Good Vibrations

Eike Drewes, PROGNOST Systems GmbH, Germany, presents a case study that illustrates effective machine protection for vertical reciprocating compressors.

Vibration monitoring systems for machine protection on reciprocating compressors employ a variety of sensor positions, depending on the machine's design. The case study discussed in this article deals with the actual damage on a vertical reciprocating compressor, and illustrates the vibration behaviour at various measuring points. It also clearly demonstrates what points must be taken into consideration when positioning sensors in order to achieve optimum machine protection.

Damage findings

While the machine was running, the top piston face struck a loosened screw, which fractured the cylinder cover of stage 1.

Application

A chemical production facility in Gelsenkirchen, Germany, uses a double acting, 4 stage, oil free, vertical reciprocating compressor for compression of polypropylene. This machine is protected with vibration sensors (accelerometers), which are mounted on each crosshead slide inside the crankcase, perpendicular to the direction of the piston movement (Figure 1). The space between the individual vibration sensors is 750 mm. In addition, the machine is equipped with dynamic pressure sensors measuring the pressure of both compression spaces from each cylinder to monitor the p-V diagram. A PROGNOST-NT online monitoring and diagnostic system evaluates the acceleration and pressure signals.

Damage sequence of events

On the morning of 3rd March 2000, gas detectors in the compressor building set off an alarm in order to shutdown the compressor. An inspection of the machine identified a broken cylinder cover of stage 1. The cause of the damage was a loosened piston nut on the top end of the piston face. While the machine was running, this screw repeatedly struck sharply against the cylinder cover in the top dead centre (TDC) position, finally destroying it. This caused the release of gas.

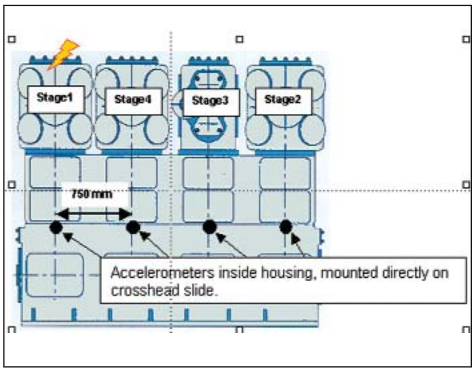

Figure 1. Compressor showing accelerometer positions at the crosshead slide.

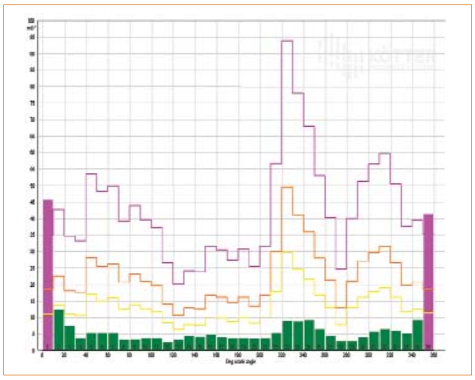

Figure 2. A segmented vibration analysis of one crankshaft revolution, where the segmented machine protection thresholds have been exceeded in segments 36 and 1.

Figure 3. RMS values of the vibration acceleration around TDC of cylinder 1 for the 1st stage (and segments 1 and 36) over six days prior to the damage event.

Analysis

On 2nd February 2000 (exactly four weeks earlier), the PROGNOST-NT system was installed and partially commissioned on this compressor. At the time of the damage event, there was only a connection between the data acquisition and analysis unit and the sensors attached to the machine. Neither the automatic shutdown function had been enabled nor was remote access to the data available, since the required data lines were not yet operative. Thus, contrary to a fully operational state, the early failure detection to alert operators was not activated. However, the vibration data were recorded in the long term trends log, where the analysed vibration values are stored as an hourly mean value.

Subsequent analysis of the trend data revealed the cause of the damage and the sequence of events. By utilising the segmented PROGNOST vibration analysis, which subdivides each revolution into 36 individual segments of 10° crank angle each, the maximum vibrations throughout all segments of one crankshaft rotation, such as those around the TDC, can be identified reliably and accurately (Figure 2).

In this case, the relevant values

with respect to machine protection are the segmented RMS values of the vibrations on the crosshead slide, as the piston reaches TDC (segments 36 and 1) until the machine breaks down, as shown here by the long term trend.

The increase in vibrations caused by the repeated impact of the loosened nut against the cylinder cover was clearly evident from the 20 fold increase of vibration values in the hourly mean value. The impact between the piston and the cylinder head was transmitted directly via the piston, piston rod and crosshead to the crosshead slide, where a conspicuous increase in vibration was noticeable, in spite of the large spatial distance from the 'cause' of the problem.

