# Technical Paper

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# Actual Damages at Compressors- Failure Mechanisms and their Root Causes

**Author:**

**Dr. Persigehl, Bernhard**
**Allianz Risk Consulting GmbH**
**Allianz Center for Technology (AZT)**
**81724 Munich –Germany**

Co-Author:

Dr. Stoiber, Johannes
Allianz Risk Consulting GmbH
Allianz Center for Technology (AZT)
Managing Director
81724 Munich –Germany

**Summary**

Four actual damages at different compressor types are shown. A liquid ring compressor suffered from manufacturing errors during the casting process. A multistage centrifugal compressor was damaged by a shockwave within the piping system. Within this compressor multiple defects during the manufacturing could also be proved. The next case shows a damaged geared compressor, since a bolt was axially overloaded. The cause for this overloading event could not be proved, only approximated. Besides, a screw compressor was damaged due to foreign object ingress.

In all four cases the laboratory investigations of the fracture pieces gave important information about the fracture mode and thus the failure mechanism.

In two of the four cases clear manufacturing failures could be proven that should have been detected before installation and operation via ultrasonic testing. In one of these two cases these manufacturing failures even caused the damage.

In three of the four cases the damage was due to an operating error. Since the recording of the operational data was limited, a definitive root cause could not be proven in all cases.

In order to prevent more severe damages Allianz Center of Technology (AZT) recommends all operators to equip their turbomachinery with a sufficient measuring and alarm system and all manufacturers to carefully review (or even to refine) the specifications of possible manufacturing defects at nondestructive testing.

## Introduction

At the Allianz Center for Technology (AZT) root cause analyses are conducted since over 80 years with special focus on turbomachinery. Basis of these investigations is the detailed knowledge of the type of equipment and the application, e.g. in a power plant or in chemical processes. For most Root Cause Analysis (RCA) laboratory investigations of the damaged components as well as operational data analyses are necessary.

In this paper, four actual damages of different compressor types are presented. This implies a detailed description of the conducted laboratory investigation, of the failure pattern, the derived failure mechanism and the identified root cause.

## Damaged Liquid Ring Compressor

The first example is a liquid ring compressor damage from a geothermal power plant, see Fig. 1:


*Fig 1: Damaged liquid ring compressor*

All blades were heavily deformed. Two blades were completely broken off, the two adjacent blades almost. From these two, the residual of the blades was cut for laboratory analyses of the manufacturer. The derived root cause was a foreign object damage.

In order to prove or disprove this theory, the debris found within the compressor casing were examined using X-Ray fluorescence analysis (RFA). The material composition of all debris was nearly identical. However, a foreign object damage could still have been the root cause, theoretically: either the foreign object completely disappeared during the damage event or the foreign object consisted of the same material than the rotor. But, the likeliness for one of these theories is limited.

Therefore, a crack detection test was conducted to find some possible original cracks. i.e. cracks that were obviously not due to a secondary failure. Here, numerous cracks could be identified. Several specimens at different positions at the rotor were cut out, see one specimen in Fig. 2: These specimens were investigated in the optical microscope.

Within the specimens a multitude of casting defects (hot cracks and welding defects form repair welding) were found. From these defects cracks were initiated.

Therefore, it must be assumed, that these manufacturing defects led to a weakening and thus to an overloading of the compressor airfoils. Based on these results, the compressor was damaged due to a manufacturing fault, not by a foreign object.

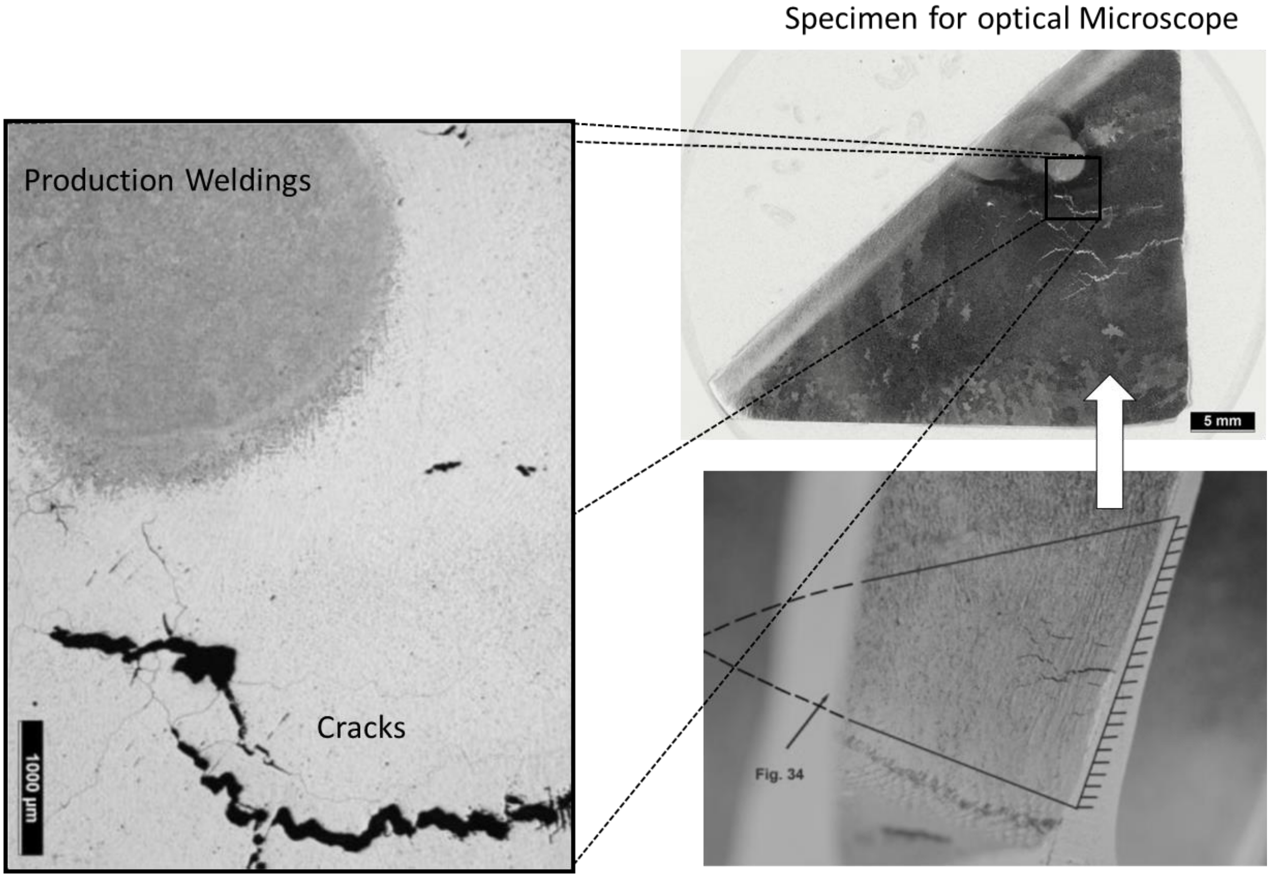


*Fig 2: Production Welding and Cracks within specimen*

In order to prevent a similar damage in the future, a better quality control of the rotor after the casting, e.g. by ultrasonic testing, should be executed. Cast defects exceeding specific limits must be detected before installation and operation.

**Damaged Multi-Stage Centrifugal Compressor**

In the next example a multi-stage centrifugal compressor damage from a fertilizer plant is shown, where a faulty opening of a valve lead to a shockwave through the piping system. The purpose of the investigation was to answer, if this shockwave was the only cause for the damage or if something else could have facilitated the damage.

The compressor consists of seven centrifugal stages. In some diaphragms the backwall was detached from the profiles. During manufacturing the backwalls are joint with the diaphragms via high temperature brazement, compare a diaphragm with or without backwall in Fig 3:

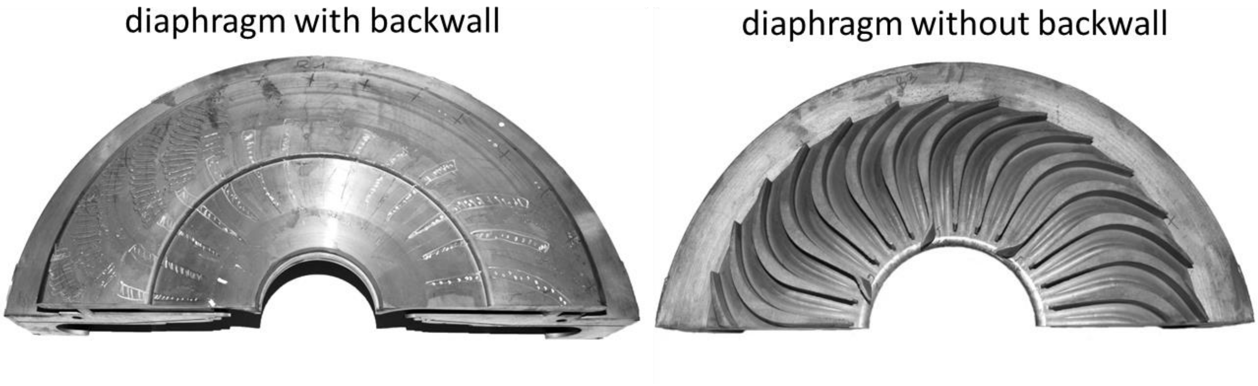


*Fig 3: Overview of the investigation parts from the multistage circumferential compressor*

Different brazed joints were analyzed by scanning electron microscopy (SEM) and optical microscopy with respect to the integrity of the brazements and the fracture surfaces of original and forced joint fractures. Two different fracture modes were identified: Forced fracture either occurred within the diffusion zone of the brazing material into the base material (right side in Fig 4) or within the center line of the brazed joints, along chains of brittle phases (left side in Fig. 4).

Since it is common understanding that brittle phases at the center line of brazed joints are to be limited and since the amount of center line cracks are high within the investigated backwalls (about 75%), the quality of the brazements has to be questioned.

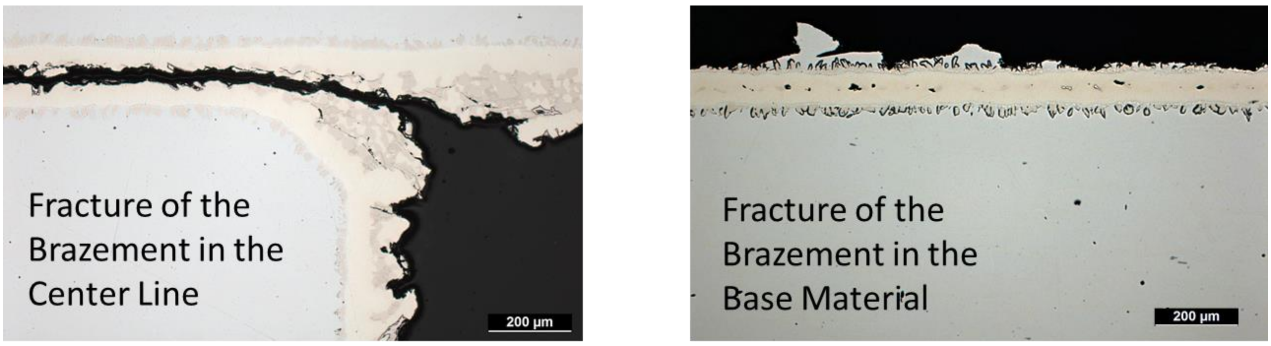


*Fig 4: Comparison of the fracture surfaces within the optical microscope*

Whether the center line cracks were already existent before commissioning or developed during operation, could not be clarified. Since the compressor was operated over a period of 8 years without failure, a “poor” brazement quality can not be the only factor for these crackings. Besides, no signs for stepwise propagation could be found. Therefore, the detachment was probably triggered by an overloading event (e.g. by the reported shock wave).

To ensure the integrity of brazements after manufacturing the diaphragms should have been tested via ultrasonic tests after the brazing process, as it was done by AZT at the beginning of the laboratory investigations. In Fig 3 left, regions with poor metallurgic joint betwe en backwall and profile are marked in white.

**Damaged Intercooled Geared Compressor**

The third example shows a damage of an intercooled geared compressor that is used for the instrument air supply within another fertilizer plant. One shaft with two centrifugal compressor wheels was heavily damaged, see Fig. 5. Both bolts to fix the rotor wheels on the shaft were broken.

