Development of an Accelerated Thermal Cycling Test for Reversible Microchannel Heat Pump Coils

Chad Bowers, Senior Research Engineer, Creative Thermal Solutions, Urbana, IL, USA;
George Baker, Principal Engineer, Modine Manufacturing Company, Racine, WI, USA;
Stefan Elbel, Chief Engineer, Creative Thermal Solutions and University of Illinois, Urbana, IL, USA;
Pega Hrnjak, President, Creative Thermal Solutions and University of Illinois, Urbana, IL, USA;

Abstract: With increasing efficiency standards and higher energy costs, the demand for reversible air-conditioning/heat pump systems is increasing throughout the stationary market. The heat exchangers in such reversible systems must perform reliably as both evaporator and condenser. The outdoor coil, specifically, is subject to very large changes in both operational pressures and temperatures. With this increase in demand and the gradual shift of the industry to aluminum microchannel coil design, comes a need to better understand and quantify the reliability of the heat exchangers. This paper presents the development of an accelerated thermal cycling test method using refrigerant at typical pressures and temperatures to demonstrate the reliability of said heat exchangers. This method employs a technique of switching the heat exchanger between condensing and evaporation modes rapidly while monitoring strain at various locations of concern to quantify any shift in strain caused by fatigue.

Key Words: Heat Pump, Reversible, Microchannel, Cycling, Thermal Fatigue

1 INTRODUCTION

The use of so-called microchannel heat exchanger technology is becoming more and more popular. Implementation of microchannel heat exchangers in place of state-of-the-art copper-tube aluminum-fin coils has been shown to improve both component and system level performance. The advantages of moving to microchannel technology as a replacement for round-tube-plate-fin are manifold and include substantial refrigerant charge reductions, possible lower airside pressure drop, higher heat-transfer coefficients on both the refrigerant and air side, greater airside surface area in a given volume, reduction in the required temperature difference between fluid streams, and possible cost reduction due to all aluminum construction. It has been shown that microchannel condensers in stationary applications can improve COP, condenser capacity, and evaporator capacity, compared to a baseline system using a round-tube condenser for all of the reasons described above. These same advantages make microchannel technology an attractive option for evaporators or heat exchangers in reversible systems that need to function both as a condenser and evaporator.

These coils differ from conventional round-tube-plate-fin heat exchangers in many ways. One of the most obvious differences is the number of tube to manifold joints required; as most microchannel heat exchangers have a plurality of parallel circuits that allow for increased air side heat transfer with little to no refrigerant side pressure drop penalties. As with all heat exchangers, microchannel heat exchangers can be subjected to a large number of thermal shocks due to system on/off cycling and defrost cycling, which causes the heat exchanger to switch operational modes. The combination of these many tube to manifold joints and the need to maintain durability over a large number of thermo-mechanical fatigue inducing events, requires sound design concepts can suitably meet this demand. Chidley *et al.* (2011) noted that the first thermally related failures in microchannel heat exchangers appear at or near tubes entering the manifold at locations near the inlet of the coil. They also created a thermo-mechanical model to predict the thermal-shock fatigue based upon infrared thermography of a microchannel heat exchanger during thermal-shock events. Zhao *et al.* (2010) pointed out that the stress-strain relationship in this type of heat exchangers can be dramatically affected by the manufacturing process, specifically the initial state of the structure, fixturing, brazing, and cool-down conditions. This was also demonstrated by Kim and Lee (2012) who saw that the fatigue behavior of the aluminum used in this type of heat exchanger can change with the brazing process used to assemble the coil. Their experiments showed that this change was larger than fatigue life differences predicted using popular methods.

With the above in mind, the objective of this work was to develop an accelerated testing method that would mimic the mode of thermo-mechanical fatigue in reversible microchannel heat exchangers due to the temperature/pressure changes experienced as part of system on/off cycling and defrost events over the life of the heat exchanger. Many current methods of performing this type of accelerated test employ a pressurized temperature controlled single phase fluid cycled through the heat exchanger. While this approach can adequately provide both specified inlet temperatures and pressures, the heat transfer and corresponding temperature profiles associated with the real mode of operation will be much different. This is due to the fact that when acting as either an evaporator or condenser, much of the refrigerant side of the heat exchanger is at a fixed temperature. When using a single phase fluid the temperature gradients seen in the material of the heat exchanger may be dramatically different than those seen in operation, leading to possible incorrect assumptions about induced stresses within the material. The test developed as part of this work uses the refrigerant that the heat exchanger is designed to operate with, R410A in this case, as well as maintaining the heat transfer function of the coil. It was determined at the outset of this work that no degradation of heat exchanger mechanical response (i.e. thermo-mechanical induced strain) at several key locations over a series of 100,000 cycles would provide high confidence of a durable design that would not fail due to thermo-mechanical fatigue. This testing was performed on two heat exchanger of the same design in parallel.

2 EXPERIMENTAL FACILITY

The accelerated thermal cycling was performed using a modified direct expansion vapor compression system that employed R410A as the working fluid. This was done in order to achieve working pressures and temperatures similar to those seen in application. A schematic of the facility operating in heating mode and cooling mode is shown Figure 1. In heating mode the hot discharge gas from the compressor is split between the support system's condenser, located in a packaged unit outdoors, and the two test units. The condensed fluid is then recombined prior to entering an electronic expansion valve, where the fluid is expanded and fed to the support system evaporator. Throttling valves downstream of the support system condenser and the test units were used to meter the distribution of the flow. The system operated in heating mode until a temperature set point of the refrigerant at the inlet of the test units, 90°C, was reached.

