Aeromechanics Simulation of Turbo Machinery

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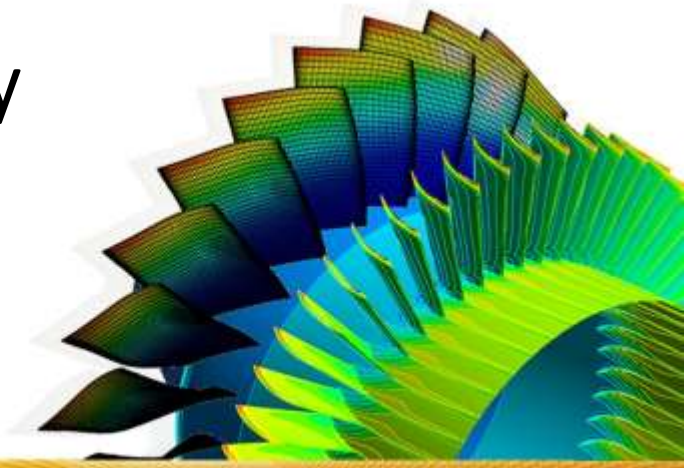
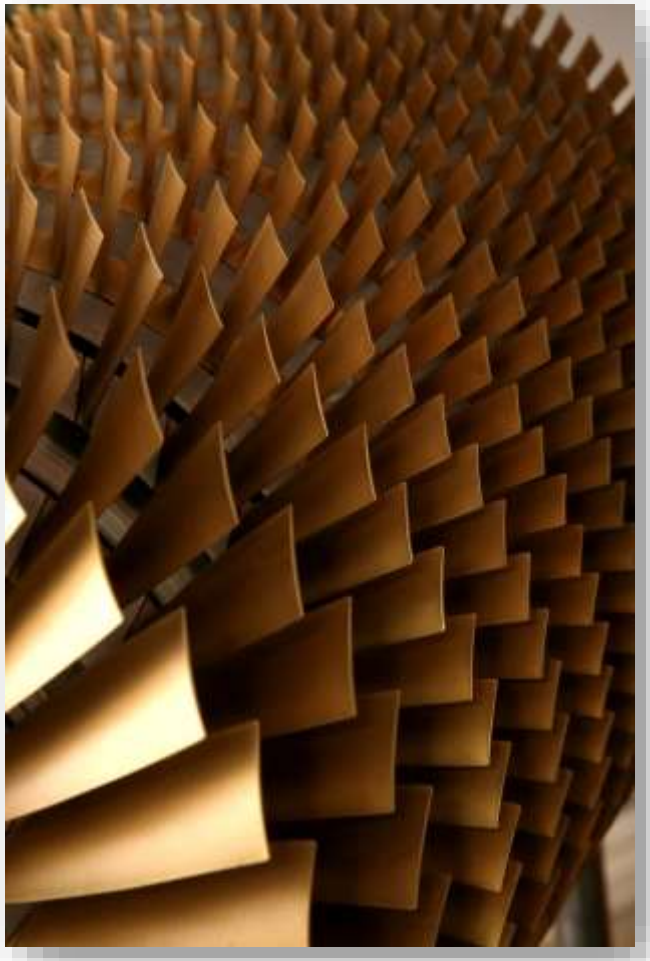
ANSYS Germany



Agenda

- Overview Aeromechanics
- Natural Vibrations
- Blade Flutter
- Forced Response
- Statistical Mistuning
- Robustness Evaluation
- Optimization



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Aeromechanics

- Fatigue and fracture of rotating components is a challenge for industry
  - High temperature environment
  - Highly stressed
  - Low damping
  - Many vibration cycles
- Industry trends point towards light weighting
  - More integrally bladed rotors (or blisks)
    - Wear or damage can demand complete replacement
    - Maintenance/downtime costs can be very high
  - Slender blades can be more susceptible to failure
- Advanced aeromechanical simulation is becoming standard practice
  - Demands a standard commercial tool that is easy to use for design engineers

Severe damage to bladed disk (*Zielinski, 2005*)

- **Pressure Oscillations → Blade Excitation**
- **Vibrations → Blade Damage**

Fluid-Structure Interaction Simulation required

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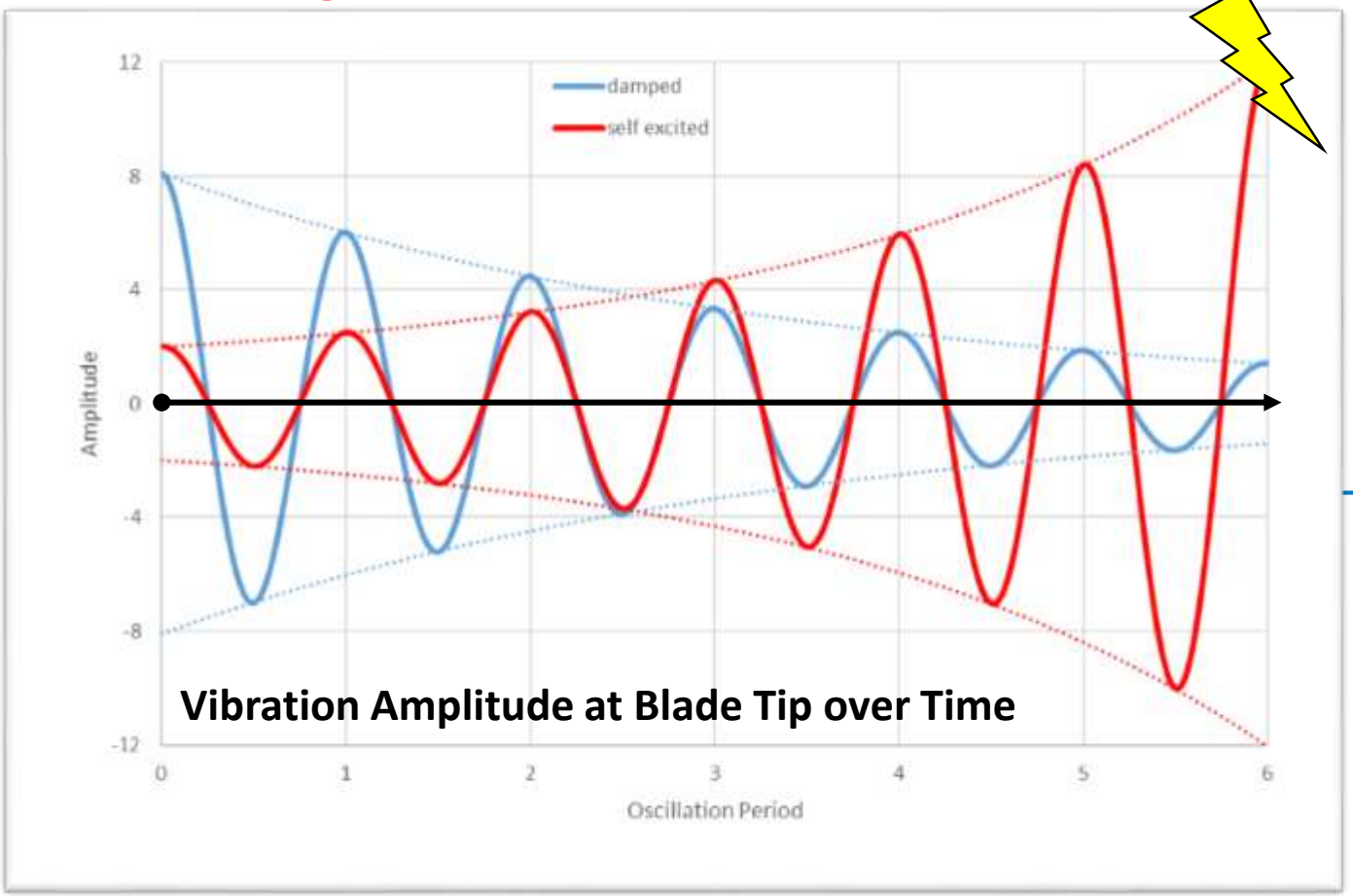
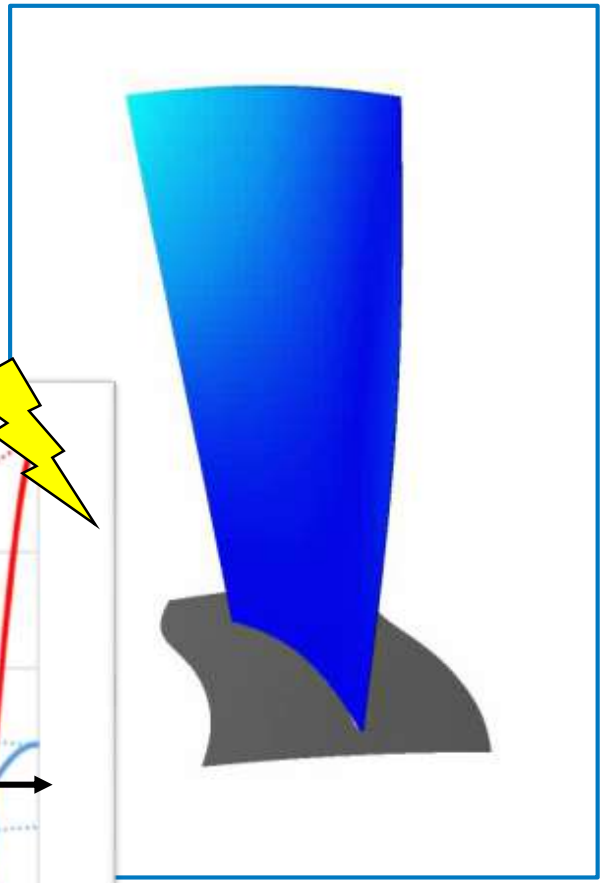
Structural Vibrations

Self excited: Vibration Amplitude is increasing over time until failure





Damped: Vibration Amplitude is reduced or limited

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Aeromechanics - Modelling

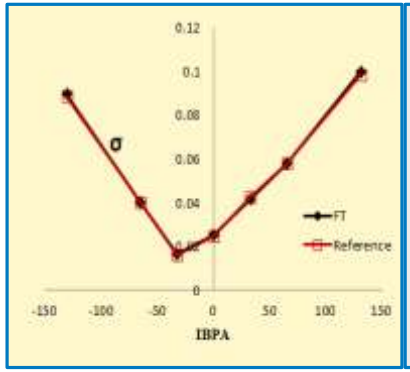
- Machine operability & durability
- Flutter margin , High Cycle Fatigue (HCF)
- Fundamentally transient Multiphysics problem → 2 Way Coupling is expensive

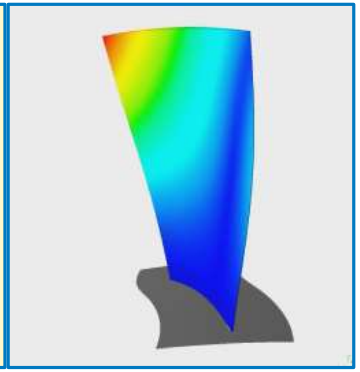
Practical & Efficient One-Way-Coupling Solution:

- **Blade Flutter:**

  Mechanical → Fluid

  Determine if the aerodynamic loads damp out blade vibration → self excited vibration?

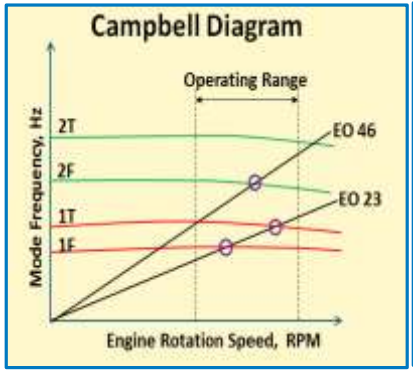


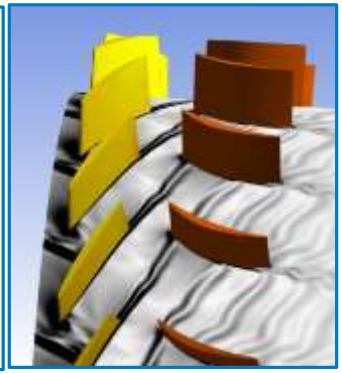


- **Forced Response:**

  Fluid → Mechanical

  Determine blade response (motion & stresses) due to excitations → Fatigue Analysis!





