Accelerated Degradation Test on Electric Scroll Compressor Using Controlled Continuous Liquid Slugging

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ABSTRACT

Refrigerant-based electric scroll compressors are often used in heat pump systems due to their ability to efficiently handle varying levels of load conditions, both for heating and cooling modes of operation. As electric compressors are considered the heart of the heat pump system, being able to determine their degradation prior to failure is of paramount importance for the health of the system. This paper provides a new testing methodology that introduces liquid slugging to degrade a compressor to the end of its useful life in an accelerated manner. The fault injection configuration consists of a modified heat pump system with the addition of two low pressure heat exchangers added in parallel to control the refrigerant quality during compressor operations. For a given refrigerant quality, the heat pump system was operated at a fixed compressor performance conditions to sustain liquid slugging for a fixed duration. Afterwards, refrigerant was controlled to be pure vapor at the compressor suction side and the compressor was controlled at several different performance conditions (i.e., fixed compressor suction superheat temperature and compressor pressure ratios, at various compressor speeds). The results show that symptoms of compressor degradation can be detected from: i) increased compressor discharge temperature, ii) increased power consumption, iii) decreased refrigerant mass flow rate, iv) decreased volumetric efficiency, and v) decreased isentropic efficiency. Teardown of the compressor and inspection of its internal components further confirmed that the methodology presented is able to produce compressor that is representative of an end-of-useful-life part.

1. INTRODUCTION

Vapor compression heat pump system is considered the preferred climate control solution for electric vehicles to improve driving range. Heat pump can supply both cooling and heating capacities with high energy consumption efficiency (Qi, 2014; Zhang, et al., 2018). Among the different parts that make up the vehicle heat pump system, the compressor is considered the most important component (Aurich & Baumgart, 2018; Staino, Abou-Eid, Rojas, Fitzgerald, & Basu, 2019).

The most widely used compressor type in battery and fuel cell vehicles is the electric scroll compressor (Aurich & Baumgart, 2018). This compressor type is driven by brushless motor, which is powered by the inverter that converts DC power from the high voltage battery or fuel cell stack to AC power. Unlike their belt driven counterparts, electric scroll compressors can operate at a range of rotational speeds, independent of engine and vehicle speeds, by varying the command to the DC/AC inverter current (Cuevas, Fonseca, & Lemort, 2012). The ability to control the speed allows electric scroll compressors to handle varying levels of load conditions in heat pump systems, both for heating and cooling modes of operation (Zhang, et al., 2018).

Due to fewer moving parts, electrically driven scroll compressors are known for their high reliability and quiet operation (Cosman, et al., 2020). They normally have smaller swept volumes and lower weights, which allow them to offer superior efficiency at high rotational speed (Aurich & Baumgart, 2018). These advantages make electric scroll compressors preferable for use in electric vehicle climate control systems as compared to other types of compressors such as the axial piston compressors widely used in conventional internal combustion vehicles (Cuevas, Fonseca, & Lemort, 2012).

As electric scroll compressor is considered the heart of the heat pump system, being able to determine degradation of the compressor prior to failure is of paramount importance to ensure that the system continues to fully perform its required functions. A prognostics algorithm can be developed to identify early signs of degradation that may lead to significant performance reduction and system failure, should corrective measures be not taken proactively (Barandier & Cardoso, 2023).

Typical failures for electric scroll compressors range from electrical faults, refrigerant leaks, to mechanical failures and overheating. Specifically, one of the primary failure modes for electric scroll compressors is mechanical damage including excessive wear, cracks, and fractures. Moreover, the main cause of mechanical damage in compressors is the high stress effects of refrigerant liquid slugging (Barandier & Cardoso, 2023). The presence of liquid refrigerant in the compression chamber is inevitable in some situations such as cold startup and defrosting process (Lin, et al., 2022). Compressing liquid refrigerant at the suction side of the compressor can result in stresses on the compressor scrolls due to the excessively high internal pressures. Moreover, the presence of liquid in the compressor can disrupt the oil recirculation and cause temporary loss of lubrication (Barandier & Cardoso, 2023).

To design an effective prognostics algorithm, behavior of the system and its components across the range of the state of health is usually characterized. This requires physical data measurements from the system ranging from healthy to degraded states. However, the time required for climate control system components, including compressors, to reach degraded states under typical field usage is excessively long (Staino, Abou-Eid, Rojas, Fitzgerald, & Basu, 2019). The standard reliability determination test, which is typically used to estimate product life under various accelerated test conditions, also requires a long test time (Chang, et al., 2016).