Upon reaching the temperature set point, the system was switched from heating to cooling mode. This was done through switching of four solenoid valves. In cooling mode, all discharge gas was sent to be condensed in the support system's condenser. The subcooled liquid was then sent through the electronic expansion valve, after which the low pressure two-phase fluid was split between the test units and the support system evaporator. Throttling valves downstream of the test units and the support system evaporator were used to meter the flow distribution between the two. This system configuration was maintained until a specified set point, 27°C, at the inlet of the test unit was reached. At this point, the system was switched back into heating mode using the four solenoid valves.

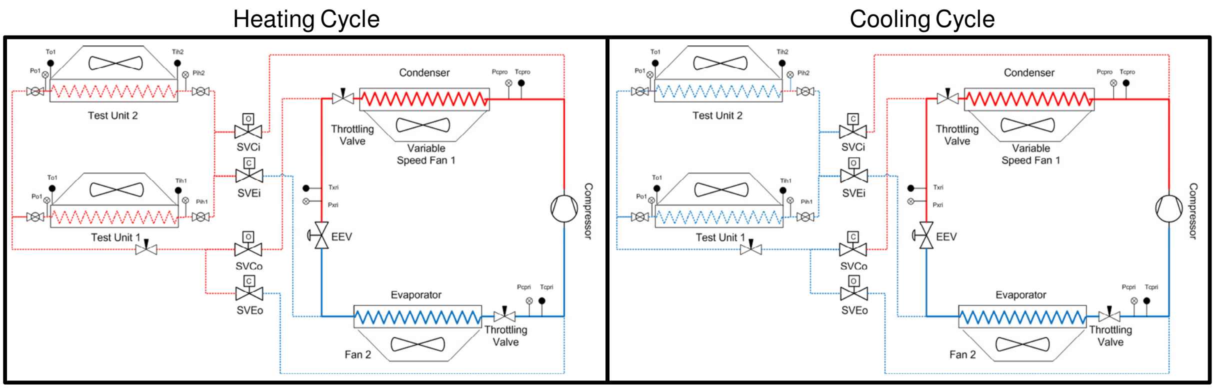

Figure 1: Heating and Cooling Mode Cycle Schematics

In order to maintain steady ambient conditions, the test units were located indoors in a facility separate from the support system. All of the valves required in the system, with the exception of the throttling valve located downstream of the support system evaporator were located in this part of the facility. The fans for the test unit were also located in this part of the facility. In addition, all controls, data acquisition, power distribution, safeties were located in the indoor portion of the facility. The support system for the testing was a modified commercial unitary type air conditioning system integrated into the building's space conditioning infrastructure. Integration into the existing building infrastructure allowed the building itself to provide a relatively stable load to the vapor compression cycle used in testing.

The test coils provided were designed to operate as both a condenser and evaporator in a reversible manner. In switching the operational mode of the heat exchanger during the accelerated testing, the refrigerant inlet and outlet locations remained constant. The direction of air flow also remained constant regardless of operational mode. This provided an overall counterflow configuration between the two fluid streams at all times.

Pressure was measured at the inlet and outlet of each test heat exchanger using piezoelectric pressure transducers having an operational range of 0-6900kPa. Prior to testing, the pressure transducers were calibrated in conjunction with the rest of the data acquisition system over the entire operating range. The refrigerant temperature was also measured at the inlet and outlet of each heat exchanger. This was done using ungrounded T-type immersion thermocouple probes. In addition to fluid pressure and temperature measurements, strain gages and thermocouples were mounted to the surface of the test unit at eight locations. Strain gages and thermocouples one through four were located on microchannel tubes on the inlet header, with one being furthest away from the inlet and four being the closest. Strain gages and thermocouples five through eight were located on the exit header, with five being furthest from the exit and eight being nearest. The strain gages provided and mounted by the heat exchanger manufacturer were a three-wire quarter-bridge configuration, with a gage factor of 2.125. In order to calculate the strain accurately, the strain gage signal voltage as well as the supply voltage were measured simultaneously. The zero voltage or unstrained voltage of each gage was considered to be the strain gage signal voltage acquired with the heat exchangers mounted in the facility and both heat exchangers and system charged with the operational refrigerant charge. It should also be noted that this zero signal was determined with the support system in idle.

Data was acquired from the above described instrumentation in two distinct ways. All pressure, temperature, and strain data were collected continuously over the entirety of the test period at the rate of approximately 0.3Hz. However, to provide higher temporal resolution to the strain response of the heat exchanger, all strain measurements were also acquired periodically at a higher sampling rate. During the course of the long term tests, high speed strain measurements were recorded for the first ten minutes of every hour at a rate of 20Hz.

3 DETERMINATION OF TARGET CYCLE CONDITIONS

Prior to the initiation of any long term cyclic testing, facility tuning was performed with sample heat exchangers provided solely for this purpose. These sample heat exchangers were used in order to ensure that no fatigue was introduced into the test coils. In addition to the facility tuning, steady state testing was performed in heating mode to characterize the operational response of the heat exchangers in conditions simulating those experienced in application, as well as ensure that there was balanced refrigerant flow through both coils.

The data presented in Figure 2 through Figure 4 show the start up and steady state operation of the facility in heating mode along with the response of one of the sample heat exchangers. In Figure 2, the pressure and refrigerant inlet and outlet temperatures are shown as a function of time during the start up and on through to steady operation in heating mode. The initial start up of the system is characterized by a dramatic increase in both pressure and temperature of the refrigerant. Initially, both inlet and outlet temperatures of the refrigerant stream rise together. As the heat exchanger begins to reject heat to the ambient, the inlet and outlet refrigerant temperatures begin to diverge. When the system comes to steady operation, the pressure and temperatures plateau to nearly constant values.

