Cause and cost

Jeffrey Phillips, Hector Bourgeois and Ed Dougherty, Fern Engineering Inc., USA, consider the cause and low cost solution to blade fatigue in large centrifugal air compressors.

When a large, four stage, 15 MW centrifugal compressor was continually plagued with major cracks and occasional loss of impeller blades, General Electric Service Shop in Pittsburg, USA, suspected fatigue as the cause. For confirmation of its suspicions and for advice on how to reduce fatigue inducing vibrations, it turned to the independent, employee owned, turbomachinery consulting firm of Fern Engineering, Inc.

Firstly, a finite element analysis (FEA) using the ANSYS software package was carried out to estimate the natural frequencies of the fourth stage impeller components. The impeller was then subjected to a series of tests with external forcing functions at various frequencies. These tests confirmed that the natural frequency of the compressor blade coincided with the 2x passing frequency of the upstream turning vanes at normal compressor operating speeds.

Because the compressor was a critical piece of process equipment and the proposed final fix required new long lead time forgings, Fern identified a simple interim modification, which was performed in the GE shop. After the modification, the compressor ran with no further failures and with no noticeable decrease in performance.

The compressor

Elliott Turbomachinery Co. originally designed the compressor. However, it was subsequently uprated from an air flow rate of 143 000 Nm^3/hr (90 000 scfm) and 2.1 barg (30 psig) discharge pressure to 210 000 Nm^3/hr (132 000 scfm) and 2.6 barg (37 psig) discharge pressure by a third party in 1995. During the upgrade, the original first and second stage impellers were moved to the third and fourth stage positions and new impellers were installed at the first and second stages. Figure 1 shows a cross section of the compressor, while Figure 2 shows the design of the shrouded fourth stage impeller.

The compressor is driven by a steam turbine with a design operating speed of 2660 rpm, however, during normal operations the speed varies between 1900 and 2850 rpm. The design operating conditions of the turbine compressor train are shown in Figure 3.

An inspection after a blade failure showed extensive fatigue induced cracking in blades on three of the four impeller stages. Additionally, on the fourth stage a portion of one blade was completely broken away, and 10 of the other 35 blades sustained damage, ranging from cuts in the leading edge (presumably from the break-away blade) to cracks up to 50 mm (2 inches) long. The equipment owner reported that the compressor had suffered more and more severe blade cracks since its uprating.

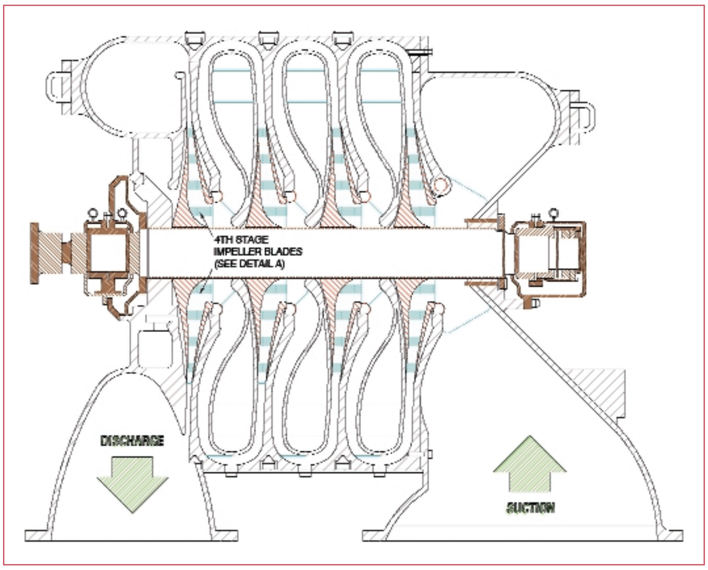

Figure 1. Outline drawing of the compressor showing the four impeller stages.