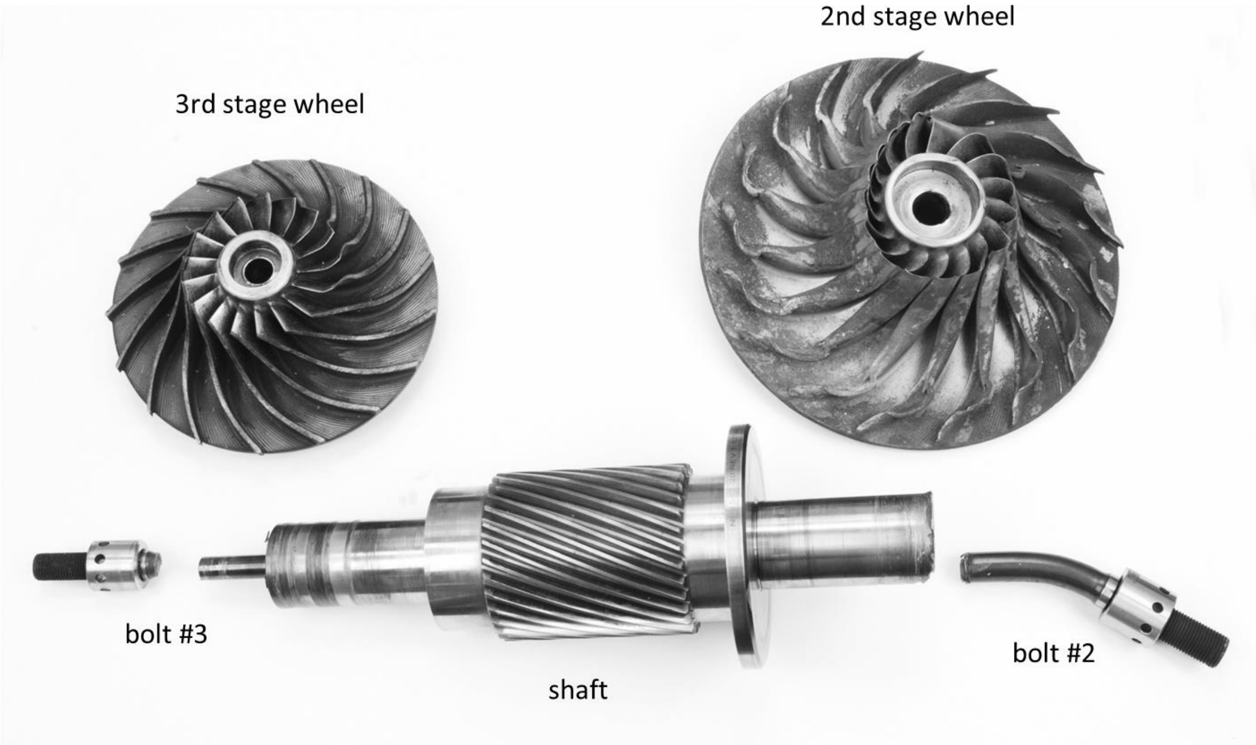


*Fig 5: Overview of the investigation parts from the geared compressor*

Based on the macroscopic findings it is very likely that the bolt at rotor #3 broke first. Due to the resulting unbalance of the shaft and the missing axial force of compressor wheel #3, a rubbing of rotor #2 versus the casing could be initiated. Thus, bolt #2 was overloaded, bent and broke. From AZT point of view, no scenario is imaginable why bolt #3 should have failed as secondary damage after a theoretical failure of bolt #2, since no hard rubbing marks were found at rotor #3. Thus, there is no reason for a secondary failure of bolt #3.