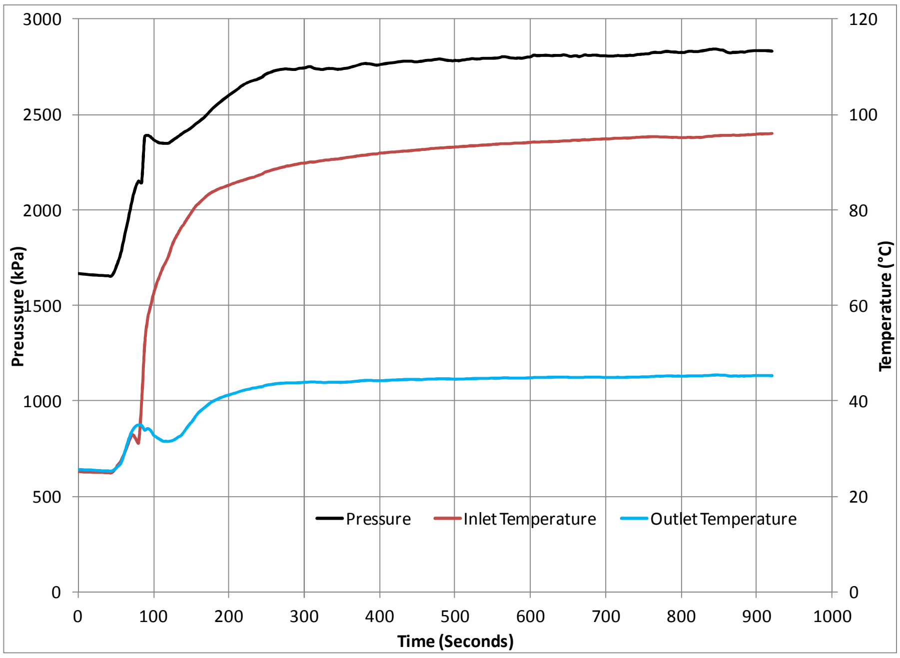

Figure 2: Refrigerant Pressure and Temperature During Start-Up and Steady State Baseline Conditions

The response of the surface temperatures at Locations 1 through 8 to the operating conditions shown in Figure 2 are presented in Figure 3. The temperature at these locations responds with a small amount of lag when compared to the fluid temperature. This is due, in large part, to the thermal mass of the heat exchanger. In addition to this time delay, there is also a noticeable offset in temperature values. This is most noticeable when comparing

Location 4, the location closest to the refrigerant inlet, to the inlet refrigerant temperature. The inlet refrigerant temperature at steady conditions was approximately 95°C, while the surface temperature at Location 4 is just over 70°C. Even though these measurements are physically close to one another, one is immersed in the refrigerant and the other is on the outside of the heat exchanger where a temperature difference must exist due to conduction through the metal and convection via the air stream. What is also apparent is that the further away from the refrigerant inlet the temperature is measured on the inlet header, the lower the temperature is. On the exit side of the heat exchanger, Locations 5 through 8, there is no difference in the surface temperatures when the system reaches steady operation.

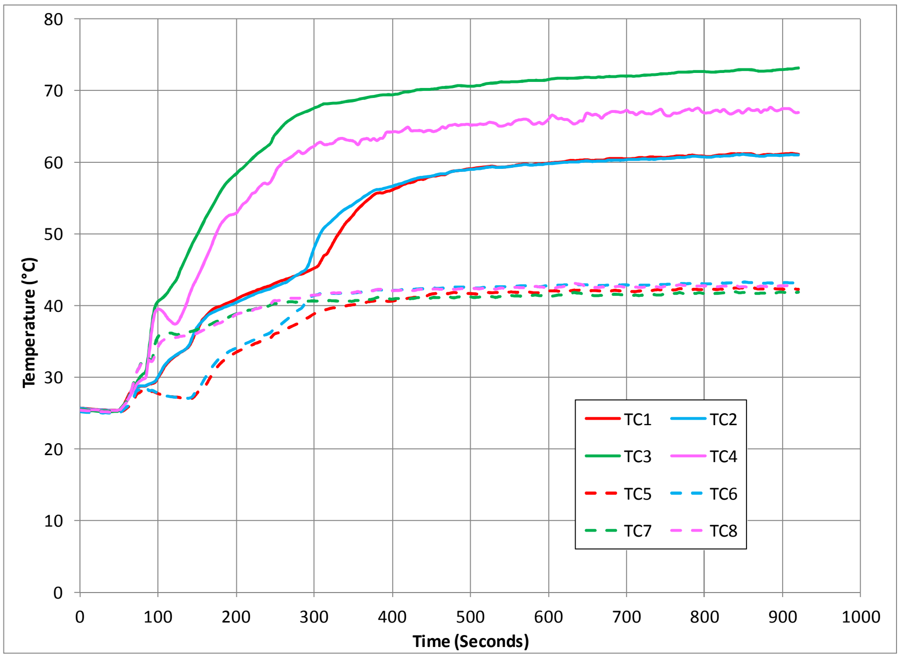

Figure 3: Surface Mount Temperature Response at Start Up and Steady Stead of Heat Exchanger Locations 1 Through 8

The other response to the system operation monitored during these baseline tests was the strain at the various locations in the heat exchanger. The strain response to the system operating in heating mode as shown in Figure 2 and Figure 3 are shown against the same time scale in Figure 4. During the initial spike in pressure and temperature, the strains have very dramatic responses. Most noticeable is the immediate increase in strain at Location 4, closest to the refrigerant inlet. In steady operation, Location 4 has strains of similar magnitude to all the other locations on that header, but during start up, the strain at Location 4 immediately increases to values close to those seen in steady operation. The strain at Location 3 shows similar behavior to this, but not as dramatic. The strain response at Locations 1 and 2 is different in that they go up briefly, come back down and then gradually increase to the steady strain value. On the outlet header, the strain behavior at all four locations is very similar. There are noticeable fluctuations during the initial increase in pressure and temperature, but as the system begins to stabilize the strains come back down to values close to what they were prior to start up.