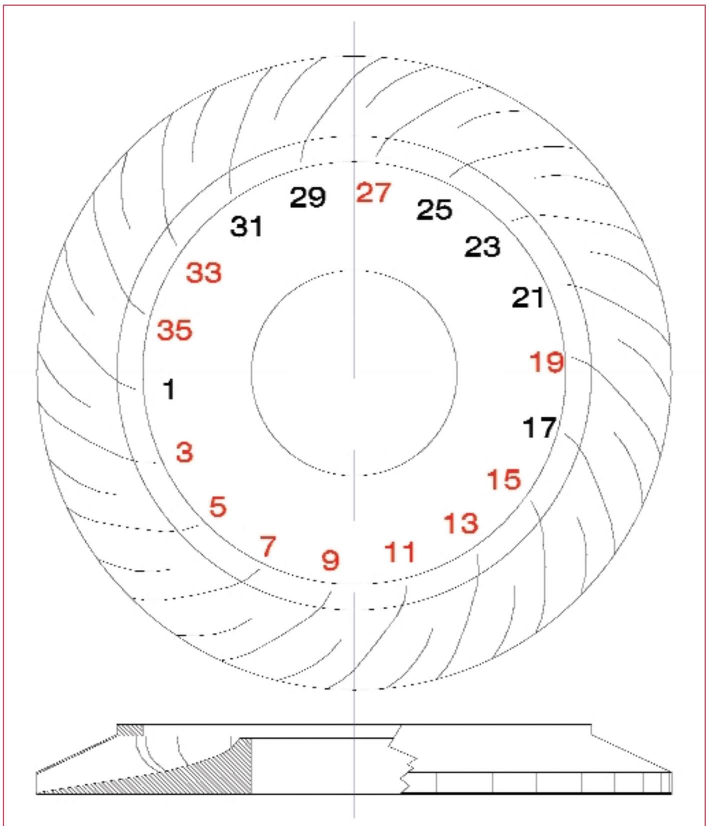

Figure 2. Diagram of the 35 blade fourth stage impeller.

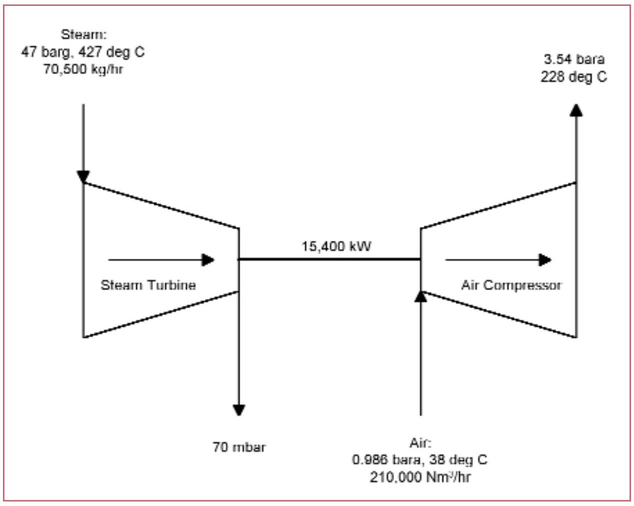

Figure 3. Design operating conditions of the steam turbine air compressor train.

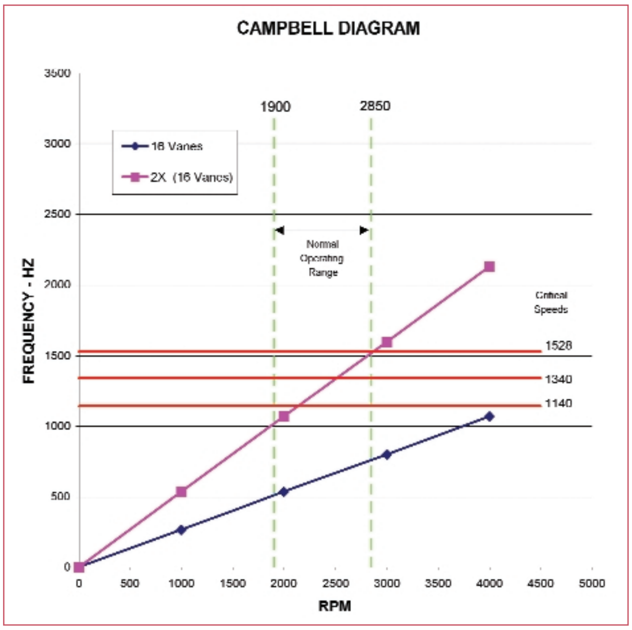

Figure 4. Campbell diagram showing relationship between impeller speed and the passing frequency of the vane wakes.

The analysis

A finite element model of the fourth stage impeller was constructed using the ANSYS FEA package. It indicated that the fundamental natural frequency of the impeller was 1517 Hz. While computer simulations can be useful in estimating the approximate range of the natural frequency, nothing competes with taking measurements on the actual piece of equipment. In addition, incremental damage to the impeller, before the final failure, may have changed the natural frequency.

Fern carried out vibration forcing function tests at its vibration lab on the actual fourth stage impeller. However, the testing was complicated by the damage sustained during operation. In order to dampen out vibrations that may not have been representative of an undamaged impeller, it was decided to shim the damaged portions with wood blocks.

The impeller was excited by Fern's custom built shaker table which features a 90 watt amplifier driven by an oscillator that can be dialled in to deliver a specific frequency of vibration. The oscillator is capable of operating over a wide range of frequencies. Based on the FEA analysis, shaker tests on the impeller were conducted over a range of 0 to 2000 Hz.

