Review of Centrifugal Fan Design Decisions Over the Past 25 Years of Operation

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This paper reviews the design and operational decisions that were made in 2005 during the replacement of a double-width, double-inlet, 75-inch-diameter, 9,000 lb., dual-speed, centrifugal fan wheel for a Q-BOP blast furnace due to excessive cracking in the wheel. The original centrifugal fan showed excessive cracking after 10 years of speed cycling duty. The replacement fan has been operating for more than 10 years and shows no signs of fatigue cracking. However, decisions were made to change the blade type and wheel material due to the original finite element analysis (FEA) and plant experience. This paper reviews the original analysis by an Algor FEA package and decisions by the plant personnel, and evaluates the process against a present-day ANSYS FEA analysis and over a total operating experience of 25 years.

A domestic U.S. steel plant operates a Q-BOP shop that utilizes two sets of two 1,100-hp baghouse induced draft (ID) fans, which were put into operation in 1995. These were 74-inch-diameter, double-width, double-inlet (DWDI) airfoil (AF) blade fans designed for 1,180 RPM and 275°F, provided by another manufacturer. The fans handle gas downstream of a baghouse.

Operation

In May 2005, one of the fans failed catastrophically. Two blades separated from the wheel, exited the steel housing, and reportedly struck a nearby building. Following this failure, the identical sister fan was inspected and found to be similarly cracked, and almost near another catastrophic failure. The fan wheels were subsequently sent to the New York Blower facility for inspection and evaluation. The visual inspection strongly suggested a fatigue-type failure. This suspicion was further supported by the fact that these fans operated cyclically via two-speed motors several times per 8-hour shift 24/7 continually between 393 and 1,180 RPM. The second set of identical fans operated at single speed and had no cracking problems.

A piece of the failed blade was sent to the steel company's research facility for analysis. The material was confirmed to be ASTM A514. Electron and optical microscopic examination revealed concerns that proper pre-heat and/or proper heat input control for this steel may not have been followed during fabrication. This led to cyclic stress concentration at or near the root of fillet welds. Design-related deficiencies, especially the use of fillet welds and excessive flexing of the blades, were believed to aid the cyclic stress concentration.

A redesign was undertaken to minimize these suspected design deficiencies. The redesign process included finite element stress analysis (FEA). The purpose of this paper is to compare those analyses with current-day practices.

The original AF wheel was modeled using ALGOR analysis software, using static stress with linear material models.

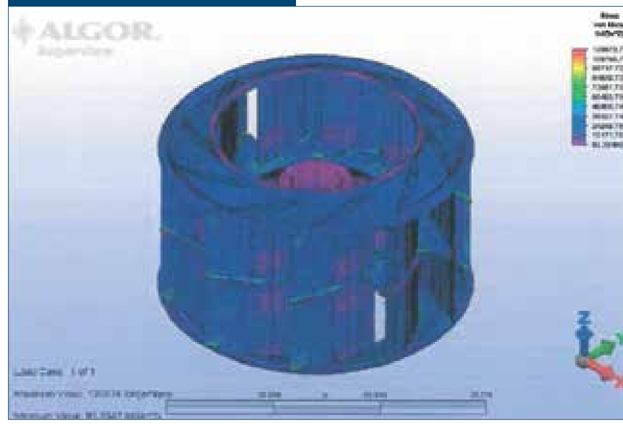
Analyses

Table 1 tabulates the properties of the wheel material used in the analyses. Fig. 1 illustrates a general overview of the model showing relative von Mises stress levels. This shows that the majority of stress is under

Table 1

Material Information	
Material model	Standard
Material source	Algor Material Library
Material source file	C:\Program Files\ALGOR\MatLibs\ algormat.mlb
Date last updated	2004/09/30-16:00:00
Material description	Quenched and tempered
Mass density	7.35e-4 lbf*s^{2}in/in^3
Modulus of elasticity	29e6 lbf/in^2
Poisson's ratio	0.29
Thermal coefficient of expansion	6.53-6 1/°F
Shear modulus of elasticity	11.2e6 lbf/in^2

Figure 1



Overall view of wheel stresses.