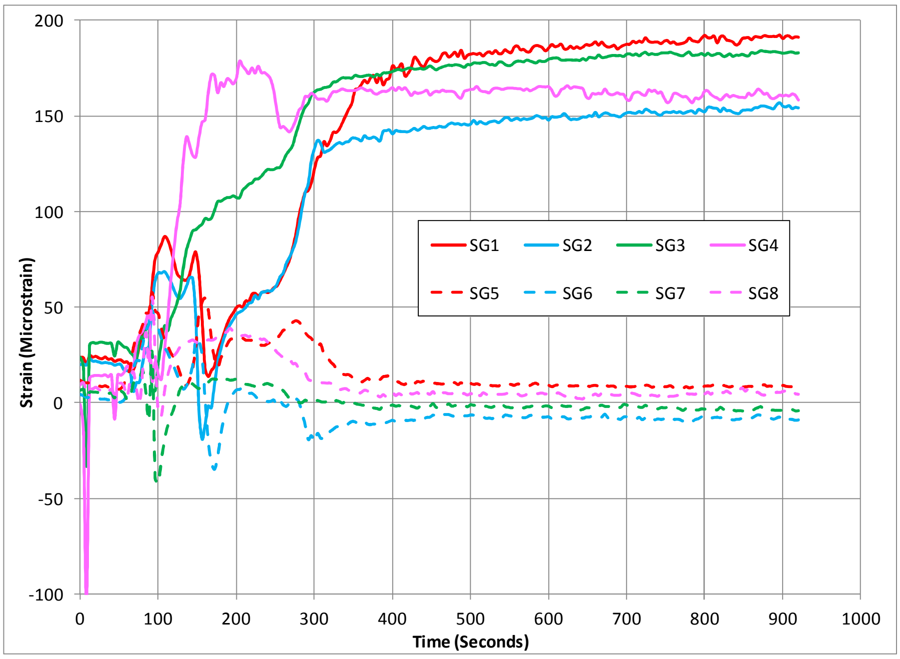

Figure 4: Strain Response at Start Up and Steady Stead of Heat Exchanger Locations 1 Through 8

4 REPRESENTATIVE CYCLING BEHAVIOR

In order to describe the thermal cycling behavior and the corresponding heat exchanger response, the following section details representative data from the fluid stream and a single heat exchanger during two consecutive cycles. The graph shown in Figure 5 presents the refrigerant temperatures and pressures for two consecutive cycles in one of the heat exchangers tested. The data presented in Figure 6 and Figure 7 represent heat exchanger response data that correspond to the fluid data shown in Figure 5.

The beginning of the first full cycle in Figure 5 starts at about 214 seconds where both temperatures and pressures go through an inflection point and begin to increase. This marks the time at which the solenoid valves were switched from cooling to heating mode. The first part of the heating cycle is marked by a sharp rise in pressure and inlet temperature over the first several seconds. After this initial sharp increase, both pressure and inlet temperature continue to increase but at a slower rate, until the heat set point is reached at approximately 242 seconds. The outlet temperature of the fluid has a much slower response to the change in operation mode. The change in temperature is also much smaller at the outlet. In addition to the pressures and temperatures, the outlet saturation temperature is shown in Figure 5. During the first six seconds of the heating cycle the exit of the coil is still superheated, but after the initial sharp rise in pressure and inlet temperature the outlet of the coil becomes slightly sub-cooled.

At approximately 242 seconds the heating set point is reached and the solenoid valves switch the system to cooling mode. Immediately, pressure begins to decrease rapidly and continues to decrease at a nearly constant rate until the cooling set point is reached about ten seconds later. The inlet temperature exhibits similar behavior, in that it drops over the cooling cycle at a nearly constant rate. There is, however, a lag of a few seconds between the pressure and the temperature. As in the heating cycle, the outlet temperature has a slower and smaller response to the change in operational mode. At the beginning of the cooling cycle, the fluid exiting the heat exchanger is still slightly sub-cooled but quickly becomes superheated as the cooling cycle progresses. Just prior to the system switching over to heating mode the exit superheat is approximately 10K.