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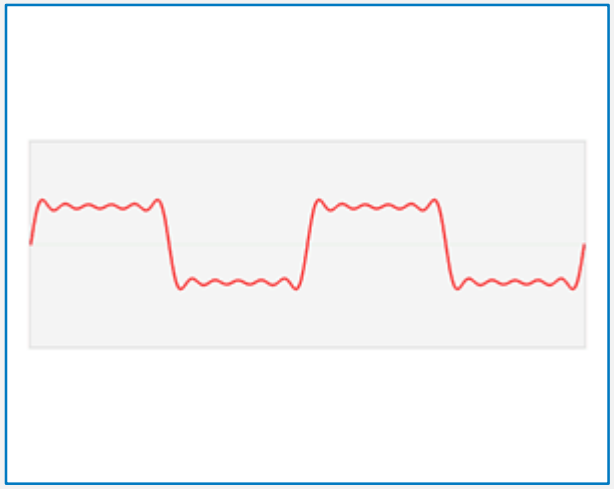
Fourier Decomposition

- The solution to transient and periodic flow can be obtained fast using Harmonic Balance method (frequency-based method), instead of using traditional time marching methods
- It assumes the solution can be represented by sin/cos based signals (Fourier-series)

$$\phi(t) = a_0 + \sum_{m=1}^{M} a_m \cos(m\omega t) + b_m \sin(m\omega t)$$

- Simple signal can be represented with few modes M (harmonics), while complex signal requires more modes to describe it.
- Originally used in microwave circuits, electromagnetic system design (i.e. ANSYS HFSS). Because transient circuit simulation is impractical.

Nice example from Wikipedia



"The Fourier transform relates the function's time domain, shown in red, to the function's frequency domain, shown in blue."

"Time-domain graph shows how a signal changes over time, whereas a frequency-domain graph shows how much of the signal lies within each given frequency band over a range of frequencies"

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Modal Analysis

Modal analysis is the study of the dynamic properties of structures under vibrational excitation

Equation of Motion (no damping, no excitation):

$$\mathbf{M}\ddot{\boldsymbol{x}} + \boldsymbol{K}\boldsymbol{x} = \mathbf{0}$$

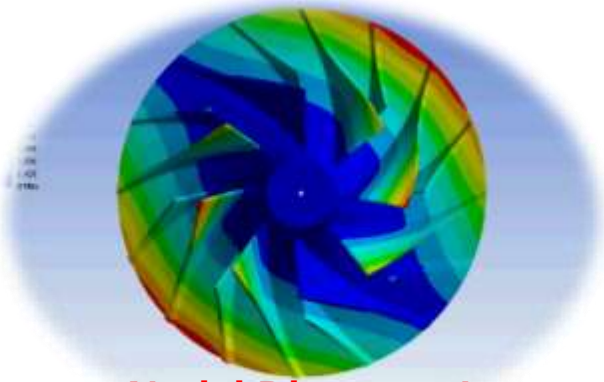
Periodic oscillation Response:

$$\boldsymbol{x}(\boldsymbol{t}) = \boldsymbol{\Phi} \cdot \boldsymbol{e}^{\boldsymbol{j}\cdot\boldsymbol{\omega}\cdot\boldsymbol{t}}$$

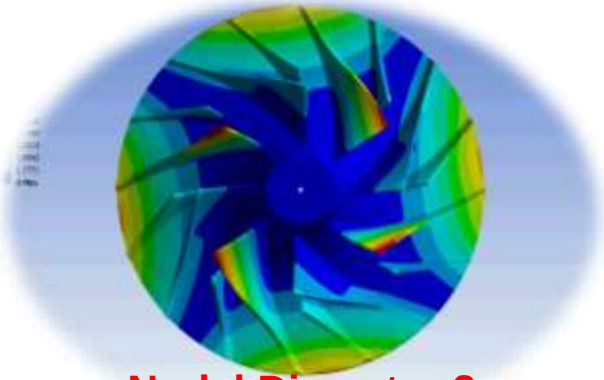
Modal Equation for Eigen-frequencies ω and – Shapes Φ

$$\left(\boldsymbol{K} - \boldsymbol{\omega}^2\boldsymbol{M}\right) \cdot \boldsymbol{\Phi} = \mathbf{0}$$

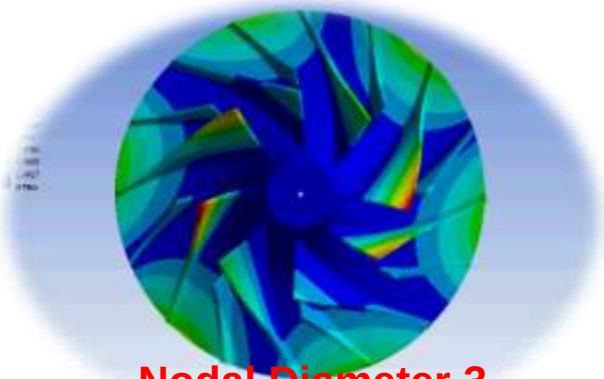
Cyclic Systems show phase lag between Blade Vibrations: IBPA=2π ND / #Blades



Nodal Diameter 1



Nodal Diameter 2



Nodal Diameter 3

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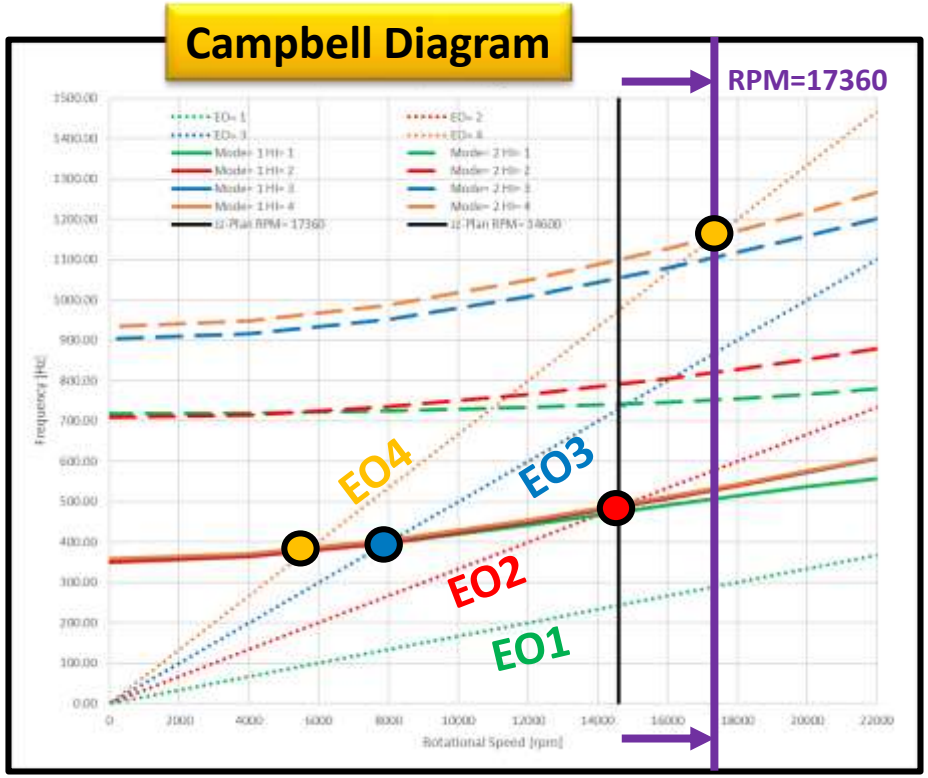
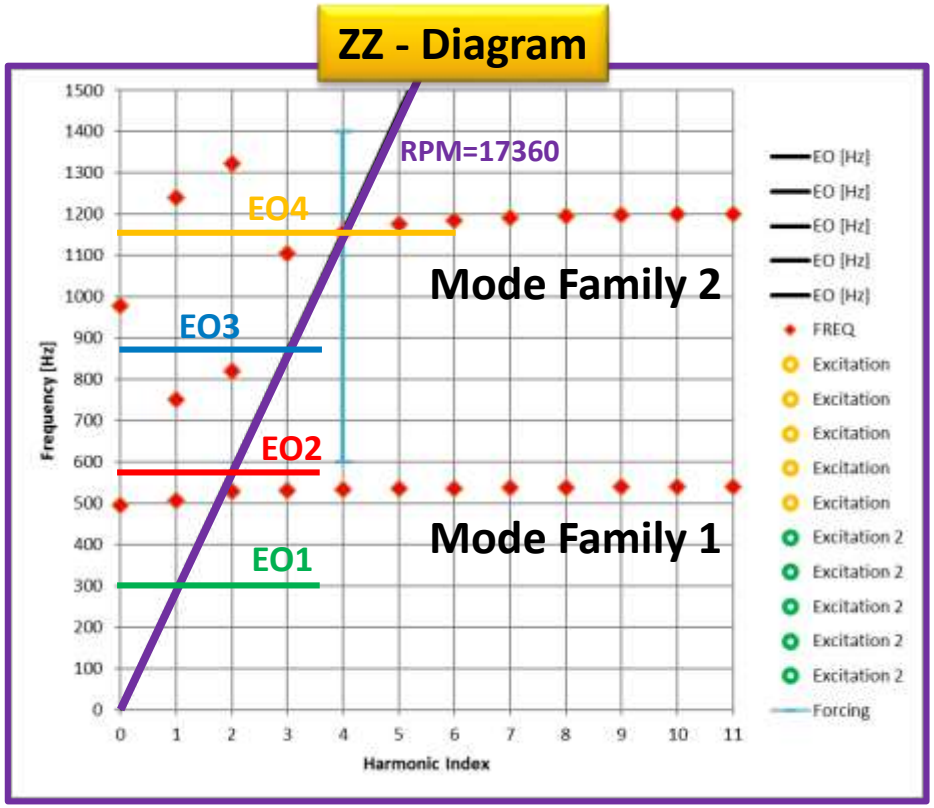
Campbell & ZZ-Diagram





Rotational Velocity vs Frequency Plane

- **EO2 crossing with Mode Family 1**
- **EO3 crossing with Mode Family 1**
- **EO4 crossing with Mode Family 1**
- **EO4 crossing with Mode Family 2**

Nodal Diameter vs Frequency Plane

EO4 is in Resonance with Mode 2, Harmonic Index 4
Mode Family 2 has large Frequency split, i.e. sensitive to Mistuning!

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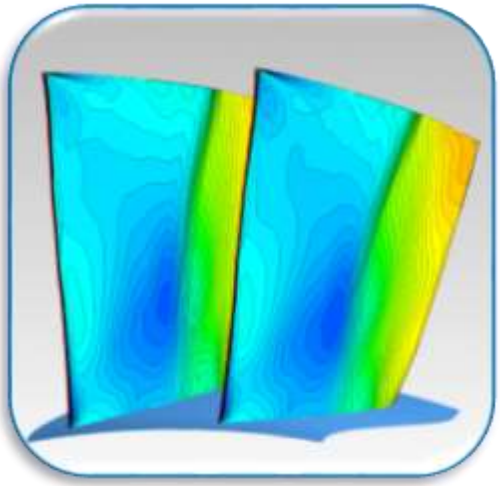
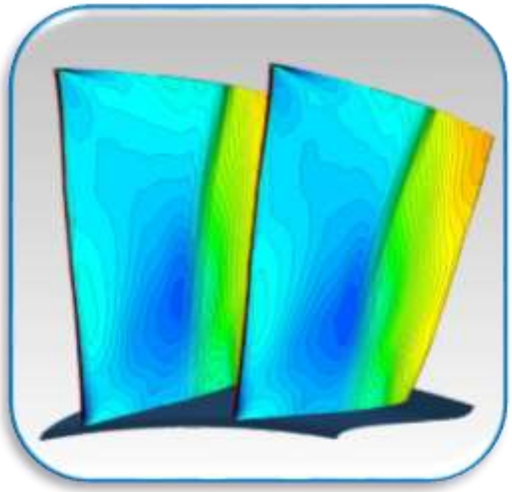
Faster Transient Blade Row Flow Solution