This paper presents a new testing methodology to degrade electric scroll compressor in an accelerated manner to the end of its useful life using controlled continuous liquid slugging at various degrees of refrigerant quality. Performance of the compressor is characterized at several breakpoints during the test to determine progression of the degradation over the duration of liquid slugging.

2. METHODS

The accelerated degradation test presented in this paper was conducted on a fault injection test bench in a laboratory-controlled environment. Note that the test bench used is not representative of the actual heat pump system on the vehicles.

In this section, description of the test bench and procedure of the accelerated degradation test are presented. Metrics that are used to evaluate performance of the compressor throughout the test are also described in detail.

2.1. Test Bench

Figure 1 shows the schematics of the test bench. The test bench consists of an electric scroll compressor with pressures and temperatures measured at the suction and discharge ports. To control the refrigerant quality at compressor inlet, the flow of liquid refrigerant downstream the condenser can be split into an evaporator and low-pressure heat exchanger, each with separate electronically controlled expansion valves and mass flow meters. Pressure and temperature measurements are taken at the inlets of the expansion valves and the outlet of the evaporator. To control the refrigerant to be pure vapor at compressor inlet, liquid refrigerant flow can be directed only to the evaporator line to ensure that the desired positive compressor inlet superheat temperature is achieved.

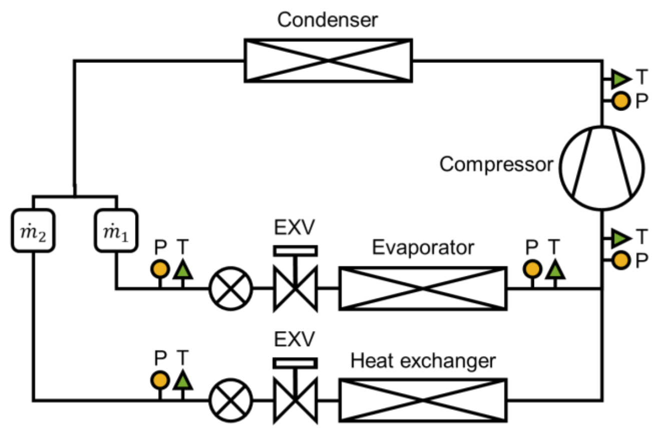

Figure 1. Schematics of the test bench.

The refrigerant fluid used in the test is R1234yf. The physical data measured from the bench during the test are listed in Table 1. Pressures and temperatures of the refrigerant at different points in the refrigeration circuit, and several variables including compressor speed, motor voltage and current, and refrigerant mass flow rate are measured using various instrumentation sensors on the bench.

Table 1. Physical data measured during the test.

Variable name	Unit	Description
P_{suc}	kPa	Compressor suction pressure
T_{suc}	℃	Compressor suction temperature
P_{dis}	kPa	Compressor discharge pressure
T_{dis}	℃	Compressor discharge temperature
n_{comp}	rpm	Compressor speed
V_{comp}	V	Compressor motor voltage
I_{comp}	A	Compressor motor current
$\dot{m}_{ref}$	kg/h	Refrigerant mass flow rate
$P_{eva,h}$	kPa	Evaporator expansion valve inlet pressure
$T_{eva,h}$	℃	Evaporator expansion valve inlet temperature

$P_{eva,l}$	kPa	Evaporator outlet pressure
$T_{eva,l}$	℃	Evaporator outlet temperature
$P_{hx,h}$	kPa	Heat exchanger expansion valve inlet pressure
$T_{hx,h}$	℃	Heat exchanger expansion valve inlet temperature

2.2. Test Procedure

The procedure for the compressor accelerated degradation test is made up of two phases. In the first phase, the refrigerant quality at which the test bench would operate to create accelerated degradation condition was determined. In the second phase, for the fixed refrigerant quality and operating condition determined in the first phase, the heat pump system was operated to sustain continuous liquid slugging for a fixed duration. Then, the system was controlled at a few different steady-state conditions with pure vapor refrigerant at compressor inlet to map the performance of the compressor.

2.2.1. Liquid Slugging Condition Determination

In the first phase of the test, the refrigerant quality at compressor inlet that would create accelerated degradation condition at a particular compressor speed and pressure operating condition was determined. Various degrees of refrigerant quality were selected based on the capability of the test bench. The operating conditions were run in the order of severity with respect to the refrigerant quality at compressor inlet from mild to severe and are shown in Table 2. Each of the liquid slugging operating conditions was run continuously for 10 hours or until signs of compressor failure were observed during the test.