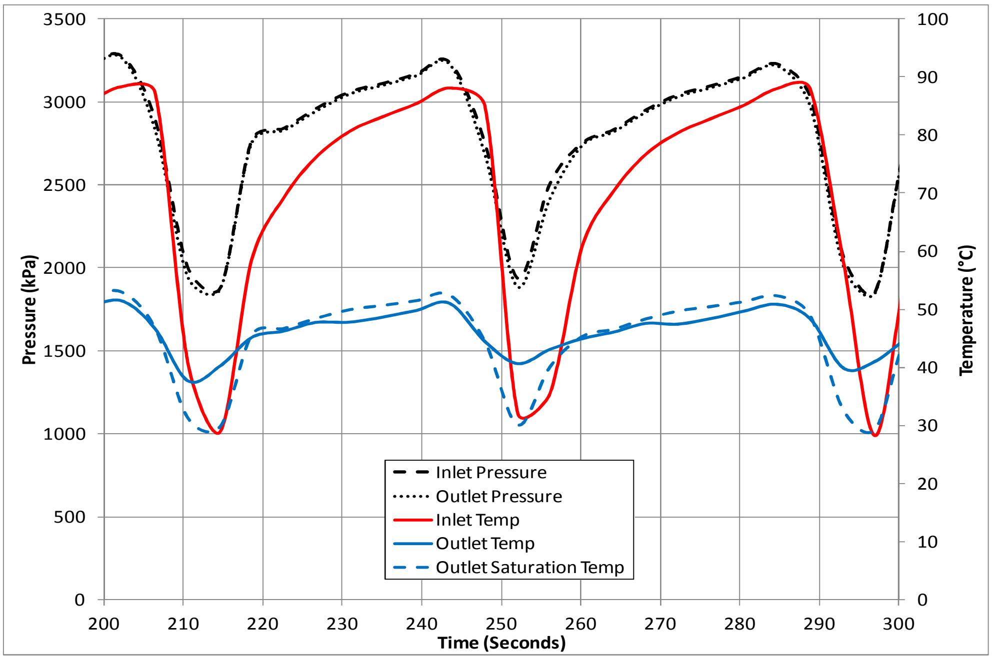

Figure 5: Inlet and Outlet Pressures and Temperatures for Typical Cycles in Heat Exchanger 1

The strain and temperature response of the test unit designated Heat Exchanger 1 at Location 1 on the inlet header is shown in Figure 6. The largest change in strain appears to occur when the system is switched from cooling mode to heating mode. The best example of this sharp change in strain can be seen at approximately 212 seconds. At the point where pressure experiences an inflection, the switching from low side pressure to high, the strain drops dramatically and then recovers to a value similar to that seen at the end of the cooling cycle. As the pressure and temperature increase in the heat exchanger the strain gradually increases, until the heat set point temperature is reached. At this point, the pressure in the heat exchanger begins to decrease and the strain also begins to decrease. The temperature response of the heat exchanger at this location exhibits noticeable lag when compared to the inlet fluid temperature. During the first full cycle shown, the inlet fluid temperature reaches a peak of approximately 90℃ at 245 seconds, while the temperature at this location reaches its peak at 252 seconds, near the end of the cooling cycle. In addition to this lag, the temperature of the heat exchanger undergoes a much smaller swing than the fluid, 47℃ to - 52℃, versus 27℃ to 90℃. Recall that Location 1 is located at the far end of the inlet header from the inlet. This distance from the fluid inlet and the thickness of the heat exchanger material combine to supress the amplitude of the temperature change at this location. As further evidence of this, the temperature at Location 4 (nearest the inlet) undergoes a 25℃ change from 50℃ to 75℃.

Locations 7 and 8 are located on the tubes connected to the exit header near the refrigerant exit. The temperature and strain responses at Location 7 are shown in Figure 7. As with all the other temperature measurements on the refrigerant exit side of the heat exchanger, there is very little variation in the temperature of the series of cycles at this location. The temperature measurement is nearly constant at a value of approximately 49℃. The strain response at this location is the most unusual when compared to all other locations measured on the heat exchanger. At the initialization of the heating cycle, there is a small increase in the strain over the course of only a few seconds. After this initial increase, the strain decreases to approximately 75microstrain. For the remainder of the heating cycle, the strain changes very little from this value. Near the end of the cooling cycle, there is a small decrease in the strain. Over the series of thermal cycles shown, the strain produced has a minimum of approximately 60 microstrain and a maximum of approximately 100microstrain. This indicates that the likelihood of a fatigue or strain related failure at this location is low.

Common sense would also indicate the likelihood of thermo-mechanical related failure at this location should be low as this will always be the place where refrigerant and ambient temperatures will be the closest and where temperature gradients within the metal should be lowest.

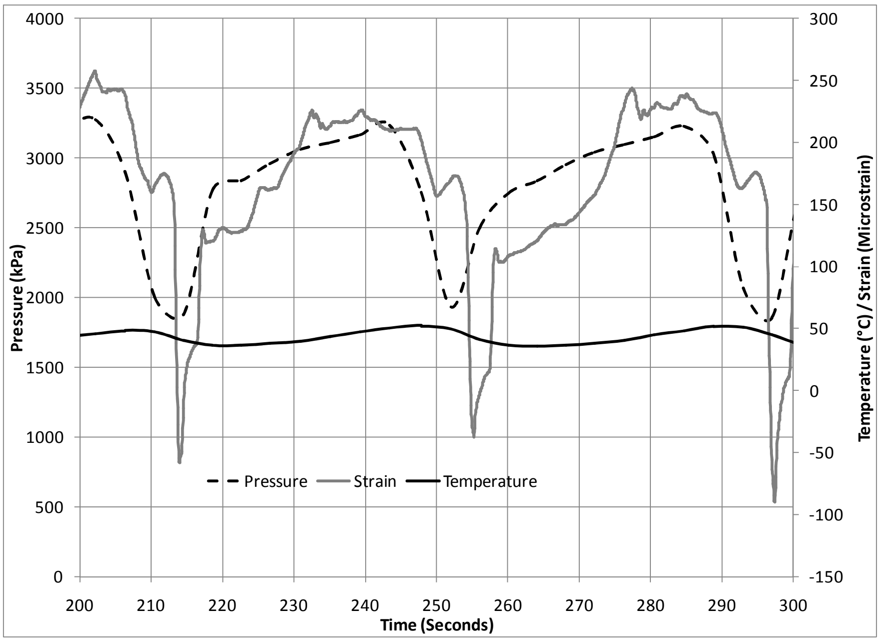

Figure 6: Strain and Temperature at Location 1 During Typical Cycling in Heat Exchanger 1

