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ABSTRACT

Based on the development of numerical analysis techniques over the past decade, the engineer can easily model the internal flow conditions and vibration characteristics of turbomachinery. When focusing on vibration response of rotating blades, it is easy to predict the resonance frequency; however, difficulty remains in the prediction of vibrational stress. The primary reasons include:

- The unsteady external excitation force and vibration response are complicated making it difficult to calculate these characteristics precisely, and
- It is likewise difficult to confirm the simulation results of vibration response since the actual measurement of vibrational stress on the rotating blades presents its own set of challenges.

The authors already introduced the measured actual stress on mechanical drive steam turbine blades at the resonance condition with nozzle wake during full load testing and compared the results of computational fluid dynamics (CFD) and finite element analysis (FEA).

On the contrary, there is not much study or discussion regarding compressor impeller resonance stress based on measuring actual stress. Considering that compressor impeller sizing has increased, it is important to deeply study the vibration phenomena relating to internal flow and blade cascade interaction excitation forces and create a clear criteria for designing large, robust impellers.

In this regard, the authors introduce a unique stress measurement technique for rotating blades at full load by specially designing flexible, high response impellers to emphasize the vibration stress.

The basic vibration characteristics of the nodal diameter interference diagram and Campbell diagram are explained. In order to estimate vibrational stress on the blades, the unsteady CFD analysis is applied for calculation of the external excitation force and the vibration response is analyzed.

The analyzed detail results are compared with the actual measured vibration stress and vibration response systematically. Through this study, the resonance with the inlet guide vanes (IGV) and a unique blade/stator cascade interaction phenomena are found. Subsequently, the cause of these phenomena is investigated, and the authors try to explain this mechanism.

Finally, based on the correlation between the measured and analyzed vibration stresses over the entire whole flow path, the strength criterion is studied as a guideline, by comparing application experience.

INTRODUCTION

Background

For the centrifugal compressor design, it is an important point to prevent failure for long term operation. From the viewpoint of impeller failure, static and dynamic stresses should be taken into consideration for proper impeller design. Prediction of static stress is relatively easy compared to dynamic stress because of two factors:
The unsteady external excitation force and vibration response are complicated, making it difficult to calculate these characteristics precisely, and

- It is likewise difficult to confirm the simulation results of vibration response since the actual measurement of vibrational stress on the rotating blades presents its own set of challenges.

Because it is difficult to predict dynamic stress when designing a compressor, the criteria for preventing fatigue failure of impeller tend to be very conservative. For the past decade, as chemical and liquefied natural gas (LNG) plant capacity has increased, so has the size requirements for the compressor. As a consequence, it is necessary to achieve a lower natural frequency for impeller in order to prevent impeller resonance.

The most important point is to accurately predict dynamic stress on impeller and then apply reasonable design criteria. This is the motivation of this study and suggestions on how to solve this question, based on analytical and experimental study, will be presented in this paper.

**Purpose of this Study**

The goal of this study is to predict the frequency and level of dynamic stress for impeller as accurately as possible. Impeller dynamic stress is caused by pressure and velocity distribution in the rotating direction. It is mainly caused by wake of the inlet guide vane, which is installed in front of the impeller. During rotation, the number of wakes times rotating speed is the frequency induced on the impeller. And when the frequency and the impeller’s natural frequency resonate, large dynamic stress fluctuations can occur.

In this study, the authors apply a combination of unsteady CFD + finite element method (FEM) analysis to solve the problem, then confirm the results through experimental measurement.

**Standard Structure of Inlet Guide Vane**

Figure 1 shows the standard structure of a centrifugal compressor. Suction flow comes from the inlet scroll to IGV. The flow is guided in the radial direction by the IGV. After passing the IGV, flow is turned to the radial direction, and turned to the axial direction in order to flow straight into the impeller.

Figure 2 shows the standard structure of a centrifugal compressor IGV. The IGV is a combination of two types of vane—short vanes and long vanes. Thus, wakes result from a combination of short and long vanes. One of the discussion points for this study is the frequency induced to impeller, which is related to number of wakes from the guide vanes. With CFD and experimentation, induced frequency for the complex guide vane is analyzed.

For this study, the authors focus on an IGV design with 28 vanes; however, the company’s standard IGV has 22 vanes. Therefore, the authors applied the nonstandard IGV for this study to avoid an overlap of the IGV induced frequency with the impeller’s own natural frequency.

**Impeller Natural Frequency**

When the impeller natural frequency is harmonized with exciting force, the impeller resonates and large stress fluctuation occurs. To predict the natural frequency resonance, it is important to study the impeller’s natural frequency as well as the mechanism of impeller resonance.

