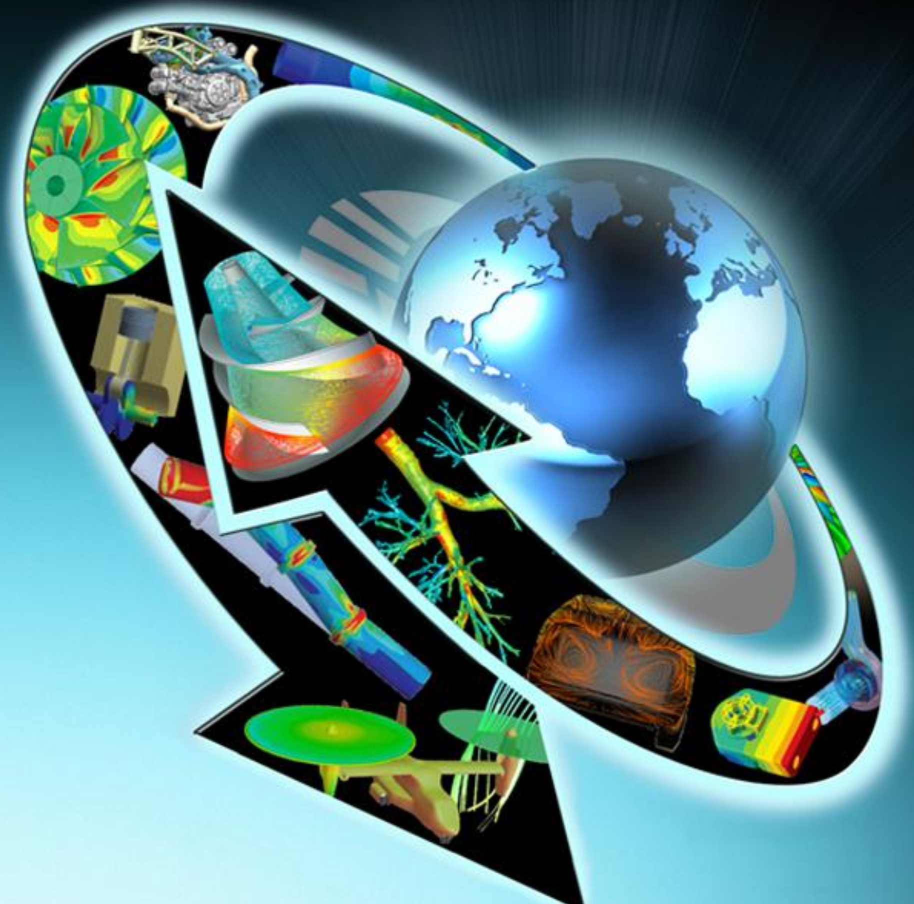
Rotordynamics with ANSYS Mechanical Solutions



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Agenda

1. Why / what is Rotordynamics
2. Equations for rotating structures
3. Rotating and stationary reference frame
4. Elements for Rotordynamics
5. Commands for Rotordynamics
6. Campbell diagram - Multi-spool rotors
7. Backward / forward whirl & orbit plots
8. Forced response
9. Instability
10. Rotordynamics analysis guide
11. Examples

Rotordynamics - why / what is rotordynamics ?

- High speed machinery such as Turbine Engine Rotors, Computer Disk Drives, etc.
- Very small rotor-stator clearances
- Flexible bearing supports – rotor instability
  - Finding critical speeds
  - Unbalance response calculation
  - Response to Base Excitation
  - Rotor whirl and system stability predictions
  - Transient start-up and stop
    - Model gyroscopic moments generated by rotating parts.
    - Account for bearing flexibility (oil film bearings)
    - Model rotor imbalance and other excitation forces (synchronous and asynchronous excitation).

Rotordynamics features

- **Pre-processing:**
  - Appropriate element formulation for all geometries
  - Gyroscopic moments generated by rotating parts
  - Bearings
  - Rotor imbalance and other excitation forces
  - Rotational velocities
  - Structural damping

- **Solution:**
  - Complex eigensolver for modal analysis
  - Harmonic analysis
  - Transient analysis

