Fatigue failures are present every day in industry, and pump parts are no exception.

In the first part of this series of articles, a short review was presented of the effect of the design forms of machine elements on their endurance limit, as well as some methods to increase the fatigue strength from the design standpoint. In this second part, several case studies are discussed as actual examples of the influence of stress raisers on fatigue failure.

Diffuser fixing screws

The first case study considers the fatigue fracture of diffuser fixing screws in a group of multistage centrifugal pumps. The failures occurred with no...
apparent cyclic stresses. The case illustrates the issue of design mistakes in the manufacture of spare parts.

Several high-pressure multistage centrifugal pumps in a power plant are designed using fixing screws to fasten the discs containing the diffuser blades to the stage casings of the pumps. The round head of these screws (Figure 1) is embedded in recessed holes in the discs and the screws are threaded in the stage casings. The bolt heads are slotted to allow use of a screwdriver. Because of the accumulation of corrosion products, it is very difficult to remove these screws after a long period in operation.

Often it is necessary to drill them out in order to disassemble the pump and new screws then have to be manufactured in a machine workshop.

The working load on these screws should be only the static tensile stress arising from the initial tightening, but given the type of head employed the tightening stresses would not be very high. The screws heads are usually tack welded in order to prevent loosening of the screws and some residual stresses from the welding remain, increasing the tensile stress. The purpose of these screws is to fasten the diffuser disc to the stage casing and there are no apparent alternating stresses acting on the screws. Working under only a static load, the newly manufactured screws should have worked indefinitely. However, most failed after a certain time in operation. The failure of every one of these screws was clearly as a result of fatigue, as is evident from the fracture surfaces (Figure 2). The appearance of these surfaces indicates the presence of a low stress concentration factor and moderate cyclic tensile overload.

![Figure 2. Fatigue fractures of three of the diffuser fixing screws.](image)

The appearance of these surfaces indicates the presence of a low stress concentration factor and moderate cyclic tensile overload.

![Figure 3. General view of the pump impeller and eroded areas of the blades.](image)
the fracture surfaces indicates the presence of a low stress concentration factor and moderate cyclic tensile overload, as can be seen from Figure 1 of the first article [World Pumps, May 2008, pp. 42-25].

The origin of the stress concentration factor was rapidly found. Plant engineers had not specified very carefully the geometric design of the screws to be manufactured and the workshop had machined right angles in the transition between heads and shanks.

But where were the cyclic stresses coming from? Although the only apparent load was of a static nature, the actual tensile load fluctuated because of the vibration induced in the diffuser disc by the rapid change in fluid velocity and pressure when circulating along the diffuser blades. This should have produced an alternating stress pattern based on a moderate mean stress (light tightening) and low stress amplitude possibly varying with high frequency, which is consistent with the fracture surface appearance shown by all the fractured screws. After the fracture of each screw, the diffuser disc is less restricted and vibration increases, making the fatigue fracture of the next screw easier.

This experience clearly reveals that even tiny and apparently irrelevant stationary parts have to be very carefully designed and manufactured in order to prevent unexpected sudden failures leading to long and expensive disassembly – repair – assembly processes.

**Repaired impeller blades**

This next study examines the case of the fatigue fracture of impeller blades after repair by surface welding. It highlights the dangers of malpractice during repair and maintenance, and careless final inspection.

Several vertical, single-stage centrifugal pumps were used in a power plant to pump cooling sea water through the main steam condensers and some other auxiliary heat exchangers. The main features of these centrifugal pumps were as follows:

- Impeller: open type, 1000 mm outer diameter.
- Impeller material: high nickel cast iron.
- Capacity: 13 300 m$^3$/h.
- Head: 9 m.
- Motor output: 450 kW.

The size of the openings in the sea water intake filters was large enough to allow small pieces of debris to enter the suction channel and to reach the impellers of the cooling water pumps. After a relatively long operating period, this resulted in erosion on both sides of the impeller’s blades. The coarse pitting on the surface of the blades (Figure 3 and 4) led to a loss in pump efficiency and even to reduction in the mechanical strength of the blades.