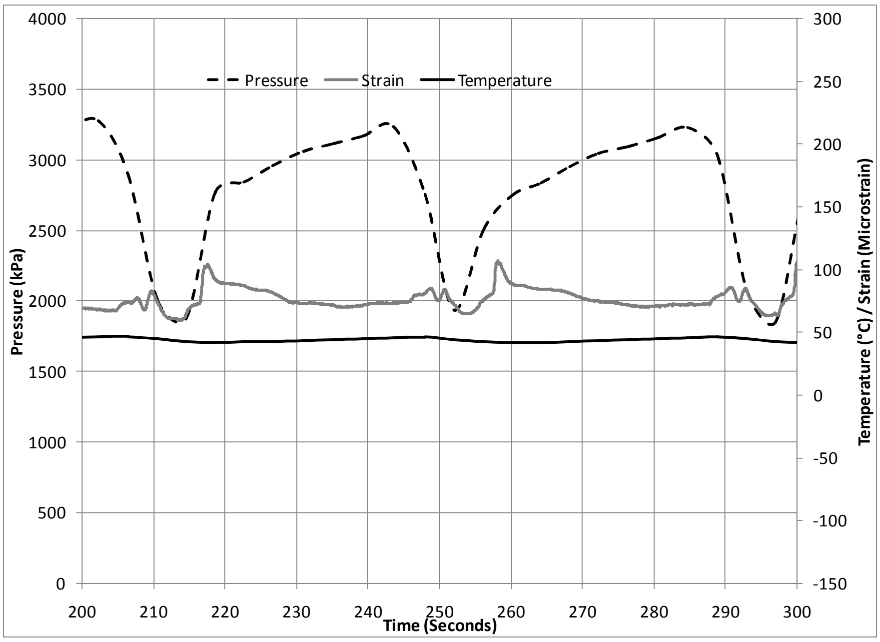

Figure 7: Strain and Temperature at Location 7 During Typical Cycling in Heat Exchanger 1

5 LONG TERM TESTING OF HEAT EXCHANGERS

After the initial facility tuning and cyclic behavior determination, long term testing was started and run until approximately 100,000 cycles were achieved monitoring the resulting temperature and strain responses of the two heat exchangers under test. In order to monitor the number of cycles achieved, the temperature results were analyzed for inflection points indicating that the system had been switched from heating to cooling or from cooling to heating. The combination of both of these inflection points indicates that the system has gone through a complete thermal cycle. In total, 102,098 thermal cycles were achieved over a period of 56 days. On average, the system provided 1823 cycles in a 24 hour period. However, the actual number of cycles varied between 1557 and 2145 through the course of testing. This variation is due to a variety of reasons, most of them being linked to ambient conditions and system maintenance.

In order to evaluate the strain distribution between the two heat exchangers being tested over a 24 hour period, histograms documenting the occurrence of various strain magnitudes at each location were created using the high speed strain data obtained during the first ten minutes of each hour. The following provides examples of representative strain responses at the various locations in both heat exchangers tested. Location 3 and Location 4 are located on the inlet header at the end closest to the refrigerant inlet. Thus the strain responses at these two locations should be similar. The histograms for both heat exchangers at Location 3 are shown in Figure 8. There is reasonable agreement in the experienced strain between the two heat exchangers. A strain of approximately 75microstrain is the most experienced over 24 hours by Heat Exchanger 1. Heat Exchanger 2 most frequently experiences strains of approximately 50microstrain. Both heat exchangers experience some level of strain below the predominant load, but very little above it. The loading at Location 4, for both heat exchangers is dramatically different than that at Location 3. This difference can be seen in the histograms presented in Figure 9. The strain loading for both heat exchangers is much higher at Location 4, with the peak occurring at approximately 230microstrain for Heat Exchanger 1 and 190microstrain for Heat Exchanger 2. While the magnitude of the predominant load is slightly different for each heat exchanger the loading behavior is very similar between the two.

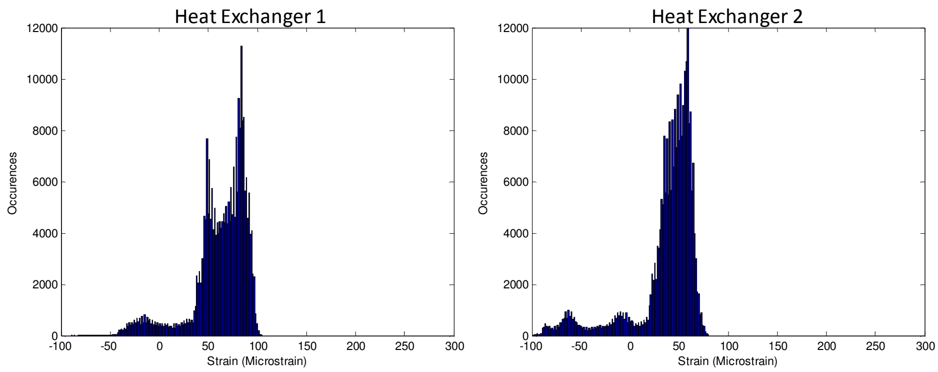

Figure 8: Histogram of 24 Consecutive Hourly High Speed Strain at Location 3 for Heat Exchangers 1 and 2

