# Disassembly of Off the Shelf Reciprocating Compressor After Employment in High-Temperature Heat Pump for More Than 1000 Operating Hours

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## ABSTRACT

High-temperature heat pumps (HTHP) will be an important technology to replace industrial heating in the temperature range from 100 to 200 °C, which is currently mainly provided by oil and gas burners. The lifetime of compressors and lubricating oil at high suction and discharge temperatures is unclear, yet crucial to understand for these applications. For this study, a reciprocating compressor was disassembled after approximately 1,000 hours of operation in a lab-scale HTHP. At least 300 hours of operation showed discharge temperatures over 120 °C and 30 hours of more than 150 °C. The cylinder head seal was damaged, colorization of the valve plate was observed, and small wear was found, possibly because of a low oil viscosity. All other parts of the compressor were unaffected, although the compressor was not specifically designed for operation at such high temperatures. Various oil measurements did not show significant deviations from the fresh oil. The study reveals that the reciprocating compressor and the oil were more resilient to the high temperatures than expected. However, more runtime is needed to gain confidence in operation in a real plant over several years.

**Keywords:** high temperature, heat pump, compressor, oil, degradation, disassembly

## 1. Introduction

The use of high-temperature heat pumps (HTHP) has risen in Europe and East Asia over the past approx. 5 years. Many developments have happened since Arpagaus et al. (2018b) described the state-of-the-art and market potential. Arpagaus (2023) listed over 30 heat pump manufacturers that offer products supplying heat at more than 100 °C. Given the typically large-scale industrial applications, manufacturers rarely choose rotary and scroll compressors. Instead, screw, reciprocating, and turbo compressors dominate the market. Reciprocating compressors have a share of approx. 50% in the list presented in Arpagaus (2023).

The developments or adaptions of the companies to their products for high-temperature applications are mostly confidential. Jürgensen (2022) gave a few glimpses into their company's development of high-temperature screw compressors. About 25 parts had to be checked or changed as the company overhauled its screw compressors to push for HTHP maturity. It was also emphasized that many experiments, visual inspections, oil tests, etc., are needed to gain the experience and confidence for a successful market launch of a product in the new application range of HTHPs. Bamigbetan et al. (2019) contributed such tests, where an off-the-shelf piston compressor was modified to endure higher temperature applications. An external manifold was crafted for the discharge gas to reduce temperature gradients across the shell, and a higher-capacity motor was chosen. With the discharge temperature limited to 140 °C, the test results were satisfying, and no major obstacles were encountered. Unfortunately, a report of a compressor inspection after operation was not included in the study.

Sun-Hee et al. (2016) tested a screw compressor at high temperatures for approximately 1000 hours. Changes in the isentropic efficiency were observed but were likely attributed to unintentional changes of the operating conditions. The study was therefore inconclusive.

To the best knowledge of the authors, this paper is the first in the open literature to discuss the wear and tear of a HTHP compressor based on a visual inspection after approx. 1,000 hours of operation under demanding temperature conditions. The entire compressor was disassembled by experienced personnel. Special attention was paid to parts or aspects that were predicted to be temperature sensitive, for example, the motor insulation and oil coking. While the motor insulation was unaffected and oil coking did not occur, other parts showed impacts of the high temperatures. The study adds first-hand experiences to the open literature by summarizing and clearly describing all findings.

## 2. Experimental setup

The test object was a reciprocating compressor employed in a HTHP. A suction line accumulator was installed to prevent any liquid from entering the compressor. However, operating conditions with saturated discharge conditions were encountered at times. An oil separator was installed downstream of the compressor with an oil return line into the compressor sump. Suction and discharge line pressure transducers were installed in proximity to the compressor. Thermocouples were of type K and installed on the surface but underneath the insulation of the respective pipes. Vibration dampeners were installed in the suction and discharge lines. **Figure 1** shows the basic schematic and a picture of the HTHP system. The compressor fan was always turned off to achieve a high discharge temperature. More details of the HTHP system are described in Arpagaus et al. (2018a) and Brendel et al. (2023a).

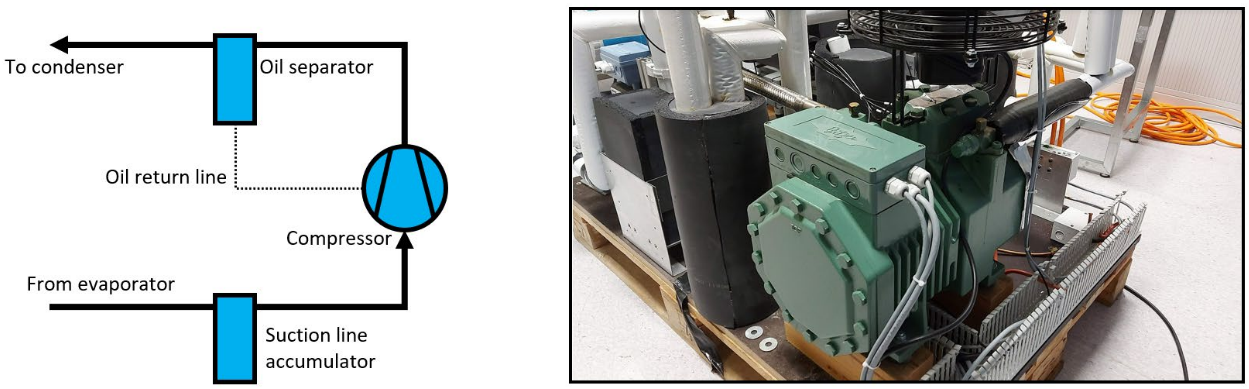


**Figure 1:** Schematic and picture of the reciprocating compressor installed in the lab-scale HTHP system.

## 3. Components and operating conditions

### 3.1 Compressor

The BITZER 2DES-3Y compressor was purchased off-the-shelf and installed unmodified. Auto-shutdown systems from internal temperature measurements were disabled. The compressor was installed from the first experimental paper on the system (Arpagaus et al., 2018a) until July 2022. The only maintenance was replacing the cylinder head sealing after a leak had been detected (see results section). The manufacturer, BITZER (Germany), emphasized that any experiments which exceeding the specified operating conditions could damage the compressor and are not covered by any warranty claims. The specified operating conditions and technical specifications are as follows:

- Model: BITZER 2DES-3Y
- Maximum evaporation/condensation temperature: 25 °C/85 °C (R-1234yf)
- Maximum discharge temperature : 140 °C
- Maximum low/high side pressure: 19 bar/32 bar
- Maximum power input: 4.6 kW
- Maximum operating current: / 8.6 A
- Number of cylinders: 2
- Displacement: 0.15 liter (0.075 liter per cylinder)
- Oil charge: Approx. 1.5 liter
- Weight: 77 kg

### 3.2 Oil

The default oil in the compressor was replaced with Reniso Triton SE 170 according to recommendations of the oil manufacturer. The oil was changed in intervals of 50 to 400 operating hours, often combined with the recovery of refrigerant. The specifications of fresh oil, according to Fuchs (2022), are as follows (note that the viscosity decreases at higher temperatures and as refrigerant is dissolved in the oil:

- Name: Reniso Triton SE 170
- Density at 15 °C: 972 kg/m$^3$
- Kinematic viscosity at 40 °C: 173 mm$^2$/s
- Kinematic viscosity at 100 °C: 17.1 mm$^2$/s
- Pourpoint: -27 °C
- Neutralization number: <0.03 mgKOH/g
- Water content: <50 mg/kg
- Flash point: 260 °C
- Color: 0.0 to 1.0

## 3.3 Total operating hours and operating hours at high temperatures

The collection of data happened in two phases. Phase 1 lasted from 2018 to October 2022. Phase 2 lasted from October 2022 to July 2023. All raw data files from Phase 2 are available with a resolution of the timestamps of one second. Therefore, the operating hours can be evaluated exactly. The data analysis shows that the compressor operated for 600 hours, including proper data collection, system start-up, shake down testing, etc. During the 600 hours, approx. 300 steady--state data points were collected. The data analysis shows that the compressor operated with discharge temperatures >120 °C for 300 hours or 50% of the time. For 18 hours (3%), the discharge temperature was higher than 150 °C. Also, for 18 hours (3%), the suction gas temperature was above 100 °C.

For Phase 1, the raw data files are not completely available anymore such that a detailed analysis cannot be conducted. However, estimating the operating hours based on the number of steady-state data points and considering that the system had slower hydraulic loops in Phase 1, the total operation time from 2018 to October 2022 is estimated to be at least 400 hours.

During Phase 1, the test matrices contained operating conditions at higher temperatures than in Phase 2. For counting the number of hours at high temperatures, it is therefore conservative to assume the same distribution as in Phase 2 (50% at discharge temperatures above 120 °C, 3% at discharge temperatures above 150 °C and suction temperatures above 100 °C. The total count for both phases is then:

- Total operating hours: 1,000 hours
- At discharge temperatures >120 °C: 500 hours
- At discharge temperatures >150 °C: 30 hours
- At suction temperatures >100 °C: 30 hours

Note that the 500 hours of operation above 120 °C include the 30 hours of operation above 150 °C.

The compressor was inverter driven and operated between 30 and 60 Hz. However, most data was collected at 50 Hz. The operating conditions were defined by various parametric system and compressor level studies as described in prior publications (Arpagaus et al., 2022; Arpagaus and Bertsch, 2020; Brendel et al., 2024b, 2024a).

## 3.4 Maximum operating conditions during steady-state

**Table 1** shows the maximum measurements for suction and discharge temperature and pressure, as well as compressor power draw considering all steady-state data points. The pressures are within the design conditions of the specific compressor. Temperatures greatly exceeded the allowed ranges, and compressor power draw exceeded the allowed range slightly. Additionally, the table shows the average values for the measurements across all steady-state datapoints.

**Table 1:** Maximum and average operating conditions.

| Measurement | Unit | Max. at steady-state | Average at steady-state |
|---|---|---|---|
| $T_{suc}$ | [°C] | 131* | 66 |
| $T_{dis}$ | | 188* | 121 |
| $P_{suc}$ | kPa | 922** | 195 |
| $P_{dis}$ | | 3232** | 1064 |
| $\dot{W}_{comp}$ | kW | 5.24 ** | 2.00 |

*The tested refrigerant was R-245fa. **The tested refrigerant was R-1234yf.

## 3.5 Compressor surface temperatures

During Phase 2 (compare with Section 3.3), thermocouples were installed at the surface of the compressor. They were attached to the compressor shell using aluminum tape. The left graphs of **Figure 2** shows temperatures of the oil sump. The thermocouple for the oil sump was attached underneath the compressor, close to the front lid where the oil centrifuge is located. It showed a maximum reading of 86 °C and an average reading of 68 °C for the data from Phase 2. Noteworthy was a much clearer correlation of the sump temperature with the discharge temperature (**Figure 2** bottom left graph) than with the suction temperature (top left). Another thermocouple was installed on the compressor shell in proximity to the cylinders. It showed a maximum reading of 124 °C and an average reading of 93 °C for the data from Phase 2. The correlation is again stronger with the discharge temperature than the suction temperature, which is to be expected since the cylinder temperature is dictated by the discharge temperature.

The reported surface temperatures are from Phase 2. The surface temperatures in Phase 1 were not measured but would have been higher since the maximum discharge temperature reached 188 °C. Extrapolation of the data in **Figure 2** (bottom left and bottom right) allows to estimate the maximum oil sump temperature in Phase 1 to 110 °C and the maximum cylinder temperature in Phase 1 to 150 °C.