Transient Full Domain



Transient with Pitch-change

Reduced geometry



Harmonic Analysis with Pitch-change

Reduced geometry
+
Faster convergence

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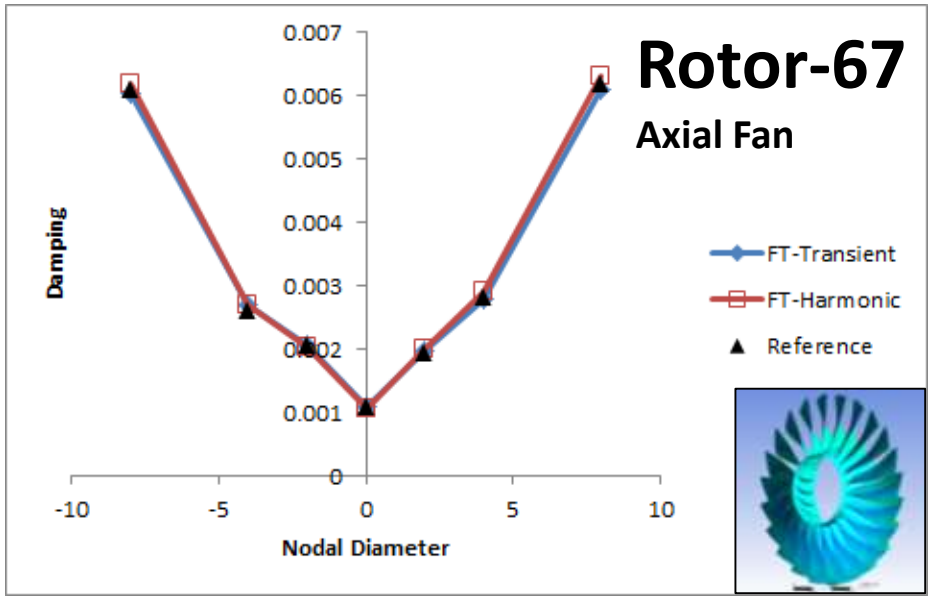
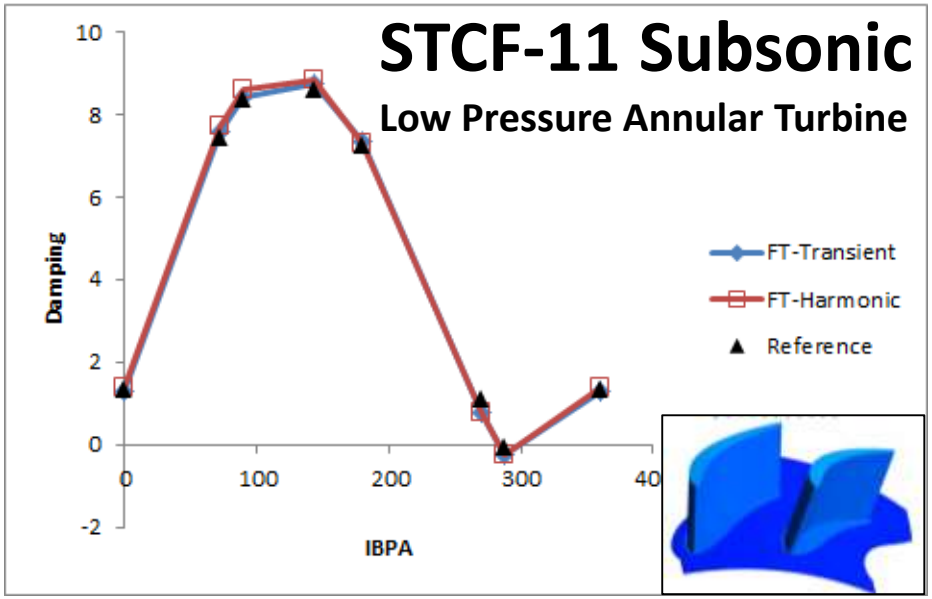
Blade Flutter & Aerodynamic Damping





- All Calculations done with: 1 mode , 15 pseudo-time-step per oscillation cycle
- starting from steady-state solution
- FT-HA (1 mode) about 20x to 30x Faster than FT-Transient (based +100 tspp)
- FT-Transient about 5x to 7x faster than Full-wheel solution

10-to-100X faster

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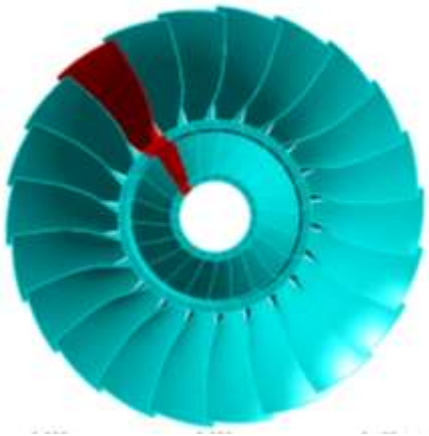
Model Order Reduction



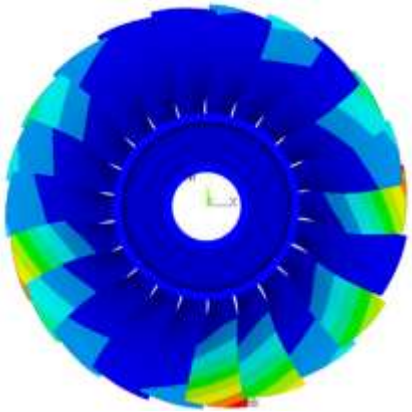
$$\mathbf{P} = \mathbf{I}e^{-j\alpha(i-1)(k-1)}$$
$$\mathbf{x} = \mathbf{P}\mathbf{q}$$

Cyclic Reduction

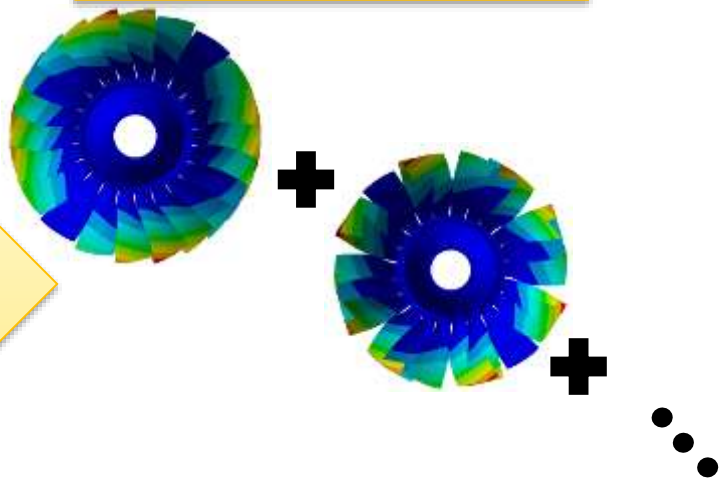
$$\mathbf{P}^{*T}\mathbf{M}\mathbf{P}\ddot{\mathbf{q}} + \mathbf{P}^{*T}\mathbf{C}\mathbf{P}\dot{\mathbf{q}} + \mathbf{P}^{*T}\mathbf{K}\mathbf{P}\mathbf{q} = \mathbf{P}^{*T}\mathbf{F}$$

$$\mathbf{M}\ddot{\mathbf{x}} + \mathbf{C}\dot{\mathbf{x}} + \mathbf{K}\mathbf{x} = \mathbf{F}$$

Modal Reduction



$$\mathbf{x} = \mathbf{\Phi}\mathbf{q}$$



$$\mathbf{\Phi}^{*T}\mathbf{M}\mathbf{\Phi}\ddot{\mathbf{q}} + \mathbf{\Phi}^{*T}\mathbf{C}\mathbf{\Phi}\dot{\mathbf{q}} + \mathbf{\Phi}^{*T}\mathbf{K}\mathbf{\Phi}\mathbf{q} = \mathbf{\Phi}^{*T}\mathbf{F}$$

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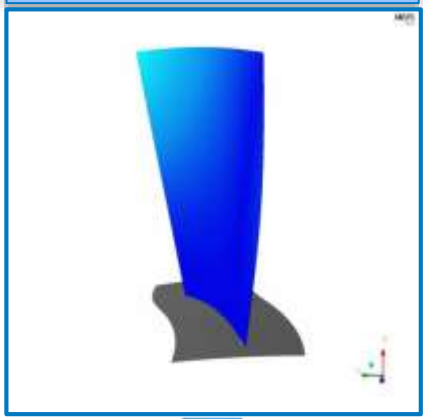
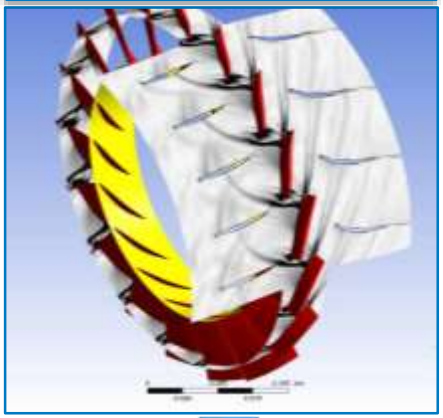
Forced Response

- **Forced Response computes Displacements and Stresses for Fatigue Analyses**
- **Frequency Domain (Forcing is oscillatory)**
  - Steady State → HCF
  - More information & faster
- **Cyclic Coordinates**
  - Less memory & faster
- **Modal Domain**
  - Faster
- **Equation of Motion:**

Aeromatrix



EO Forcing



$$\left\{-\Omega^2\mathbf{I}+i\Omega\left[\left(\beta+\frac{1}{\Omega}g\right)\mathbf{\Lambda}^s+\alpha\mathbf{I}\right]+\mathbf{\Lambda}^s+\sum_n \mathbf{q_n}^T\delta_n\mathbf{K_o}\mathbf{q_n}+\mathbf{K_a}\right\}\mathbf{a}e^{i\omega t}=\left\{\sum_n F^T_{n,EO}\tilde{\Phi}^T_h\mathbf{F}e^{i\phi(n-1)}\right\}e^{i\omega t}$$

Excitation Frequency | Reduced Mass | Reduced Damping | Reduced Stiffness | Mistuning Contribution | Aero Stiffness | Projection to modal space and expansion from cyclic coordinates | Single sector engine order forcing

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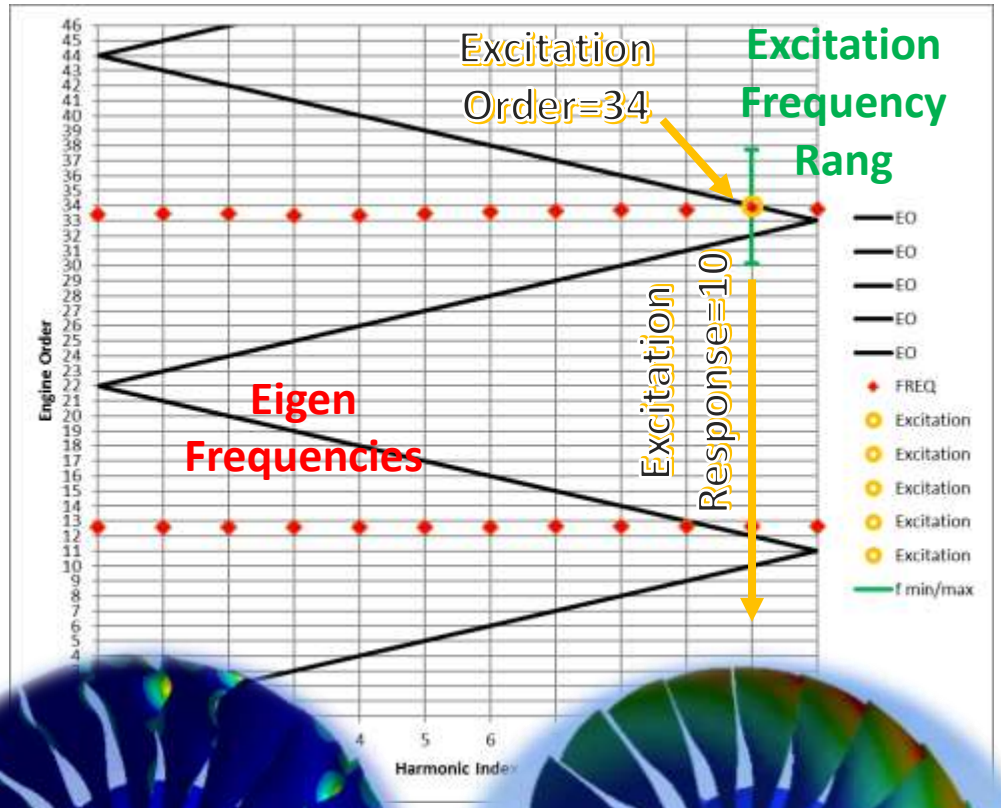
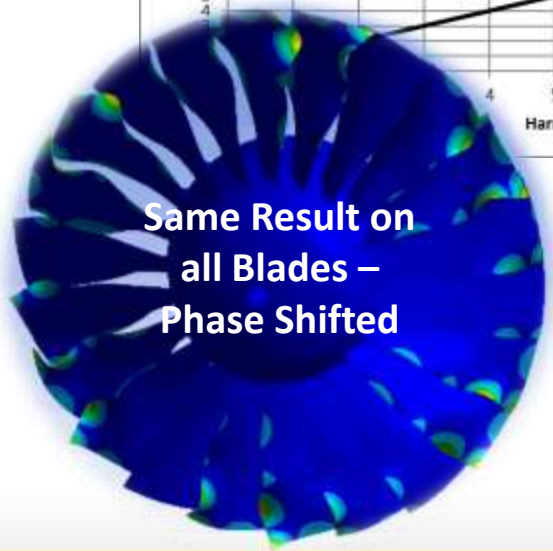
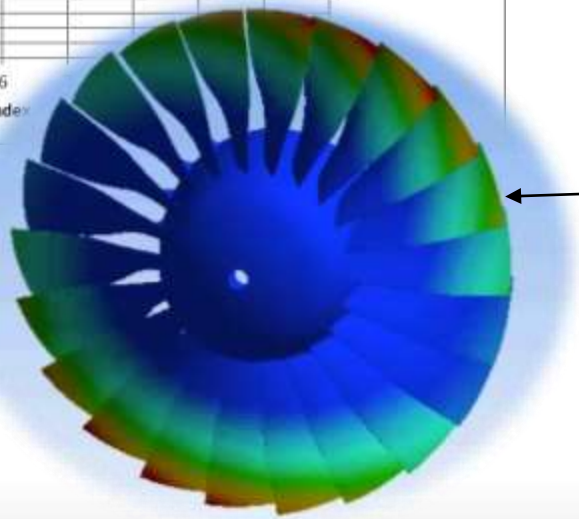
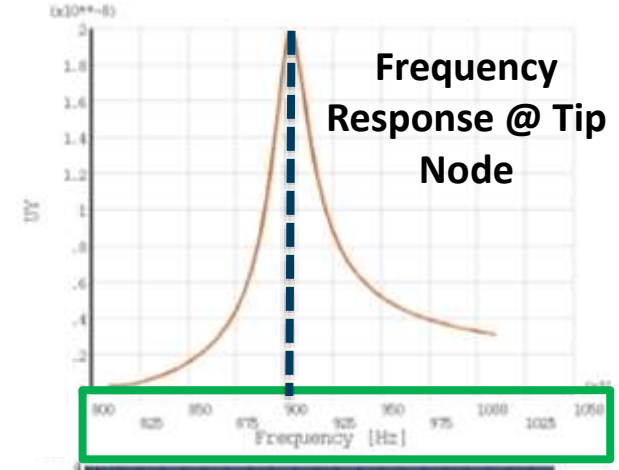
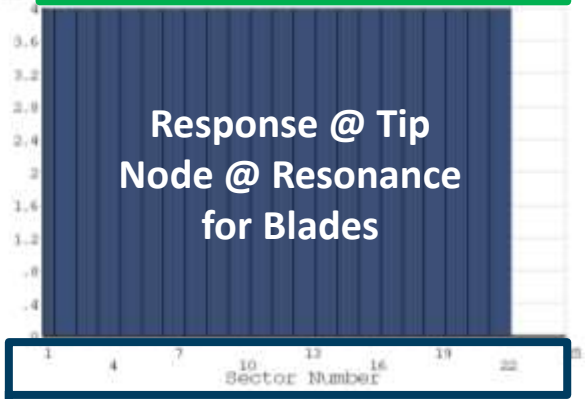
Forced Response – Tuned

