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ABSTRACT

The formation of joints is critical to the long-term reliability with leak-free operation of heating, ventilation, air conditioning, and refrigeration (HVAC&R) systems. Leakages commonly occur due to fatigue failure developing in joined materials from continuous pressure and temperature cycling with mechanical vibration and refrigerant under pressure inside the system. In particular, rapid pressure and temperature changes happen frequently (e.g., multiple times per day) when the system is switched on and off. Therefore, it is important to have an automatically controlled fastresponse pressure and temperature cycling (PTC) test stand available to test the performance of refrigeration joints to evaluate new bonding technologies. An innovative PTC test stand with hot-gas bypass control was designed, built, and demonstrated that eliminated the need for an evaporator and ensures rapid transition between different operating conditions. Tests were performed to demonstrate test stand functionality using R410A as the refrigerant to provide pressure and temperature cycles from 600 to 4500 kPa and 5 to 80 ℃. A 50-cycle, 5-hour demonstration test was performed with both adhesively bonded and brazed joints following standardized joint testing guidelines. Both joint types survived the test without leaking, suggesting that the adhesive joints have sufficient thermal fatigue resistance along with the conventional brazed joints. Throughout the demonstration, the test stand accurately controlled the setpoint temperatures and pressures while switching the test section between these conditions. The test stand serves as a new approach for pressure and temperature cyclic fatigue testing of joints in HVAC&R systems.

KEYWORDS

Fatigue Testing; Pressure and temperature cycling; HVAC&R; Adhesive joints; Hot gas bypass control

1. INTRODUCTION

In the heating, ventilation, air conditioning and refrigeration (HVAC&R) industry, a typical system consists of various individual components connected together by tubes and joints to form a complex piping system for energy generation or distribution. At every point where two individual parts meet, joining technologies must produce a mechanically adequate bond between the surfaces without any damage to the original part. These joints must meet structural requirements for the initial weight and operational pressure of the system and allow leak-free refrigerant flow through their path as designed. A strong and reliable joint can provide a stable, efficient, and leak-free system assembly. The failure of joints causes leakage and a series of negative consequences including danger to the system, energy inefficiencies, and pollution to the environment. Gschrey et al. (2011) projected the contribution of fluorinated gases to CO_2 emissions will increase from approximately 1.3% (2004) to 7.9% (2050) in a business-as-usual scenario. Many of the refrigerants used in HVAC&R systems are fluorinated gases (Bauer et al., 2015) and the leakage of refrigerant from the systems contributes to CO_2 emissions in many ways (Koronaki et al., 2012). As reported by the United States Environmental Protection Agency (2015), many categories in the HVAC&R industry are facing leakage issues, among which several common equipment categories such as cold storage, residential unitary air conditions (AC), and mobile air conditioners have leakage rates of 10% per year or more. Annual leakage rates vary from one refrigeration system to another (Coulomb, 2008). In particular, supermarket refrigeration systems can have up to 30% leakage per year due to the nature of field installing long line sets (Beshr et al., 2015). It is also reported that joint failure caused by faults in joining techniques is one of the major reasons for the leakage (EPA 2015; Francis et al., 2017).

New joining technologies need to be explored, developed, and evaluated to address these leakage issues. There are several alternative joints proposed to replace the traditional brazed or soldered joints, including two major types: mechanical joints and adhesive joints. Brazed or soldered joints rely on wetting and spreading of a molten filler material on the surface to form a metallurgical bond between the filler and substrate. Mechanical joints including compression joints, press joints, push joints, and similar. A report by ASHRAE (RP 1808; Elbel et al., 2018) compared the assembly and reliability of brazed joints, compression joints, press joints, and flare joints and concluded that press fitting works better than the compression and flare joints; brazed joints have the minimum leakage rate if assembled properly. In general, mechanical joints are highly dependent on the mechanical forces and some types need to be retightened for leak-free operation (Elbel et al., 2018). Adhesive joints rely on the process of bonding two materials with the aid of an adhesive, a substance capable of holding materials together by surface attachment (ASTM D 907, 2015). Adhesives have been proven reliable and widely used in industrial joining process including the automotive, aerospace, and electronics industries. It can be applied without using any specially designed tools or fittings, which makes it flexible for use on different geometries (Banea et al., 2018; Campilho et al., 2009; Devries and Adams, 2002). However, applying adhesive joints in HVAC&R systems is relatively new proposition and there is an almost complete lacking of evaluation under the operating conditions specific to this industry, especially in fatigue failure with respect to pressure and temperature cycling. Fatigue in engineering is a loss of structural integrity over long-term operation under the influence of repeated or continuous application of stress. Studies in material behavior shows that fatigue failure is common to most types of materials and it has been estimated that 80% of all engineering failures can be contributed to fatigue (Dowling, 1998). For adhesive joints in HVAC&R systems, the static stress in most of the tube-to-tube joints is not very high, as these joints serve a sealing function rather than a load-bearing function. However, temperature and pressure changes along with the vibration of the system applies a continuously changing stress in the bonding area, which may cause thermal fatigue failure even if the stress is much less than the critical static stress.