The fracture surfaces of both broken bolts #2 and #3 were investigated using a SEM and an optical microscope. Neither bolt #2, nor bolt #3 failed due to a fatigue fracture. The causal bolt #3 broke due to a forced fracture. The fracture mode is a “cup and cone” fracture. The fracture pattern of the conducted tensile test (with a specimen cut out of the undamaged part of one bolt) was the proof that the specified material cracked as “cup and cone” fracture, see Fig. 6:

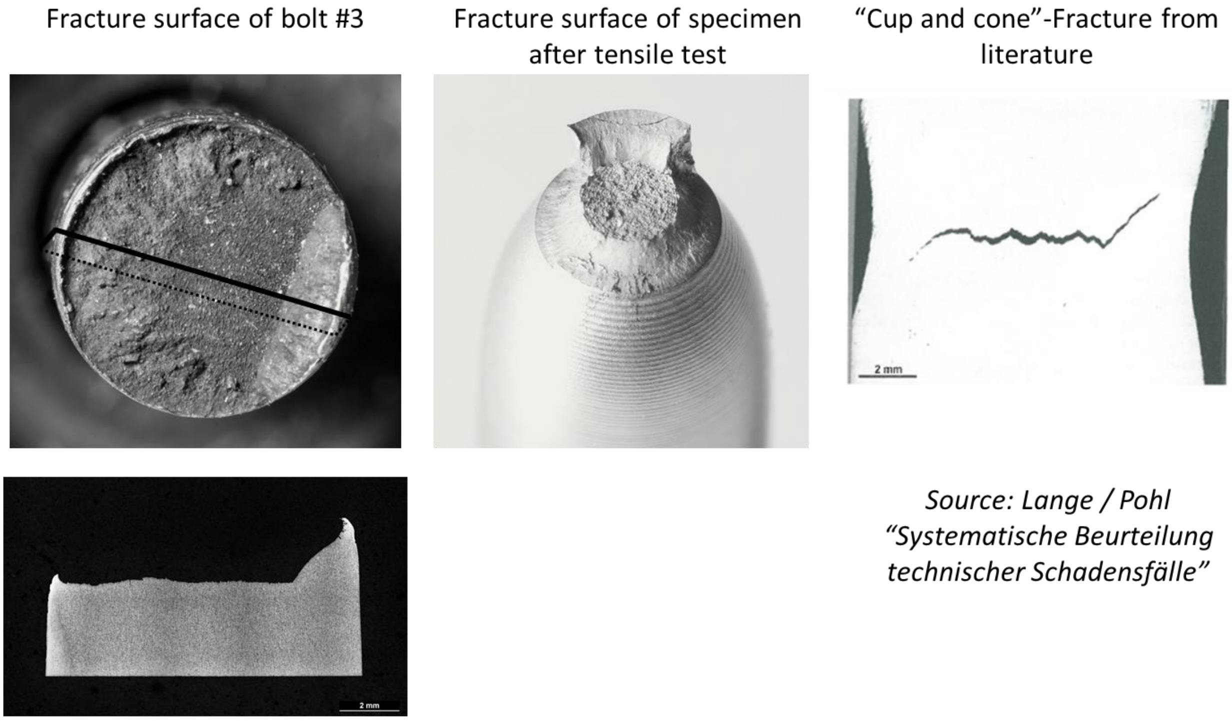


*Fig 6: Comparison of fracture surfaces*

This fracture mode is typical for axial overloading. Different causes were discussed for this overloading event. As most likely cause a malfunction of the compressor blow off valve behind stage #3 was identified. Due to such an opening the back pressure of the compressor would suddenly drop down to zero, resulting in an axial force on compressor wheel #3. However, a direct proof for this theory could not be found, since the amount of operational data (as the position of the mentioned blow off valve) was limited.

**Damaged Screw Compressor**

At the end, a damaged screw compressor is shown from an air supply for a test facility. Both rotors rubbed axially against the casing, see in Fig. 7, left hand side. On the front side of the rotors the annealing colors can be seen from the rubbing event. One rotor could only be disassembled with high forces, since the connection between rotor and casing resembled a cold welding.

In order to investigate the cause for the bridging of the axial clearance several possibilities were discussed.

- A manufacturing error was very unlikely, since the compressor was operated over 15 years without a problem.
- An assembly error was a relevant issue, since a revision of the complete machine was conducted approx. 100 operating hours before the damage event. However, a check of the working and clearance protocols disproved this theory.
- Thus, the likeliness for an operating error was very high. Here, two possibilities were discussed:
  On the one hand, it was investigated if a trip or an abnormal operating situation like a sudden drop of the suction or the discharge pressure could be responsible for the

damage. In all of these imaginable operating situations a remaining axial gap of at least 33µm between the rotors and the casing remained, see an example in Fig. 7, right hand side.
On the other hand, the possibility was discussed during which operating situation a foreign object could have caused the damage. Here, a reasonable damage scenario could be found.