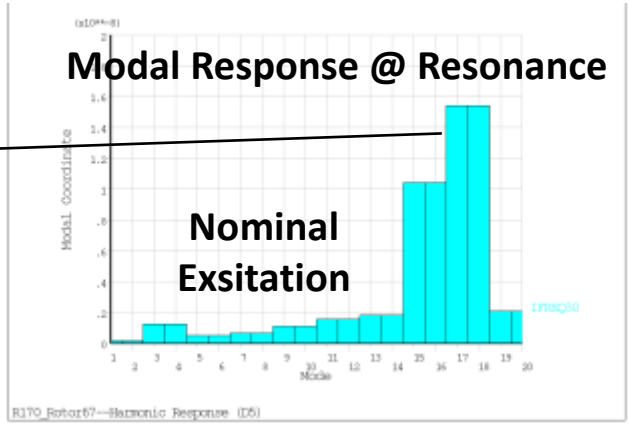












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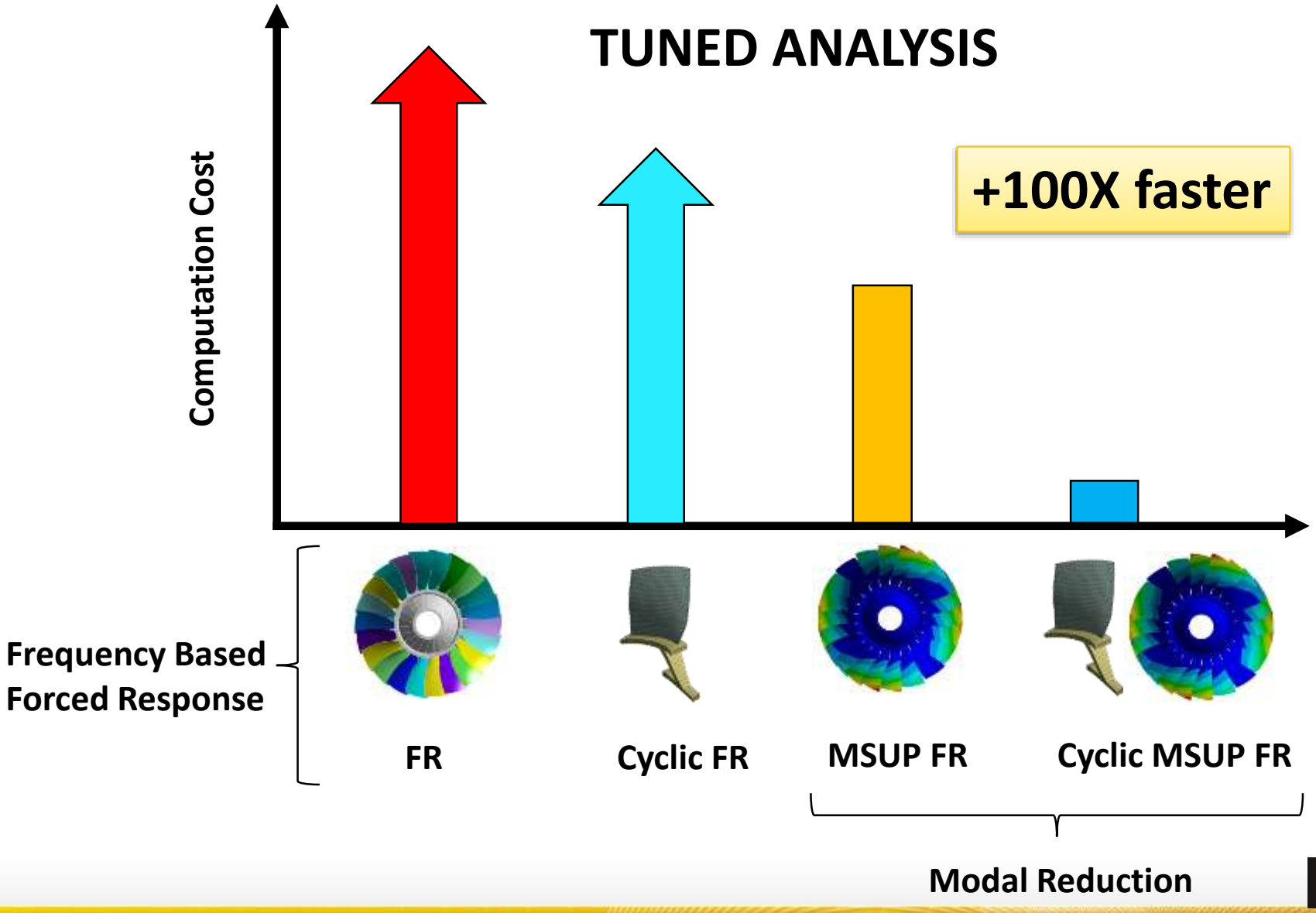
Fast and Efficient Forced Response Method



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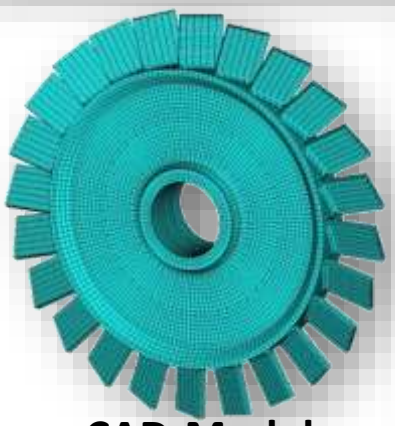
What is Mistuning?

- Why does Blade x break?
  - Local Production Error?
  - Local Material Error?
  - Local Overload?
  - Local Erosion?
  - …
- Non cyclic System due to
  - Allowed Production Tolerances
  - Small Erosion
  - …
- → Mistuned System

Rotor Damage at Blade x



CAD-Model (=Tuned System)

Real-Model (=Mistuned)

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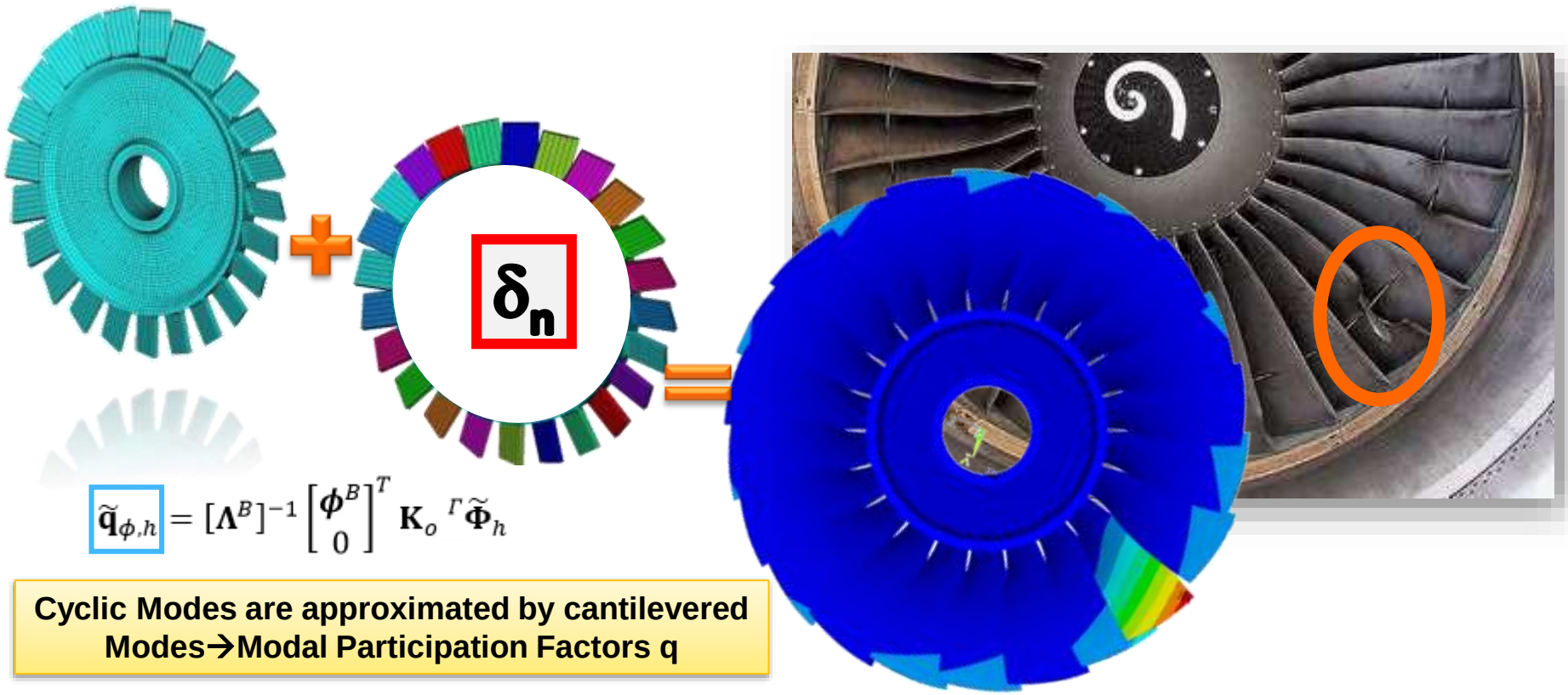
Aeromechanics – Mistuning



$$\tilde{\mathbf{q}}_{\phi,h} = [\mathbf{\Lambda}^B]^{-1} \begin{bmatrix} \boldsymbol{\phi}^B \\ 0 \end{bmatrix}^T \mathbf{K}_o \, {}^{\Gamma}\tilde{\mathbf{\Phi}}_h$$