Figure 3 shows the typical mode of impeller natural frequency for the centrifugal compressor. This figure shows the impeller deformation at the 5 nodal diameter (ND) mode natural frequency. And the number of blades (Zb) is 17.

When ND, Zb and harmonics (H: frequency of excitation/rotating speed) follow the relationship below, impeller resonance can occur.

\[
H = i \cdot Zb \pm ND
\]

where i is an integral number (0, 1, 2,…). So for this case, possible harmonics of resonance are 5, 12, 22, 29, ….

Figure 4 shows the natural frequency versus nodal diameter of the impeller, and operation range of the model test. The number of impeller blades is 17. So the impeller is expected to resonate at
3 nodal diameter with 14 harmonics (caused by the wake of long IGV vane) or at 6 nodal diameter with 28 harmonics (caused by wake of the long IGV vane).

Figure 4. Nodal Diameter Family.

APPROACH

The authors applied two approaches to predict the dynamic stress on the impeller:

- Unsteady CFD for the flow field with IGV + impeller and response vibration analysis for the impeller with FEA
- Experimental study for real-scale test to confirm the analytical study

There are many combinations of IGV/impellers for this application, and it is difficult to apply a precise analysis or test for every compressor impeller. So as a final result of this study, the authors propose a stimulus factor that can be applied for every case of impeller design. Commercially available software is available for both CFD and FEA.

Steady CFD Analysis

As a first step, the steady CFD analysis is applied for IGV + impeller to confirm the velocity and pressure field around impeller, and estimate the main component of the effective wakes as an exiting force on impeller.

Analysis Condition for CFD

Steady and unsteady CFD analysis are applied for IGV + impeller. Figure 5 shows the model for analysis. The analysis consists of four flow paths:

- Inlet piping for the compressor suction nozzle (outside of compressor),
- Inlet volute + IGV,
- Impeller and
- Diffuser scroll.

Because dynamic stress induced by the lack of a uniform flow field to the rotational direction of impellers, all dimensional factors for uniformity of flow field, including IGV, are taken into consideration.

Figure 5. Model for Analysis.

Boundary condition of this analysis is as follows.

- Uniform flow at inlet of piping
- Uniform pressure at inlet of piping

Flow rate and pressure correspond to the test condition as follows (refer to Figure 12).

Steady CFD Analysis Result

Figure 6 shows the outlet velocity distribution of the IGV at shroud/middle hub sides of impeller location. Remarkable points of the result are as follows.

1. At the shroud side and middle, the interval of velocity distribution is small. It means that main source is short vanes. On the contrary, at the hub side, the interval of velocity distribution is large. It means that main source is long vanes.

2. At the top side (near the suction nozzle side) the velocity distribution is larger than that of the bottom side (far from suction nozzle) and at the inlet of impeller, the interval of wake is a little bit larger than that of the IGV.
Unsteady Analysis

Prediction Method of Vibrational Stress

Sequence of stress prediction is as follows:

1. Unsteady pressure distributions on the impeller from CFD result are broken down to each harmonic by fast Fourier transform (FFT).
2. Impeller natural frequency and mode shape are simulated by FEM.
3. Harmonics and resonate natural frequency are predicted by Equation (1).
4. Effective external force on each vibration mode is calculated from pressure distribution for the frequency of CFD result and mode shape (distribution of modal deformation) of FEA result.
5. Maximum stress on impeller for each mode will be predicted by FEA using the calculated effective external force.

Analysis Result

Table 1 shows the summary of the result for relative stress vibration of CFD+FEA analysis. For this analysis, four harmonics (14, 27, 28, 29) are taken into consideration. Each result is normalized by the result of 14 harmonics. From this result, 14 and 27 harmonics are the main components of impeller resonance.

<table>
<thead>
<tr>
<th>Harmonics</th>
<th>Relative stress</th>
</tr>
</thead>
<tbody>
<tr>
<td>14</td>
<td>1</td>
</tr>
<tr>
<td>27</td>
<td>0.94</td>
</tr>
<tr>
<td>28</td>
<td>0.17</td>
</tr>
<tr>
<td>29</td>
<td>0.29</td>
</tr>
</tbody>
</table>

For short vane’s resonance, not 28 harmonics (same as vane number), but 27 harmonics (−1 of vane number) is the main component. This may be caused by the large interval wake at the top side that is strongly influenced compared to the small interval wake at the bottom side (Figure 7). This may have an impact for designers wanting to prevent impeller resonance since not only the number of IGV, but also the −1 component should be taken into consideration for induced frequency. And not only the long vane wake but also the short vane wake should be considered. Figure 7 shows the result of pressure fluctuations on the impeller blade surface.

Figure 7. Pressure Fluctuations on Impeller Blade Surface.

