Aeromechanics Considerations for a Centrifugal Compressor Impeller operation in High Density Fluids

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EBARA ELLIOTT ENERGY

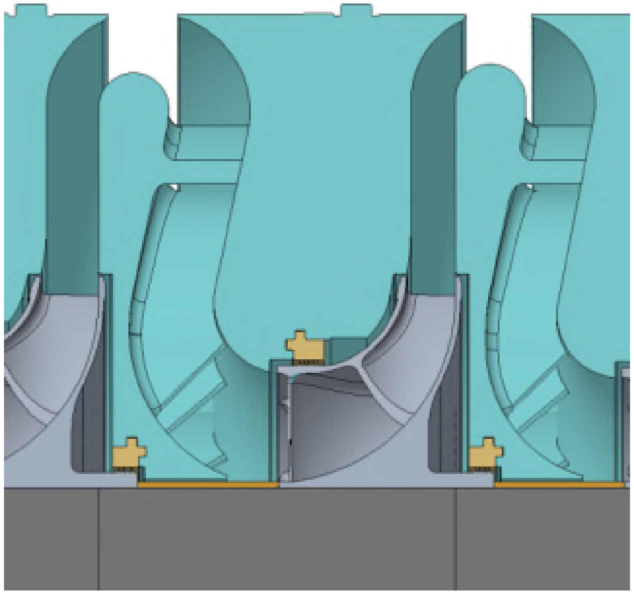
Outline

- Introduction and Motivation
- Impeller Design Consideration
  - Resonance Avoidance
  - High Cycle Fatigue Analysis
- Impeller-Diffuser Interaction Case Study in sCO2 application

Introduction & Motivation

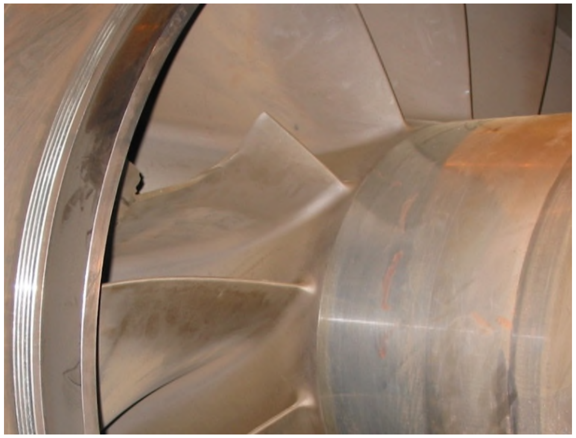
- Impellers have natural frequencies that can be excited by the interaction of flow non-uniformity with impeller blades.

- Sources of excitation (flow non-uniformity):
  - wakes from upstream vanes
  - Reflections from downstream diffuser vanes
  - circumferential pressure variation.



Introduction & Motivation

When an excitation frequency coincides with a natural frequency (forced excitation), there is potential for high cycle fatigue.



Impeller blade failure (Pettinato, et al., 2013*)
Causes: Choke operation, liquids, resonant operation (?)

Impeller disk cover failure (Kushner, et al., 2024**)
Causes: Resonant operation, liquids

Self excited (flutter) failure mode is rare in centrifugal compressor applications….

Pettinato, B., et al.,2013 "Experimentally Based Statistical Forced Response Analysis for Purpose of Impeller Mistuning Identification."Middle East Turbomachinery Symposium, https://hdl.handle.net/1969.1/172623
Kushner F, Pettinato B, Pechulis M, Thorat M., 2024,"Modal Analysis and Resonance Avoidance of Bladed Disks," Turbomachinery Symposium, Texas A&M University, https://hdl.handle.net/1969.1/1582470

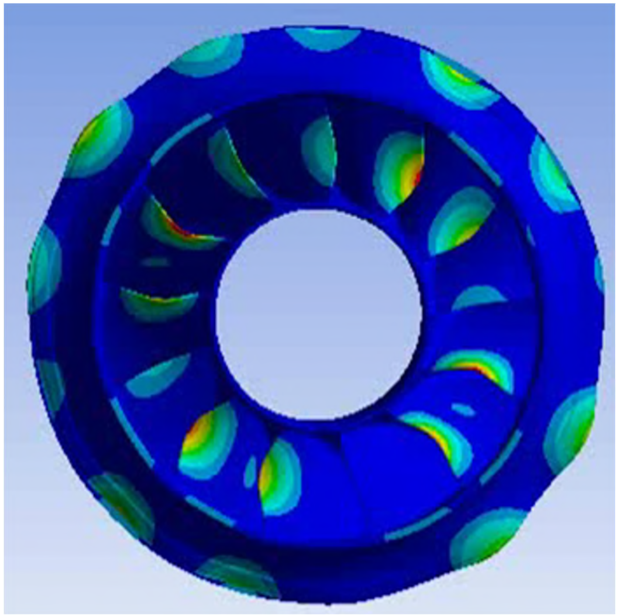
Impeller Design – Aeromechanics Considerations

