VIBRATION FAILURES

DYNAMIC TESTING AND FINITE ELEMENT MODELING HAVE BEEN SUCCESSFUL DIAGNOSTIC METHODS

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A well-practiced approach is needed to find solutions to failures due to vibration. This approach typically includes passive vibration analysis methods, strain gage results, and dynamic pressure measurements. Field test data can be used to calibrate a Finite Element Analysis (FEA) model that mimics the problem. The model can then be used to establish a solution without resorting to trial-and-error problem solving. The below case history demonstrates that a combination of comprehensive dynamic testing and FEA can be a cost-effective approach to solving difficult, chronic mechanical problems.

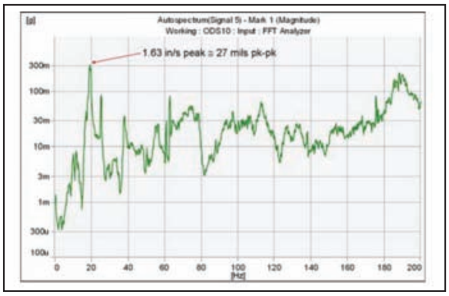
Finding the root cause

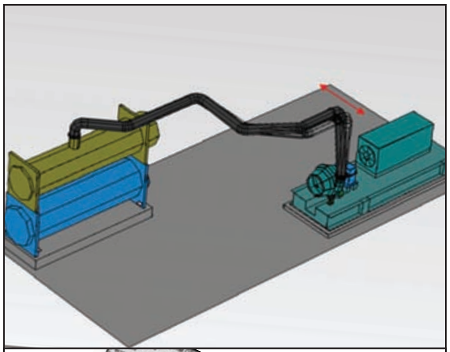
A single-stage, overhung type centrifugal compressor was commissioned in 2001 for a refrigeration plant in New York City. Resident businesses depended on the system primarily to maintain environmental control of critical computational equipment and secondarily for building space air conditioning.

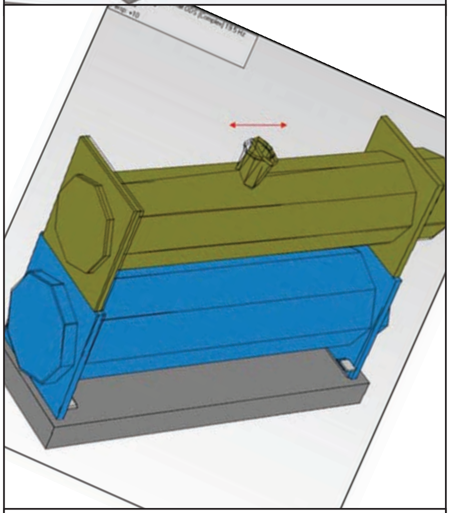
Since installation, the discharge pipe failed three times, the first time at the compressor end at the welding location where the piping meets the flange. The last two failures took place at the welding location where the nozzle meets the condenser shell.

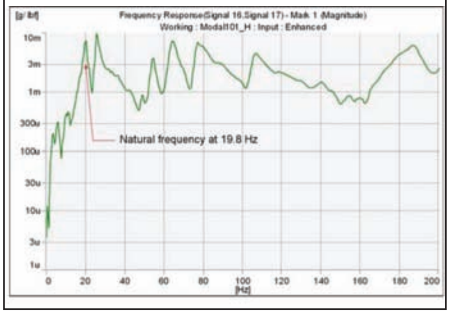
The unit was driven by a V-16 Caterpillar engine (maximum speed 1,500 rpm or 25 Hz) through a speed increaser gear box (2.4:1 gear ratio), and through an additional speed increaser built into the compressor (2.24:1 gear ratio). The maximum impeller speed was 8,047 rpm (134 Hz). The compressor wheel had 18 rotating blades, 9 pre-rotation inlet guide vanes for capacity control, and the diffuser was vaneless with a “tongue” volute arrangement.

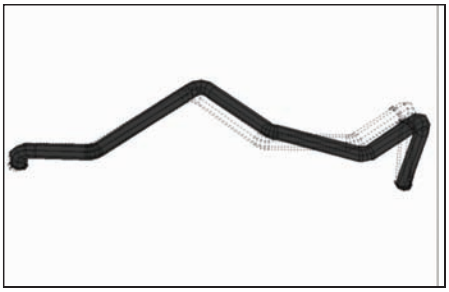
To determine the root cause of the failures a thorough vibration test was performed. This consisted of a combination of impact modal testing, Operating Deflection Shape (ODS) evaluation (p. 21, March/April 2006), and strain gaging to measure the bending and shear stress in the discharge pipe-nozzle where the failures had taken place.