50,000 PSI, greater than a 2:1 safety factor on yield strength.

Fig. 2 shows that the maximum stress anywhere on the wheel is 120,000 PSI, which is found in the notched area of the centerplate (web plate). Interestingly, no cracking occurred here.

Fig. 3 shows a close-up view of the blade next to the shroud connection. The peak is 75,000 PSI, which is the second-highest stress found on the wheel. Although the crack did not initiate here, the cyclic flexing was thought to contribute to the cracking in the shroud close to this peak blade stress.

At the request of plant personnel, a new wheel was designed using a single-thickness backward-curved (BC) blade profile instead of the airfoil profile. Analyses used the same program, modeling technique and material properties as the original AF fan. A few iterations resulted in a design that reduced peak stresses and flexing of the blades. A key feature is the inclusion of blade-stiffening rings midway between the centerplate and shrouds on each side of the double-width wheel. One is at the blade leading edge and one at the trailing edge.

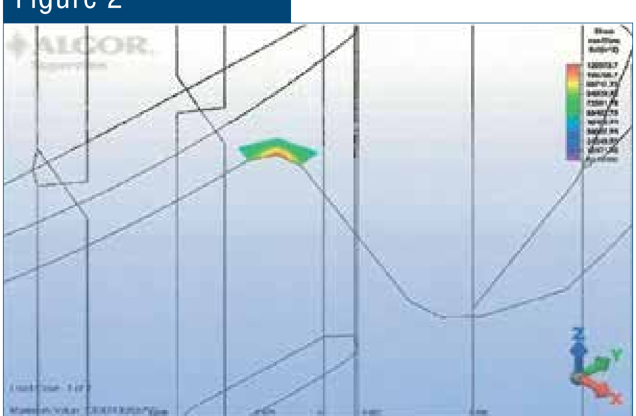
Fig. 4 shows the overall stresses of the BC wheel. Figs. 5 and 6 show close-up views in the key areas of blade tip and intersections of the blades to the shroud, centerplate, and stiffening rings. Peak stresses in these areas are less than 50,000 PSI.

All four fans were subsequently replaced with wheels of this design and have been operating for more than 14 years.

Since 2005, analysis software and other practices have improved. How would this problem be approached today?

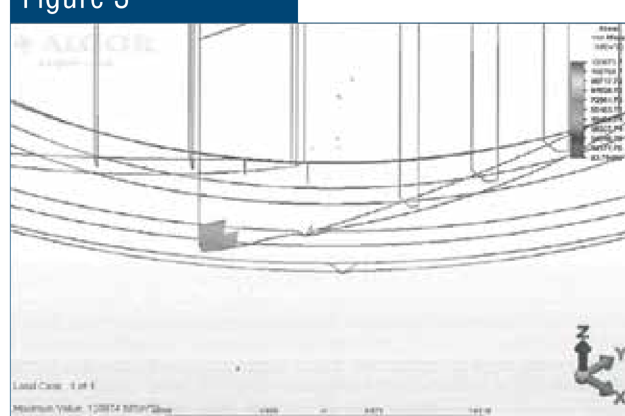
The analysis tools today are much more refined. The current ANSYS program allows for manipulation of mesh size, methods, and the ability to refine the mesh in areas of interest. This greater control over

Figure 2



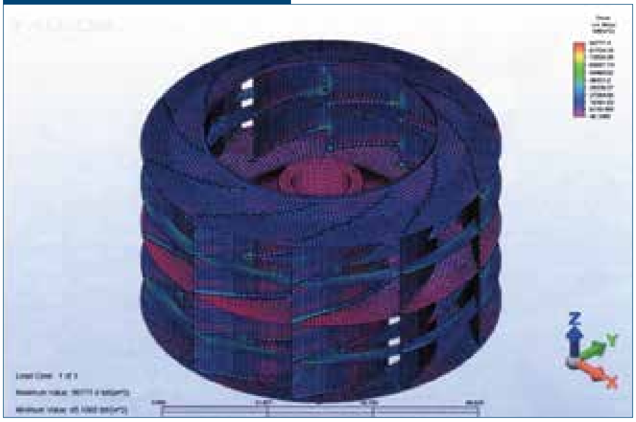
Peak stress anywhere on wheel.