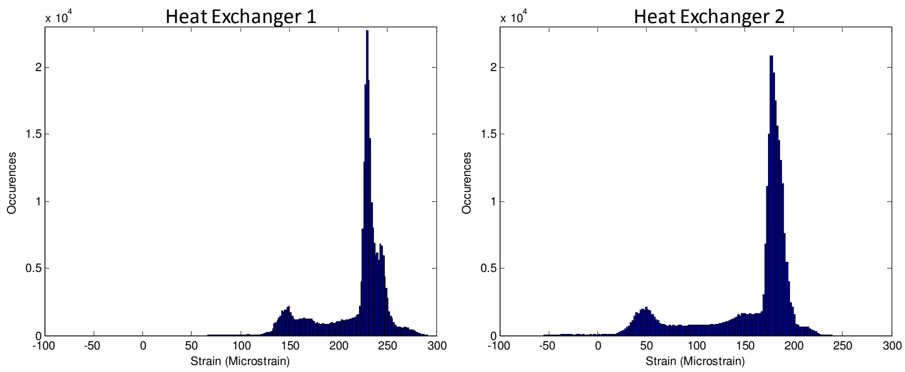

Figure 9: Histogram of 24 Consecutive Hourly High Speed Strain at Location 4 for Heat Exchangers 1 and 2

Location 7 and Location 8 are the final measurement locations on the heat exchanger and are located on the exit header at the end nearest the refrigerant outlet. Histograms for the strain over the same 24 hour period are shown in Figure 10 and Figure 11, respectively. The shape of the strain distribution is similar for both heat exchangers; however, at Location 7 the magnitude of the predominant strain is higher in Heat Exchanger 1 while at Location 8, the magnitude of the predominant strain is higher in Heat Exchanger 2. While the loading of the heat exchangers does not appear to be uniform from location to location, it does seem that loading of the heat exchangers is very similar at the same location. This indicates that both heat exchangers were exposed to a series of thermal cycles, within this 24 hour period, that should cause similar fatigue or damage to the coil.

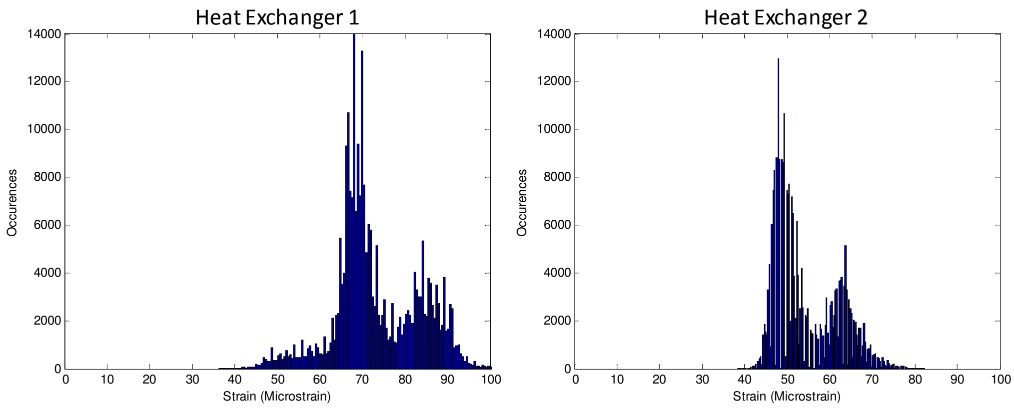

Figure 10: Histogram of 24 Consecutive Hourly High Speed Strain at Location 7 for Heat Exchangers 1 and 2

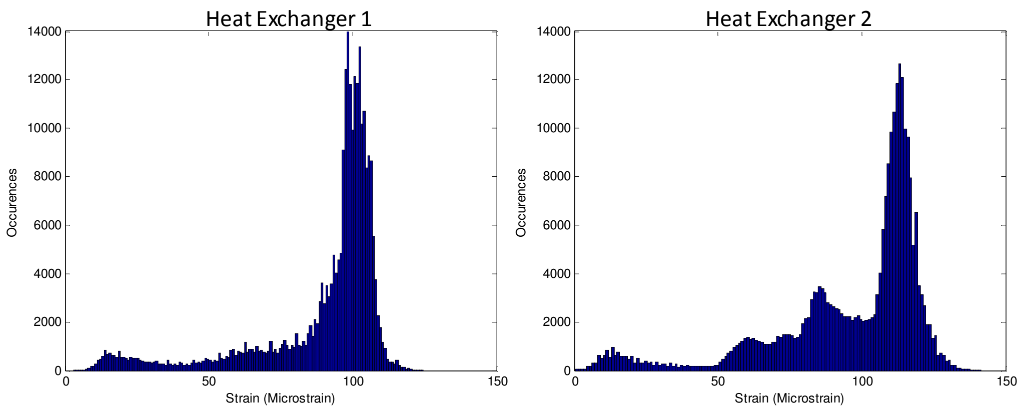

Figure 11: Histogram of 24 Consecutive Hourly High Speed Strain at Location 8 for Heat Exchangers 1 and 2

In order to understand how consistent the response and loading of the heat exchangers was over the entirety of the test period, the strain distribution over a 24 hour period was examined for six distinct 24 hour periods resulting in a given amount of cumulative cycles. The results of this analysis for Heat Exchanger 2 at Locations 3 are shown in Figure 12. In the 24 hour period resulting in a cumulative cycle count of 17,000 cycles the predominant strain is approximately 50microstrain, with very little strain occurring above this value. The form of the strain distribution as well as the magnitude is very similar for all but one of the six days examined. The 24 hour period resulting in a cumulative cycle count of 46,000 cycles has a bimodal distribution of the strain with modes at approximately 40microstrain and 90microstrain. While there is a difference in the distribution of the strain on this day compared to the others, magnitude of the experienced strains is still in the same range. This being said, the distribution and magnitude of the strain after 102,000 cycles appears very similar to that seen after only 17,000 cycles; suggesting strong repeatability of the loading and little wear of the heat exchanger at these locations.