Over the past few decades, there are numerous studies of thermal fatigue failure reported for piping and tubing in thermal systems that have large and frequent temperature changes in the system. For example, Poursaeidi and Bazvandi (2016) analyzed the thermal fatigue life of gas turbine casing due to the emergency shut down. Du (2016) simulated the thermal stress and fatigue fracture of a single tube for the solar tower molten salt receiver to find the minimum heat flux and the critical crack total length. While it is well recognized that fatigue failures in piping and tubing are of critical concern to system operation, there has been little research done on tube-to-tube joints in thermal systems.

In this study, in order to evaluate the thermal fatigue resistance of joints due to the pressure and temperature cycling of working fluid as in the HVAC&R industry, review of the thermal fatigue testing approaches and standard requirements were performed firstly. Based on the review, an innovative pressure and temperature cycling (PTC) test stand with hot-gas bypass control is proposed, designed and tested using adhesive joints as a demonstration.

2. REVIEW OF THERMAL FATIGUE TESTING OF JOINTS

Thermal stress is created by changes in the temperature of a material. In adhesive joints, the different thermal expansion of adhesive and adherends is the main reason for the thermal stress. Thermal stress also contributes to the fatigue failure, especially when the joints undergo a temperature change. A review on fatigue in adhesively bonded joints by Wahab (2012) revealed that this topic has received limited attention in the literature.

Banea and da Silva (2010) reported that adhesively bonded steel joints exposed to high temperature (tested at 80 ℃) have a decrease in strength by as much as ~30%. Gao et al. (2011) tested and simulated the fatigue lifetime of adhesive films in both hydrothermal aging and thermal cycling, and they found a decrease in fatigue life after longer aging times. As for the thermal cycling, they found that the fatigue life had an initial increase when going through increased thermal cycling, but this eventually decreased. Wu et al. (2016) investigated the effect of thermal exposure on the fatigue characteristics of the adhesive bonded aluminum joints. They found that the fatigue resistance decreased slightly in a high-cycle regime loaded at 40% of the maximum quasi-static strength ($> 10^6$ cycles) and significantly degraded in a low-cycle regime loaded at 80% of the maximum quasi-static strength (~ $10^3 - 10^4$ cycles); they argued that the reason for degradation was due to adhesive oxidation.

In order to understand and evaluate the fatigue performance of adhesive joints, a test stand must simulate the joint working conditions of an HVAC&R system. In a review of the available standards, it is found that the specific requirement for adhesive joints in HVAC&R application is given by the ISO standard 14903 (ISO, 2017). Adhesive joints need to go through pressure, temperature, and vibration cycling tests to be qualified for use. The chemical compatibility of any adhesive with the refrigerant, lubricant, etc. must also be evaluated, similar to any new material that is introduced into a system. The test stand developed in this paper offers a new approach to address the temperature cycling component of the standard; a method to address the vibration testing component of the standard was described separately (Liu et al., 2021). As shown in Figure 1, the joint under test needs to be subjected to temperature/pressure swings for a certain number of cycles. The pressure and temperature ranges are to be determined by the manufacturer or application requirements. In the cycling test, a 2-minute dwell period should be maintained after reaching the designated maximum or minimum temperature. One complete cycle contains one heating and one cooling process and the complete test should contain 50 cycles. This needs to be followed by a 200-cycle pressure test and 2,000,000-cycle vibration test to fully prove the joint has good thermal fatigue resistance.

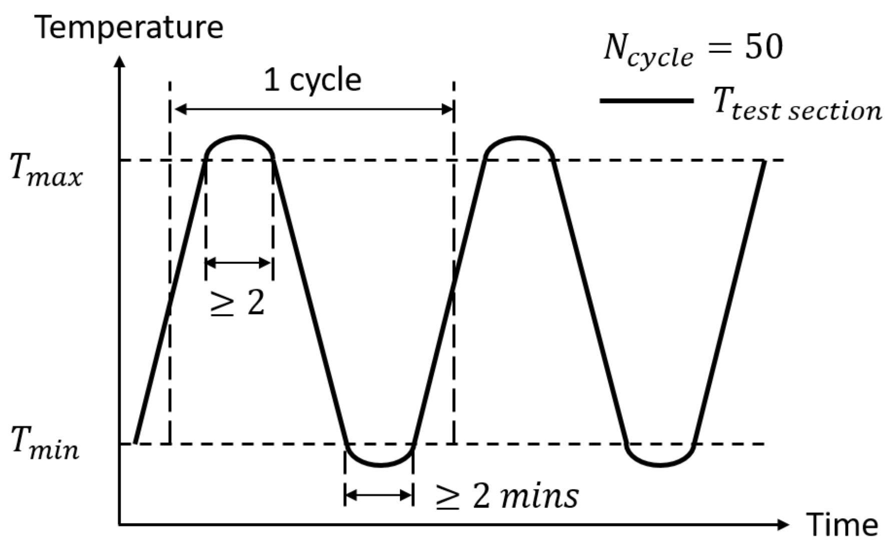

Figure 1: Pressure and temperature cycling (PTC) test example adapted from ISO 14903 (2017).