Cyclic Modes are approximated by cantilevered Modes→Modal Participation Factors q

$$\left\{ -\Omega^2 \mathbf{I} + i\Omega \left[\left(\beta + \frac{1}{\Omega} g \right) \mathbf{\Lambda}^s + \alpha \mathbf{I} \right] + \mathbf{\Lambda}^s + \sum_n \mathbf{q_n}^T \delta_n \mathbf{K_c} \mathbf{q_n} + \mathbf{K_a} \right\} \mathbf{a} e^{i\omega t} = \left\{ \sum_n F^T_{n,EO} \tilde{\Phi}^T_h \mathbf{F} e^{i\phi(n-1)} \right\} e^{i\omega t}$$

Excitation frequency | Reduced mass | Reduced damping | Reduced stiffness | Mistuning terms | Aero stiffness | Projection to modal space and expansion from cyclic domain | Single sector engine order forcing

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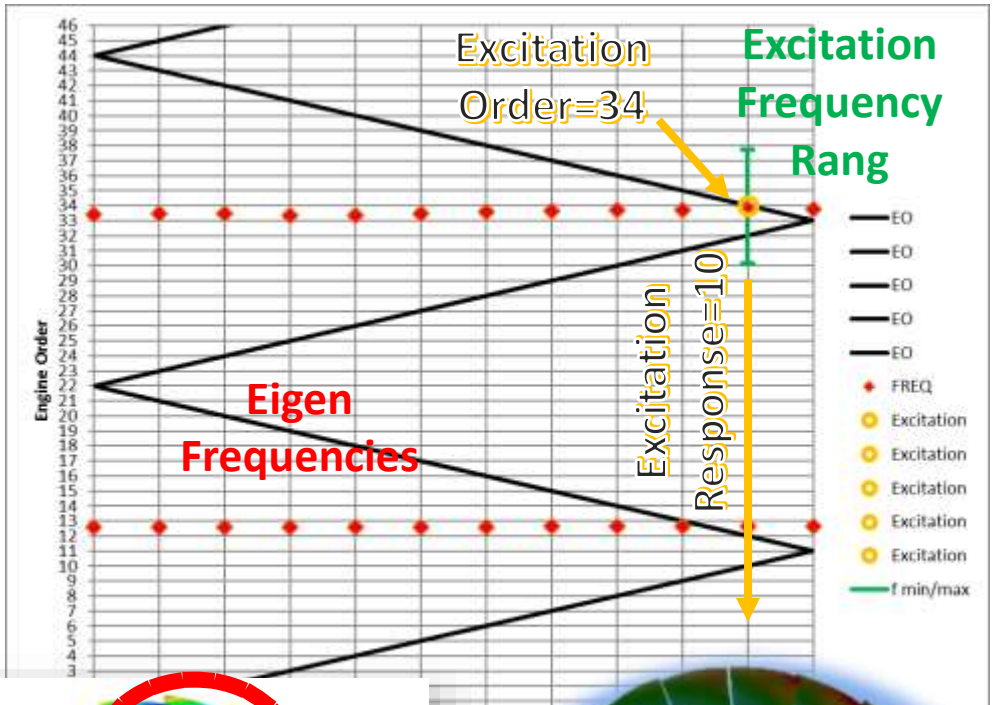
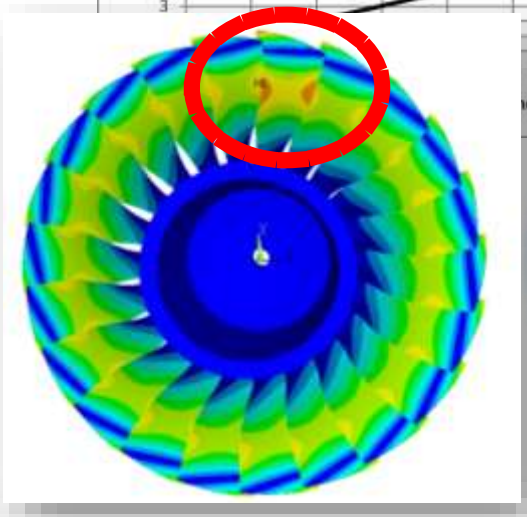
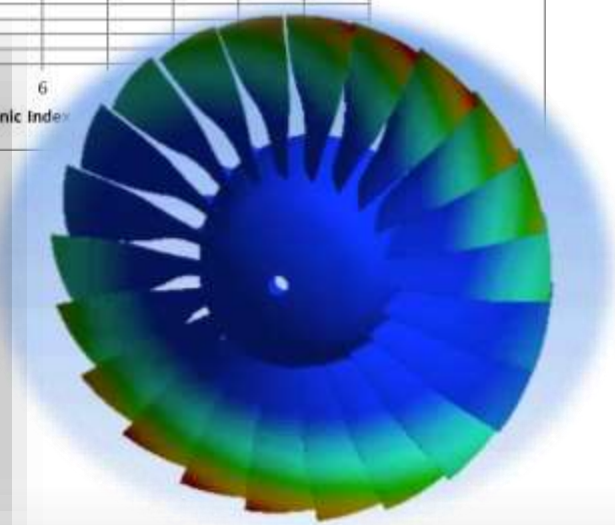
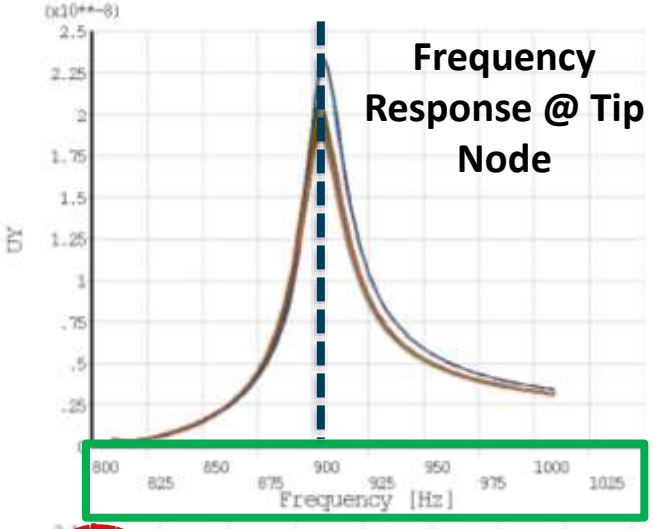
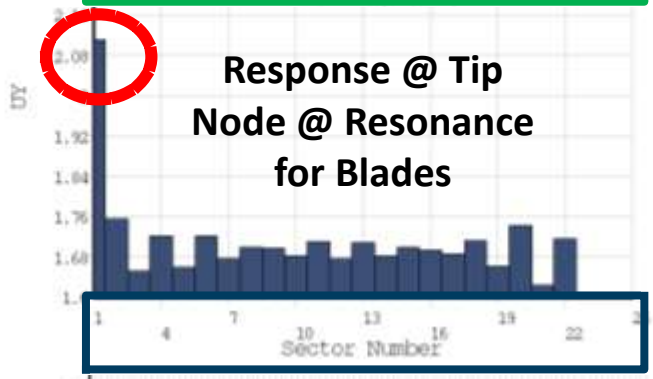
Forced Response – Mistuned

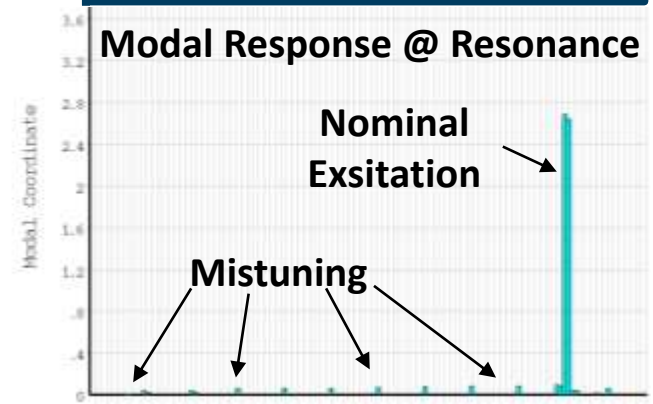












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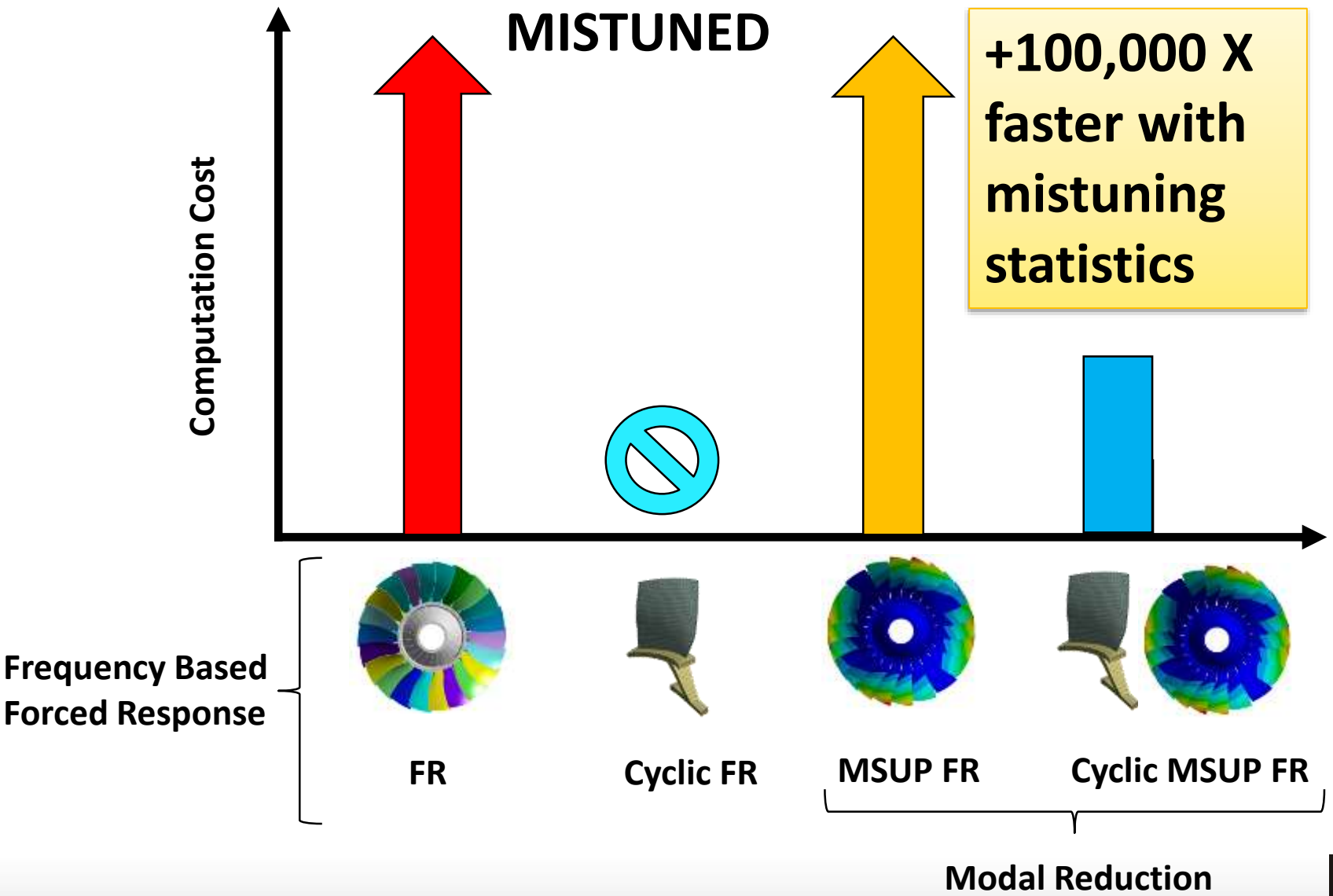
Fast and Efficient Forced Response Method



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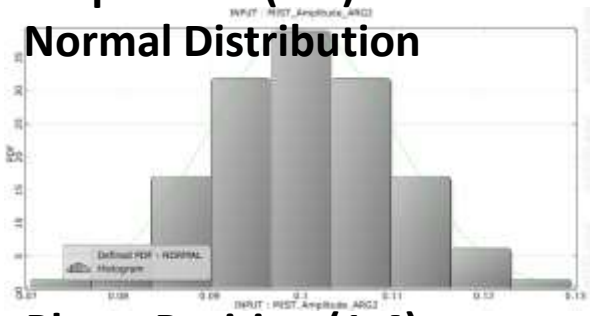
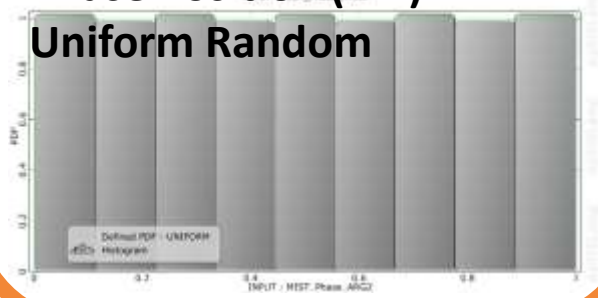
Robustness Evalution

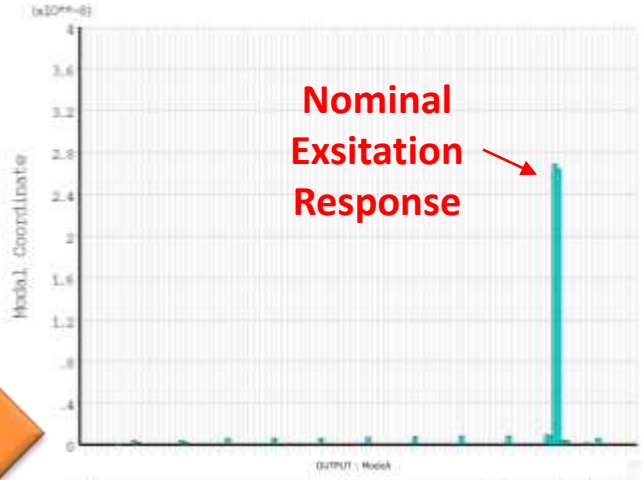
Amplitudes (1-4)
Normal Distribution



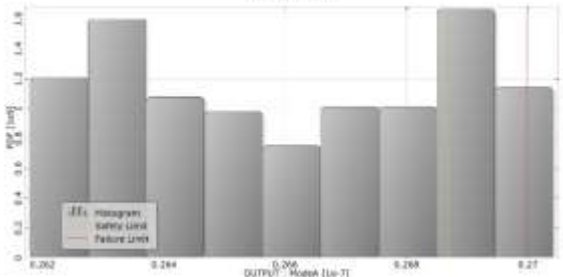
Phase Position (1-4)
Uniform Random



2x4 Parameter, 100 Design Points

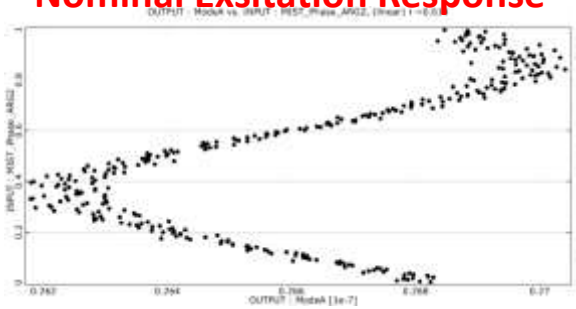


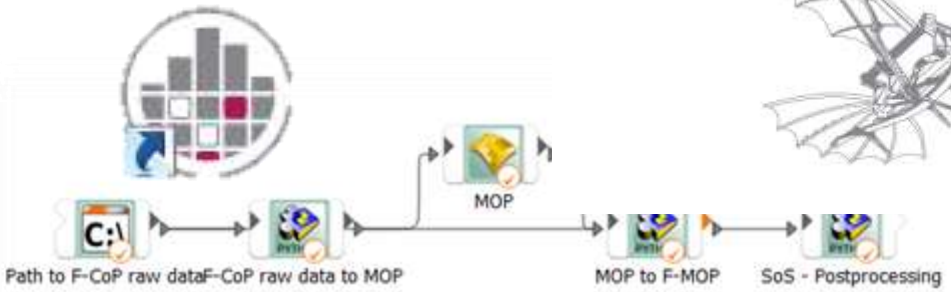
Probability



Nominal Exsitation Response

Imperfection Phase 2





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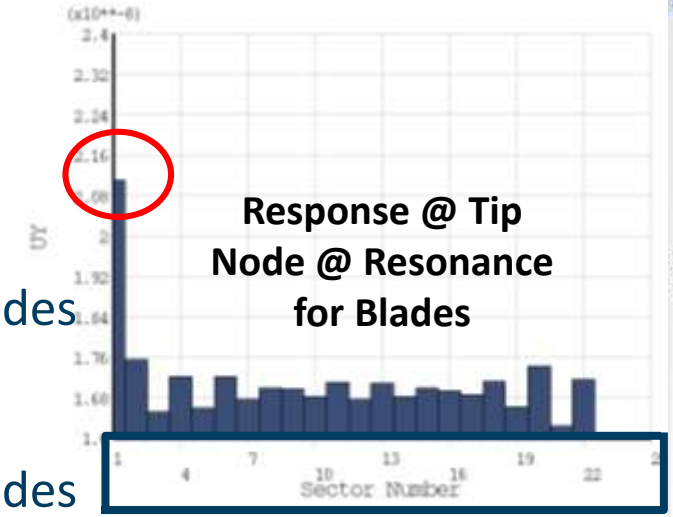
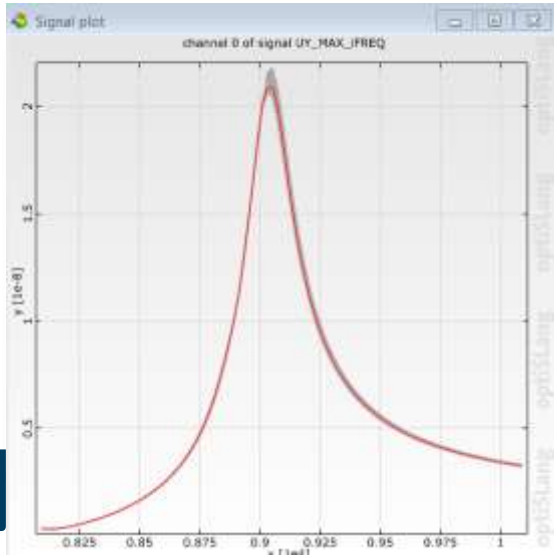
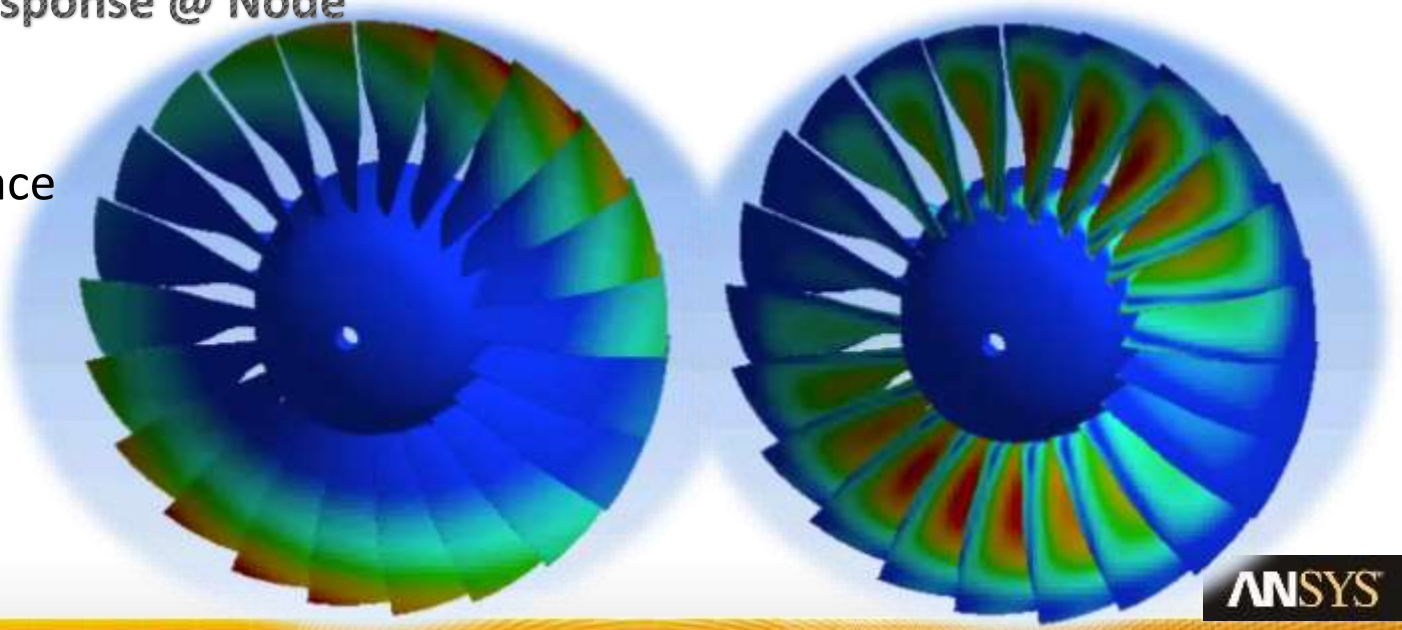
Parameterization – Output

- Scalar
  - Global Maximum
  - Local Maximum @ Blades
- Signal
  - Local Maximum @ Blades
  - Frequency Response @ Node
- Field → SoS
  - Value @ Surface







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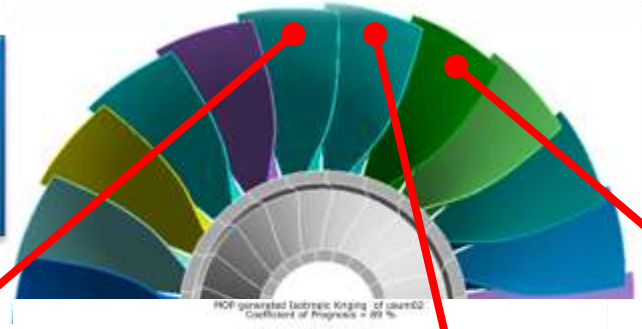
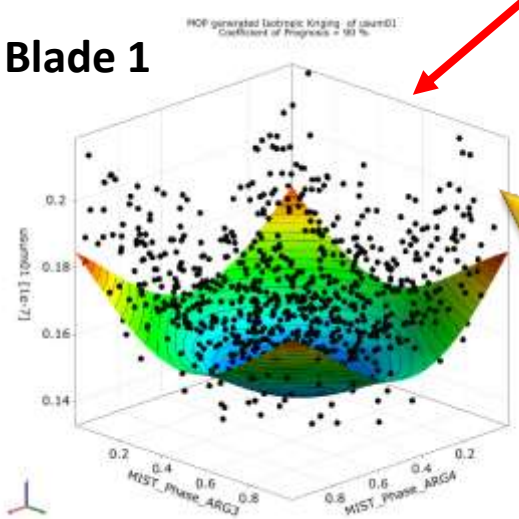
Parameter Impact & Response Surface

MoP is Phase Shifted!

