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A new phase of increasing oil production on existing installations is going on worldwide

Retrofit: Increasing the Efficiency of Pumps

A new phase of increasing oil production on existing installations is going on worldwide. Sulzer has completed multiple upgrades of pumping equipment in major installations over the past years. The performance and efficiency of the pumps have been significantly increased, also resulting in lower CO_2 emissions to the atmosphere.

Uprating of process machinery can be carried out at many levels, from small increases in capacity and improved reliability of the equipment to major upgrades of complete operating systems. An important goal is to increase the efficiency of the pumps: this not only has a big economical effect but also an ecological one. Because less power is needed from the gas turbine driver, less fuel will be burnt, and therefore the emissions of CO_2 to the atmosphere will be reduced. Industrial countries have pledged a reduction in the emission of CO_2 and other greenhouse gases in the Kyoto Protocol.

The most flexible design for retrofit is the barrel casing pump which allows the cartridge to be interchanged with the up-graded design. However, impressive upgrade results are also achieved on axially split multistage pumps. The reason for the uprate can vary from modernisation of old or obsolete equipment to changes in operating expectations and/or under performing equipment. The retrofit should enhance the eco-efficiency of the pump. The essence of all upgrades is to maintain the existing boundary parameters and utilize the maximum amount of the original equipment with considerable, consequential savings in time and costs. Therefore, in many cases, notable benefits to the process are possible with little or no impact on the original footprint area, the drive system, the utility supplies, all skid/site interfaces, as well as the control and instrumentation.

The mechanical characteristics of the pump such as vibration levels, thrust loading, operating temperatures, etc. will also remain unchanged from the original specifications. These can be proven along with the new performance during factory tests in much the same way as the original equipment with the utilization of a test barrel and associated equipment. The upgraded cartridges can be tested to industry standard codes and specifications as per the original equipment. More recently, clients are using the thermodynamic method for conducting site tests, thereby further reducing the delivery time.

Uprating pumps on oil production platforms increases the efficiency and also reduces the CO_2 emissions to the atmosphere.

The retrofit principle shows the greatest flexibility on the multistage barrel casing/ cartridge design shown on left.

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TURBOMACHINERY DEVELOPMENT

A retrospective review of some troublesome turbine blade failures

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1. ABSTRACT

This note was originally written in 1996 to record experience with blade failures on two turbines at ICI's Wilton factory. The history of the failures, the results of investigations into the causes, and the measures adopted to prevent further repetitions are given. Some comments are made about blade failures in general.

2. INTRODUCTION

Turbine blades are not normally a problem. However when troubles do occur they can be difficult to cure. This is partly because the rotating blades are exposed to quite high steady stresses. A typical blade root with a mean stress of 250MPa will be likely to yield locally at overspeed conditions due to stress concentrations. Yet fatigue is the failure mechanism of almost all turbine blade failures. This is because the flow field into a rotating turbine blade is by definition unsteady. Wakes from the preceding stationary blade row (the nozzles) impose a strong excitation equal to approximately 100% of the steady bending load at nozzle passing frequency. Partial arc admission on control stages imposes excitations of similar magnitude but lower frequency. Since this partial arc loading has a typical 'square wave' form the resulting excitation of the rotor blade covers a broad range of frequencies. More subtle excitations come from the non-uniformity of nozzle spacing, obstacles upstream and downstream of the blade row and non-uniformities due to connections into or out of the machine. Flow instability can also provide an excitation for compressors and for longer turbine blades (typically in the last stages of condensing machines).

Turbine blades of any size have many natural frequencies in a frequency range which can be brought to resonance by these excitations. Thus variable speed machines have to be designed to endure resonance. This requires conservative blade design (low bending stresses and careful blade detailing), control over the excitation from the flow path, and control over the response of the blading, including damping. Fixed speed machines are sometimes designed to avoid specific dangerous resonances, and may have specially tuned blades for this purpose.

3. PLANT NO. 1 - A VARIABLE SPEED MACHINE

This plant had a 10MW steam turbine driving a recycle gas compressor. The machine was commissioned in late 1979.

In April 1982 after about 16,000 hours operation on rotor no. 2 the machine suddenly stopped. When a restart was attempted very heavy unbalance was found on the turbine at low speed so the turbine was opened. One blade was found to be missing from the last stage of this 4 stage turbine (see figure 1). It was concluded that the vibration following the blade failure had been so violent that it caused the turbine mechanical overspeed trip to operate.

The blade fracture surface was dominantly fatigue. On disassembly another 3 blades in stage 4 were found to be cracked. The relative position of the cracked blades and the broken blade are shown in figure 2. The 70 blades were grouped into 10 packets (or groups) of 7 blades by a rivetted cover-band. It can be seen that the cracked blades were all at the end of a packet. Also the cover-band had interlocking 'tabs' at the ends of each packet, and these tabs had fretted, indicating relative motion between the blade packets. It seemed clear that the blade failure was due to blade packet vibration. Extensive investigations of blade, packet and disc natural frequencies were carried out by the owner.