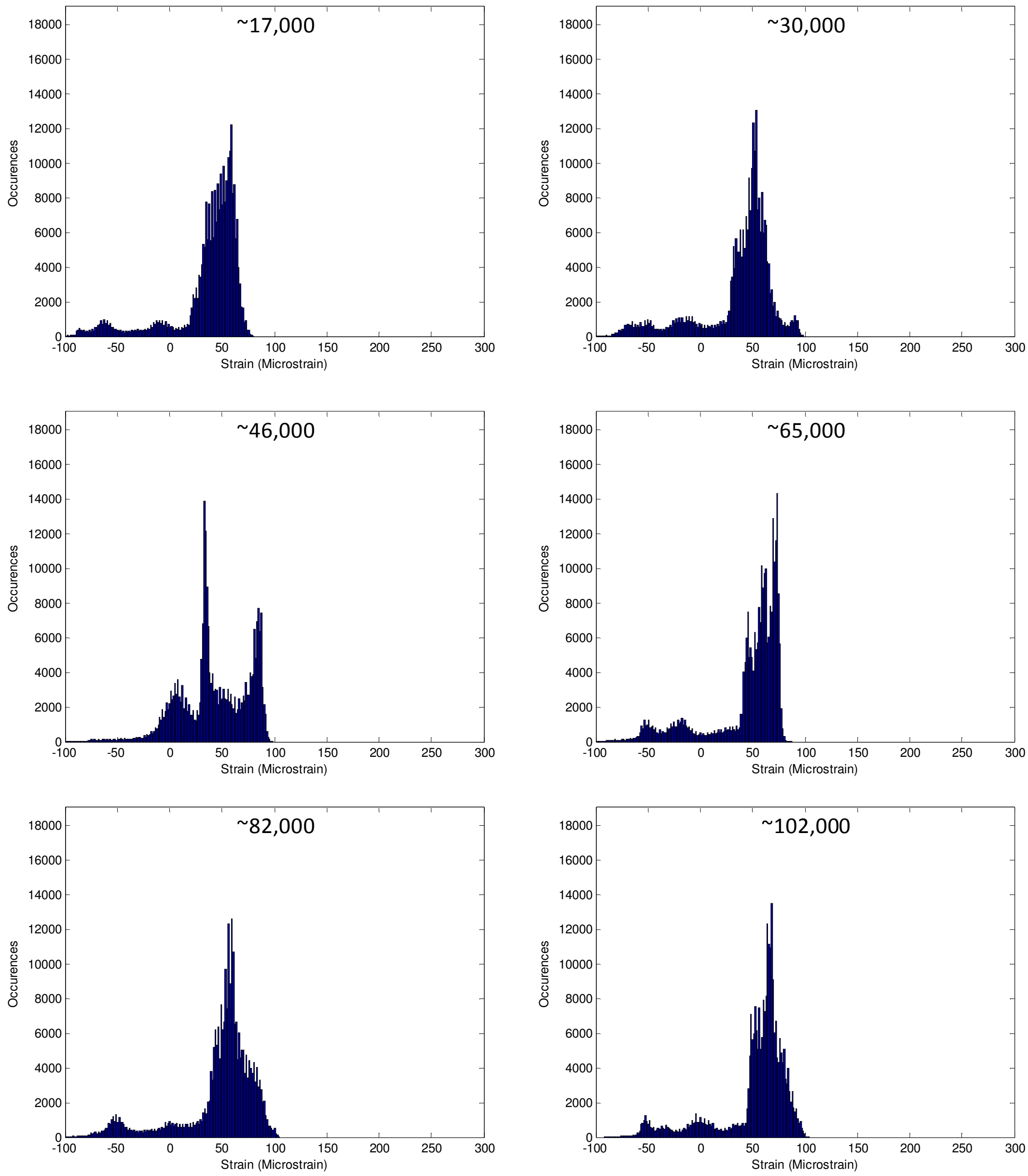

Figure 12: 24 Consecutive Hourly High Speed Strain at Location 3 for Heat Exchanger 2 for Days Ending With The Cumulative Cycle Count Shown

6 CONCLUSIONS

Microchannel technology has many attractive features from both a performance and cost perspective and has been adopted widely in some applications. Use of this technology in reversible heat pump system is the next logical step in its widespread adoption. Ensuring the durability of such a product, especially when being used in a new application, is important for the adoption of any technology. Ensuring this durability in a relevant and accelerated manner can be a difficult task. The work above described the development of a new accelerated thermo-mechanical fatigue test for microchannel heat exchanger being used in reversible air-conditioning and heat pump equipment. This novel accelerated test uses the refrigerant R410A to cycle the heat exchange through conditions that replicate operation of the coil as both an evaporator and a condenser. The results of the test of the two heat exchangers used show that over a series of approximately 100,000 cycles there were no measured mechanical responses indicating any thermo-mechanical related fatigue that would lead to a failure of the coils. Similar accelerated testing could be performed on all the system subcomponents used in reversible heat pump such as valves, line sets, and connectors.

7 REFERENCES

Chidley, A., Roger, F., Traidia, A., 2011, "A Thermal-Shock Fatigue Design of Automotive Heat Exchangers", *World Academy of Science, Engineering and Technology*, Vol. 49, pp1034-1038

Kim, H.H., Lee, S.B., 2012, Effect of a Brazing Process on Mechnical and Fatigue Behavior of Alclad Aluminum 3005, *Journal of Mechanical Science and Technology,* Vol 27, pp2111-2115

Zhao, H., Elbel, S, Hrnjak, P.S., 2010, "Transport Phenomena Involved in Controlled Atmosphere Brazing of Microchannel Aluminum Heat Exchanger", International Refrigeration and Air Conditioning Conference, Paper 2406