To solve these two problems, maintenance engineers decided to repair the impellers by surface welding. An electrode for EMAW (electric metal arc welding) was correctly selected and an appropriate welding procedure was prepared, including carefully cleaning the surface of the blades before welding and grinding to a smooth surface finish after welding. The impellers were repaired and put back into service for another year.

During the next maintenance period one year later, the pumps were dismantled and inspected again. The welding deposit on the blades was in fairly good condition. However, there were visible cracks adjacent to some of the weld beads, in the border between the welding and the base metal.

Although a good welding procedure had been prepared for the repair job a year earlier, the maintenance engineers who performed the final inspection failed to detect the undercut remaining along some weld beads and also failed to identify its harmful effect. The grinding job after welding had not been careful enough and some sections of undercut had remained.

The load on the impeller blades during the operation of the pump can not be considered as absolutely steady but rather as an irregular load varying with time. In spite of the low notch sensitivity of cast iron, the undercuts acted as stress concentration factors, locally increasing the stress up to values high enough to create surface cracks. The bottom of these cracks also had a sufficient stress raiser effect to make the crack develop through the blade thickness. The surface on both sides of the crack clearly featured a fatigue failure.

Only the timely inspection executed one year after the repair prevented the complete failure of the cracked blades, which surely would have been a major disaster for the whole pump.

**Pump shaft fracture**

The final case study in this part considers the fatigue fracture of a pump shaft and further demonstrates the risks of repair and maintenance malpractice. The case concerns the same group of pumps, with the same features, as those in the second case study.
On account of the very large size of these pumps, the shafts are made in two parts (4880 and 2458 mm long) assembled by a shrink-fit coupling. The two shaft parts and the coupling piece are made of AISI 304 stainless steel.

Thrust load is supported from the upper motor bearing and through a rigid motor–pump coupling. Three journal rubber bearings are placed along the pump shaft into the pump column pipe. Shaft sleeves are used on the shaft at the locations of the bearings and the packing. The shaft sleeves were originally locked in position by two socket head set screws each. For an unknown reason, during a scheduled maintenance job on one of the pumps, the two set screws were replaced by plug welding and the pump was put back into service.

After a period of several months of operation, the pump suddenly failed to supply sea water (no discharge pressure at all) and the power consumption dropped sharply. An initial check on site revealed that the rotor (at least what was visible of the rotor) rotated freely – much too freely – when moved through the motor–pump coupling.

The pump was taken out of the pit and placed where it could be disassembled and inspected. Even before dismantling, a fractured shaft was found. The longer bottom shaft was fractured about 50 mm below the shrink-fit coupling. After totally dismantling the pump, the shaft fracture was found to be caused by fatigue, which occurred in a surface perpendicular to the axis, under the shaft sleeve corresponding to the upper bearing. The initiation point of the fatigue fracture was clearly located at the position of the plug welding fixing the shaft sleeve. Although the shaft is made of AISI 304 steel whose weldability is very good, the shaft sleeve is made of cast stainless steel whose weldability is not good. It is suggested that some micro cracks could have appeared during cooling after welding and later propagated to the shaft sleeve and also to the shaft. Even a small misalignment between pump and motor could induce bending cyclic stresses that caused fatigue crack growth and ultimately the fracture of the remaining section of the shaft. The failure process would have been accelerated by the effect of crevice corrosion caused by the stagnant sea water trapped between the shaft and the shaft sleeve.

This case is a clear example of how harmful repair malpractice can be.

Summing up

Several case studies involving fatigue failures have been discussed. The causes of failure have been identified as arising from inadequacies in design, manufacture or repair. Consequences from the economic and production standpoints can be figured out by taking into account the scope of the repair or replacement jobs and the shutdown time required to carry them out.

In the third and final part of this series of articles more case studies will be discussed and final conclusions will also be presented.

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