Table 2. Liquid slugging operating conditions.

Condition No.	n_{comp} (rpm)	P_{suc} (kPa)	P_{dis} (kPa)	Quality $(-)$
1	6000	550	2500	0.9
2	6000	550	2500	0.8
3	6000	550	2500	0.7
4	6000	550	2500	0.65
5	6000	550	2500	0.6

After running the compressor at the operating conditions specified in Table 2, audible metal-to-metal noise started to occur at the fifth operating condition. Scroll failure was confirmed following the teardown of the compressor. Wear marks are apparent on the fixed and orbiting scroll surfaces, and fracture occurred on the orbiting scroll. Consequently, refrigerant quality of 0.6 at compressor speed of 6000 rpm and compressor suction and discharge pressures of 550 and 2500 kPa was selected as the continuous liquid slugging condition to degrade the compressor in the second phase of the test.

2.2.2. Continuous Liquid Slugging and Performance Mapping

The second phase of the accelerated degradation test comprises cycles of continuous liquid slugging and performance mapping as shown in Figure 2. The liquid slugging operating condition, which was determined in the first phase, was sustained continuously for a fixed duration of 2.5 or 5 hours in each cycle to degrade the compressor.

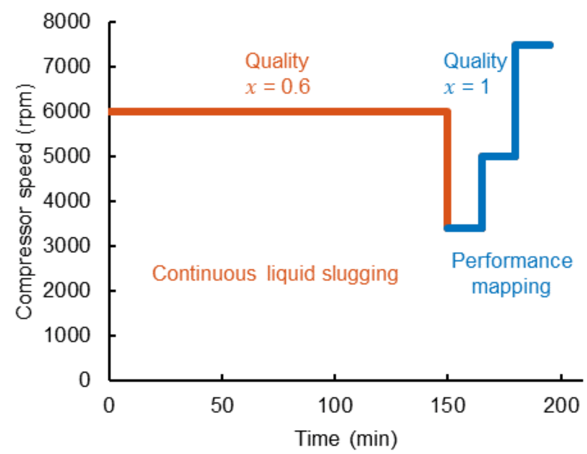

Figure 2. Continuous liquid slugging and performance mapping.

Afterwards, refrigerant was controlled to be pure vapor at the compressor suction side and the compressor was controlled at several different performance conditions (i.e., fixed compressor pressure ratio and compressor suction superheat temperature at various compressor speeds), so as to duplicate conditions known to us from the compressor component data sheet for an ideal electric scroll compressor. The steady-state operating conditions for performance mapping are shown in Table 3.

The liquid slugging operating condition was run for a total duration of 45 hours. Twelve mapping points were run at several breakpoints throughout the test to determine the performance of the compressor at the end of each continuous liquid slugging cycle.

Table 3. Performance mapping steady-state points.

Condition	n_{comp} (rpm)	P_{suc} (kPa)	P_{dis} (kPa)	$T_{suc,SH}$ (℃)
A	3400	325	1900	5
B	5000	325	1900	5
C	7500	325	1900	5

2.3. Compressor Performance Metrics

The performance metrics mapped at breakpoints comprise quantities that are obtained by direct measurements using instrumentation on the test bench and calculated values. Measured performance metrics include compressor discharge temperature (T_{dis}), power consumption $(\dot{W}_{comp})$, and refrigerant mass flow rate $(\dot{m}_{ref})$. The refrigerant mass flow rate and compressor power consumption are normalized with respect to their values at time equal to zero.

Calculated performance metrics include volumetric (η_{vol}) and isentropic efficiencies (η_{is}). Volumetric efficiency is a measure of how efficient the scroll compressor chamber compresses the refrigerant, whereas isentropic efficiency is the ratio between the ideal isentropic work and the actual work going into the compressor (Staino, Abou-Eid, Rojas, Fitzgerald, & Basu, 2019).