In addition, the affected cylinder could be precisely recognised, since this noticeable increase was identified at only one of the four vibration measuring points on the crosshead slides. Figures 4 and 6 illustrate the long term trend of the RMS values at the time of TDC in cylinder 1, over the same period as illustrated in Figure 3 for the remaining cylinders from stages 2 - 4.

It is conspicuous that no increase is noticeable on the remaining measuring points. Even the immediate adjacent sensor on the cylinder of the 4th stage (Figure 6) registers a barely noticeable increase of approximately 10%. This is rather strange, since this sensor is positioned only 750 mm away on the same base frame. In spite of the compact compressor frame design and the large mechanical excitation that the piston impact produced on the cover, the vibration transfer to the sensors on the adjacent crosshead slides is barely measurable. This is especially conspicuous when additional vibration parameters are included in the analysis.

A concurrent comparison of the analytically 'more sensitive' absolute peak values of the vibrations for the measuring points on the 1st stage and the 4th stage cylinders also shows no significant increase. Figure 7 shows a comparison of the trends in vibration peak values around the TDC of cylinder 1 over a period of approximately six days prior to the damage event. It provides no evidence of vibration transfer between the two adjacent stage 1 and stage 4 cylinders. The increase in vibration on the stage 4 measuring point with a factor of 1.03 is significant, however, compared to a factor of 15 on stage 1.

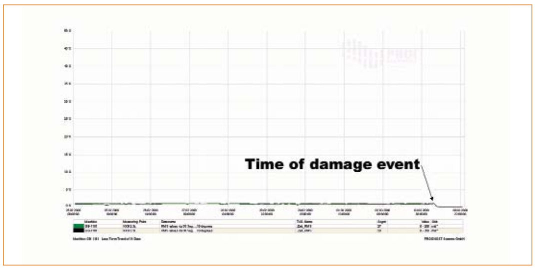

Figure 4. RMS values of the vibration acceleration around TDC of cylinder 1 for the 2nd stage (segments 27 and 28) over six days prior to the damage event.

Figure 5. RMS values of the vibration acceleration around TDC of cylinder 1 for the 3rd stage (segments 18 and 19) over six days prior to the damage event.

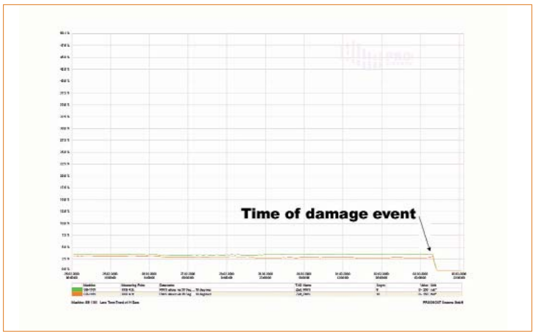

Figure 6. RMS values of the vibration acceleration around TDC of cylinder 1 for the 4th stage (segments 9 and 10) over six days prior to the damage event.

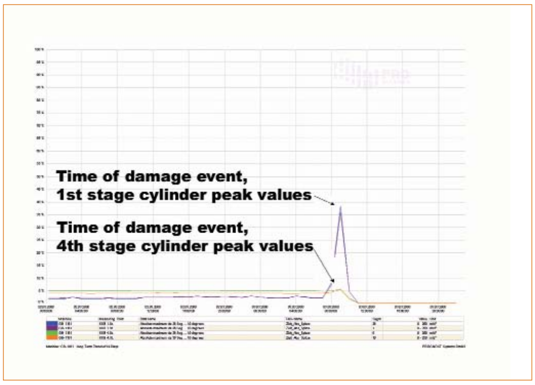

Figure 7. Peak values of the vibration acceleration around TDC of cylinder 1 for the 1st and 4th stage (segments 9 and 10) over six days prior to the damage event.

In addition, the hourly mean values in Figure 7 also reveal that the machine operated with elevated vibrations for more than two hours. Even the last mean value ahead of the maximum vibration peak was already elevated approximately three fold. It would have been possible to have shutdown the machine earlier based upon the elevated vibration values, since the PROGNOST machine protection feature monitors these parameters for each single crankshaft rotation, which in this case would amount to 497 analyses per minute.

Monitoring with a minimum

The damage analysis determined that the loosened nut on the top piston face resulted in a huge increase in vibrations on the crosshead slide. This case proves that even the components on the cylinder that are relevant for machine protection can be monitored with a single sensor per crank on the slide, an interesting factor from both an economic and a technical perspective. Had the PROGNOST system been fully commissioned at the time, the damage would have been reported immediately after it occurred. Presumably this would have prevented the total fracture of the cylinder cover.

A distinct vibration transfer from the crosshead slide of one cylinder to the crosshead slide of an immediate adjacent cylinder is not measurable even in the presence of strong impulsed excitation, despite the single piece solid cast iron housing and the small spatial distance. Therefore, one sensor for each slide must be considered essential for total machine protection.