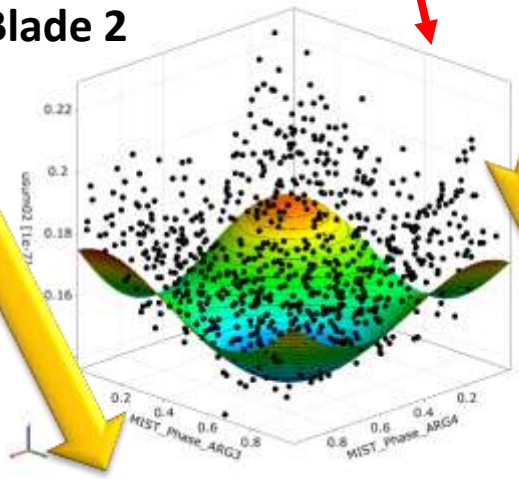
Phase is more important than Amplitude



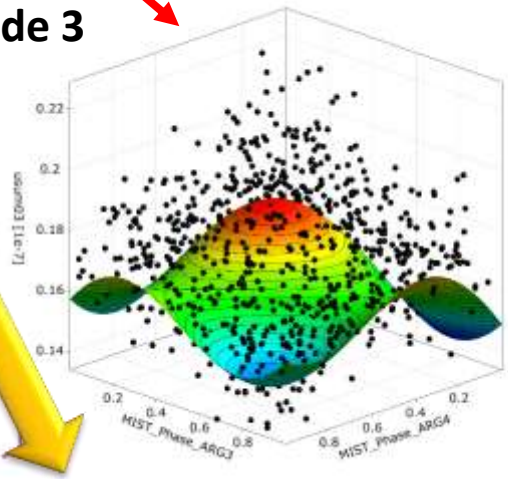
Blade 1

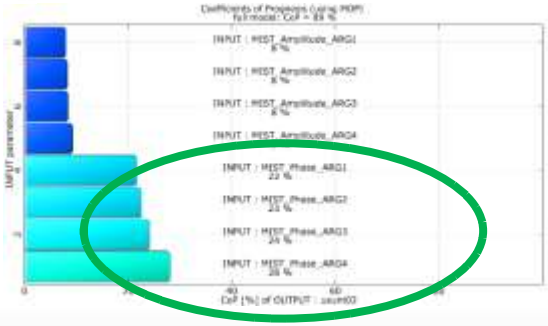


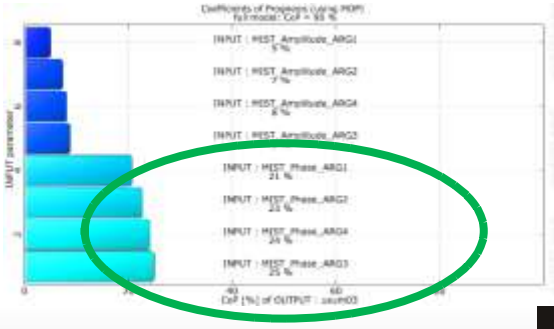
Blade 2



Blade 3







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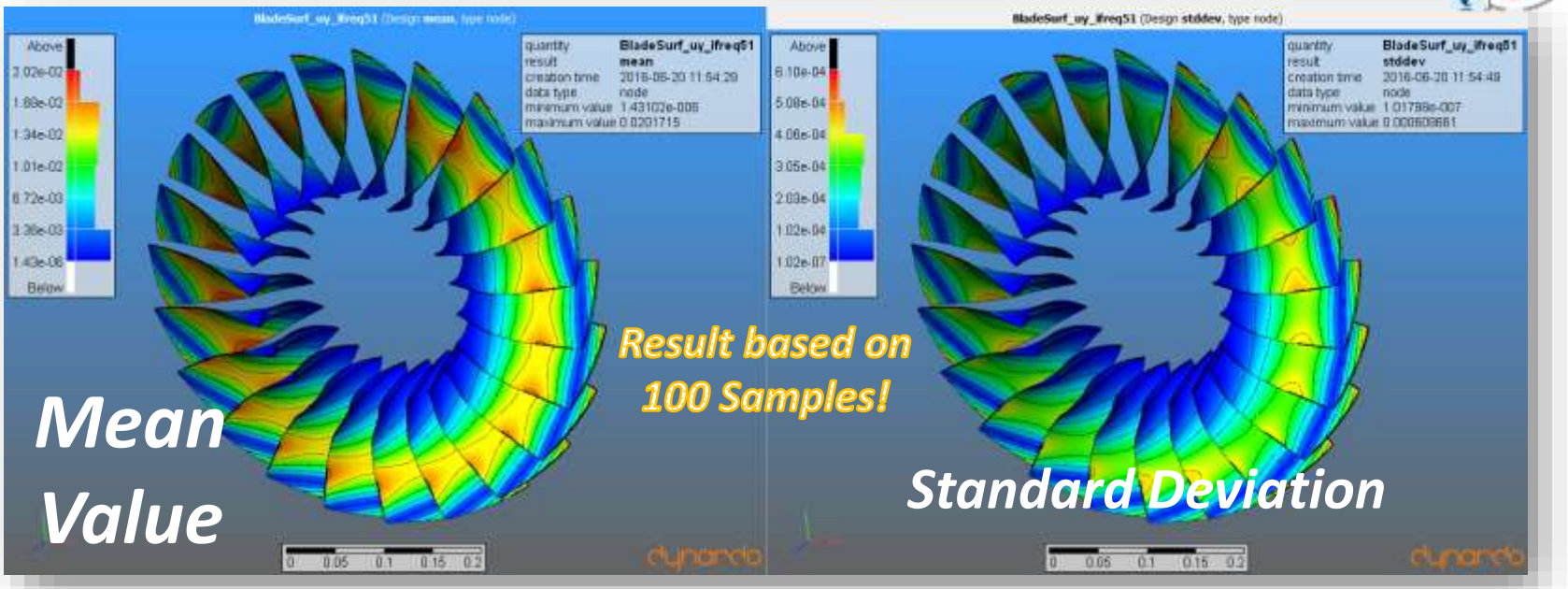
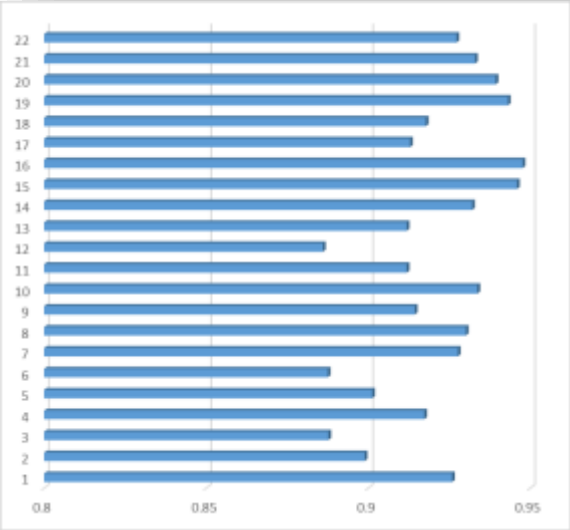
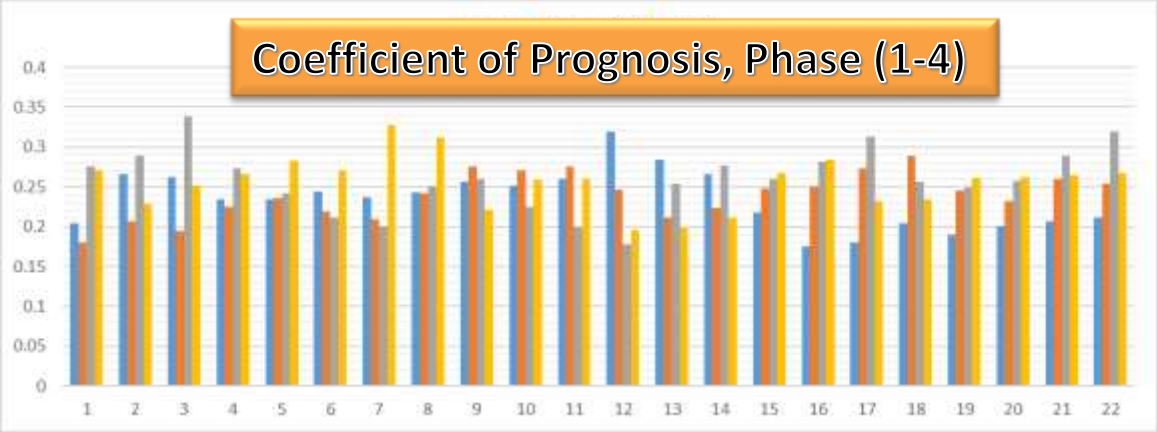
Final Result with Statistic on Structures



Coefficient of Prognosis, Total





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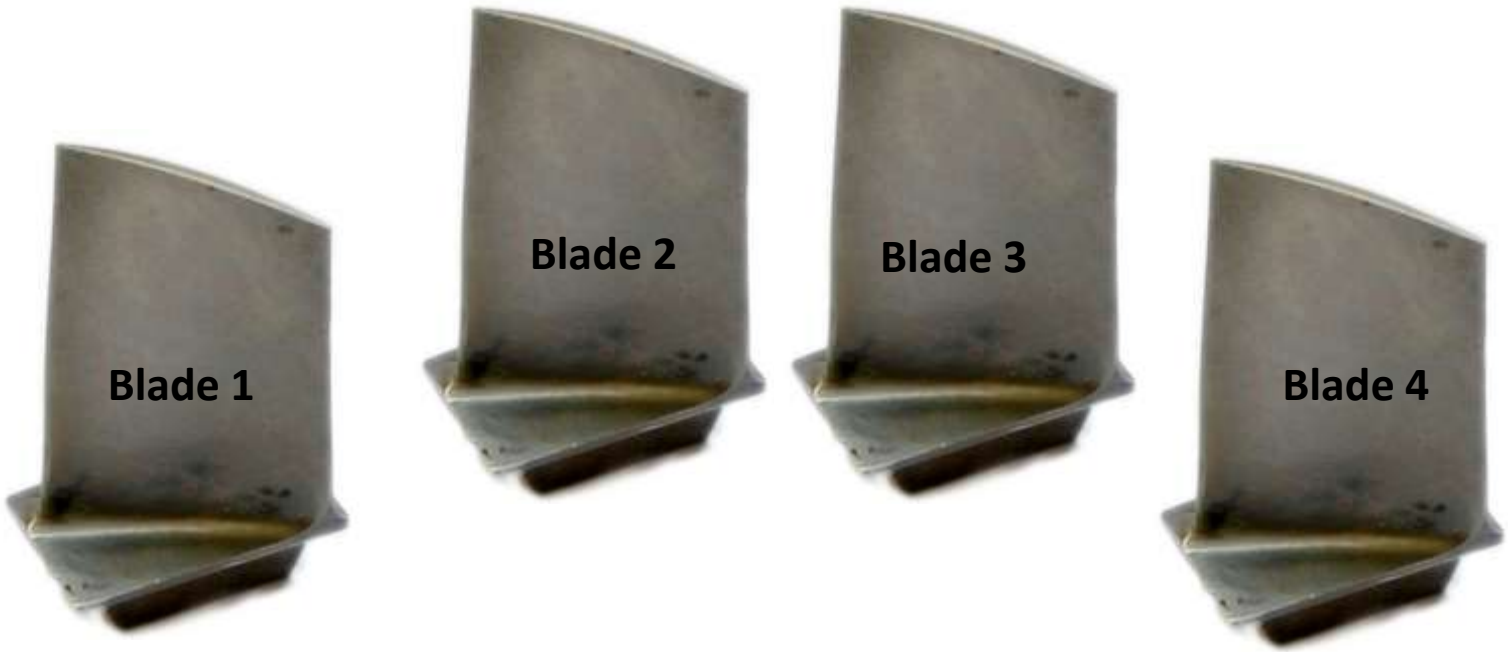
Optimization with Mistuning

- Blades of a rotor are manufactured and passed quality assurance
- All blades are different, mistuned
- What is the effect of mistunung?
- Is there a optimal configuration?
- Combinatoric is required (Workflow and Optimization Algorithm)



...

[Bilder: Wikipedia]

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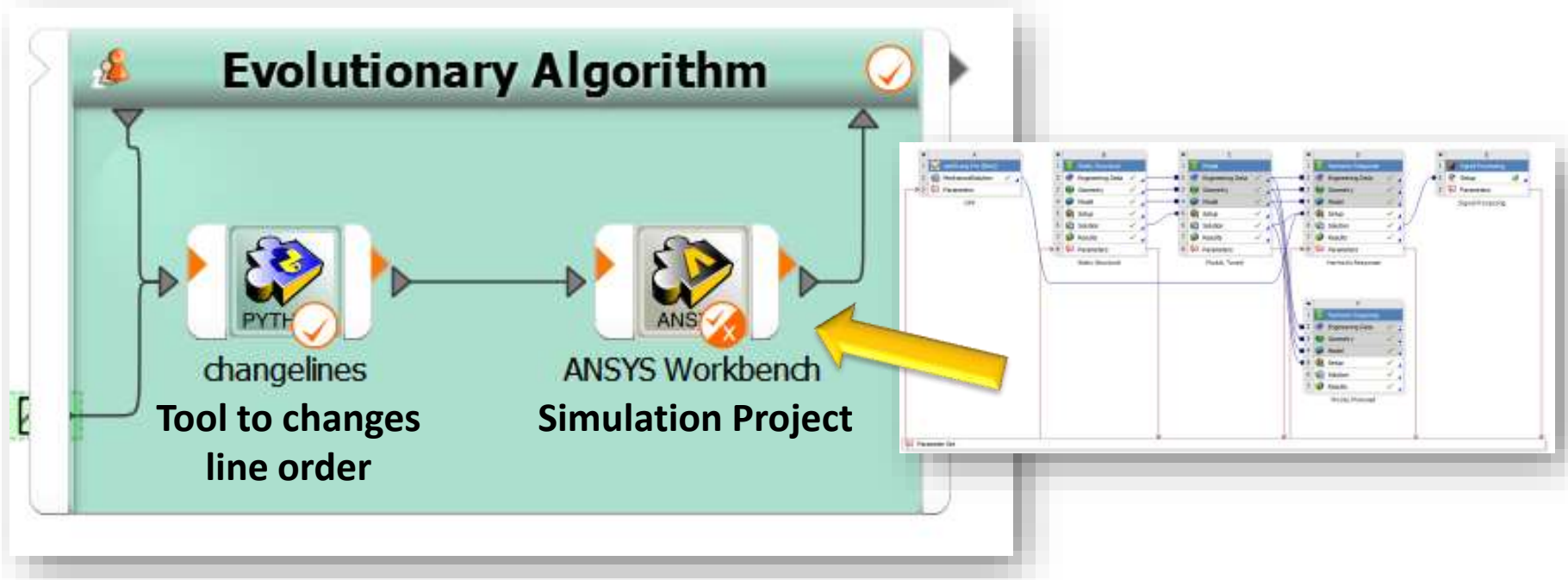
Workflow in optiSLang