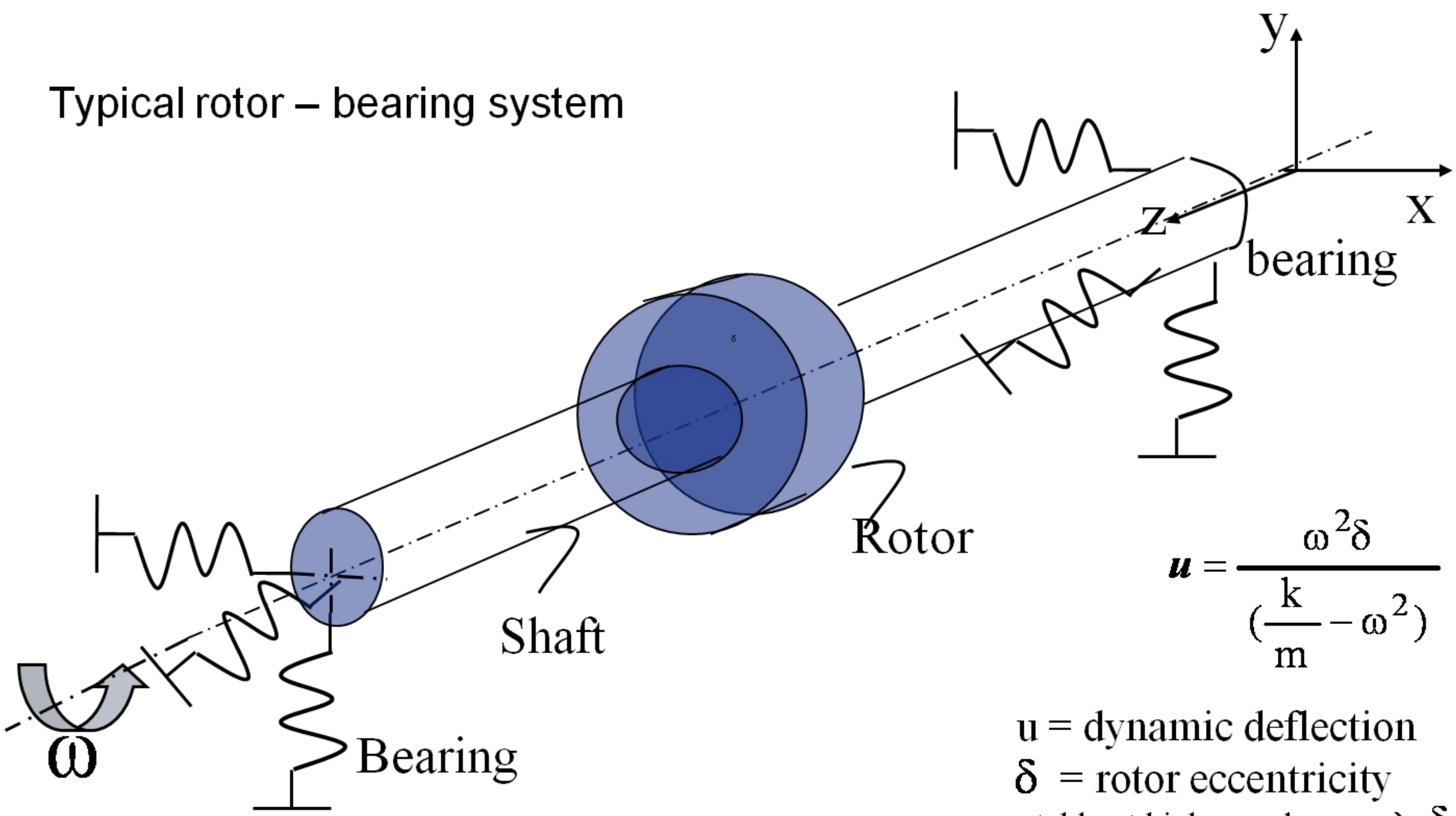
Rotordynamics features

- **Post-processing**
  - Campbell diagrams
  - Mode animation
  - Orbit plots
  - Transient plots and animations

- **User's guide**

- **Advanced features:**
  - Component Mode Synthesis for static parts

Typical rotor – bearing system



$$u = \frac{\omega^2 \delta}{\left(\frac{k}{m} - \omega^2\right)}$$

u = dynamic deflection

δ = rotor eccentricity

stable at high speeds as u → δ

- In a stationary reference frame, we are solving the following equation:

$$[M]\{\ddot{u}\}+([C]+[G])\{\dot{u}\}+([K]+[B])\{u\}=\{f\}$$

- ***M, C & K*** **are the standard mass, damping and stiffness matrices**

- ***G & B*** **represent respectively the gyroscopic and the rotating damping effect**

Dynamic equation in rotating reference frame

$$[M]\{\ddot{u}_r\} + ([C] + [C_{cor}])\{\dot{u}_r\} + ([K] - [K_{spin}])\{u_r\} = \{F\}$$

Coriolis matrix in dynamic analyses:

$$[C_{cor}] = 2\int \rho\, \Phi^T\, \overline{\omega}\, \Phi\, dv$$

$$\overline{\omega} = \begin{bmatrix} 0 & -\omega_z & \omega_y \\ \omega_z & 0 & -\omega_x \\ -\omega_y & \omega_x & 0 \end{bmatrix}$$