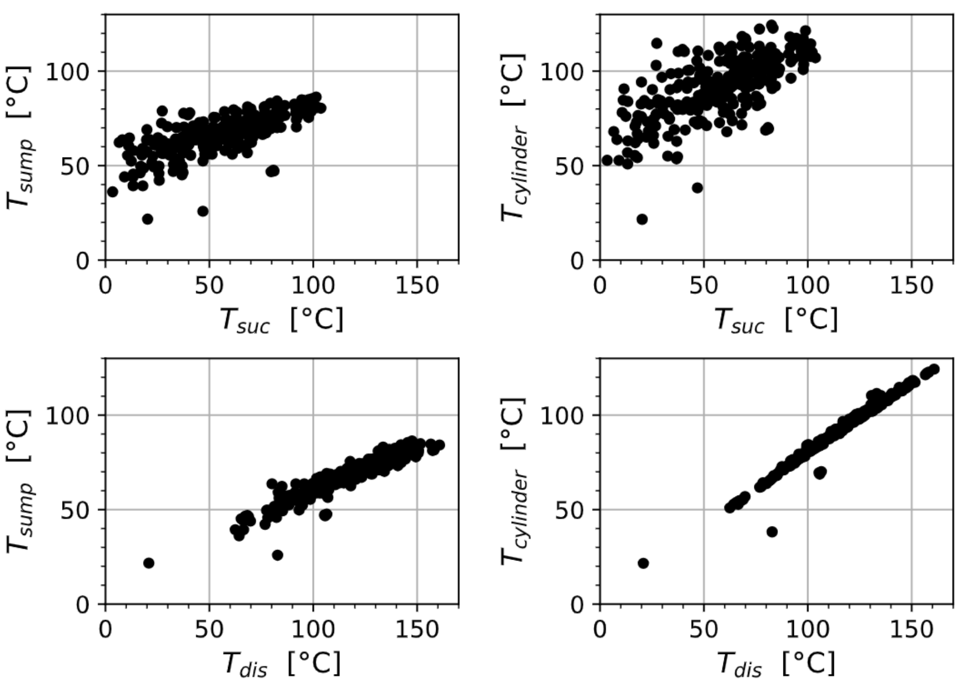


**Figure 2:** Correlation of compressor sump and cylinder temperature (measured on surface) with compressor suction and discharge temperature.

## 3.6 Start-up procedures

Before the compressor start-up, the oil sump heater was run until an oil sump temperature of 30 °C was measured on the surface of the compressor. This contrasts with typical operating procedures, where the oil sump heater is on whenever the compressor is off. The short pre-heating time may have caused the compressor to start up with substantial amounts of dissolved refrigerant. This is a potential cause for some wear and tear not attributed to the operation at high temperatures. The compressor was usually started with a frequency of 50 Hz.

## 3.7 Compressor efficiency

The efficiency of the compressor did not seem to be negatively affected by the operation at high temperatures. Data from Phase 2 was used to fit a compressor correlation published in Brendel et al. (2023b). The correlation fits the overall isentropic efficiency as a function of the suction pressure and pressure ratio and was established as follows:

$$\eta_{ois} = a_0 - \frac{0.6}{(P_r - a_1)^{a_2 \cdot P_{suc}}} - a_3 \cdot P_r^{1.8} \tag{1}$$

$$a_0 = 0.66981\,[-], a_1 = 0.01466\,[-], a_2 = 0.00838\,[kPa^{-1}], a_3 = 0.00102\,[-]$$

$\eta_{ois} = \dot{m}(h_{2s} - h_1)/\dot{W}$ is the overall isentropic efficiency, $P_r$ the pressure ratio, $P_{suc}$ is the suction pressure and $a_0$ to $a_3$ are fitted coefficients. The correlation predicts the efficiency for approximately 129 test points with an average absolute deviation of 0.013 over a broad range of temperatures and pressures. Thus, operation at high temperatures did not show differences compared to operation at lower temperatures when looking at the efficiency.

Similarly, the volumetric efficiency as a function of the pressure ratio was fitted as follows:

$$\eta_{vol} = 1 - b_0 \cdot (P_r - 1)^{b_1} \tag{2}$$

$$b_0 = 0.08244\,[-], b_1 = 0.72773$$

$\eta_{vol} = \dot{m}/(NV_{swept}\rho_{suc})$ is the volumetric efficiency, N the rotational speed of the compressor, $V_{swept}$ the swept volume of all cylinders and $\rho_{suc}$ the suction density.

**Figure 3** shows the trends of the overall isentropic efficiency as a function of pressure ratio (x-axis) and suction pressure (legend entries) and the volumetric efficiency as a function of the pressure ratio.

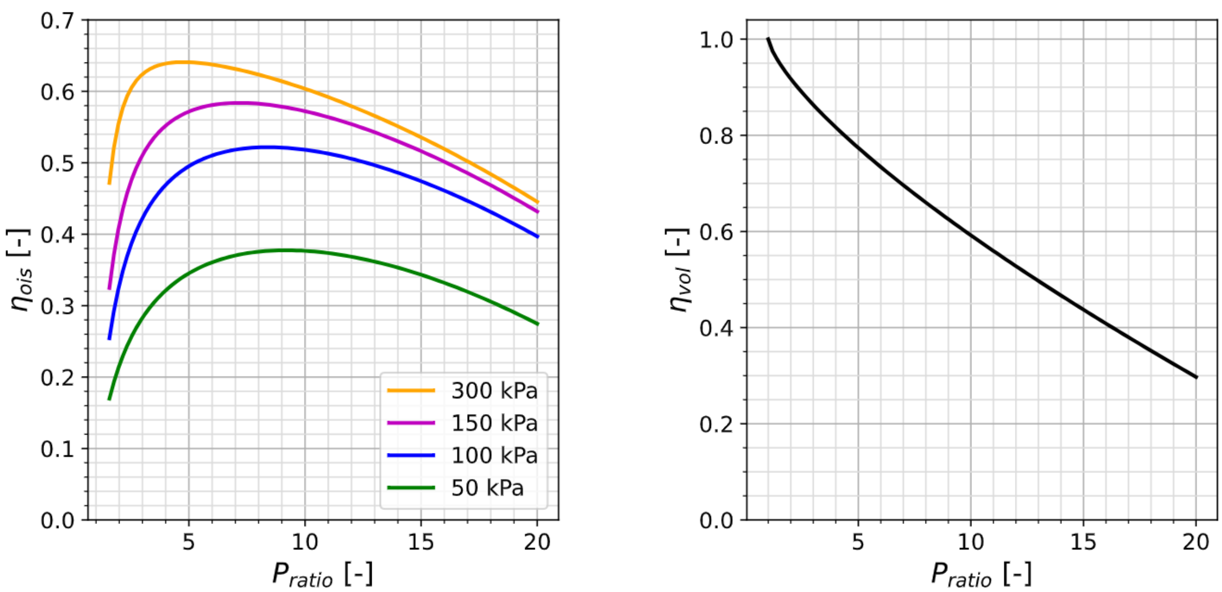


**Figure 3:** Overall isentropic efficiency according to correlation in Brendel et al. (2023b).

## 3.8 Refrigerants

The heat pump was originally designed to assess HFO and HCFO refrigerants to replace the HFC refrigerant R-245fa. In Phase 2, it was also employed to evaluate various refrigerant mixtures. All refrigerants and mixtures that have been tested in the heat pump and, thereby, in the compressor from 2018 to July 2023 are shown in **Table 2**.

**Table 2:** Tested refrigerants and refrigerant mixtures.

| **Tested pure refrigerants** | **Tested mixtures** |
|---|---|
| R-1336mzz(Z) | R-1234yf/1233zd(E) |
| R-1336mzz(E) | R-1234yf /1336mzz(Z) |
| R-1224yd(Z) | R-32/1224yd(Z) |
| R-1233zd(E) | R-32/1224yd(Z)/1336mzz(Z) |
| R-245fa | R-32/1224yd(Z)/1234yf |
| R-1234yf | R-32/1234yf/1336mzz(Z) |

# 4. Compressor examination

In July 2023, the compressor was completely disassembled in a laboratory for inspections from the manufacturer, BITZER. The disassembly was conducted by highly experienced personnel who had previously disassembled hundreds of other compressors, always searching for wear and tear or causes of failure. The following two sections describe all parts that showed discoloration or wear and tear and some parts that were not impacted by the high temperatures.