The vibration data was acquired using a PC, compatible with an Iotech Daqbook 216, equipped with a DBK 18 - 4 channel low pass filter card with a data rate of 20 kHz. The data was processed using the Iotech Data Acquisition System Laboratory (DASYLab 4.1). A Fast Fourier Transform (FFT) analysis of the data was carried out using the built in FFT module.

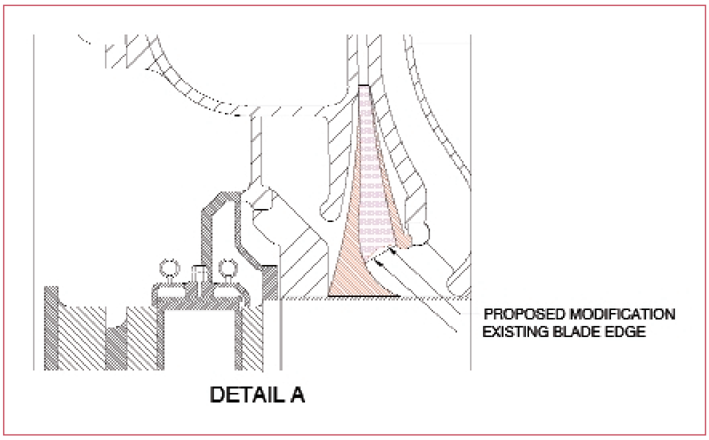

Figure 5. Drawing of the original impeller blade with Fern's recommended modification shown.

Data was collected both with and without the dampening wood blocks. The undamped impeller had a large peak at 1528 Hz, near the natural frequency predicted by ANSYS. There were also smaller vibration peaks ranging from 1140 to 1340 Hz. The damped impeller had its largest peak at 1540 Hz (slightly higher than the undamped rotor) with additional smaller peaks ranging from 1100 to 1460 Hz. The conclusion was that the type of damage sustained by the rotor would slightly lower its natural frequency. It should also be noted that the tests were conducted with the impeller at room temperature, whereas the temperature of the air discharging from the fourth stage of the compressor is typically 228 ˚C (442 ˚F). The modulus of the elasticity decreases as the temperature of metal increases, and this will also cause the natural frequency to decline.

The most logical source of a forcing function for vibrations on the impeller during normal operation are the 16 upstream vanes arranged around the circumference of the inlet duct to the fourth stage. As the impeller rotates, its blades pass through the wake left by each of the 16 vanes. If these 16 wakes per cycle match up with the natural frequency of the impeller, they could drive the rotor into significant vibration.

The relationship between the impeller rotating speed and the passing frequency of the vane wakes is presented in Figure 4. The diagram shows that if the impeller rotated at its lowest operating speed of 1900 rpm, each blade would pass a vane wake at a frequency slightly above 500 Hz, and if it rotated at its maximum speed of 2850 rpm, the passing frequency would be 760 Hz. These values are far below the measured natural frequency of the impeller or any of the other minor vibration peaks identified during the lab tests. The natural frequency and the other peaks are shown nominally by the horizontal lines in the Campbell diagram.

However, it is important to also look at multiples of the passing frequency, since hitting a wake at exactly every other cycle of the natural frequency of the impeller will also excite the blades. It can be seen in the Campbell diagram that the 2X line (the equivalent of having 32 vanes) in the region of the normal compressor operating speeds does intersect the frequencies where subordinate vibration peaks where measured and comes close to the lower natural frequency measured on the damaged impeller.

Taking into account the fact that the hotter metal temperatures would have lowered the natural frequency below the levels measured in the lab and that a forcing function does not have to be exactly on the natural frequency to induce significant vibrations, it was concluded that wakes from the inlet vanes probably did excite the natural frequency of the impeller and lead to the eventual failure by fatigue.

The solution

Based on Fern's analysis, it was obvious that the natural frequency of the impeller blades needed to be increased to well above its current value. One way to accomplish this would be to rebuild the impeller with 50% thicker blades. This is the long term correction that GE adopted. However, because the air compressor was a critical piece of equipment in the owner's process, and because this approach requires a fairly long time to implement, an interim fix had to be found.

Fern's hardware design manager recommended that a small portion of the leading edge of each impeller blade be scalloped as indicated in Figure 5. In addition to increasing the natural frequency because of the configuration change, the newly rounded leading edge softens the effects of the upstream vane wakes since the wakes would be hitting the edge at slightly different times. The modification had the potential to degrade compressor performance slightly, but any degradation would be modest and was offset by the simplicity of the solution.

The recommended modification was quickly and easily implemented at GE's shop and the compressor was returned to service. Reports from the field indicated that no measurable degradation in performance could be observed, and no blade damage has been noticed since.

Enquiry no: