AN IMPELLER DYNAMIC RISK ASSESSMENT TOOLKIT

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ABSTRACT

A standardized and cost-effective risk assessment toolkit is described in this paper to comprehensively analyze centrifugal compressor impellers under steady and dynamic loads. The first step in this process occurs in the proposal stage where the gas loading levels are appraised using a preliminary screening computer program. The application engineers execute the product configuration program for each proposal that invokes the screening program. All designs are screened for steady gas bending stress levels for each client-required operating point. An additional dynamic audit is performed if these stresses are above a given threshold level. The alternating loads are derived from upstream and downstream flowpath disturbances. Natural vibration modes that need to be avoided are evaluated using the SAFE diagram and harmonic response analysis, followed by automated results postprocessing. The overall minimum factor of safety is determined for all impeller model locations and known excitations using a fully automated process.

The use of this advanced analysis approach, which has been developed using a blend of automated computer simulation and broad experience base, results in greater reliability, profitability, and reduced risk for both client and manufacturer. Practical real-world examples of impellers analyzed using this approach will be presented in this paper.

INTRODUCTION

Centrifugal compressor impellers are being engineered today with higher power densities, increased gas pressures, smaller package sizes, and the need to minimize initial project costs. At the same time, the requirement to maintain and improve impeller reliability must be preserved. A dynamic assessment of impeller stages is required to achieve these objectives. An impeller is a key component of a compressor. It is subjected to inlet and exit flow variations through the stage, and, therefore, it must be designed to withstand the alternating pressure loads due to these variations in addition to withstanding steady loads. While the requirement for reliable and robust impeller design can be addressed with elaborate engineering analyses and testing, including static/dynamic finite element analysis (FEA) and transient computational fluid dynamics (CFD), it is still subject to time and cost constraints. These analyses could become cost prohibitive for the custom compressor manufacturer that designs each machine uniquely for client requirements. The historical approach is to satisfy steady-state stress requirements and consider modal interferences on special designs or field issues. The use of an interference or SAFE diagram is an excellent first step in the dynamic process, but impellers will often show multiple interferences between mode shape and excitation shape with no clear method to resolve the severity. Although this will definitely help improve reliability, the absence of the analytical component dealing with the evaluation of alternating loading and resulting stress at potentially resonant frequencies remains an issue. In order to calculate alternating stress levels, it is essential to estimate the dynamic pressure forcing functions. The loading on an impeller can be easily influenced by several aerodynamic flow phenomena such as swirl, wake formation, flow separation, stall, surge and other fluctuations.

Marshall and Sorokes (2000) have described the various flow phenomena causing forced vibrations of centrifugal compressor rotors. Impeller vibrations are adversely affected by the same phenomena. Due to the inherent complexity of these phenomena, accurate estimation of the alternating forcing function becomes difficult without a high-end transient CFD analysis and/or testing. Another approach to estimate the forcing function has been to use a portion of the steady loading for the alternating load component. The turbine industry has long used steady stage horsepower for estimating the excitation levels. However, this approach must be validated through various channels, including CFD and field experience, in order to be successfully applied to different impeller families with variable geometry.

IMPELLER STRESS DESIGN CHALLENGES

Steady Stress Considerations

A first key element of any impeller stress design process is to ascertain the steady load carrying capacity of a particular design. This, in turn, is governed by the properties of materials used for the impeller. Various approaches have been used to predict the centrifugal load deflection and stress on impellers including 1D, 2D, and peak nodal stress in 3D solid element FEA models. Cameron, et al. (Cameron, Geise, Abbott) have presented a process where impeller overspeed limits are established based on steady-state FEA impeller models using the peak element stress. The elemental stress approach based on a standardized mesh has been shown to provide excellent correlation with test data.

Modal Analysis

In the subsequent steps of the process, the impeller natural frequencies are calculated by means of a modal analysis. In turbomachinery design, the Campbell diagram has been used for many years, but the SAFE diagram (Singh, et al., 1988; Singh, et al., 2003) represents a significant improvement with additional mode shape information.