There have been a few pressure and temperature cycling test stands designed in the literature to evaluate refrigerant joints and fittings. Hourahan (1998) described separate test methods to evaluate the pressure and temperature cycling fatigue for refrigerant lines and fittings. Hot gas from a natural gas burner and chilled water from a refrigeration system were used to generate the heating and cooling needed for the thermal cycling. It was observed that a total number of 62,000 cycles without failure could prove that the thermal fatigue failure was not an issue. However, no refrigerant was present in the test section (given that water was used as the working fluid) which is not recommended by the ISO standard. Wilson and Bowers (2014) designed a test stand for accelerated fatigue testing that was used to evaluate the fatigue performance of flame-free refrigeration fittings. Two separate test stands were designed and built. The thermal shock test stand used a standard vapor compression cycle with a set of solenoid valves to split either hot high-pressure refrigerant or cold low-pressure refrigerant to the test section. In order to control the temperature of the test section in the designed range, an additional heat exchanger was installed between the test section and the vapor compression cycle to provide extra heat transfer capacity for condensing the hot gas through the test section without using the condenser in the supporting vapor compression cycle. However, the additional heat exchanger brings challenges in system control and extra cost. The operating range of pressure and temperature are also limited by the hot and cold reservoirs.

3. EXPERIMENTAL TEST STAND

As found in the literature review, the fatigue failure of adhesive joints is highly determined by the joint shape, geometry, materials, and operating conditions. For the purpose of this study, a pressure and temperature cycling (PTC) test stand is developed to investigate the thermal fatigue of adhesive (or other) joints in HVAC&R systems. The test stand generally follows the guidance of the ISO 14903 standard, with some interpretation and modifications to better simulate and investigate the joint performance in real HVAC&R systems.

Joints in a standard vapor compression system will experience both steady-state operation and stop/start cycles, as most systems are designed to cycle on and off based on the load. For example, air conditioners, heat pumps, and refrigerators in domestic applications can all easily cycle on and off many times a day. In these systems, several common refrigerants are used, including R22, R134a, and R410A. In this study, R410A is selected as the working fluid for the test stand because it has the most extreme operating temperature and pressure ranges of these common refrigerants. A conventional R410A air conditioner or heat pump can experience severe operating conditions with large variation and high absolute values for pressure, where the high side pressure is approximately 4 times the pressure of R22 or 8 times the pressure of R134a, operating at similar temperature conditions. Testing using R32 and CO_2 are also of interest; however, the current test stand is not designed to handle flammable refrigerants and the high pressure required by CO_2 with all copper tubes. Compared to other studies published in the open literature with the same research objectives, which used a complete vapor compression cycle to perform the pressure and temperature cycling, a new type of PTC test stand is introduced here that leverages a hot-gas bypass vapor-compression cycle to induce the cycling, which simplifies the system architecture and enhances the controllability and operating range.

3.1 Test Stand Design

The test stand was designed to provide combined pressure and temperature cycling. To evaluate the reliability of joints at different temperature and pressure typical in HVAC&R systems, a test stand must control both the evaporating and condensing temperatures as desired. A standard vapor compression cycle as described previously can be used for this testing, but with extra difficulty in changing between the hot and cold source temperature due to the unbalanced condensing and evaporating load in different modes. Splitting of flow between the test section and vapor compression cycle is critical to the safe operation of the test stand, with an additional heat exchanger required. It also needs two heat exchangers connected to two different constant-temperature sources for the condenser and evaporator. In order to provide the hot and cold gas for the cyclic testing, an alternative hot-gas bypass (HGB) method is used. The HGB method removes the evaporator from the standard vapor compression cycle by adding a HGB line with three sets of valves used to control the high-side pressure, low-side pressure, and mass flow rate in the system (Hubacher et al. 2002). It is a well-studied technology and widespread for compressor development and R&D activities in both subcritical refrigerants such as R134a (Zhang 2018), R410A (Schmidt 2018) and supercritical refrigerant CO2 (Kurtulus et al., 2014). Its comparability with various refrigerants and easy-to-control feature fits the PTC test purpose perfectly. The hot and cold gas can then be simply redirected to a test section containing the joints using two sets of solenoid valves in the discharge and suction lines, without flow splits between the test section and the vapor compression cycle. The full heating or cooling load are used in the test section without additional heat exchanger. There are three operating modes for the system: a non-testing mode where the test section is sealed off from the normally operating hot-gas bypass loop; a heating mode in which the solenoid valves direct the hot gas through the test section; and a cooling mode in which cold gas is directed through the test section. The corresponding schematic figures for each of these operating modes are shown in Figure 2. Note that not all the supporting in-line parts are shown in the schematic figures. Non-critical valves and sight glasses used for practical start-up, control, and operation of the facility are omitted for simplicity.