## 4.1 Parts with wear and tear

The following wear was found:

1. In 2022, a leakage was discovered at the cylinder head sealing. The sealing was replaced with an identical spare part. The removed sealing showed severe discoloration (**Figure 4** → A1). After replacing the cylinder sealing, the compressor was completely leak tight again until the disassembly in July 2023.
2. Strong discoloration was detected at the lift limiters and valve plate (**Figure 4** → B1 and B2), clearly due to high discharge temperatures. No deposits from potential oil coking were found.
3. Small deposits of microparticles were found on both pressure vanes (**Figure 4** → C1). This is unusual and caused by a low oil viscosity following the high temperatures at the discharge vane.
4. Light wear was determined in both the small and big ends of the connecting rod (**Figure 5**→ D1). Additionally, there was a light smoothing of the shaft at the position of the main bearing. The wear at the connecting rod and shaft may be from low oil viscosity. Still, it could also be from insufficient preheating or demanding operating conditions in general, independent of the high temperatures.

In general, discoloration, wear, and tear clearly due to high temperatures were found at the cylinder head sealing, the valve plate, and on the discharge vane. Nevertheless, it is estimated that the compressor could have operated much longer at the described operating conditions.

## 4.2 Parts without wear and tear

Several parts were found to be completely unaffected by the demanding operating conditions. Among these are the following:

1. Most sealings other than the cylinder head did not show discoloration (**Figure 6**→ F).
2. The entire motor and its insulation showed no discolorations or damage. All internal electrical resistances were as expected (**Figure 6** → G).
3. No signs of wear were found on the honed cylinder bore (no picture).
4. The oil centrifuge and compressor shell were unaffected (**Figure 6** → H).
5. The edges of the cylinder rings were not particularly sharp, indicating no excessive friction loads.

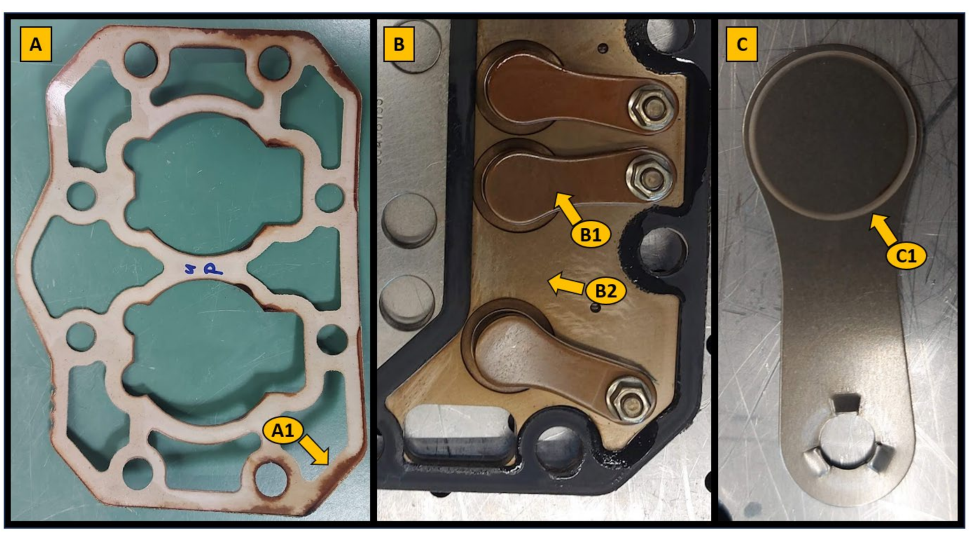


**Figure 4:** Compressor parts showing wear: A: Cylinder head sealing. B: Valve plate with lift limiters. C: High-pressure vane.