The SAFE diagram is used to check if vibration modes fall within the speed range of a machine. If they do, the most prudent solution would seem to be to make modifications to the design so as to move the natural frequencies out of the speed range. However, for a complex structure such as a closed impeller, this usually proves to be a challenge. In most cases, there are several mode frequencies that could be excited by the forcing from stationary structures upstream and downstream of the impeller. In those cases, the risk usually can be minimized by changing the vane count. Although this solution may address the problem for a simple case, there are several cases where a change in the count of stationaries does not provide an interference-free situation and still results with some modes ending up in the speed range at either the fundamental excitation or at higher harmonics of that excitation. This situation then needs to be addressed by evaluating how much of a risk any given interference poses. This can be achieved by conducting a harmonic response analysis for the resonant conditions of concern and then evaluating the alternating stress levels at those conditions.

Excitation Estimation

The challenge in doing a harmonic response analysis comes from the difficulty in accurately estimating the alternating load levels on an impeller for each excitation type. While high-end CFD analysis or testing can be employed to estimate the loading, it is highly unlikely to be used on every impeller being fabricated at a manufacturerís facility mainly due to constraints of time and resources. On the other hand, the stage horsepower can be successfully used to estimate the alternating loads on an impeller without additional time or cost to the overall project. Figure 1 shows the estimation of alternating load from the steady horsepower load on a stage.

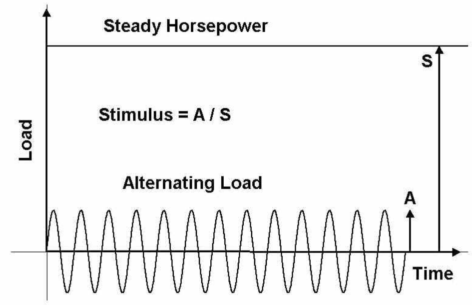

Figure 1. Alternating Load Estimation.

As shown in Figure 1, the derived stimulus is computed as a ratio of the alternating load "A" and the steady load "S." An initial investment in an effort to validate this method with CFD analyses, testing, and experience will provide returns by enabling higher power density impeller designs and reducing warranty costs by improving reliability in the long term.

Harmonic Response Analysis

A commercially available FEA code will apply a specified harmonic loading across a frequency range to any structure and predict the resulting displacements and stress levels. Of particular interest are the solutions at each natural frequency. Unless a code with cyclic symmetry response capability is used, a full 360-degree model is required. A typical impeller may have inlet guide vanes (IGV) upstream and a downstream diffuser with low solidity diffuser (LSD) vanes. The analyst must map these loads on the 360-degree FEA model and solve for the alternating stresses at each resonance. The analyst or external program must also postprocess the enormous quantity of output data generated by these solutions. Recent advancements in technology have resulted in increased computing power and disk capacities on computers for processing and storage of these enormous data. High capacity disk configurations can be utilized to store and process the output generated by the solutions.

TOOLKIT DEVELOPMENT

Historical Results Analyzed

An important step in developing a process for risk assessment was the evaluation of historical data. This was achieved by collecting vital field experience, including field incidents in order to assess the details of operation and levels of stress in each case. A large impeller geometry database was created to include all the field experience data. It should be noted that it is essential to have data of both successful and failed impellers in order to relate the loading history to any acceptability indicating parameter. An automated application was created to execute the evaluation run on all cases in the database. Once this was accomplished, the challenge was then to interpret the results in order to relate to the trends of success and failure. This was achieved by relating the trends as a function of gas bending load on the impeller. The analysis levels were then established using these trends. Figure 2 shows the dynamic assessment value comparison due to the analysis performed.

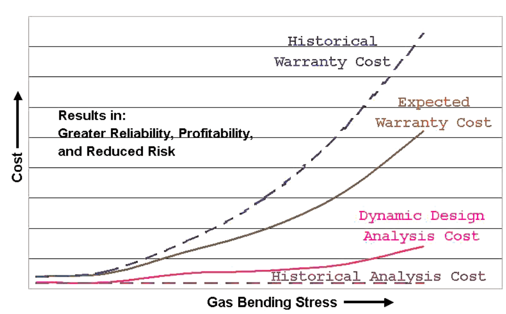

Figure 2. Dynamic Assessment Value.

When no detailed advanced analysis was conducted, the historical warranty costs were maximum with the historically associated analysis costs at a minimum. The implementation of this new toolkit is expected to significantly lower the warranty costs while adding a low fraction of the warranty savings to the dynamic design analysis that is required as a result of the new process.

Risk Assessment Toolkit

One of the key elements of this toolkit is the risk assessment filter screen that is accessed at the proposal stage in the sales regions of order quotation. Figure 3 shows the flow of this process.

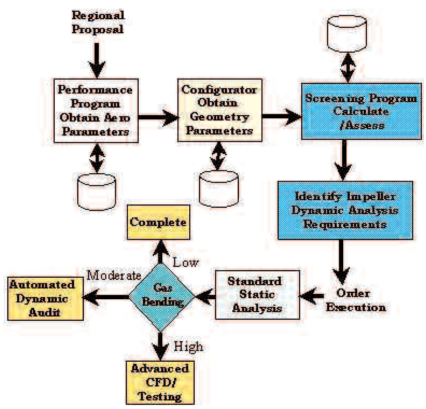

Figure 3. Risk Assessment Process Flow.

The order quotation personnel in the sales regions are the first important link in this chain. By helping identify the gas bending stress levels in the selected impellers based on aerodynamic data for that stage, this tool will facilitate application engineering personnel in making the right selection of impellers for a particular application at the start of the cycle. As shown in the flow diagram, all impellers are screened based on input from the aerodynamic performance database and an impeller geometry database. The design screening program then calculates the gas bending stress levels in the impeller based on a “worst case” loading situation estimated automatically based on various performance parameters. Once the stress levels are calculated, a decision must be made on the acceptability of the stress levels for a particular design. Once the highly loaded impellers are identified, they are flagged for further analysis. This approach facilitates multiple dynamic analysis levels based on gas bending stress, construction method, materials selected, flow coefficient, and other design parameters. As shown in Figure 3, depending on the load level and other parameters, the future dynamic analysis direction is chosen:

- No additional analysis,
- Standard dynamic audit including modal and harmonic response analysis, or
- High-end transient CFD analysis or testing followed by a response analysis.

The results of this early assessment are automatically communicated to various departments involved through electronic mail notification leading to a streamlined process in the project cycle.

Automated Dynamic Audit

The core process of this toolkit is the standard dynamic audit described in more detail below. Once a notification for performing a risk assessment is received, the next step is to generate a finite element (FE) model for the impeller to be assessed. The model being generated must have a mesh density that is adequate for the accurate computation of impeller natural frequencies. Impeller modal tests were conducted followed by correlation studies to establish acceptable levels of FE mesh density to be used on models. Figure 4 shows a typical mesh density used for a closed impeller. The entire analytical process is automated and follows the flow shown in Figure 5.

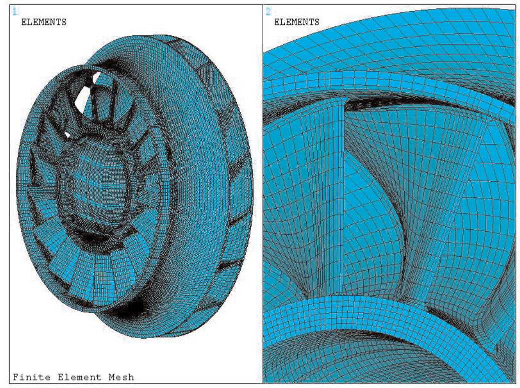

Figure 4. Typical Impeller Mesh Density.

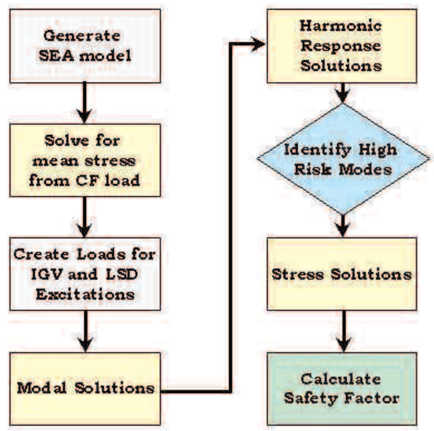

Figure 5. Automated Dynamic Audit Process.

Each solution routine is conducted by a commercially available FEA code. The automated process starts with a static solution at the maximum compressor speed with the available FEA model for an impeller, and the mean stress at each node in the model is computed. Alternating pressures resulting from upstream and downstream vanes are placed on one impeller blade pitch and converted to nodal forces. Once complete, the model is ready for a modal solution to calculate impeller natural frequencies of vibration including prestress resulting from centrifugal speed and shrink load. The modal solution will compute impeller natural frequencies that are used to generate a SAFE diagram in order to help identify any interference with excitations. The single blade nodal forces are copied to all blades in the 360-degree model for the nodal diameter pattern required with each excitation. The displacement harmonic response is then computed for all excitations. A response stress solution then calculates the alternating stress levels for all the high-risk modes. The mean stresses for each interference are calculated by scaling them from the centrifugal stresses previously calculated at maximum compressor speed. Finally, a factor of safety is calculated at each node in the FE model at each identified frequency. An additional program feature is the ability to consider the weld material properties in the calculation of the factor of safety. The minimum factor of safety is summarized for all vibration modes and all excitations within the machine speed range. The model generation process is standardized and the entire

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remaining process is fully automated so that operator intervention is not normally required. The dynamic audit process has resulted in well over an order of magnitude reduction in solution effort and detailed dynamic calculations on production impellers can now be performed efficiently.

CASE STUDIES

Case Study 1

The intent of this case study is to examine the successful application of the risk assessment toolkit to a field problem that was resolved by redesign of the impeller before the release of this toolkit. Several trains with the identical compressor using similar impellers were in service at various locations. An impeller had experienced field incidents with two issues. A scallop-shaped piece broke off at the outer diameter (OD) of the disk and there were crack indications near the blade leading edge. This impeller design (Case A) had 15 blades and 16 IGVs. Figures 6 and 7 show the typical mode shapes associated with this kind of indication pattern.

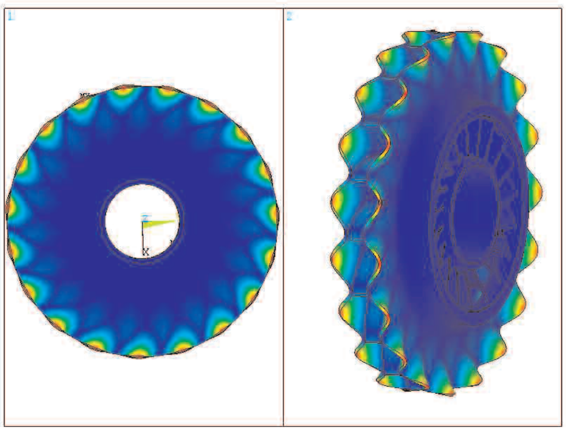

Figure 6. OD Scallop-Out-of-Phase Mode.

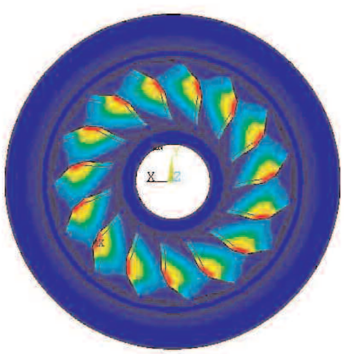

Figure 7. Blade Leading Edge Mode.

This impeller design was modified (Case B) with 17 blades and 16 IGVs. This impeller also experienced incidents but only with blade leading edge cracks. A third redesign modified the impeller geometry and reduced the number of IGVs from 16 to 12 (Case C). This impeller was put in service and has been operating successfully for several years.

This case was selected to apply the risk assessment toolkit methodology in order to compare all three designs (A, B, and C). Models were created for the impellers and run through the automated routines to calculate the factors of safety. Figure 8 shows the factor of safety plot of all three impeller designs against the compressor speed range.

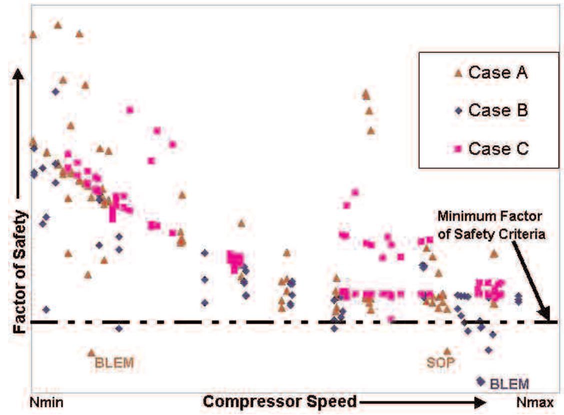

Figure 8. Dynamic Audit Results—Case Study 1, Cases A, B, and C.