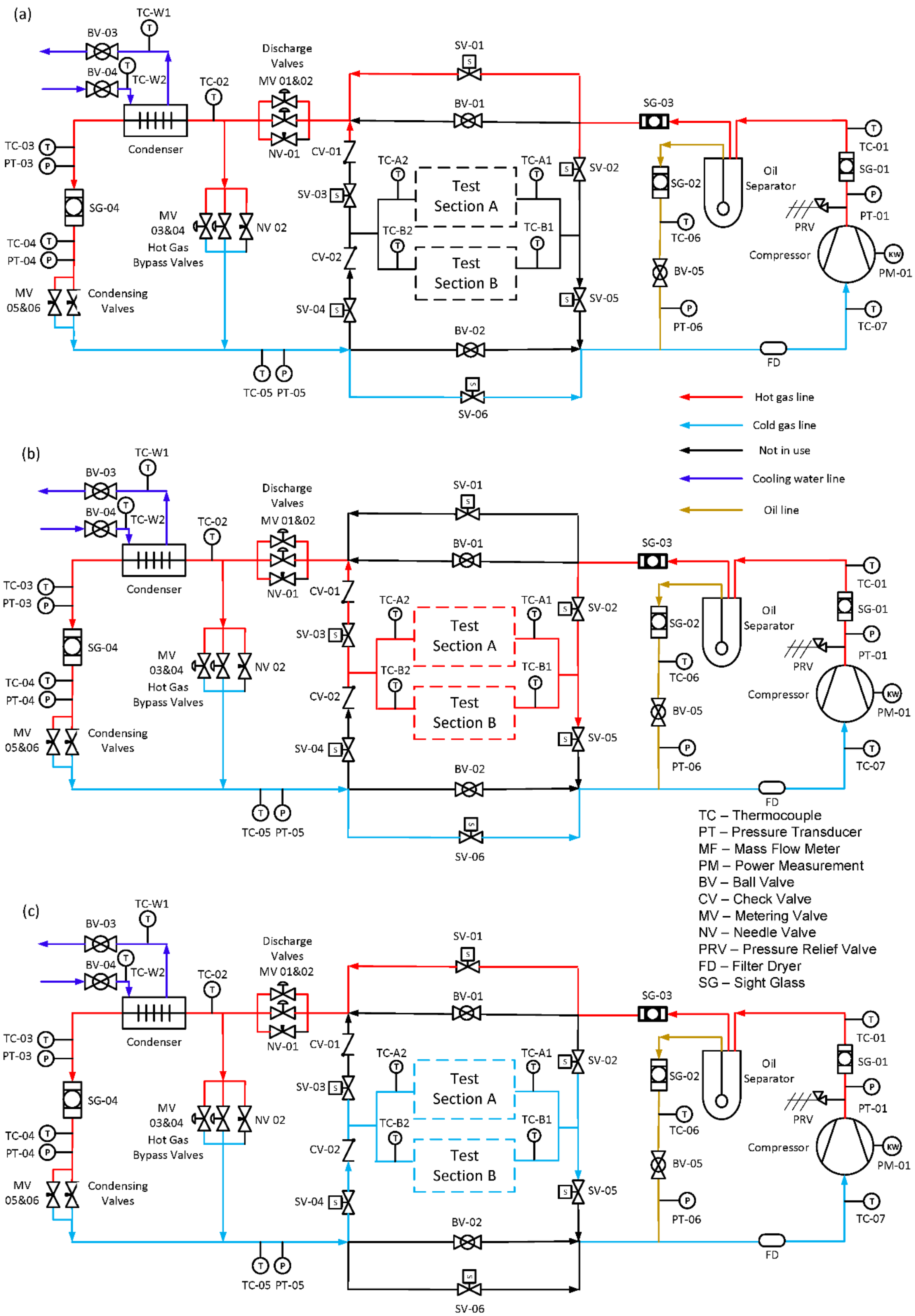

Figure 2: Schematic diagram of the developed pressure and temperature cycling test stand with hot-gas bypass control under (a) non-testing, (b) heating, and (c) cooling modes.

Regardless of the operating mode, the HGB loop itself works in the same way outside the test section, which is best illustrated by the red (high temperature/pressure) and blue (low temperature/pressure) flow lines in the schematic diagram in the non-testing mode (Figure 2 (a)) where all flow is diverted around the test section. High pressure and temperature gas comes out of the compressor into the oil separator. At the oil separator outlet, the first set of solenoid valves (SV01 and SV02) controls the hot gas into the test section. After the test section, there is a set of discharge valves (MV 01&02 and NV 01) to control the discharge pressure and temperature. After the discharge control valves, the hot gas is split into two streams: one to the condenser liquid line and one to the hot-gas bypass line. A set of hot-gas bypass valves (MV 03&04 and NV 02) are placed after the discharge valves and in front of the suction line, and a set of condensing valves (MV 05&06) are placed after the condenser outlet. Refrigerant from these two lines mixes downstream of these control valves and becomes the single suction gas line to the compressor with low pressure and temperature.

These sets of valves are used to control and maintain three different pressure levels within the HGB loop: the discharge pressure, intermediate (condensing) pressure, and suction pressure. The condenser is cooled by a constant-temperature cold-water loop, which fixes the condensing pressure based on the temperature. With this relatively constant intermediate pressure, the discharge pressure and suction pressure can be changed freely over a relatively large range. For example, if a higher discharge pressure and temperature is needed, reducing the opening of the discharge metering valve will raise the pressure (increase the pressure difference between the condensing and discharge pressure). Similarly, reducing the opening of the suction valves can decrease the suction pressure as well as the suction temperature.