From these results, some interesting phenomena are observed. At the impeller shroud side of inlet, the pressure fluctuation is observed both for 27 and 14 harmonics. But the pressure fluctuation of 27 harmonics is larger and wider than that of 14 harmonics. At the middle and hub side, there is no major pressure fluctuation for 27, but at the edge of inlet, small fluctuation is observed for 14 harmonics. As a result, the modal force of 14 and 27 harmonics is the main component for the effective force for vibration of the impeller.

Experimental Study

Tapping Test

Before the actual load test, a tapping test for the subject impeller was carried out. Figure 8 shows the test result. The test result of natural frequency is well matched with the FEM result.
To verify the impeller resonance, it is necessary to measure the impeller response at rotating condition during the load test. The fluctuation of stress is measured by using a strain gauge installed on the impeller with a slip ring system. Figure 11 shows the stress measuring point of the impeller. The measured signal is transmitted to the shaft end by wiring in the middle of the shaft. The signal is then transmitted to the stator side by using a slip ring. The point of measurement is corrected from mode shape of natural frequency by FEM.

Figure 11. Stress Measurement Point.

Figure 12 shows the test condition. Suction pressure and flow rate are fixed with closed loop and the rotating speed is changed from maximum to minimum. During this sequence, stress fluctuations, resonance frequency and rotating speed are measured.

Figure 12. Test Condition.

Figure 13 shows the Campbell diagram of the results. At the speed of 27 harmonics and seventh nodal diameter mode of impeller natural frequency resonance, a large vibrational stress occurs. On the other hand, at 28 or 14 harmonics and sixth or third nodal diameter mode, the impeller may resonate in the sequence, but no large vibrational stress was observed. Therefore, as predicted by numerical simulation, 27 harmonics is the main component for the effective force for impeller vibration for this condition.

Figure 13. Campbell Diagram.
This result means that for the combined IGV with long/short vanes, the long vane wake does not have an impact on impeller excitation at the impeller’s natural frequency, but the short vane wake does affect impeller resonance. Furthermore, it is not the short vane harmonics number, but the \(-1\) number that strongly excites the impeller natural frequency.

**Estimation of Vibration Stress**

By simulation/experiment, the main component of impeller resonance and the stress fluctuation are revealed. In this paragraph, to make these results useful for compressor design, the stimulus factor is taken into account.

**Definition of Stimulus**

Stimulus (S) is the ratio of dynamic/static force. It is easy to predict static force on an impeller, but difficult to predict dynamic force. So from the result of this study, the authors calculate Sc for the prediction of future compressor designs. S is defined as follows:

\[
S = \frac{F_V}{F_S}
\]  

where:  
\(F_V\) = Dynamic modal force  
\(F_S\) = Static modal force

Modal force is calculated as integration of local pressure × modal deformation. 

From dynamic modal force, vibration stress on the impeller is calculated as follows:

\[
\sigma_V = \frac{F_V}{m_i \omega_i^2} \frac{\pi}{\delta_i} \sigma_{V \_rel}
\]  

where:  
\(m_i\) = Modal mass  
\(\omega_i\) = Velocity  
\(\delta_i\) = Log decrement  
\(\sigma_{V \_rel}\) = Relative stress

From these equations, S is calculated as followed based on the test result, static pressure analysis and mode analysis.

\[
S = \frac{\sigma_V \cdot \delta_i \cdot m_i \cdot \omega_i^2}{\pi \cdot F_S \cdot \sigma_{V \_rel}}
\]

**Estimation of Stimulus**

From the result of experiment, stimulus for the tested condition is summarized as Figure 14. Stimulus and flow coefficient have a strong relationship, and as flow coefficient increase, stimulus increase. Based on these data, stress fluctuation can be easily estimated. This result can be very efficient for impeller design of impeller strength for prevention of fatigue failure.

**SUMMARY AND SUGGESTIONS**

By simulation and experimental study, impeller resonance with IGV wake is verified. Also, the main component of resonance frequency is determined to result from the wake of \(-1\) number of the short vane by simulation and experimental study.

For compressor designing, it is important to consider the following:

- Maintain separation of the impeller natural frequency from the short guide vane harmonics.
- Not only is the number of guide vanes important, but also \(-1\) of the number is necessary to avoid impeller natural frequency.
- Stress fluctuation can be easily estimated from the stimulus value verified by model test.

**NOMENCLATURE**

- \(Z_b\) = Number of blade  
- \(H\) = Frequency of excitation/rotating speed  
- \(S\) = Stimulus  
- \(F_V\) = Dynamic modal force  
- \(F_S\) = Static modal force  
- \(m_i\) = Modal mass  
- \(\omega_i\) = Velocity  
- \(\delta_i\) = Log decrement  
- \(\sigma_{V \_rel}\) = Relative stress

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