Volumetric efficiency is given by Eq. (1), while isentropic efficiency is given by Eq. (2). $\dot{V}_{disp}$ is the volumetric flow rate of the refrigerant, and ρ_{suc} is the compressor inlet density. $h_{is,dis}$ is the compressor outlet isentropic enthalpy, h_{suc} is the compressor inlet enthalpy, and h_{dis} is the compressor outlet enthalpy.

$$\eta_{vol} = \frac{\dot{m}_{ref}}{\dot{V}_{disp} \cdot \rho_{suc}} \tag{1}$$

$$\eta_{is} = \frac{h_{is,dis} - h_{suc}}{h_{dis} - h_{suc}} \tag{2}$$

3. RESULTS AND DISCUSSION

Figure 3 shows compressor discharge temperatures at the performance mapping points. It can be observed that the temperatures start to increase after 32.5 hours of continuous liquid slugging. The compressor discharge temperature increases from 86.5 ± 1.5 °C to 92.8 ± 1.0 °C at operating condition A, from 84.5 ± 1.3 °C to 89.5 ± 1.1 °C at operating condition B, and from 84.7 ± 1.7 °C to 90.0 ± 1.4 °C at operating condition C.

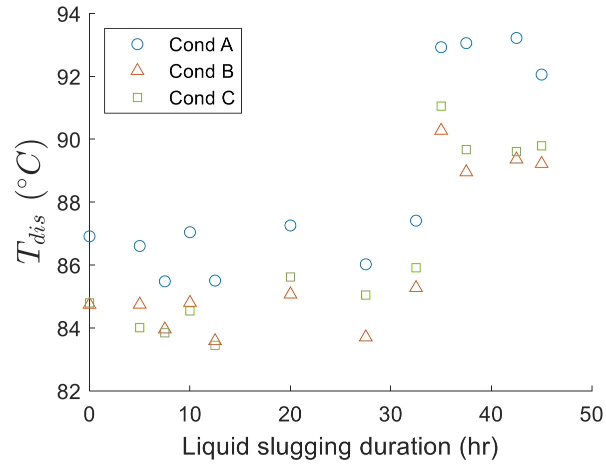

Figure 3. Compressor discharge temperature.

Compressor power consumptions, normalized with respect to their initial values, are shown in Figure 4. Similar to discharge temperature, compressor power consumption increases as degradation level of the compressor becomes more severe. After 37.5 hours of continuous liquid slugging, normalized compressor power consumptions increase to 1.05 ± 0.02, 1.02 ± 0.00, and 1.02 ± 0.01 for operating conditions A, B, and C, respectively.

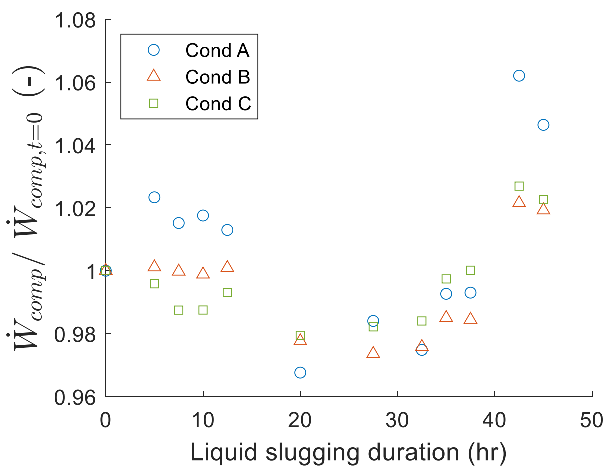

Figure 4. Normalized compressor power consumption.

Figure 5 shows refrigerant mass flow rates at the performance mapping points, normalized with respect to their values at time equal to zero. Opposite to the trends observed in compressor discharge temperature and power consumption, refrigerant mass flow rate tends to decrease slightly with increased degradation level. Following 32.5 hours of continuous liquid slugging, normalized refrigerant mass flow rates decrease to 0.969 ± 0.033 and 0.977 ± 0.018 for operating conditions B and C, respectively.

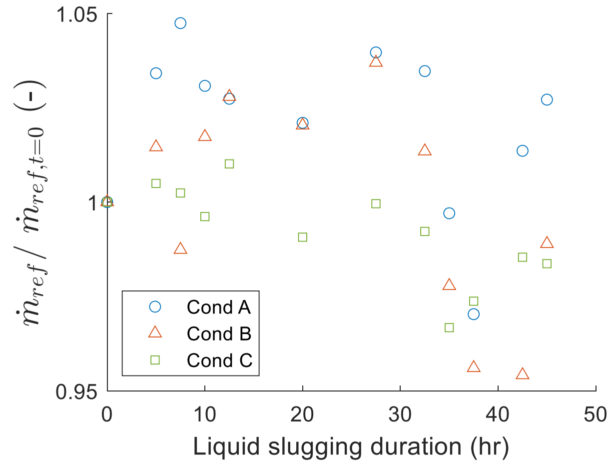

Figure 5. Normalized refrigerant mass flow rate.