Figure 1

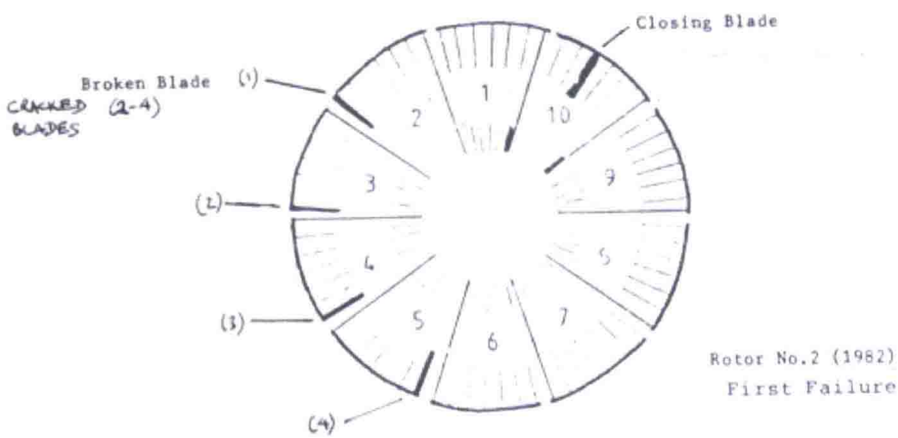

Figure 2

The blades had cracked at the upper platform of the double hammerhead root (see fig. 3).

Figure 3

The crack initiation was at the suction side corner. On further investigation it was found that the blade airfoil was stacked so that centrifugal loads imposed significantly higher stress on this side of the root. In addition, the blade root platforms were flat, while the disc groove is curved, hence all the centrifugal load was concentrated on the corners of the blades.

The failed blade was examined under the scanning electron microscope (SEM) by the vendor who concluded that the fracture surface indicated fatigue 'with the influence of corrosion'. Recommendations from the vendor included changing to a material of somewhat improved strength and corrosion resistance, re-stacking the blade profile to distribute the centrifugal load more equally, and increasing the radius between root and blade shank to reduce the stress concentration in the blade root.

This solution was not accepted technically or commercially by the owner, mainly because it was felt that the vendor was not investigating the failure seriously and because there was no guarantee available against a further failure. The turbine was re-bladed by a third party blade manufacturing specialist, supported by their consultant engineer. The blades were re-engineered with the following changes: improved material similar to that proposed by the turbine vendor, rolling radius for the root to eliminate corner loading, longer cover bands to damp out per rev excitation of the lower packet modes, better radial stacking of the blade profile and shot peened roots. The re-bladed 4th stage had four packets, two of 18 blades and two of 17 blades.

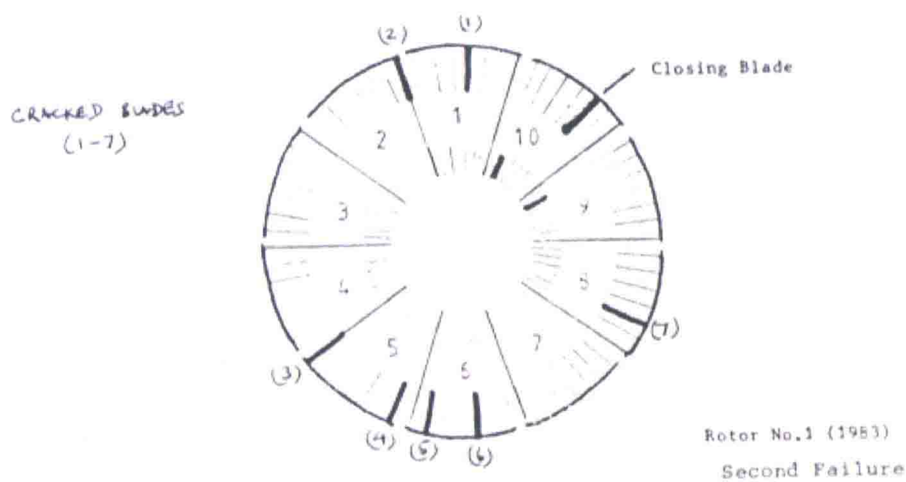

Figure 4

In 1983 after about 15,000 hours operation, turbine rotor no. 1 (the original spare) was removed from service and the 4th stage blades removed for examination. A number of blades were found to be cracked, but this time in different locations in the packets (see figure 4).

One of these cracks was over 75% through the critical root section of the blade (see figure 5). Once again the cracks were caused by fatigue. Obviously we had been very lucky to avoid another failure in service.

Figure 5

In Oct 1984 after about 12,000 hours operation the first modified (Mk2) rotor experienced a blade failure, of the single (unique) closing blade. No other cracks were found. This rotor had operated at significantly higher speed and load than either of the Mk1 rotors. The blade manufacturer's consultant suggested that the failure was due to vibration in the 1st tangential out of phase mode. However it was subsequently discovered that there were also some manufacturing anomalies with this blade (the serrated teeth had been re-machined during manufacture).

At this point it was decided that the original vendor should once again be involved to see if a more robust design could be achieved. After extensive discussion in which all the resources of the vendor were involved, the following package of changes was proposed:

- retain better radial stacking from Mk 2 design
- retain rounded root land from Mk 2 design
- retain shot peening from Mk 2 design
- diffuser plate between 1st and 2nd stage to reduce any partial arc excitation
- cut back the 18 piers on the stage 4 diaphragm to try to reduce the excitation from this source
- alter the stage 4 nozzle exit angle slightly to reduce the blade loading on the 4th stage rotor blade (at the expense of the 3rd stage)
- remove the two blank nozzles from the diaphragms on stages 3 and 4, and replace them with special nozzles to reduce the flow disturbance
- reduce the 4th stage nozzle length slightly, and hence the 4th stage blade length (to match the new nozzles). This produced a slight reduction in tensile load on the blade which compensated for the next change
- install a rotor blade with an integral cover plus a cover band, to increase damping in the blade assembly. The cover band was to be longer than the original, but shorter than Mk2 design (14 blades per packet)
- increase the fillet radius at the root to shank transition to reduce the stress concentration. This involved machining the rotor disc.