Failure Avoidance

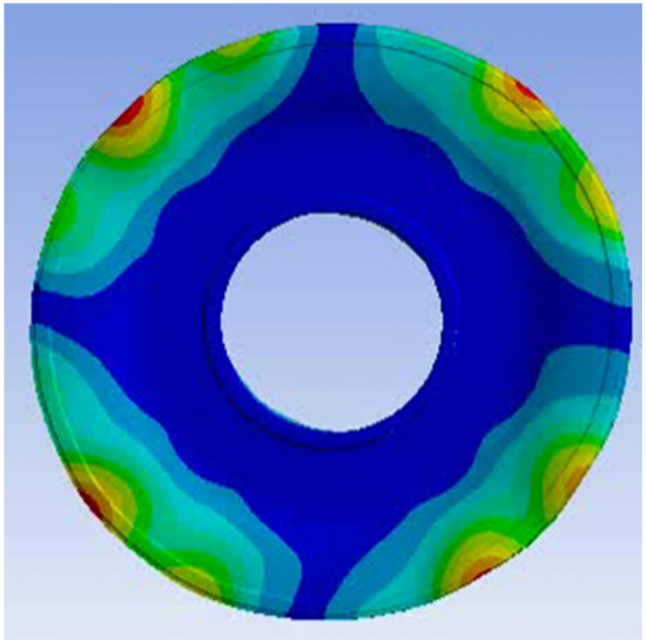
- Avoid disk critical speeds
- Avoid excitable natural frequencies in the operating speed range
- Keep stresses below an acceptable vibratory stress limit

Impeller Design – Aeromechanics Considerations

Impeller Vibration Patterns



2 Nodal Diameter Blade Leading Edge

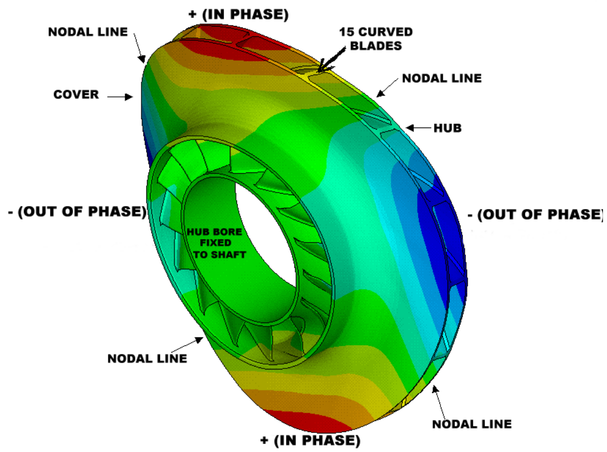


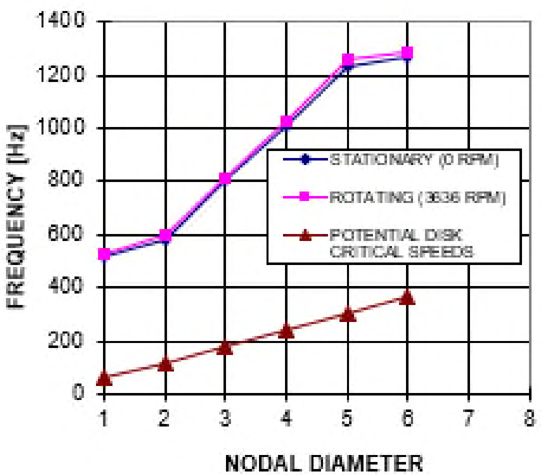
2 Nodal Diameter Disk Mode

Disk Critical Speed Avoidance

Disk Critical Speed Avoidance

- Very few disk critical speed failures. All reputable OEMs avoid these.
- $N_{cr} = \frac{f_r}{n}$
- A disk critical speed will exist when a natural frequency with "n" diameters coincides with "n" x running speed.
- As shown in the example figure, this impeller does not have a disk critical speed when operating at 3636 RPM.

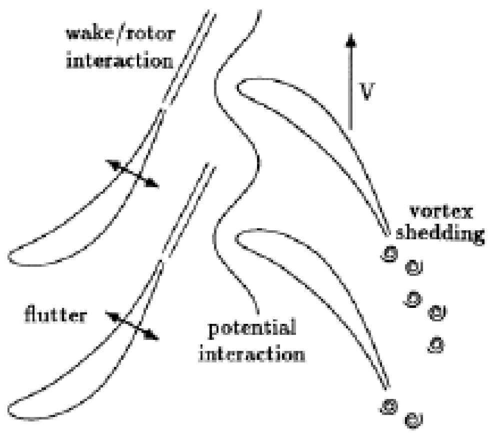


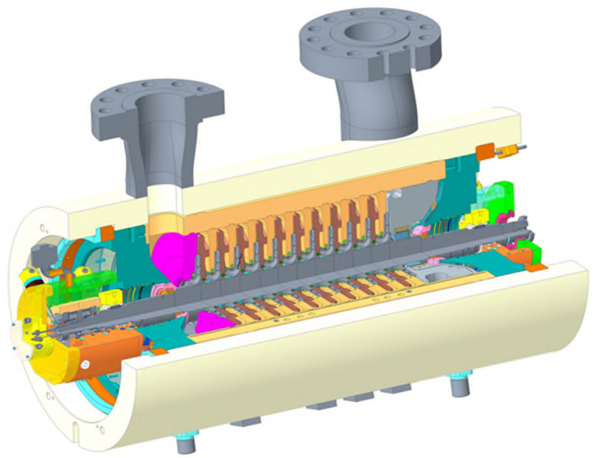


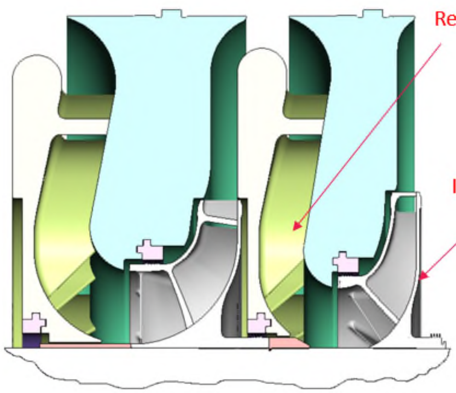
Impeller Interaction Resonance Avoidance

Typical Interactions:

- Upstream Vane Interactions
  - Inlet Guide Vanes
  - Return Channel
- Downstream Diffuser Vanes
- Inlet Distortion (Inlet/Side Stream Inlet Bends)
- Acoustic Coupling