Each symbol on the plot is a worst case node for one mode/excitation combination. On a typical impeller there can be 20 to 30 resonances with three to eight excitations per resonance. Case A impeller had a blade leading edge mode (BLEM) interfering with the fundamental excitation from the 16 IGVs and a scallop-out-of-phase (SOP) mode interfering with the same excitation. Both vibration modes were identified by the dynamic audit tool for assessment and the factors of safety computed were below the acceptable criteria. Case B impeller also showed a BLEM with a factor of safety below the required criteria but no scallop modes were identified. Case C passed the factor of safety criteria for all mode/excitation combinations. These results were consistent with the field incident behavior of all three cases, thereby validating the risk assessment toolkit methodology.

Case Study 2

This case study shows a successful impeller design that has been operating in the field for several years. The impeller FE model was run through the risk assessment toolkit for verification. Figure 9 shows the resulting factor of safety plot versus compressor speed.

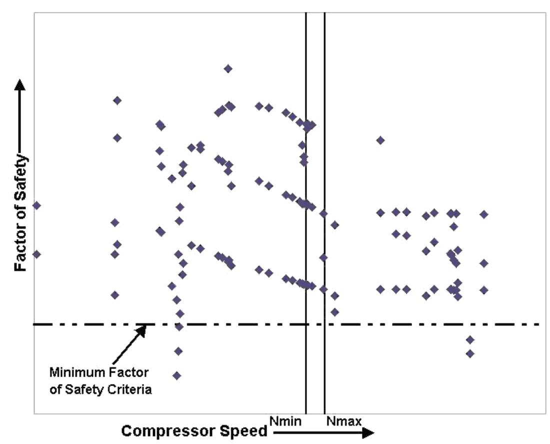

Figure 9. Dynamic Audit Results—Case Study 2.

Note that the speed range for this case is smaller than that of case study 1. The speed range from minimum compressor speed, Nmin, to the maximum compressor speed, Nmax, is of interest for assessing the risk for this impeller. Though there are several responsive modes in the speed range that would show up on the SAFE diagram as interferences, the factors of safety for all the mode/excitation combinations are above the minimum criteria line. The fact that this impeller has been running successfully for several years corroborates the toolkit criteria limits, and, thus, demonstrates the successful application of the toolkit methodology.

Case Study 3

This case study shows an impeller designed for maximum reliability by selecting a speed range that has no responsive mode/excitation combinations. Figure 10 shows the results of a dynamic audit performed on a field impeller.

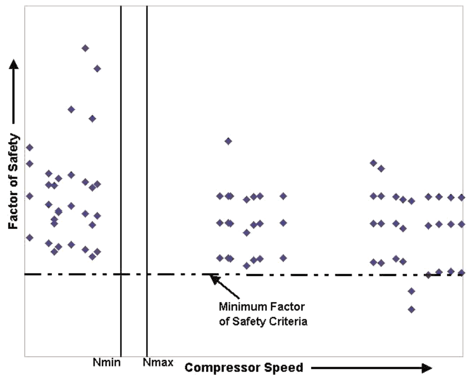

Figure 10. Dynamic Audit Results—Case Study 3.

While it is difficult to have the speed range of a compressor fully governed by maximum impeller reliability, this still would be an option for some compressors with a fixed count of stationary vanes upstream and downstream of the impeller. As shown in Figure 10, there exist bands in regions of the plot where the impeller does not have a strong enough mode/excitation combination so as to generate a response. The area within the speed range from Nmin to Nmax is free of any responsive modes that could lead to high alternating stresses and a low factor of safety. Selecting such a speed range would then result in a highly reliable impeller design.

CONCLUSION

The purpose of this paper was to present a risk assessment toolkit developed by the authors and in use at their company. The existing static analysis approach has been greatly enhanced by this toolkit and a fatigue-based approach can now be used with much reduced effort and time to address impeller reliability. Historical incidents were used to establish levels of analysis required as a function of impeller loading. Each impeller chosen by the application engineers is screened using the computer program at the proposal stage and identified for potential risk. The dynamic audit tool is fully automated to perform modal analysis and harmonic response analysis, and provides a consistent approach to calculate a factor of safety for all impellers built by the manufacturer. In summary, the use of this toolkit results in better understanding of impellers in a dynamic environment and provides greater reliability, profitability, and reduced risk for both the client and the manufacturer.

NOMENCLATURE

BLEM	= Blade leading edge mode
CFD	= Computational fluid dynamics
FEA	= Finite element analysis
HP	= Horsepower
IGV	= Inlet guide vane
LSD	= Low solidity diffuser
OD	= Outer diameter
RPM	= Rotations per minute
SOP	= Scallop out of phase

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