Another important feature of this test stand design is that the mass flow rate can also be controlled by adjusting the three sets of the valves. Changing the metering influences not only the pressure drop across the valve but also the mass flow rate. This is utilized to control the superheat at the inlet of the compressor and the mass flow rate in the system. When the refrigerant vapor is compressed, work input by the compressor also increases the temperature of the refrigerant. If the refrigerant were to be expanded back to the inlet pressure of the compressor without a commensurate amount of heat rejection out of the cycle, the refrigerant would continue to become hotter. To account for this heat input to the refrigerant, and to achieve the appropriate superheat at the inlet to the compressor, the ratio of mass flow of the refrigerant through the condensing line and through the hot-gas bypass line is adjusted by actuating the respective valves. Because refrigerant in the bypass line is higher temperature vapor, opening the hot-gas bypass valves will increase the refrigerant temperature into the compressor. Conversely, when more refrigerant is routed through the liquid (condenser) line, the compressor inlet temperature is decreased. The relative ratio of these two mass flows of different enthalpies determines refrigerant enthalpy at the compressor inlet. This balance is used to achieve the desired superheat at the compressor inlet and also determines the total mass flow rate of the system.

After the HGB loop reaches the desired operating condition, the solenoid valves within the test section are used to switch between the heating and cooling modes to perform a cycling test. The heating condition is achieved by connecting the test sections with compressor discharge tube to flow the hot gas into the section before going into the condenser by opening SV-01, SV-02 and SV-03, as shown in Figure 2(b). After finishing the heating condition, SV-02 and SV-03 close and disconnect the test sections from the compressor discharge. The cooling condition starts by turning SV-04 and SV-05 on to flow the mixed cold gas into the test section, as shown in Figure 2(c). The mixed cold flow will then go back to the compressor to close the loop. The next heating condition follows the cooling condition. The cycle is repeated by switching the valves alternately. In both heating and cooling modes, the HGB loop can operate stably and continuously due to the relatively small thermal capacity from the test sections.

3.2 Test Section and Conditions

There are two identical test sections installed in the test stand. At the inlet and outlet of each test section, connectors are installed to ease the connection of new samples into and out of the test stand. In the current study, in order to have a reference sample for evaluation of the fatigue resistance of the adhesive joints, brazed joints are tested in one of the test sections. The adhesive joints in the other test section are manufactured and assembled by 3M with a pre-machined fixture using a toughened, two-part epoxy structural adhesive. The brazed joints are made at Purdue University by a qualified technician. U-tube joints are selected as our testing samples due to the large number of them in the tube-and-fin heat exchangers. The U-joint has a tube diameter of 0.375

mm (3/8) inch based on industry recommendation. The center-to-center distance between the two tubes is 25.40 mm (1 inch).

A 3D model for a test section is shown in Figure 3. Each test section has a 10 U-bends in series (20 total joints) that are connected by straight tubes that are clamped down in a fixture plate. The lengths of the tubes are kept as short as possible (50.8 mm) to minimize the heating/cooling time (i.e., thermal capacity) of the test section, and also to decrease the amount of hot/cold gas trapped in the test section when switching between the heating and cooling condition. All the test sections are leak-checked using water-immersion method with nitrogen charged to ~1700 kPa (250 psi) before testing to ensure that they were leak-free.

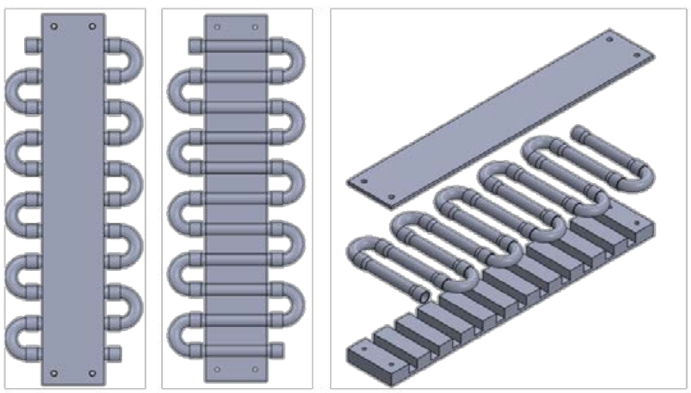

Figure 3: 3D model of the test section with bonded U-bends and fixture plate.

The pressure and temperature test conditions for the test section of the hot-gas bypass test stand is mainly determined by the selected compressor and valve openings. Based on the test stand size and capacity, a 2.5-ton single stage reciprocating compressor designed for R410A is used. Based on the data and compressor map provided by the manufacturer, the possible test conditions are calculated using Engineering Equation Solver (EES) assuming no pressure drop. The results are summarized in Table 1. These calculations assume operation with a 20 K superheat at the compressor inlet.

Table 1: Range of possible operating conditions of the hot-gas bypass test stand for the selected compressor.