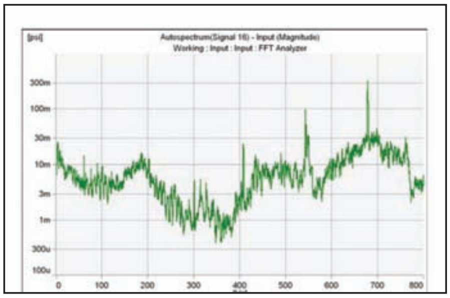


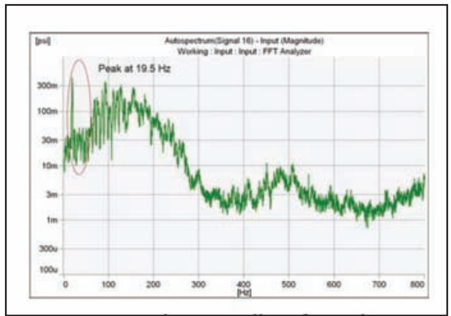
Clockwise from top
Figure 1: Vibration spectrum measured at the discharge valve after the first elbow from the compressor in the horizontal direction
Figure 2: ODS test results during rotating stall of the compressor (50% load), showing motion of the discharge piping pivoting on the compressor at 19.5 Hz
Figure 3: ODS tests showing the motion of the condenser nozzle pivoting on the shell structure at 19.5 Hz
Figure 4: Typical frequency response spectrum of the compressor discharge pipe during the impact modal testing measured in the horizontal direction
Figure 5: Mode shape of the discharge piping at natural frequency of 19.8 Hz with the compressor non-operating. Rocking motion seen on the piping pivoting on the flange

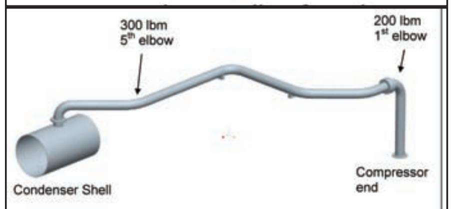
Data for the modal test were acquired using tri-axial — three orthogonal directions — accelerometers at approximately 200 locations/direction, distributed on the compressor, gear, and discharge piping. The impact modal testing determined natural frequencies and their mode shapes while the unit was not operating.

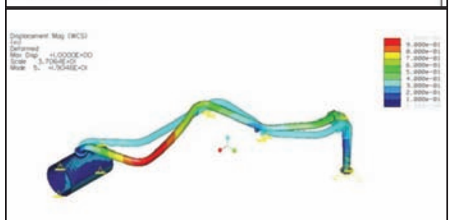
This modal testing technique can be applied with the system running, but the procedure takes more time. Therefore, only limited in-operation modal tests were performed, mainly to confirm natural frequencies did not shift during operation. The forced response testing — ODS testing — was performed at the maximum load of the compressor, with data acquired at approximately 530 locations/direction in three orthogonal directions. Testing was carried out at different loads, too.

During full load operation of the compressor, the maximum vibration amplitude was measured at approximately 0.16 inches/sec peak at the engine speed (1,500 rpm or 25 Hz). The peak vibration for single compressor operation was located at the









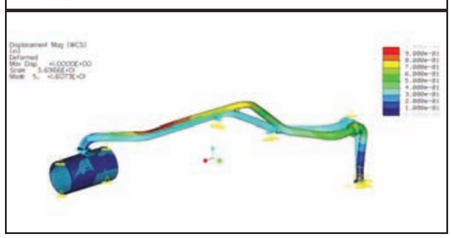


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Figure 6: Dynamic pressure transducer reading from the compressor discharge during full load of the unit

Figure 7: Dynamic pressure transducer reading from the compressor discharge during rotating stall of the unit

Figure 8: Solid model of the discharge piping and the proposed modification by attaching 200 lbm at the 1st elbow and 300 lbm at the 5th elbow

Figure 9: Mode shape of the offending mode at 19.1 Hz calibrated approximately with the field test result (19.8 Hz)

Figure 10: Mode shape of the offending mode shifted down to 16.1 Hz after adding masses at the 1st and 5th elbow

inboard casing of the engine, which was found to be acceptable, as was the bending stress level on the piping (3.45 ksi).

However, when the compressor was operating in parallel with a sister unit, sharing the refrigeration load, the discharge piping of the tested compressor started to vibrate violently, increasing the bending stresses at the location where previous failures took place. The maximum vibration amplitude was measured on the discharge piping with approximately 1.63 in/s peak at 19.5 Hz (1,170 cpm) or 26.6 mils peak-to-peak in displacement (Figure 1).

The bending stress of the piping at the compressor end increased from ±3.45 ksi to ±13.8 ksi (including the stress concentration factor), which was close to the endurance limit of the material used for the piping (approximately 15.0 ksi for an ASTM A53 Gr. B seamless black steel). The ODS plot at a “strong” subsynchronous frequency of 19.5 Hz showed the discharge piping rocking and pivoting at the compressor discharge flange, while the rest of the unit remained stationary (Figure 2). Figure 3 shows the ODS plot of the condenser-evaporator shell stack during the same sub-synchronous frequency. Rotating stall in the compressor stage was determined to be the cause of this phenomenon.

