2 operates between the first-stage and second-stage. While HX3 operates between the Stirling second-stage and an expansion device. HX2 and HX3 are identical in geometry so HX3 is shown operating at two different temperature ranges; the higher range is used to compare its performance to HX2 while the lower temperature range is closer to its nominal operating conditions.

None of the three tested heat exchangers have a performance as high as expected. Design effectiveness was 98% for HX1 and 99% for HX2 and HX3. While HX2 and HX3 are identical in design, they exhibit substantially different performance in the current testing.

All the exchangers had a substantially higher than expected pressure drop. After some time spent troubleshooting, it was determined that some of the flow channels had been plugged when the inlet/outlet tubing was brazed on. While internal features were added to create a braze stop, an error in the design created a wicking feature that pulled braze to the flow channel entrance. It cannot be determined which flow passages are blocked, and this has a large impact on the performance since some flow can be ‘trapped’ without a way to transfer its heat into the other flow stream. By examining the pressure drop data, the number of flow channels can be estimated, and the flow channel calculations follow the experimental data in that HX1 and HX3 have more blocked channels than HX2. HX2 does have blocked channels although it is possible that the blockages are in more favorable positions allowing higher performance.

A new method to attach the inlet/outlet tubes has been developed that is strong and reliable. This method welds the tubes in place. It does place some limitations on the possible positioning of the tubes but these are not very restrictive only limiting a tube from passing directly over the opening into the heat exchanger.

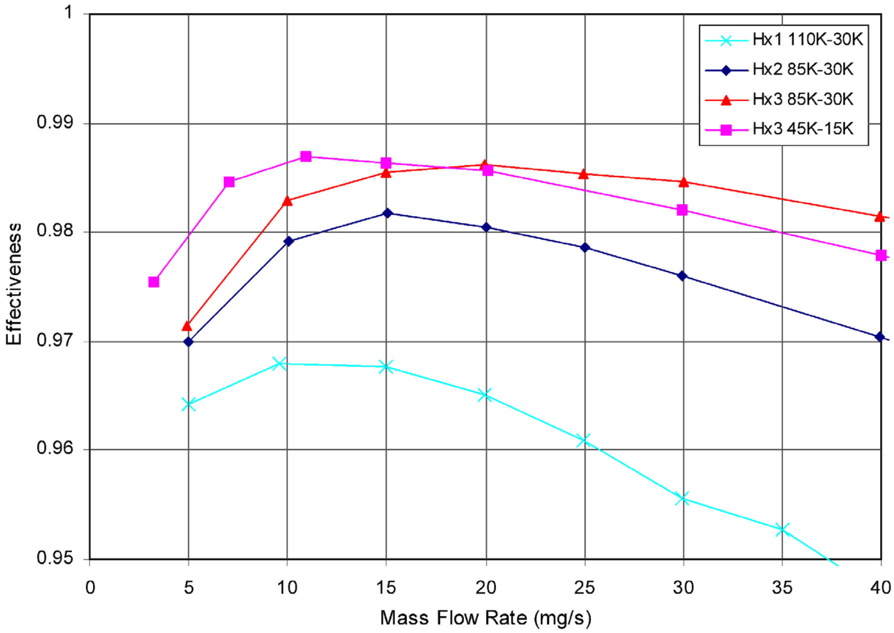

Figure 4. Performance results. Heat exchanger #3 is also tested at the operating temperature of exchanger #2 to compare performance since they are identical in geometry but will be used at different operating conditions.

FUTURE WORK

Seven additional microplate heat exchangers are currently being manufactured. These include four unique designs allowing for comparison not only of different designs but also in consistency in performance of the same designs. These exchangers will be used in three different cryocoolers but unfortunately due to cost and schedule constraints all the exchangers must be manufactured concurrently so we will not benefit from lessons learned for the additional units. These new units include all our lessons learned on previous exchangers including the bridge bond design, hermetic diffusion bond sealing, and the new port configuration. All exchangers will be performance tested, and the results will be presented at a later date.

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