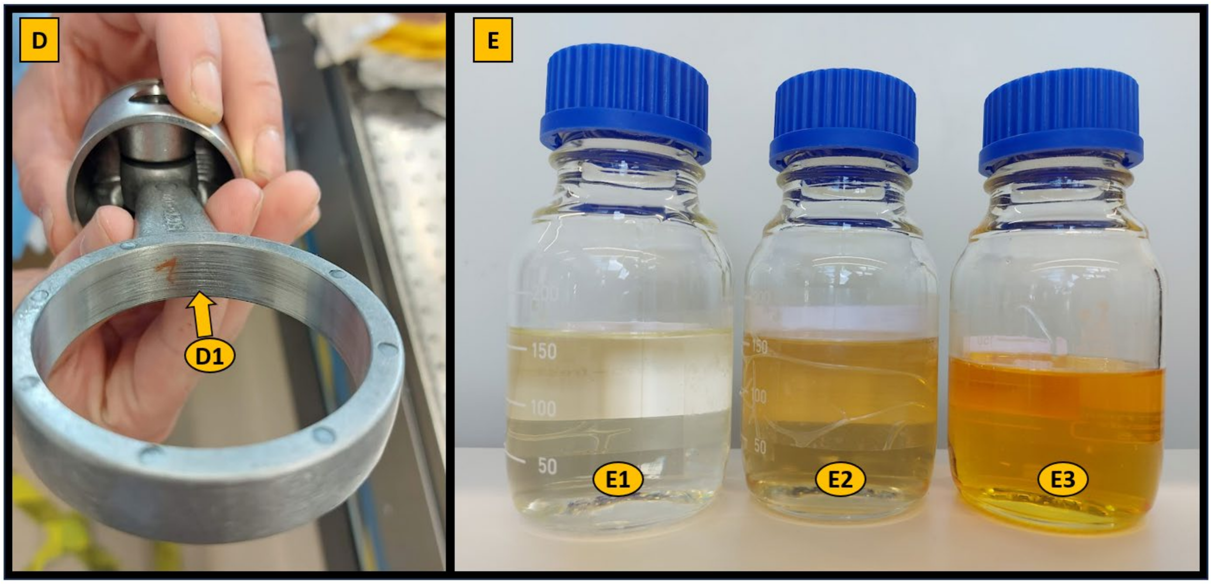


**Figure 5:** D: Small wear at the large end of the connecting rod. E: Fresh oil (E1), Oil after 396 hours of operation in heat pump (E2), artificially aged oil with neutralization number of ≈3 (E3).

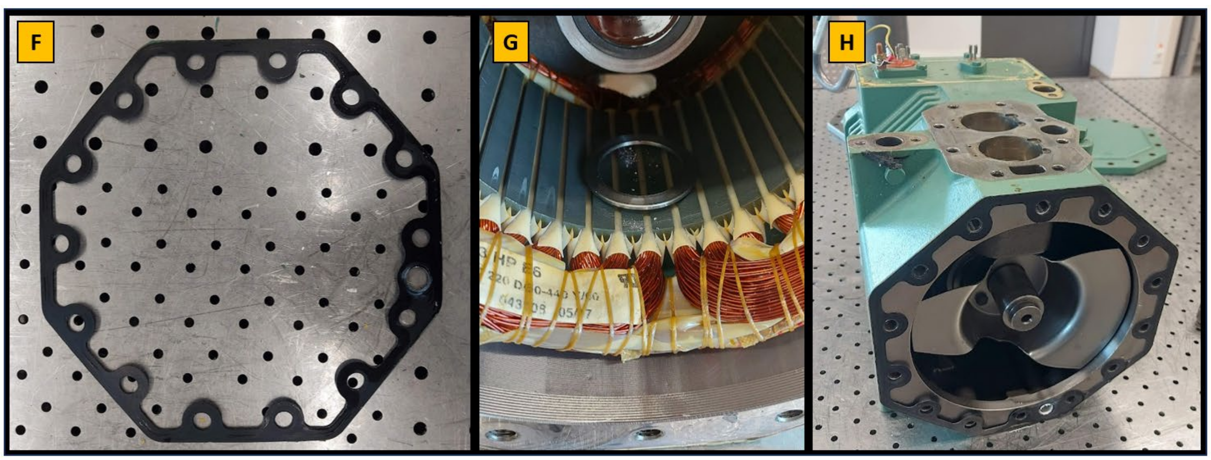


**Figure 6:** Compressor parts not showing any wear and tear. F: Sealing of bearing lid. G: motor windings and insulation. H: Compressor shell and oil centrifuge.