Experimental testing

Experimental modal analysis testing was conducted by producing an impulse excitation at a high-leverage location for modes of potential interest, and measuring the frequency response at locations enveloping the main components of the unit individually: Compressor, gear box, and discharge piping. The impulse was provided through a specially instrumented, modally tuned impact hammer, and the response was measured using tri-axial accelerometers placed on the unit while the compressor was not operating.

Figure 4 shows a typical spectrum of the frequency response measured on the piping in the horizontal direction. The spectrum shows the discharge piping structural natural frequencies at 5.9 Hz, 8.0 Hz, 11.4 Hz, and 19.8 Hz. The natural frequency at 19.8 Hz had the strongest response.

Figure 5 shows the plot of the mode shape in the horizontal direction, where the discharge piping is rocking side to side pivoting on the compressor flange as seen during operation. The compressor and gear box modal testing by itself did not reveal any harmful structural natural frequency on those structures at or close to the running speeds.

Impact of instability

A dynamic pressure transducer was temporarily installed at the discharge of the compressor. During full load of the compressor, the spectra of the pressure transducer did not reveal any anomaly (e.g., Figure 6).

However, when both the engine-driven compressors operated simultaneously at part load (around 50% each), with independent evaporator-condensers, and with two chillers in service, the evaluated compressor became aerodynamically unstable. The characteristic of the spectrum changed completely, showing a strong peak at 19.5 Hz (Figure 7) as a result of rotating stall of the compressor.

This aerodynamic instability recirculated the discharge gas, and generated pressure pulsations at 19.5 Hz. This pressure pulsation tuned into the 19.8 Hz structural natural frequency of the piping, dramatically increasing the vibration level of the piping, and therefore the piping stresses, in a classic case of fluid-structure interaction.

At the compressor end, the peak bending stress generated during the rotating stall of the compressor was measured to be on the order of 13.8 ksi. Therefore, the mechanism of the failures could be described as accumulation of alternating stress cycles under a resonant condition of the piping system. Since the condenser-end nozzle was repaired recently, before conducting the vibration testing, with application of a large amount of weld material, its stiffness had increased, reducing the stress level. Therefore, the discharge piping at the compressor end was considered the weakest point with regard to failure, due to its measured elevated stress.

The operation of these compressors at partial load had been considered a valid performance point by the OEM, especial-

Natural frequencies from field testing (Hz)	Natural frequencies from FEA Baseline (Hz)	Mode Shape description
5.87	6.15	4th elbow vertical bouncing & horizontal overall motion
8.0	8.51	Vertical pipe twist + 4th elbow bouncing + horizontal motion
11.4	14.3	2nd and 3rd elbow vertical bouncing
19.8	19.1	Vertical pipe pivoting + 5th elbow bouncing

Table: Piping Natural Frequencies from the Field Testing and FEA

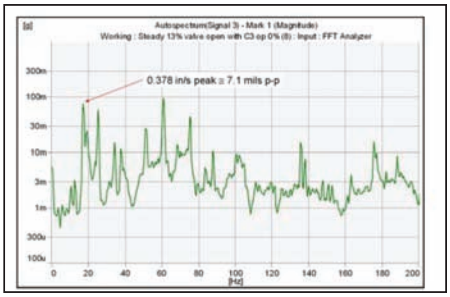

Figure 11: Vibration spectrum after implementing the modifications show reduced vibration levels

ly during the spring and fall seasons. It was possible to redesign the compressors, with vanes of variable diffuser angle to avoid rotating stall. However, this upgrade would have required a high investment and down-time of the units to implement the appropriate modifications. Instead, FEA was performed to determine a potential alternate fix and detune the offending natural frequency.

A solid model of the discharge piping capturing all the details of the pipe dimensions, flanges, a portion of the condenser, and spring-loaded-support locations were constructed using a Computer Aided Design solid modeling program. The solid model of the piping was then transferred into a finite element program for the analysis, and closely calibrated, based on field test data. Table shows a comparison between the natural frequencies registered during the testing and the natural frequencies predicted in the analysis (baseline) with a brief description of the mode shape.

The piping vertical bouncing frequency could not be as accurately predicted as the other test-determined natural frequencies, because of dimensional tolerances and imperfections in flange face-to-face contact. Nevertheless, the results were accurate enough to assess relative benefits of proposed options, particularly for the 19.1 Hz mode.

A quick fix that works

Based on iterative FEA calculations, by adding 200 lbm attached at the 1st elbow, and 300 lbm at the 5th elbow from the compressor discharge piping (Figure 8), the calculated structural natural frequency shifted downwards from 19.1 to 16.1 Hz (Figures 9, 10). By the third quarter of 2007, the OEM and the end-user implemented this fix recommendation on a trial basis by tack welding several split-plates around the piping to provide the required weight, and the vibration level reduced considerably and sufficiently (from 1.63 inches/sec peak to 0.38 in/s peak), particularly during the rotating stall of the compressor.

Figure 11 shows the spectrum of the piping after modification and operating both gas engine driven compressors operated simultaneously at part load (~50% each). This relatively easy and quick fix allowed the end-user to continue uninterrupted operation without shutting down the compressor or without need of additional capital investment. TI

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