$T_{suction}$(°F/°C)	$T_{discharge}$(°F/°C)	$P_{suction}$(kPa)	P_{cond}(kPa)
10/-12.2	263/128.3	351.6	2297.3
30/-1.1	238.1/114.5	529.9	2798.6
50/10	229.7/109.8	770.1	3388.8
60/15.6	234.9/112.7	917.7	3834.9
60/15.6	246.7/117.9	917.7	4074.8

The specific set points during testing are decided from this possible range of operation of the system so as to understand the fatigue caused by temperature and pressure variations. The discharge temperature and suction temperature are the most extreme possible temperatures that the test section can experience. The lowest temperature should be above the freezing point of water to avoid freeze/thaw cycles that can potentially have other unintended influences on the fatigue failure. Also, the highest temperature is limited by the compressor discharge overheat protection, which shuts down the compressor when the temperature is too high. Based on this limit, system temperatures typically do not exceed 100 ℃ in vapor compression cycles with common refrigerants. Thus, the operating condition in the third row of Table 1 can be a potential test condition for the given test stand, which thereby will cycle the test section (at maximum) between 10 ℃ and 109 ℃. However, note that the actual temperature swing will be smaller than this due to the heat transfer between the environment and tubes leading to the test section. Ultimately, the test condition temperature swings should be decided by the application. During the test, the condensing temperature, suction temperature, subcooling, and superheat are adjusted to achieve this desired testing condition.

3.3 Cycling Procedure and Instrumentation

In order to achieve the designed cyclic function described in Section 3.1, the control algorithm for one single cycle is described below and summarized in Table 2. The steps are repeated to carry out the pressure and temperature cycling test.

Because the solenoid valves in the test sections are operating directly between the high-side and low-side pressures, there is a high pressure difference across the valves. Solenoid valves designed for R410A are selected due to their high maximum operating pressure difference, selecting the appropriate valves to be compatible with the suction and discharge tube sizes. The coils are selected to give the largest maximum operating pressure difference of 31 bar (450 psi), which satisfies the test stand requirements. Preliminary testing found that the solenoid valves would not always fully close fully in the presence of a high pressure opposite to the flow direction. In order to ensure consistent full closure of the solenoid valves and to avoid backflow when switching between the test modes, two additional check valves are installed in the test section lines (CV 01 & 02).

The refrigerant temperature and pressure in the test sections are monitored and recorded. Pressure transducers (PT 01 & 06) are installed at the inlet of the test section. The temperatures are measured with two thermocouples (TC A1 & A2 and TC B1 & B2), one placed at the test section inlet and the other one at the outlet. The HGB test stand is equipped with several additional T-type thermocouples to measure the temperature at various locations, gauge pressure transducers to measure the pressures, and a power meter to measure the compressor power consumption. A Coriolis-effect mass flow meter is used to measure the refrigerant vapor mass flow rate after the oil separator. The sensor specifications are listed in Table 3.

Table 2: On/off status of solenoid valves under different test conditions

	Non-Testing Mode	**Heating Mode**	**Cooling Mode**
SV-01	ON	OFF	ON
SV-02	ON	ON	OFF
SV-03	OFF	ON	OFF
SV-04	OFF	ON	OFF
SV-05	OFF	OFF	ON
SV-06	OFF	OFF	ON

Table 3: Sensor specification for the hot-gas bypass test stand

Sensor	**Model**	**Range**	**Accuracy**
Thermocouple	Omega TMQSS-125T-6	0 – 350 ℃	1.0 ℃ OR 0.75% Full Scale
Pressure Transducer	Honeywell PX2AF1XX500PAAAX	0 – 500 psia	0.25 %
Pressure Transducer	Omega PX176-1KS5V	0 – 1000 psig	1.0 % Full Scale
Power Meter	Ohio Semitronics GW059-EG	0-20 kW	0.04% Full Scale
Mass Flow Meter	MicroMotion CMF050	0-0.6055 kg s^{-1}	0.5%

4. TEST STAND PERFORMANCE

After test stand construction and leakage testing, multiple tests with different operating conditions were carried out to confirm the operational capabilities of the test stand. The first testing operated the HGB flow loop at steady state, without engaging the test section, to confirm that it could achieve the designed range of testing conditions. At steady state, the temperature can be controlled to achieve as low as ~0 ℃ at the compressor inlet and as high as 110 ℃ at the compressor outlet, with respective inlet and outlet pressures of ~350 kPa and ~4600 kPa. Note that in the HGB test stand, these pressures are intrinsically coupled to the temperature ranges based on the choice of the refrigerant. After confirmation of this functionality, the cyclic testing capability of the facility is evaluated following the procedure described in Section 3.3. Figure 4 shows the representative transient temperature profiles during several cycles. The figure shows the temperature cycles at the inlet and outlet of the test section (T_{in}, T_{out}), along with the discharge (T_{dis}) and suction (T_{suc}) temperature of the compressor that are intended to be kept steady. In this figure, the compressor discharge state is controlled at ~70 ℃ and ~3750 kPa, while the suction state is at ~ 3 ℃ and ~700 kPa. In this testing, the compressor discharge and suction temperature stays stable except during mode switching. The temperature of the test section oscillates when mode switching happens approximately at 800 s, 1000 s, and 1300 s, as indicated by the vertical dashed lines in Figure 4. The heating process raises the test section from ~10 ℃ to ~50 ℃ in ~120 s, while the following cooling process takes approximately 80 seconds. The hot-gas bypass test stand ensures that the duration of the heating and cooling parts of the cycle are determined only by the thermal mass of the test section itself. The thermal dynamics of the transition is not as crucial as the difference between the maximum and minimum temperatures reached at steady state. However, a very rapid temperature change could play a secondary role in the fatigue failure. However, from a perspective of minimizing the test time to enable accelerated failure testing, the goal was to minimize the transition time. Following the standards introduced in Figure 1, these lengths of time satisfy the minimum two-minute dwell period at the desired temperature. One full cycle period is approximately 7 minutes, with a 10 - 50 ℃ temperature swing; the 50-cycle test as required by the standard then takes approximately 6 hours to complete.

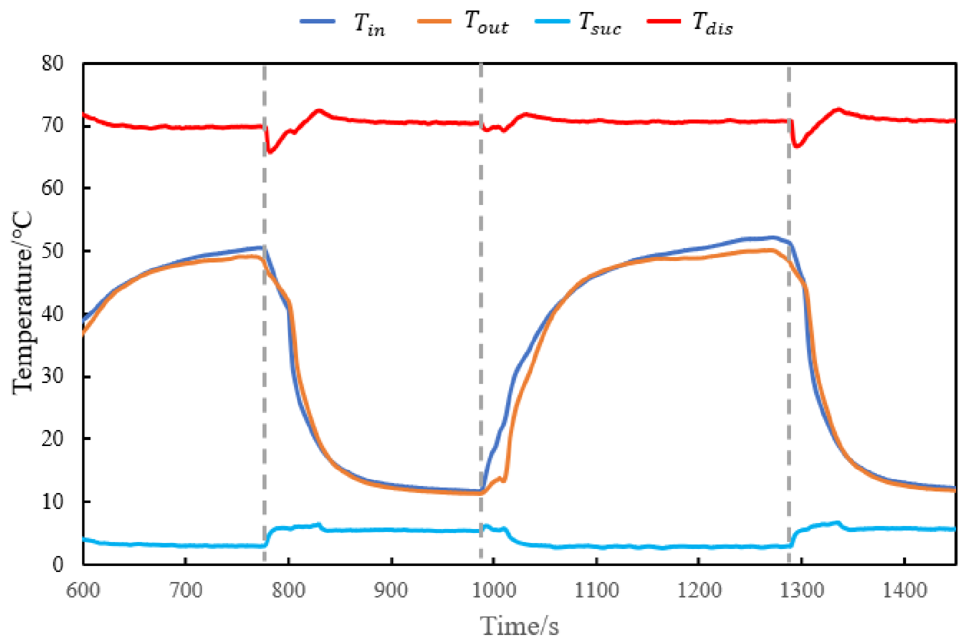

Figure 4: Temperature of the compressor inlet and outlet as well as the temperature of the inlet and outlet of the test section for one complete cycle

There is a slight difference between the test section inlet and outlet temperatures when the test section is at a relatively high temperature, which is due to heat losses to the environment from the test section. With two identical test sections with 10 U-joints each, only ~1℃ of difference was observed during the testing. The system is capable of testing increased number of joints but this may require a slightly longer waiting time, as increasing the number of joints also increases the thermal mass. Further, the dwell time is considered complete only once the last joint reached desired temperature. If the thermal mass is excessively large, the system will not reach steady state with frequent switching due to the unbalanced heating and cooling capacity. In this case, an additional heat exchanger will be needed as done in Wilson and Bowers (2014) or the hot-gas-bypass mass flow rate could be changed accordingly in heating and cooling mode to balance the needed heat transfer.

Also, there are temperature fluctuations in the suction and discharge lines each time the test stand switches between modes, which is caused by the release of hot/cold refrigerant from the test section into the system. It is noteworthy to point out the temperature difference between the hot/cold gas lines in the system and the test section temperatures. For example, in Figure 4, the refrigerant temperature at the compressor discharge is ~20 ℃ higher than the maximum test section reached at the end of the heating mode. Likewise, in the cooling mode, the refrigerant at suction of the compressor is ~10 ℃ lower than that at the test section. The major reason for this temperature difference is the heat transfer to the environment along the refrigerant lines. Although all the tubes and components are well-insulated, there are still heat transfer due to the large temperature difference with respect to ambient, especially from the compressor discharge to the test section, where the refrigerant must travel through the oil separator and addition tube lengths. The highest test section temperature that can be achieved in the current facility is 80 ℃ with a compressor discharge temperature at 105 ℃.

The discharge temperature used in this study is not very high considering applications such as high-temperature heat pumps. However, the setup can still provide large temperature swings on the joints, larger than those experienced in many practical systems. In addition, this is not the limit of the test stand, which can be operated at more than 100 °C at the compressor discharge as discussed in Table 1, but rather a demonstration of one of the tests performed with the test stand. The lowest temperature achieved is -5 ℃ in the test section but it can go much lower with decreasing the suction pressure. However, as discussed previously, the low temperature is maintained above 0 ℃ to avoid freezing and thawing of the joints.

After demonstrating temperature cycling of the test section and determining the cycle time, a full 50-cycle test was performed following the test standard. The test stand performs as designed, as indicated by the results in Figure 5, which shows the temperature and pressure of the compressor inlet (T_{dis}, P_{dis}) and outlet (T_{suc}, P_{suc}) as well as the temperature of the inlet (T_{in}) and outlet (T_{out}) of the test section. The total testing time required for all cycles is only 5 hrs. The average discharge temperature of the compressor is 56.2 °C at a pressure of 3364 kPa. The average suction temperature and pressure are 3.6 °C and 771 kPa, respectively. In the test section, the highest temperature measured during the cycling is 43.4 °C and the lowest temperature is 9.2 °C. During the testing, a time-periodic fluctuation in the suction/discharge temperatures and pressures can be observed due to the switching of the solenoid valves.