Figure 3



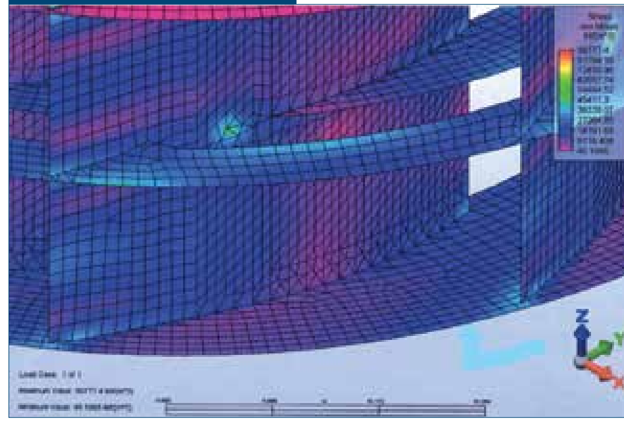
Peak stresses on blade.

Figure 4



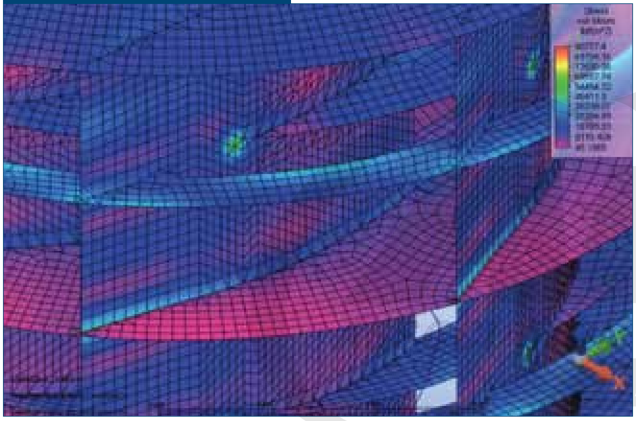
Overall view of backward-curved (BC) wheel stresses.

Figure 5



Close-up view of BC wheel stresses.

Figure 6



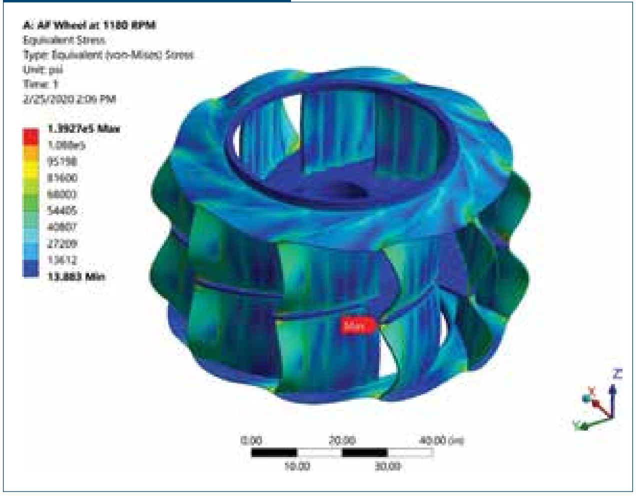
Close-up view of BC wheel stresses.

meshing results in more accurate simulations. ANSYS also offers many post-processing options that can be used to isolate fan components, such as the blade, for further inspection of stress distribution. In addition to stress values, other parameters can be calculated, such as deflection. These provide deeper insight into the operating mechanical integrity of the fan.

Figs. 7–12 show the results of the stress and modal analyses for both the original AF wheel and the replacement BC wheel using contemporary techniques, which can be compared to the previous analyses.

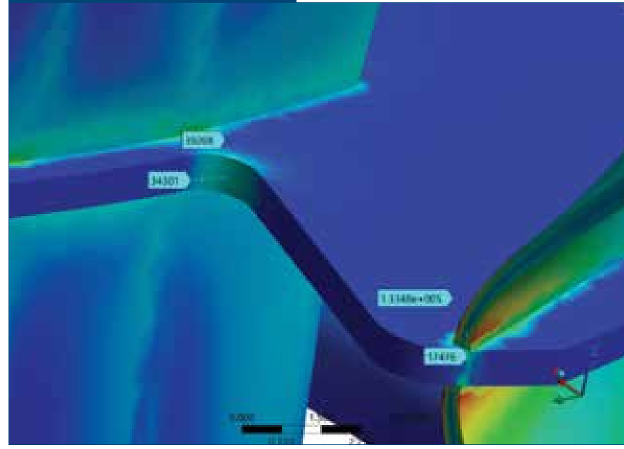
Fig. 7 shows a general overview of the original AF wheel using ANSYS. Note the peak stress occurred in the blade, adjacent to the centerplate, not at the failure point.

Figure 7



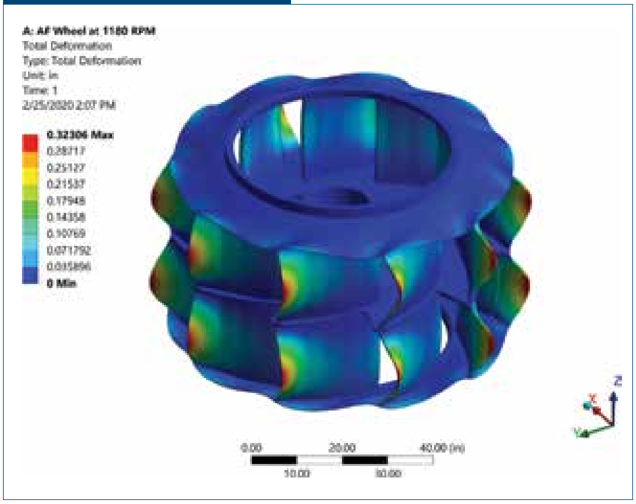
Overall view of airfoil (AF) wheel stresses via ANSYS.

Figure 8



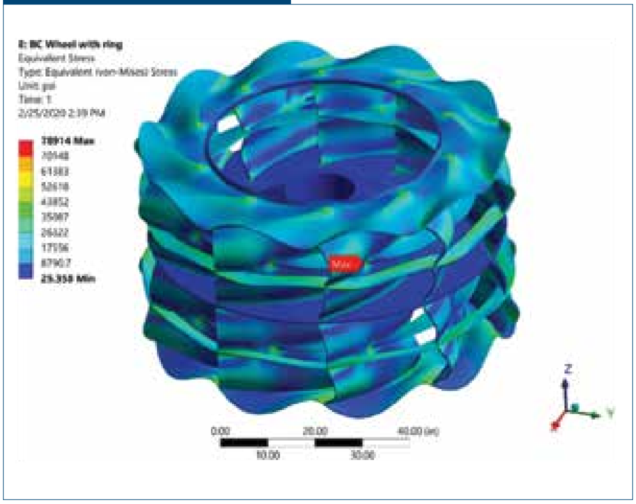
Comparative view to Fig. 2 via ANSYS.

Figure 9



Deformation of AF wheel via ANSYS.

Figure 10



Overall view of BC wheel stresses via ANSYS.

Fig. 8 is the same view as Fig. 2. The peak stress of the original analysis decreased from 120,000 PSI stress to a value here of only 39,208 PSI.

The deformation within the wheel is represented in Fig. 9. The peak value is almost 3/8 inch, located at the center of the blade tip. This provides insight into why the crack initiated in the shroud adjacent to the blades due to the frequent speed cycling.

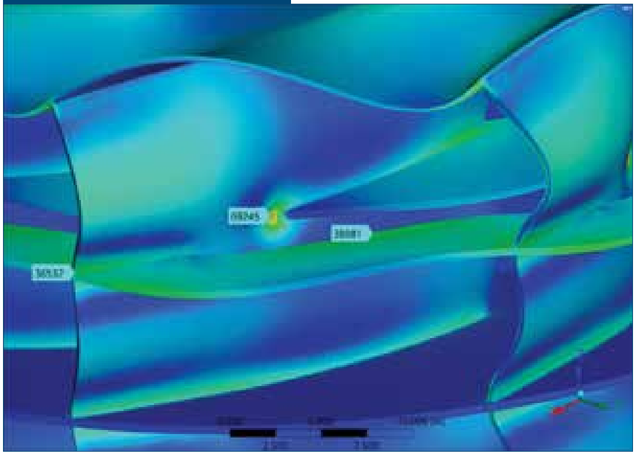
Figs. 10 and 11 are the comparative views for the BC wheel to Figs. 6 and 8 using ANSYS. The peak stress appears to be in the same location as the original analysis at a similar magnitude of approximately 80,000 PSI.

The deformation for the replacement BC wheel is shown in Fig. 12. Note that the maximum value compared to the AF wheel decreased significantly, from 0.323 inch to 0.081 inch. This comparison helps demonstrate the improvement of the replacement wheel design.

Modern analyses include modal analysis, which determines frequencies and mode shapes of the resonances within the wheel assembly. Should a resonance be found near an operating frequency, the design can be modified to eliminate it or move it to a frequency that will not be excited during operation. This is particularly important for the set of fans driven by two-speed motors since there are twice as many possible operating frequencies than the single-speed fans. Plus, any resonances between the two speeds could be excited as the fans cycle between them.

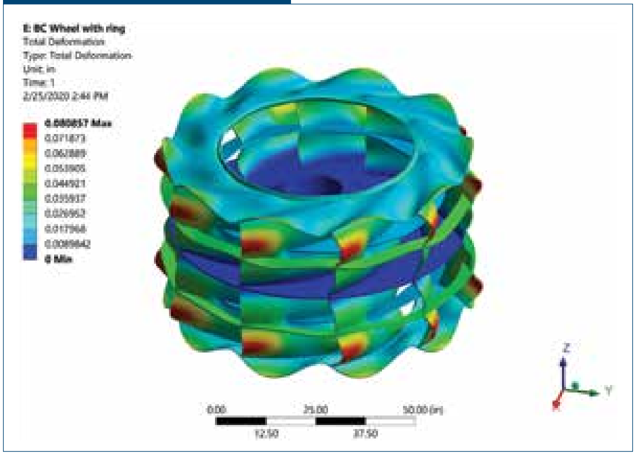
Table 2 lists the frequencies of the primary modes for each wheel design. The mode shape for each is pictured in Figs. 13–16. The excitation frequency for the first diametral, torsional and umbrella modes is the

Figure 11



Close-up view of BC wheel stresses via ANSYS.

Figure 12



Deformation of BC wheel via ANSYS.