Impeller Interaction Resonance Avoidance

Interaction Resonance

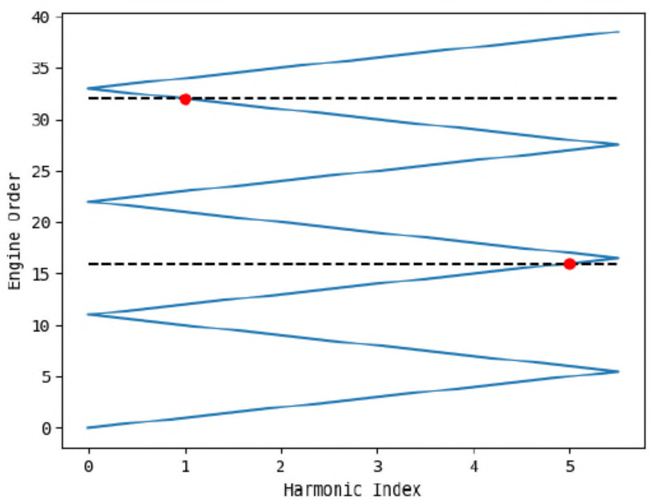
- Vane count/Blade number excites specific nodal diameter patterns (phase cancellation) due to aliasing

•Kushner's Equations (1979)* for disk interaction resonance

$$|\, y \cdot S \pm z \cdot B \,| = n \qquad 1(a)$$

$$y \cdot S = h \qquad 1(b)$$

$$f_r = y \cdot S \cdot \omega \qquad 1(c)$$



y & z = integers — n = No. diameters

S = No. stationary vanes — h = No. harmonic of speed

B = No. rotating blades — f_r = resonant frequency

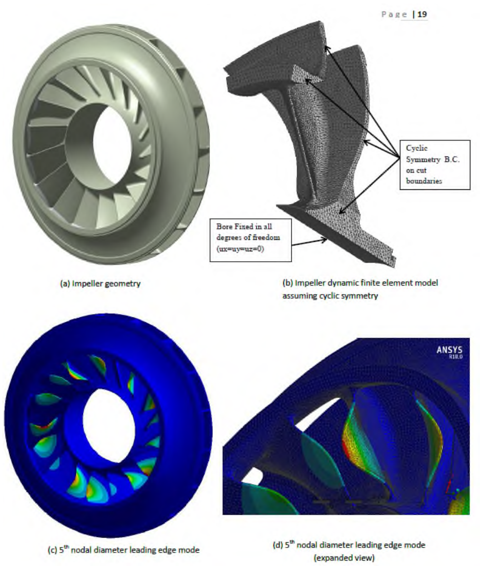
$y \cdot S \cdot \omega$ = multiples of vane passing frequency

*Kushner, F., 1979, "Disc Vibration - Rotating Blade and Stationary Vane Interaction," ASME Paper No. 79-Det-83

Operating in Resonance

- Managing Stress (below acceptable limits)
- High Cycle Fatigue (HCF) Problem
- Impeller Evaluations typically require High Fidelity FEA and CFD analysis

Operating in Resonance



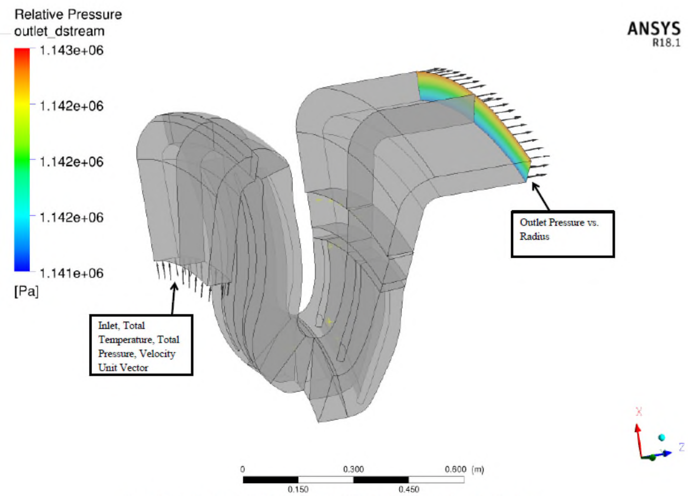
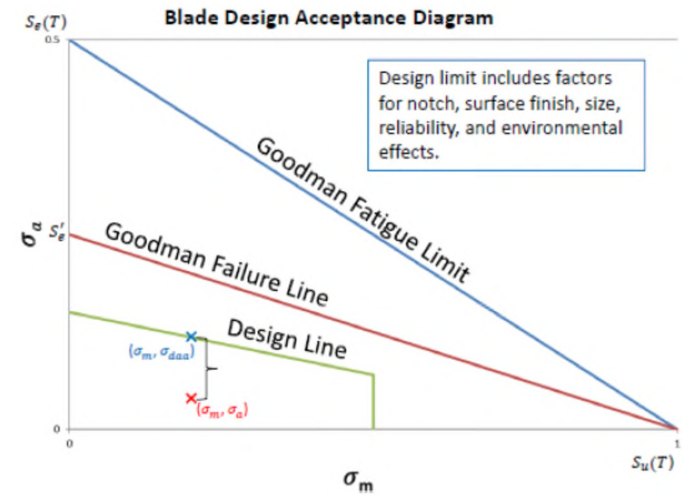


Figure 13 Impeller Stage CFD Model, Unsteady CFD – Double Passage, Boundary Conditions



FEA:
- Steady Loading: Centrifugal + Gas Bending
- Unsteady Forced Response: Loading from CFD Analysis + Assumed/or damping from CFD

CFD:
- Steady Loading
- Transient Harmonic Loading
- Estimation of Aerodynamic Damping