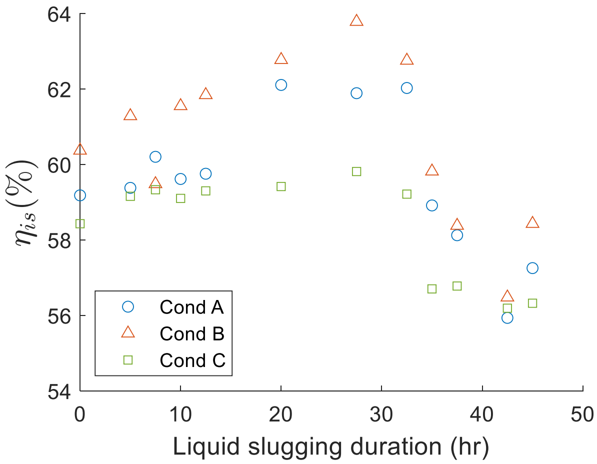

Figure 7. Compressor isentropic efficiency.

Compressor volumetric and isentropic efficiencies are shown in Figure 6 and Figure 7. From the figures, it can be observed that both volumetric and isentropic efficiencies start to drop after 32.5 hours of continuous liquid slugging. Volumetric efficiency decreases by 2.8%, 4.4%, and 2.7% for operating conditions A, B, and C, respectively. Isentropic efficiency undergoes losses of 3.0%, 3.5%, and 2.7% for operating conditions A, B, and C, respectively.

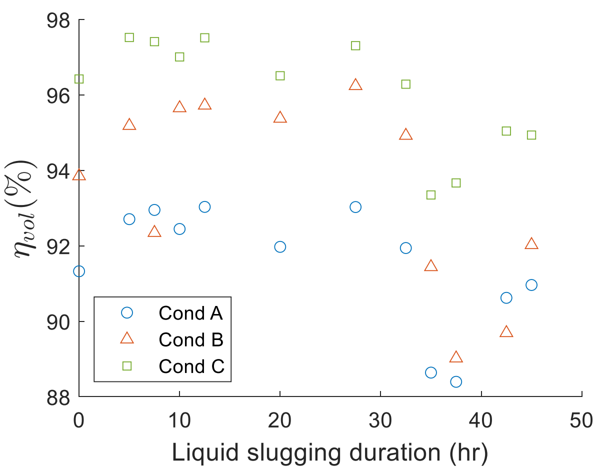

Figure 6. Compressor volumetric efficiency.

To determine the actual state of health of the compressor at the end of the continuous liquid slugging degradation test, a complete teardown was performed. Darkening or blackening is observed on the discharge chamber in the rear housing of the compressor. This is likely due to the high compressor discharge temperature during operation. Signs of heavy scuffing wear are observed on both the fixed and orbiting scroll surfaces, likely due to poor lubrication during operation. Coating material on a large portion of the orbiting scroll surface was also worn off, with no coating remaining on some portions of the surface. Based on the results of the teardown inspection, it was determined that the compressor is at the end of its useful life following the degradation test.

To evaluate the effectiveness of the methodology presented in this paper, further studies will be conducted to characterize the performance of the compressor. The measured and calculated performance metrics discussed in the previous section can be utilized to construct health indicators that will be used in prognostics algorithms to estimate the state of health of the compressor at the various steady-state performance mapping points throughout the degradation test.

4. CONCLUSION

In this paper, a new testing methodology was presented to degrade electric scroll compressor in an accelerated manner to the end of its useful life using controlled continuous liquid slugging. When compared to other approaches, this method significantly reduces the time required for the accelerated aging test to bring compressor to a degraded state. The test bench employs an evaporator and low-pressure heat exchanger working concurrently in parallel to control the refrigerant quality at compressor inlet. Compressor performance mapping was performed at 12 breakpoints throughout the degradation test to assess the symptoms of degradation.

The results show that compressor discharge temperature and power consumption increase as compressor degradation level becomes more severe. In contrary, refrigerant mass flow rate tends to decrease slightly with increased degradation level. Following the same trend, volumetric and isentropic efficiencies drop as the compressor becomes more degraded. These measured signals and calculated metrics can be utilized in further research to construct health indicators that can be used in the development prognostics algorithms to estimate the state of health of the compressor.

The actual state of health of the compressor at the end of the accelerated degradation test was determined by a complete teardown. Inspection of the internal components show that the compressor tested is representative of part that has reached its end of useful life.

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