- Cantilevered blade frequencies are measured and stored in a input file
- Each line represents a physical blade
- Changing lines in the input file represents another blade configuration on the disc
- Optimizer varies line order
- Objective: minimize maximal stress/displacement in frequency response



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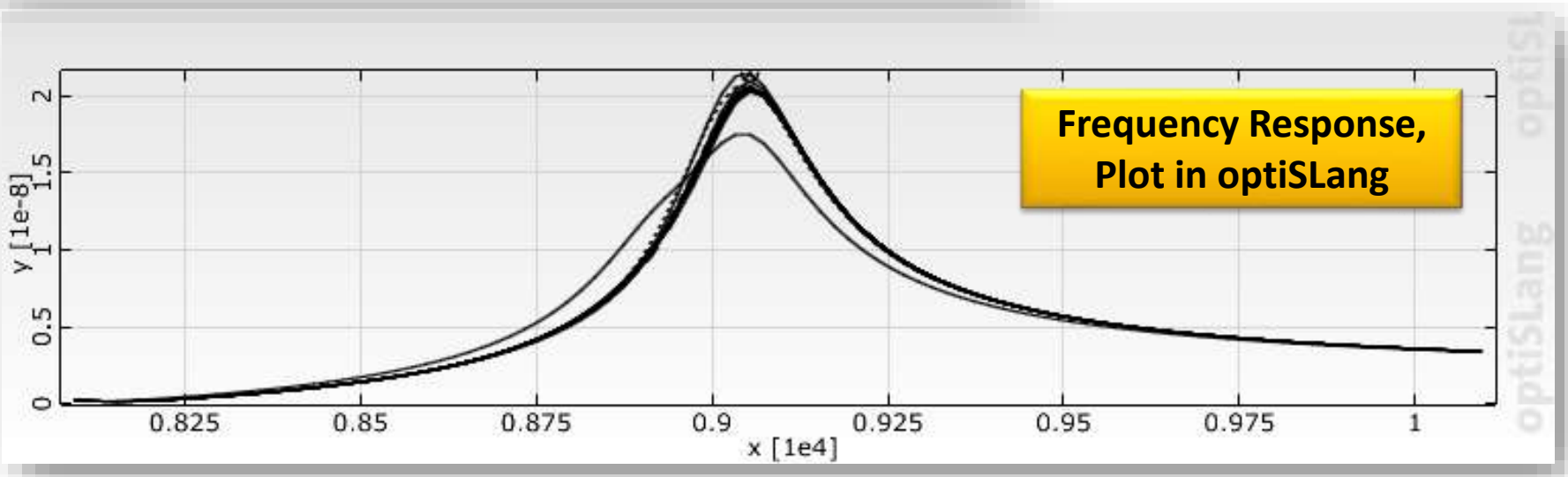
Workflow Details

- Frequency response for all blades are output signals
- Maximal values at each frequency represent response hull

```
Amplitude values for node     411 UY  RSYS=  1811
,Max,Max,Max
Frequen,Value,  Node#,Sector#,Sector  1,Sector  2,Sector  3,Sector  4,Sector
  8091.0    , 0.20897E-09,     411,    22, 0.15191E-09, 0.20734E-09, 0.20717E-09,
  8111.0    , 0.17495E-09,     411,    22, 0.14010E-09, 0.17330E-09, 0.17315E-09,
  8131.0    , 0.14891E-09,     411,    22, 0.14873E-09, 0.14738E-09, 0.14729E-09,
  8151.0    , 0.17678E-09,     411,     1, 0.17678E-09, 0.13627E-09, 0.13630E-09,
  8171.0    , 0.21866E-09,     411,     1, 0.21866E-09, 0.14544E-09, 0.14566E-09,
  8191.0    , 0.26967E-09,     411,     1, 0.26967E-09, 0.17360E-09, 0.17400E-09,
  8211.0    , 0.32712E-09,     411,     1, 0.32712E-09, 0.21512E-09, 0.21566E-09,
  8231.0    , 0.38973E-09,     411,     1, 0.38973E-09, 0.26538E-09, 0.26605E-09,
  8251.0    , 0.45687E-09,     411,     1, 0.45687E-09, 0.32183E-09, 0.32264E-09,
```

Frequency Response, i.e. Output from Simulation Process



Frequency Response, Plot in optiSLang

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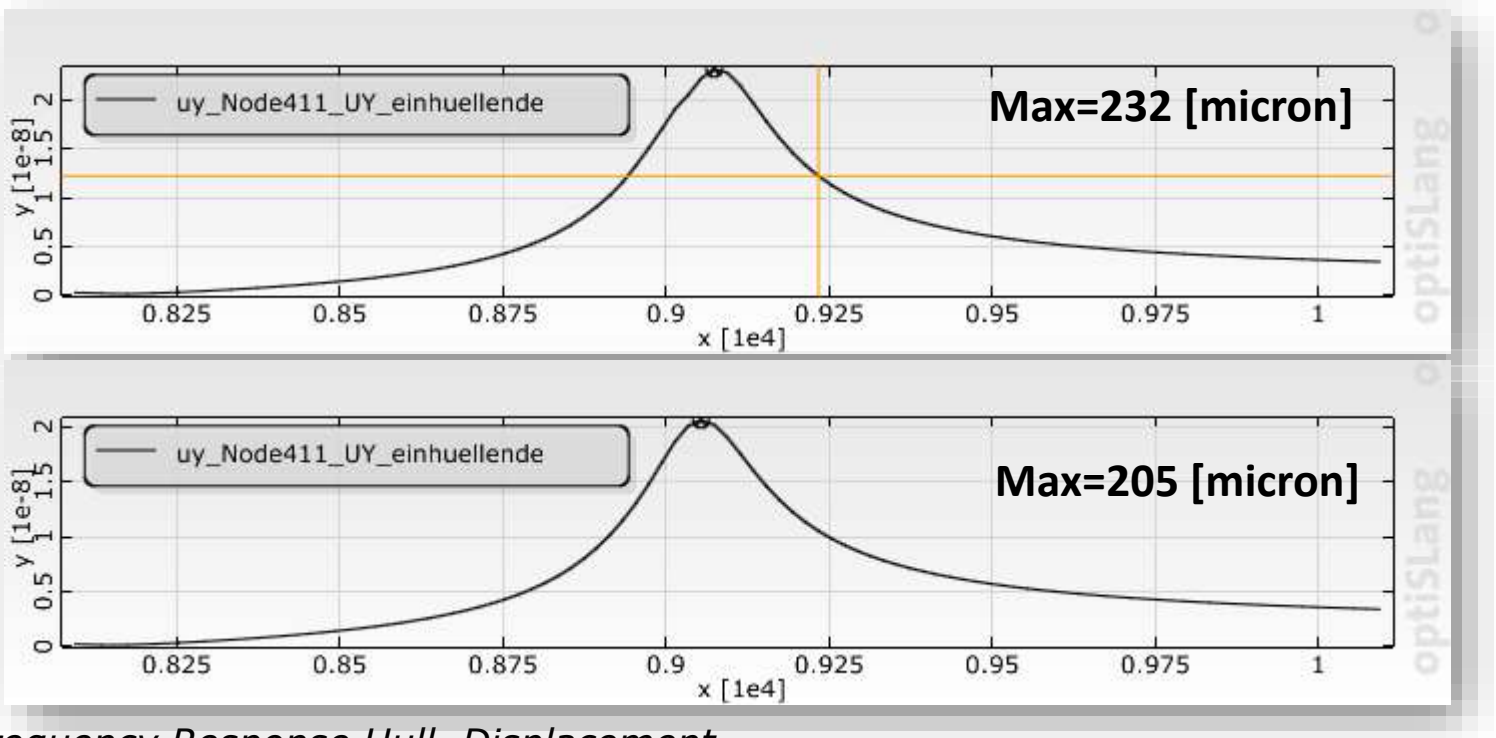
First Result

- Best Configuration vs. Worst Configuration

→ Minimize vs. Maximize



Frequency Response Hull, Displacement

Worst/Best Configuration = 1.13, i.e. 13% Variation!

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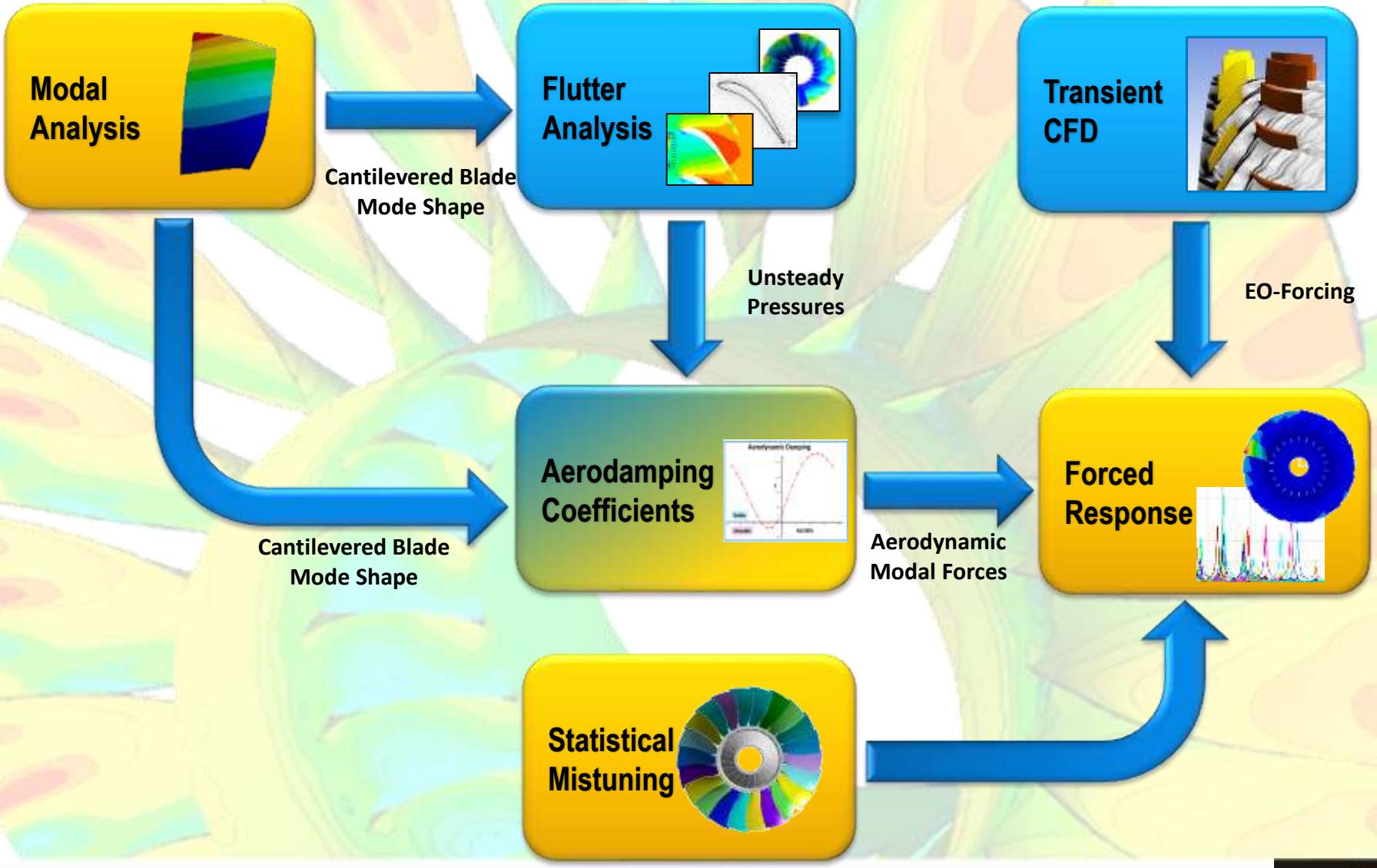
Summary Aeromechanics



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