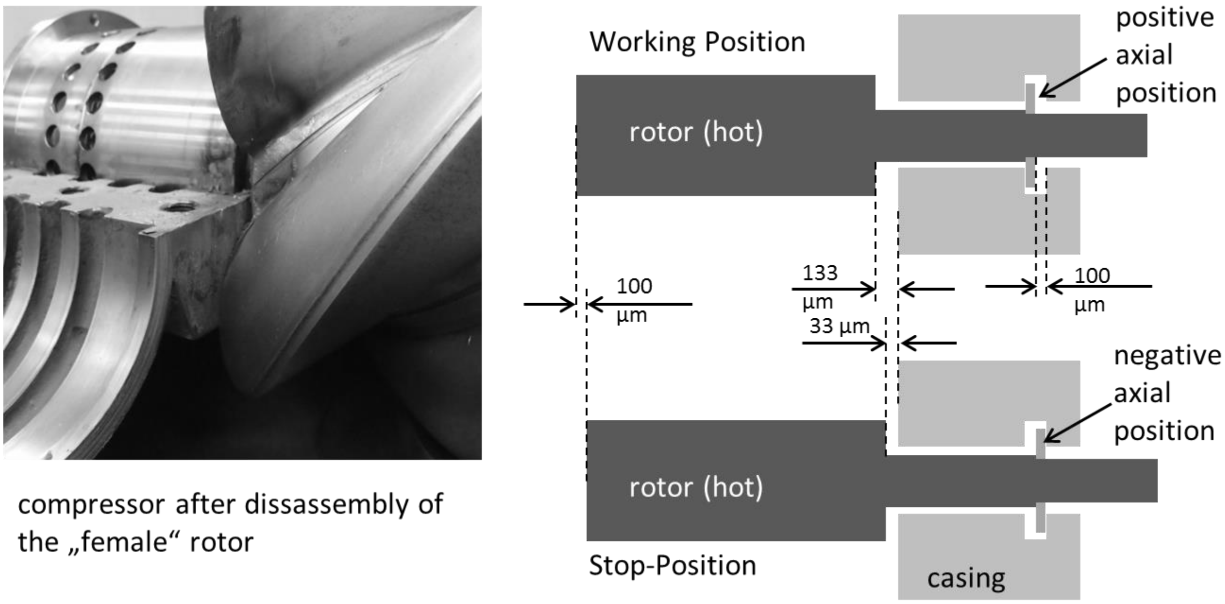


*Fig. 7: Damaged screw compressor (left) with annealing colors, different axial rotor positions (right)*

The root cause has been approximated on a rather theoretical basis, as follows: A foreign particle entered the axial clearance between the (cold) rotor and the casing during the last start-up of the machine. During this start-up, the pressure difference between inlet and discharge side was low. Besides, the rotational speed was still low. Thus, the flow forces and the centrifugal forces on the foreign particle were limited enabling generally the ingress of the foreign particle into the axial gap.

As remedial action an air filter with a smaller mesh size was installed at the facility as well as a recording of the operating data including an automatic control of operations.

## Lessons Learnt from the Presented Damages

The metallurgic analyses of the fracture surface gave important hints on the fracture mechanism in all four cases. Based on these results, the root cause could be approximated. Since however there was no sufficient measuring and recording of the operational data, a definitive proof for the root cause could partly not be found.

Thus, as lessons learnt of this paper, the operators of turbomachines are requested to increase the effort for recording and monitoring of operational data. The choice of the recorded data is dependent on the particular turbomachine. In general, the following data are recommended: shaft vibrations, rotational speed, inlet and outlet fluid parameters (pressure and temperature). By additional implementing of an alarm system into this monitoring system several damages could have been avoided or at least their severity could have been reduced.

Besides, in two of the four presented cases the turbomachines were operated with clear manufacturing defects, that should have been identified by ultrasonic testing before installation and operation. Thus, the manufacturers of turbomachines are requested to review, if the specifications of possible manufacturing defects are reasonable or if they should be refined.