Table 2

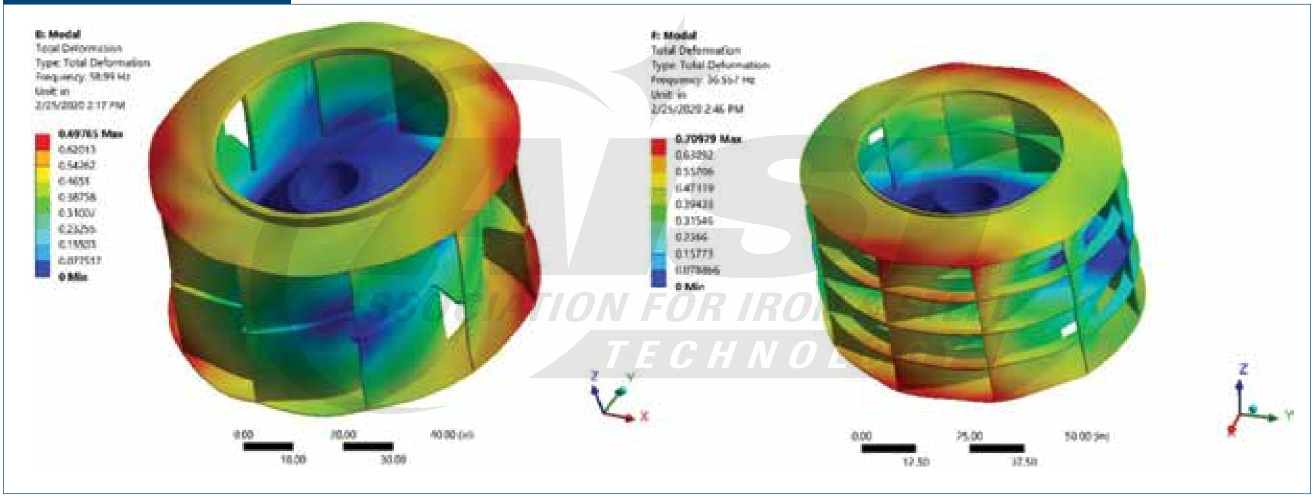
Modal Frequencies for Each Design

Modal frequencies, Hz	First diametral mode	Second diametral mode	Torsional mode	Umbrella mode
Original airfoil wheel	58.990	174.440	168.230	103.000
Replacement backward-curved wheel	36.557	123.680	122.000	55.952

fan operating speed, which is 6.55 Hz at 393 RPM and 19.67 Hz at 1,180 RPM. All modal resonant frequencies are well above these. The excitation frequency of concern for the second diametral mode is six times operating speed, which is 39.3 Hz at 393 RPM and 118 Hz at 1,180 RPM. The second diametral mode resonant frequencies are well above these. Had any of these resonant frequencies aligned with the corresponding modal frequencies, high-cycle fatigue would likely have caused failure much sooner.

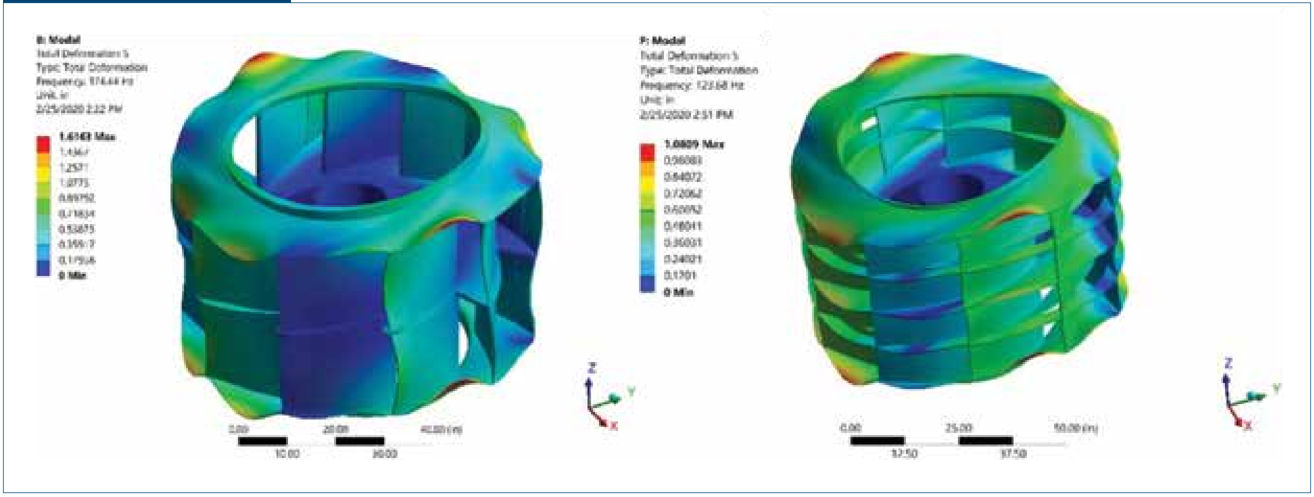
Another critical analysis is fatigue life. Relatively low stress levels can initiate cracking if they are cyclic. The fatigue calculations using the stress results from the ANSYS FEA predict that the original AF wheel will start to yield and form microcracks at the joint between the centerplate and the blade after 17,235 cycles between 393 RPM and 1,180 RPM. This equates to 2 years, assuming continuous operation at one cycle per hour. The BC wheel is estimated to have infinite life at the joint between the centerplate and