High Cycle Fatigue Evaluation
- Material properties
- Steady and alternating stresses at critical locations

Considerations in a high gas density fluid

- Higher excitation magnitude

- Higher damping

- Added mass effect (stronger fluid/structure coupling)- typically rare in turbomachinery applications

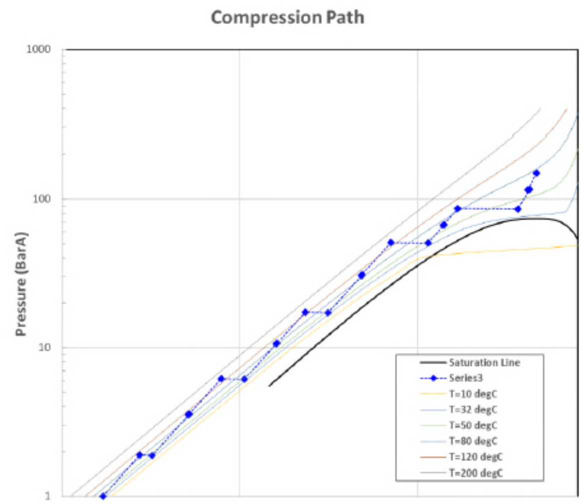
- Aeroacoustic interaction* may need to be considered

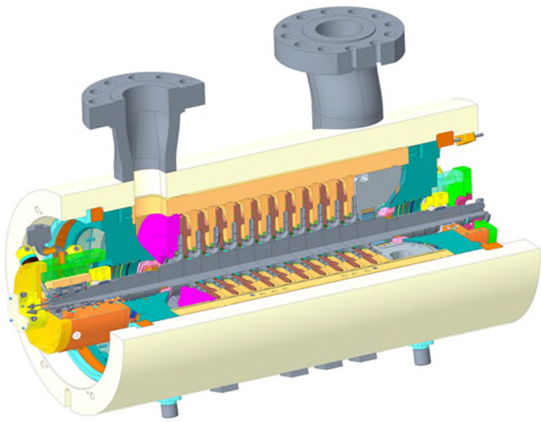
* Konig, Sven, Nico Petry, and Norbert Wagner.,2009, "Aeroacoustic Phenomena In High Pressure Centrifugal Compressors-A Possible Root Cause For Impeller Failures." Turbomachinery Symposium, Texas A&M University, https://hdl.handle.net/1969.1/163083

Test Case

Impeller Diffuser Interaction Study

- Last stage impeller of multistage inline compressor
- Overall process compression from 1 bar to 150 barA (three compressors)
- Last two stages of last compressor supercritical

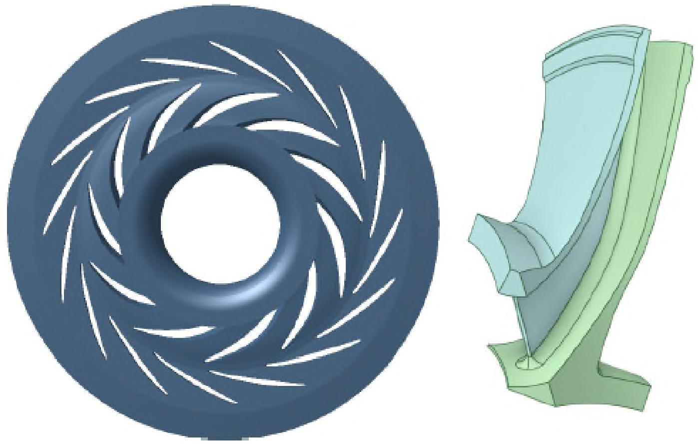




Test Case

Diffuser Vane Count = 16

Impeller Blade Count = 11

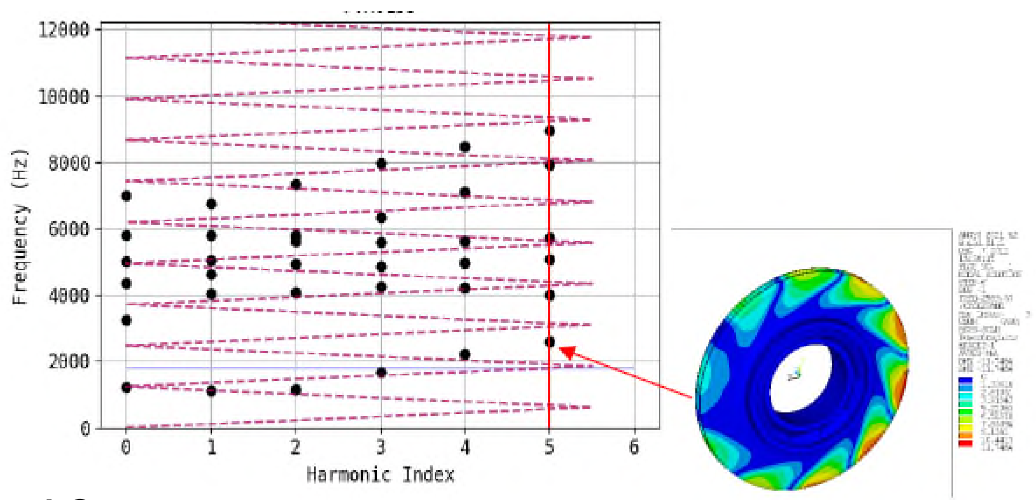


(a) Impeller/Vaned Diffuser Sectional View (b) Impeller Sector Model

Table 1: Boundary conditions for the impeller

Flow Coefficient	0.013
Inlet Mach Number	0.56
Inlet Total Pressure	112.8 bar
Inlet Total Temperature	329.55 K
Mass Flow Rate	176.63 kg/s
Rotor Speed	6747 rpm

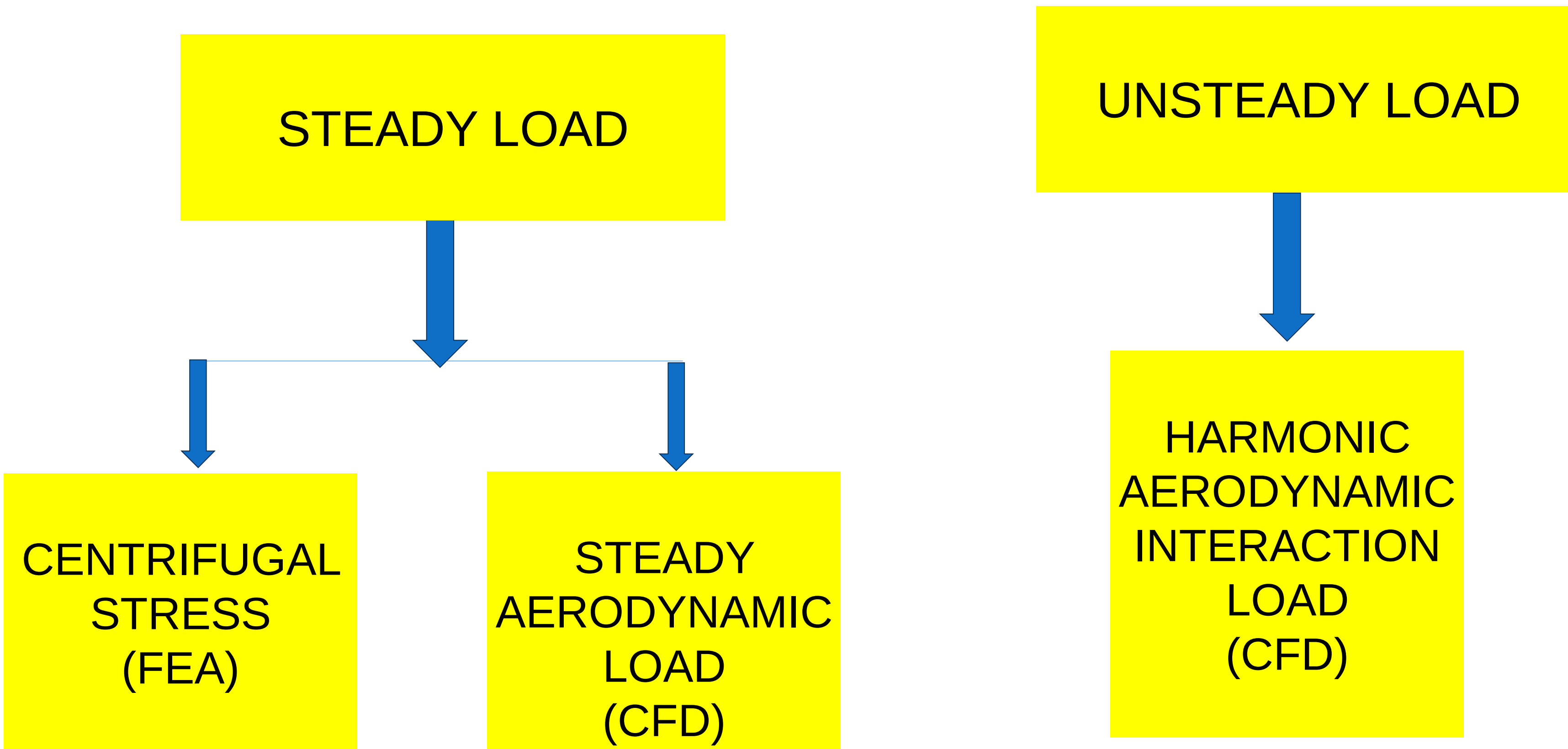
Structural Modal Analysis



Diffuser Vane Count = 16
Impeller Blade Count = 11

- 5 ND pattern has potential for excitation
- Adequate separation margin (44%) for nearest 5 ND mode
- If operating in resonance, will the stresses be acceptable?

High Cycle Fatigue Analysis



Steady and Unsteady Aerodynamic Loading determined using Numeca Fine™/Turbo 18.2.
Structural stresses determined using ANSYS Mechanical 2021

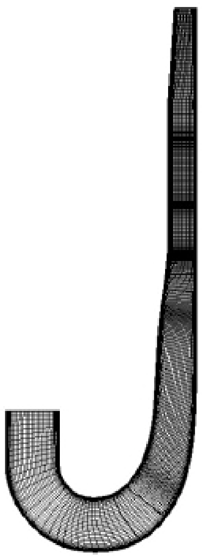
CFD Analysis

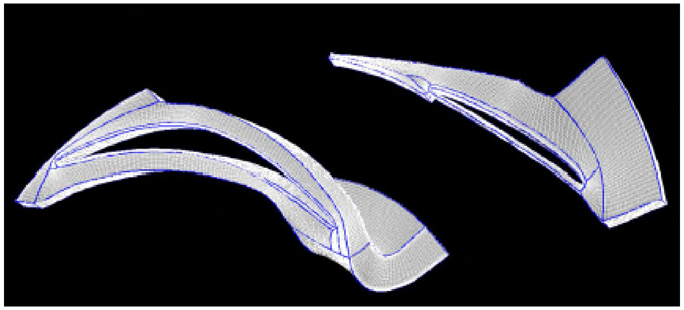
Numeca Fine^{TM}/Turbo 18.2 NonLinear Harmonic Method

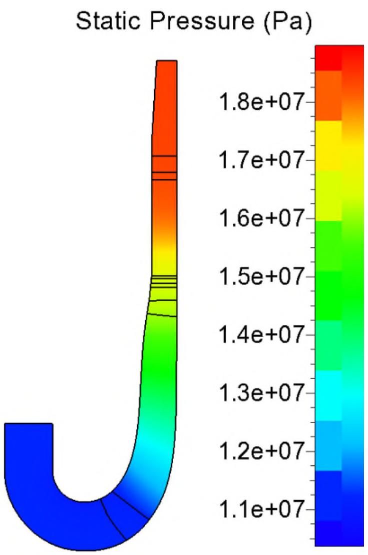
Sector Model 1.096 x 10^6 Nodes and 1.005 x 10^6 Elements

sCO_2 gas properties using TabGen (part of Numeca and based on NIST REFPROP)

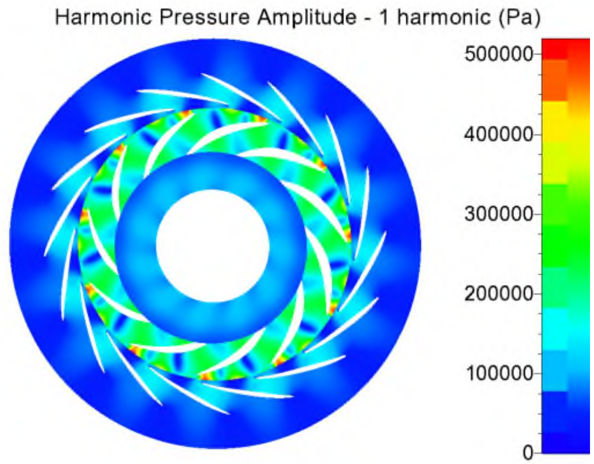
Spalart-Allmaras Turbulence Model



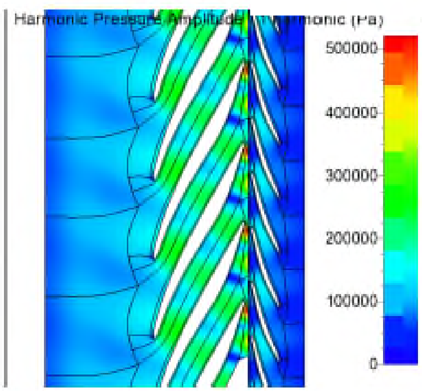




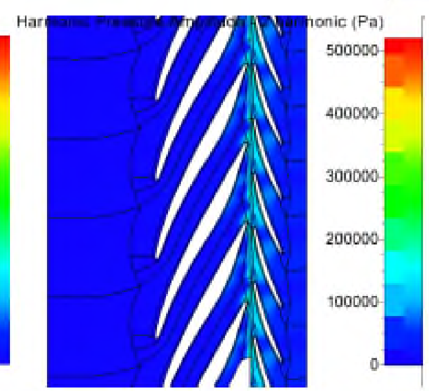
CFD Analysis



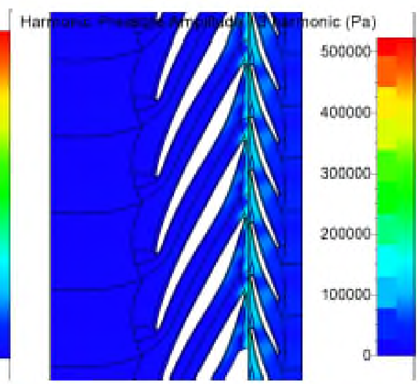
1st Harmonic (max = 5.16 bar, 4.6% of inlet pressure, P_{in})



1st Harmonic
4.6% of P_{in}



2nd Harmonic
3% of P_{in}

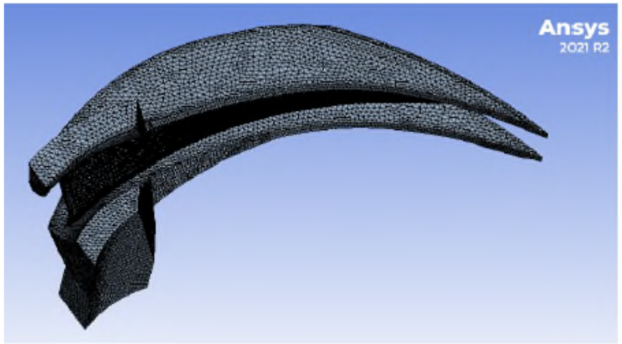


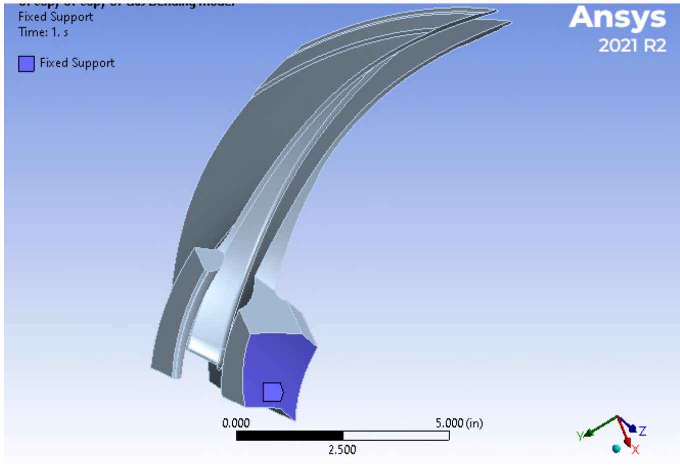
3rd Harmonic
1.6% of P_{in}

Structural Analysis

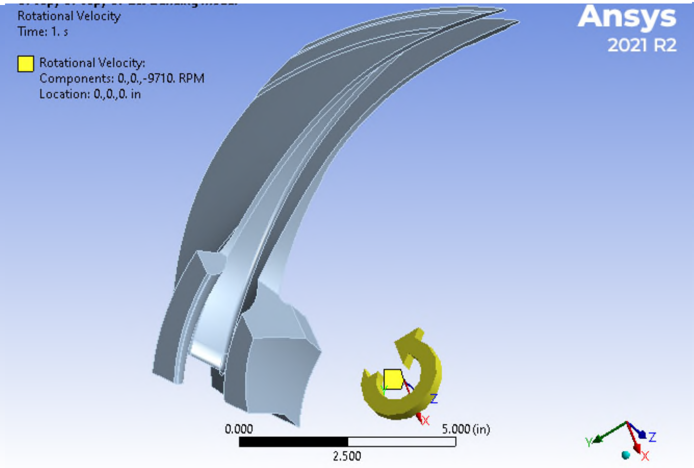
ANSYS Mechanical 2021 R2

Sector Model of Impeller

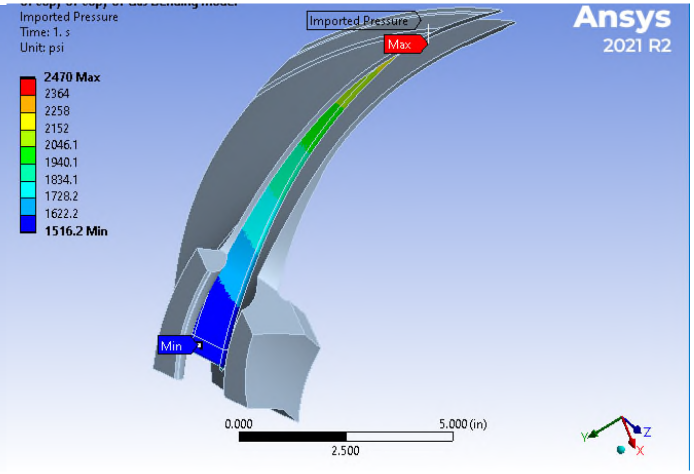




Fixed Support at Impeller Bore



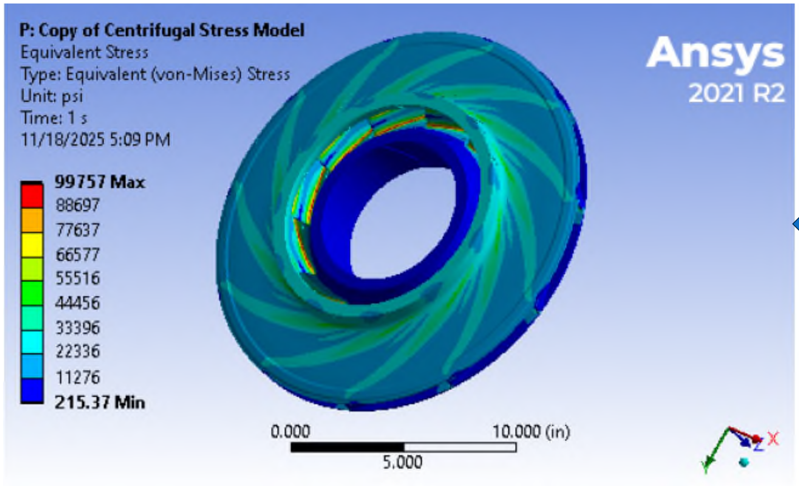
Centrifugal Load



Blade Pressure Load

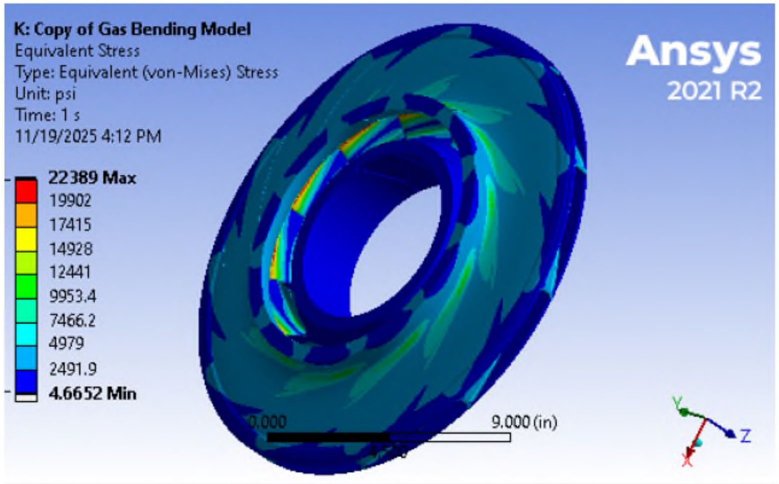
Structural Analysis

Steady Loading

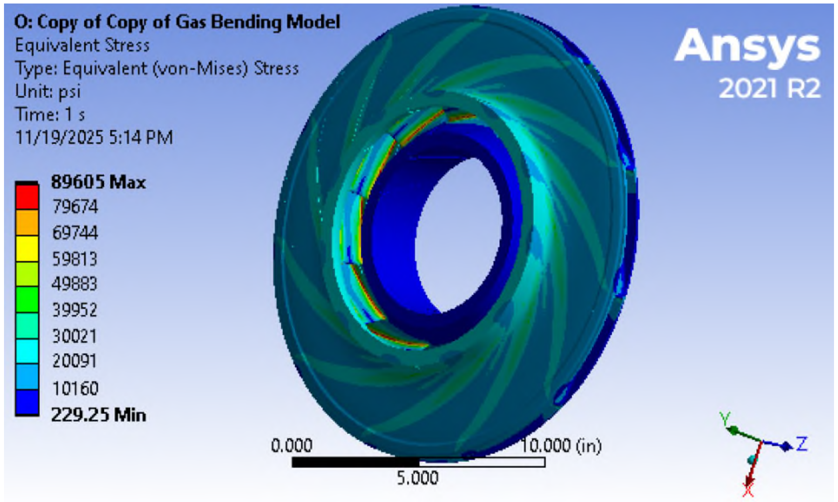


Maximum Centrifugal Stress = 99.7 ksi

Maximum Pressure Load Stress = 22.4 ksi



Structural Analysis
Steady Loading



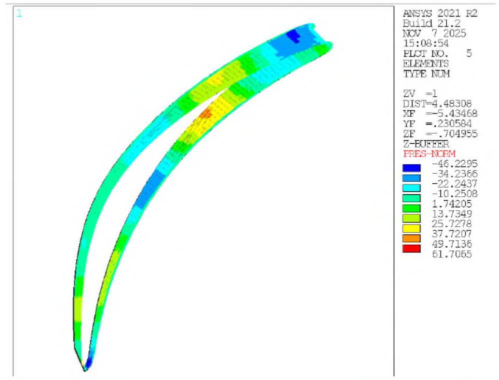
Combined (centrifugal + gas load) stress = 89.6 ksi

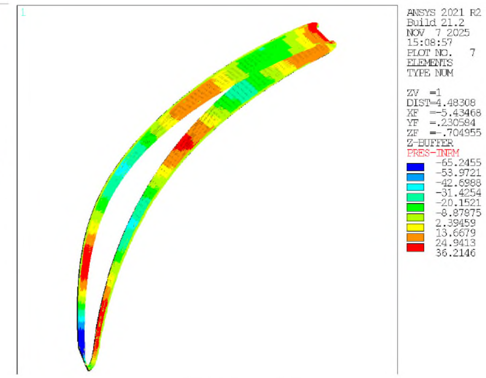
Gas Loading Stress is not additive to Centrifugal Stress!
Max. of combined load is less than Centrifugal Stress only.

Structural Analysis

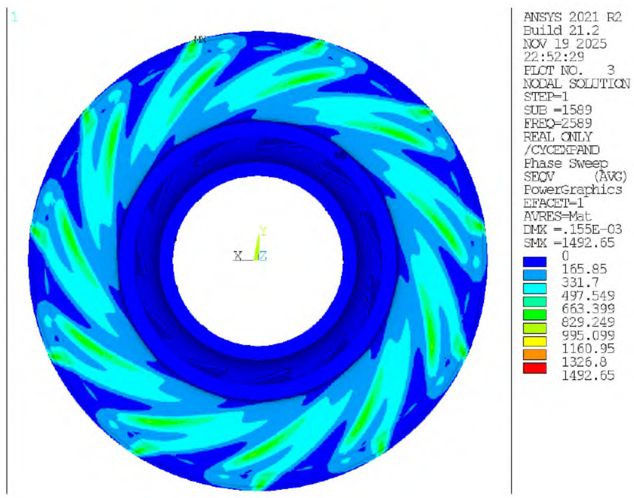
Unsteady Loading

Unsteady Harmonic Load transferred from CFD to FEA





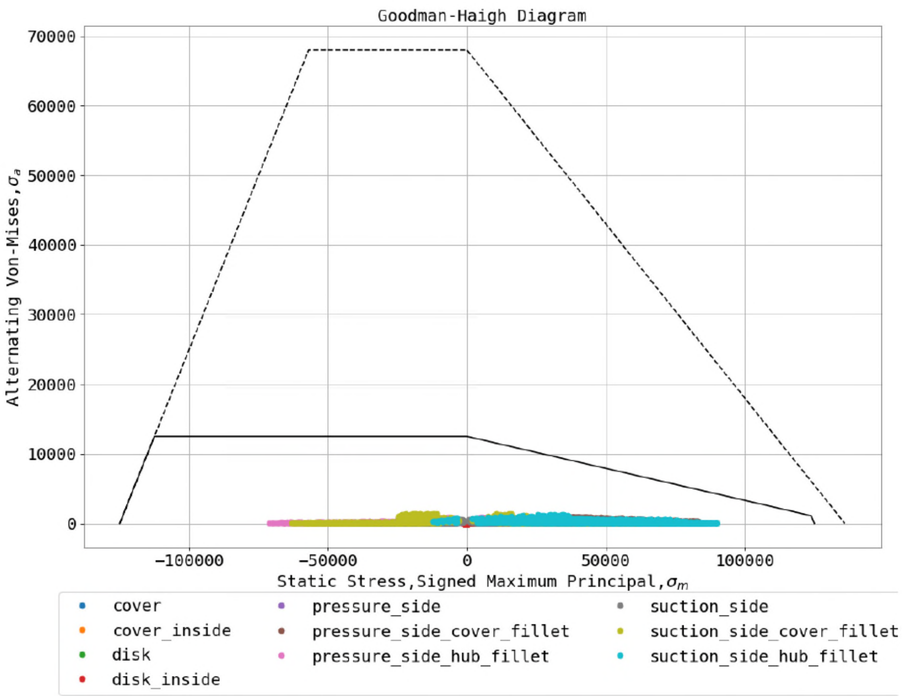
FEA Harmonic Response Analysis (assumed 0.3% modal damping)



High Cycle Fatigue Analysis

Stresses plotted on Goodman Haigh Diagram for HCF

High Strength Steel:
σ_u=135 ksi
σ_y=125 ksi



Summary

- Aeromechanic failures in centrifugal compressors typically involve forced excitation mechanisms mainly vane-blade interactions
- Impeller-diffuser interaction in sCO_2 application presented
- Impeller designed for resonance avoidance
- Analysis presented for intentional resonance shows acceptable stresses based on High Cycle Fatigue Analysis