## 5. Oil examination

### 5.1 Visual inspections

The oil was changed in intervals of 50 to 400 operating hours. A discoloration was always found from clear to yellow. However, the oil quality cannot be determined from the visual inspection. As a reference, **Figure 5** shows three oil samples: → E1 is fresh oil with a clear color. → E2 is oil extracted from the compressor after 396 operating hours with a yellow color. → E3 is artificially aged oil. The sample was heated in an oven for a long time but without the addition of metals or refrigerant and without any mechanical loads. The sample was heated until the neutralization number was approximately 3. The sample has a darker yellow color than the oil after 400 hours of operation. However, such a loose comparison is only for illustrative purposes and curious readers. It does not allow to gain much insight.

### 5.2 Oil quality measurements

Several oil samples were analyzed. The oil sample which had accumulated the most operating hours was from July 18th, 2023. The oil had been in the HTHP for 396 total operating hours. 196 hours were operation with a discharge temperature of more than 120 °C, and 5 hours were operation with a discharge temperature of more than 150 °C. The oil was used for mixtures of R-1336mzz(Z), R-1234yf, R-32, and R-1224yd(Z). The analysis showed no significant changes except for an increased water content, possibly introduced between the recovery and charging of the heat pump with refrigerant when it was sitting idle and ambient air may have been in the heat pump:

- Density at 15 °C: 972.9 kg/m$^3$ (fresh: 972 kg/m$^3$)
- Kinematic viscosity at 40 °C: 168.8 mm$^2$/s (fresh: 173 mm$^2$/s)
- Kinematic viscosity at 100 °C: 16.99 mm$^2$/s (fresh: 17.1 mm$^2$/s)
- Neutralization number: 0.04 mg KOH/g (fresh: <0.03 mgKOH/g)
- Water content: 207 mg/kg (fresh: 50 mg/kg)
- Color: 0.5 (fresh: 0-1)
- Comparison of the infrared spectrum: No change

In summary, the oil had not been affected by the operation at elevated temperatures. Future work should test oil for longer periods at yet higher temperatures. The result also shows that the wear and tear observed in the compressor did not result from a degraded oil.

## 6. Conclusions

An off-the-shelf reciprocating compressor was employed in a laboratory HTHP for approx. 1,000 operating hours from 2018 to 2023. Operating conditions reached up to 188 °C discharge temperature and 130 °C suction temperature, both greatly exceeding the design operating conditions. Data assessment revealed that the compressor operated for 30 hours at discharge temperatures greater than 150 °C and 30 hours with suction temperatures greater than 100 °C. The cylinder head sealing had to be replaced after a leak was detected and was found to have experienced strong discoloration. Discolorations of the valve plate and deposits on the discharge vane were other effects owing to the high temperatures but were not detrimental to the compressor operation. The motor and its insulation were unaffected. Deposits from oil coking were not found. Oil analyses comparing fresh and used oil did not show significant oil degradation after up to 400 operating hours. Future work should test the compressor for longer periods at yet higher temperatures.

## NOMENCLATURE

| | | |
|---|---|---|
| $a_i$ | Coefficients in correlation | Various |
| $\eta_{ois}$ | Overall isentropic efficiency | - |
| HTHP | High-temperature heat pump | - |
| HFC | Hydroflurocarbon refrigerants | - |
| HFO | Hydrofluoroolefin refrigerants | - |
| HCFO | Hydrochlorofluoroolefin refrigerants | - |
| $P_{dis}$ | Compressor discharge pressure | kPa |
| $P_{suc}$ | Compressor suction pressure | kPa |
| $P_r$ | Pressure ratio | - |
| $T_{suc}$ | Compressor suction temperature | °C |
| $T_{dis}$ | Compressor discharge temperature | °C |
| $T_{cylinder}$ | Compressor cylinder temperature (measured on surface) | °C |
| $T_{sump}$ | Compressor sump temperature (measured on surface) | °C |
| $\dot{W}_{comp}$ | Compressor power draw | kW |

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