Figure 13



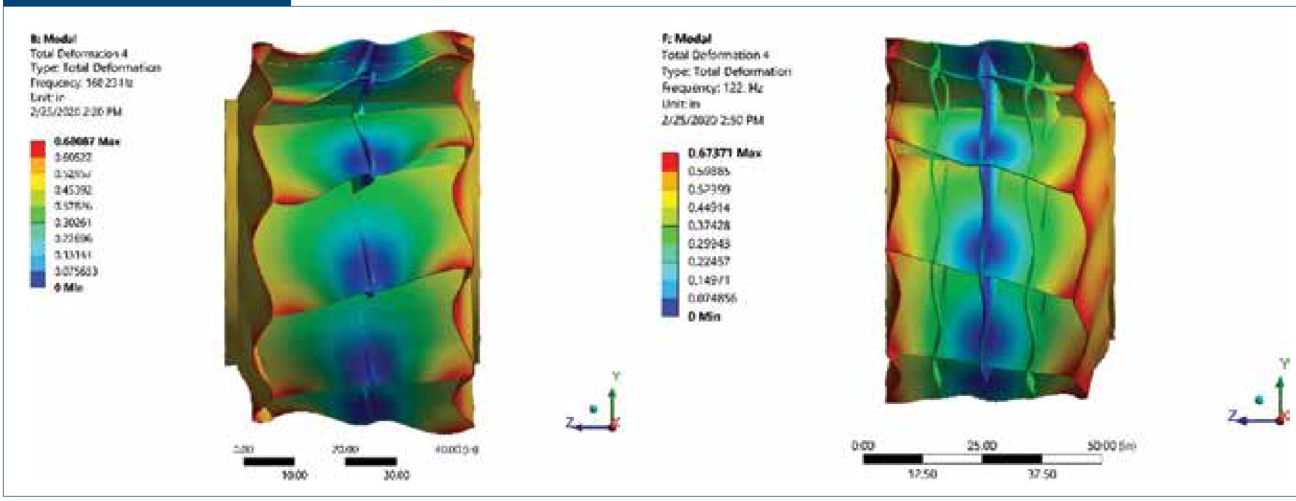
First diametral mode shape of AF and BC wheels, respectively.