Before and after the 50-cycle PTC test, both the brazed joints and adhesive joints in the test sections are leaked checked using the water-immersion bubble test method as suggested by ISO 14903. No bubbles were observed to leave the test subjects throughout the duration of test, suggesting that the leakage rate is not measurable using this method. Tracer gas testing would be required to measure the actual leak rate per fitting per year.

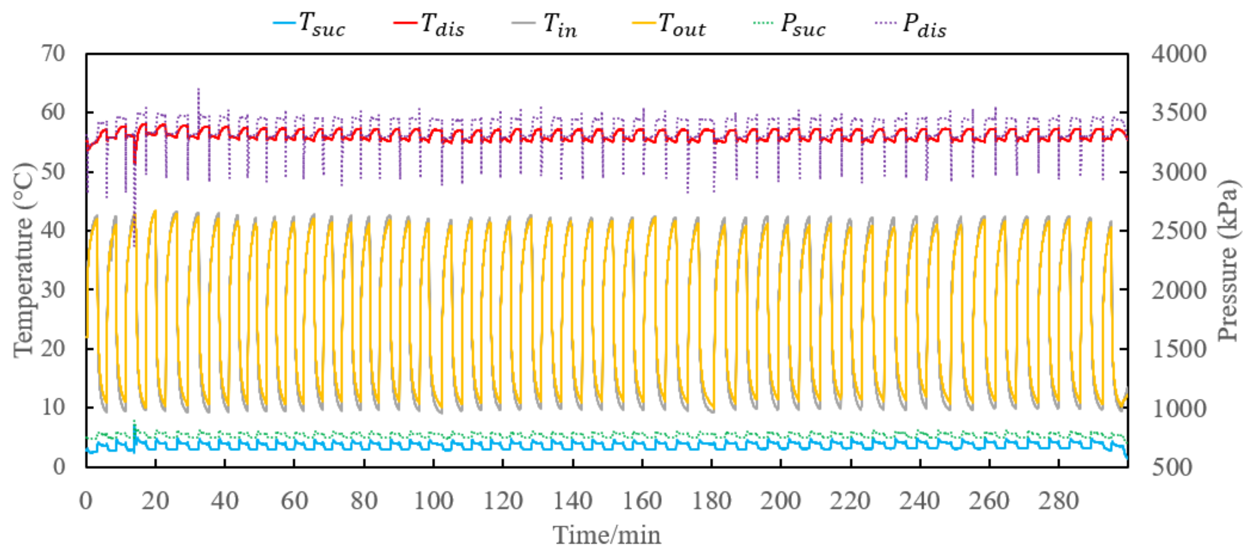

Figure 5: Temperature and pressure of the compressor inlet and outlet as well as the temperature of the inlet and outlet of the test section

It is worth noting that the cycling schedule as specified by the available test standards does not reflect the actual loading of systems under normal operation. The failure under normal operating conditions is a complicated phenomenon and there are currently no general guidelines or predictive methods available for correlating accelerated failure testing to practical system lifetimes. While the focus of this work is not to speculate on the correlation between the test standard and practical operation, one cycle of the PTC test stand might correspond to one on/off cycle in a real system. In the test condition shown in Figure 5, 1 hr of testing contains ~10 cycles, which corresponds to 10 on/off cycle in a real system. Depending on the type of system, this can be equivalent to different number of hours of real system operation. A domestic refrigerator may turn on and off several times in one day while a chiller may run 24/7 without switching between on and off state. Using the current test stand, it is possible to create the cyclic operating conditions according to an established standard and enable researchers to acquire data of this kind in order to better correlate lab testing to failure under normal conditions as future work.

5. CONCLUSIONS

In order to design and evaluate new joining technologies that meet the requirements for the HVAC&R industry, it is important to developing testing methods to evaluate the influence of pressure and temperature cycling on fatigue of the joints. An innovative test stand is designed, built, and demonstrated to simulate the pressure and temperature cycling of joints in an HVAC&R system in an accelerated manner. The test stand applies hot-gas bypass control to reduce the cost and time of cycling operation. The test stand is shown to successfully perform the pressure and temperature cycling test as designed for the assessment and evaluation of adhesive joints, while satisfying the ISO standard 14903. Testing confirms that a full cycle of heating and cooling the test sections can be finished in a period 7 min with controlled temperature and pressure oscillations. In a demonstration of the technique, a 50-cycle PTC test was performed with temperature cycles from ~10 ℃ to 40 ℃ and finished in 5 hr. Both the brazed joints and adhesive joints installed in the test sections were confirmed to be leak-free after the testing, which indicates that adhesive joints have thermal fatigue resistance under the given testing condition. A suggested future modification based on the current results is to reduce the heat loss from the test stand lines by using better insulation and reducing the tube length, which would allow evaluation of the joints up to higher maximum temperature. Solid state relays can be easily used to control the solenoid valves in the test stand for long-term testing of joints using this facility. The test stand developed in this work serves as new research infrastructure for the pressure and temperature fatigue testing of joints and allows for research and development of new joining technologies and other components for HVAC&R applications.

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