Figure 14



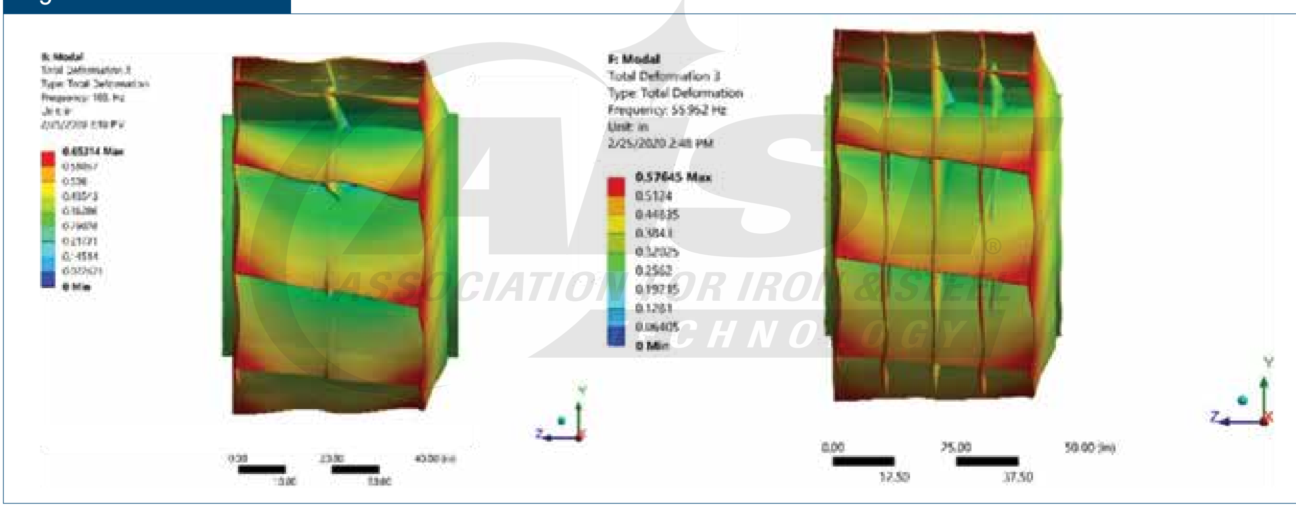
Second diametral mode shape of AF and BC wheels, respectively.

Figure 15



Torsional mode shape of AF and BC wheels, respectively.

Figure 16



Umbrella mode shape of AF and BC wheels, respectively.

the blade, but will start to yield and eventually fail at the joint between the blade and the front plate after 69,587 cycles, or 8 years. The actual history shows that the wheel life far exceeds these estimates. The more realistic conclusion is that the new BC wheel has a design fatigue life that is four times greater than the original AF fan.

Other Considerations

The fabrication of the fan wheel must adhere to the appropriate weld procedures in compliance with AWS D14.6, Specification for Welding of Rotating Elements of Equipment.[1] Had the proper pre-heat, heat input and post-weld cooling control been followed on the original wheels, it is possible that the failures may not have occurred.

During and following welding, non-destructive testing (NDT) is required to confirm the weld integrity. This includes magnetic particle inspection and/or radiographic examination of all welds.

Any analysis includes inherent uncertainties. Assumptions have to be made and variations in fabrication occur. One valuable test that can be used to validate the analyses is a "bump" test of the impeller. While sitting idle in the shop, the test will measure the natural frequencies identified in the modal analysis.

Beyond the mechanical aspects of the fan, consideration must be given to the effect on aerodynamic performance of any design changes. In this application, the change from an AF to single-thickness BC blade would have an effect. The blade shape was decided upon between plant personnel and New York Blower engineers based on reasonable judgment and experience. Today, the design would be validated through a laboratory model test in accordance with AMCA Publication 802, Establishing Performance Using Laboratory Models.[2]

A critical point, regardless of the analytic, fabrication, NDT and testing processes, is to include customer plant personnel in the discussions and decisions. The people that operate and maintain the equipment have the best perspective of the fan's history and can provide valuable insight into what is required for replacement parts.

Conclusion

Advances in analysis and testing technology, fabrication practices and NDT processes described in this work demonstrate that better decisions can be made today when evaluating fan mechanical integrity problems and implementing solutions.

Acknowledgments

The authors would like to thank Brian Schuetz of BJ Associates LLC.

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1. AWS D14.6, Specification for Welding of Rotating Elements of Equipment, American Welding Society, Miami, Fla., USA.
2. AMCA Publication 802, Establishing Performance Using Laboratory Models, Air Movement and Control Association International, Arlington Heights, Ill., USA. ✦

This paper was published in the AISTech 2020 Conference Proceedings. AIST members can access the AISTech 2020 Conference Proceedings in the AIST Digital Library at digital.library.aist.org.