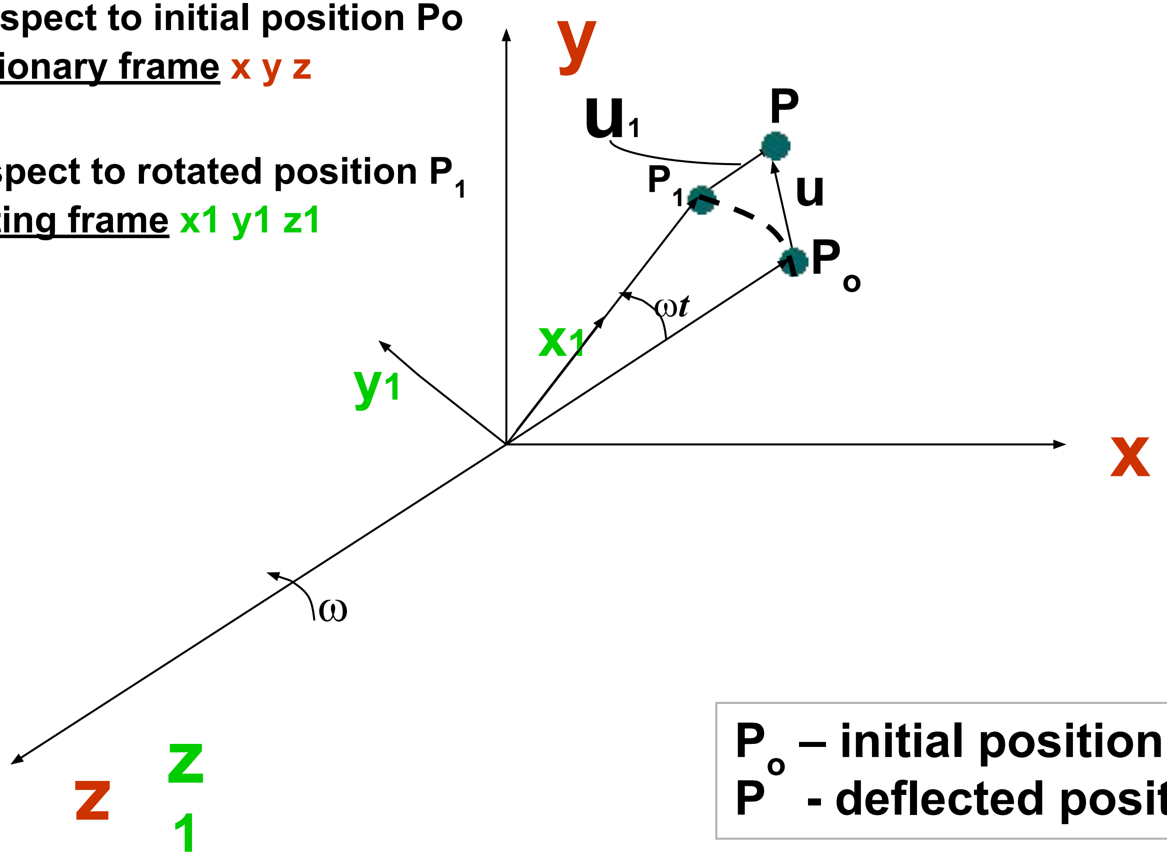
By extension, the Coriolis force in a static analysis:

$$\{f_c\} = [C_{cor}]\{\dot{u}_r\}$$

Rotating point mass P (small displacement)

u – disp. with respect to initial position Po
stationary frame x y z

u_1 – disp. with respect to rotated position P_1
rotating frame x1 y1 z1



P_o – initial position
P - deflected position

Acceleration of point mass P (rotating frame)

$$r = R + r_1$$

$$\dot{r} = \dot{R} + \dot{r}_1 + \omega \times r_1$$

$$\ddot{r} = \ddot{R} + (\ddot{r}_1 + \omega \times \dot{r}_1) + (\dot{\omega} \times r_1 + \omega \times \dot{r}_1 + \omega \times \omega \times r_1)$$

For constant R and ω

$$= \underbrace{\ddot{R}}_{0} + \ddot{r}_1 + 2\omega \times \dot{r}_1 + \underbrace{\dot{\omega}}_{0} \times r_1 + \omega \times \omega \times r_1$$

Acceleration of point mass due to deflection Po – P
(small displacement - rotating frame)

$$r = r_0 + \delta r => r = r_0 + u \quad r_1 = r_{10} + \delta r_1 => r_{10} + u_1$$

Acceleration vector

$$\ddot{u} = \ddot{u}_1 + 2\omega \times \dot{u}_1 + \omega \times \omega \times u_1 + \omega \times \omega \times r_{10}$$

Coriolis — **spin softening** — **centrifugal**

Rotordynamics - reference frames

- Rotordynamics simulation can be performed in two different reference frames:
  - *Stationary reference frame:*
    - Applies to a rotating structure (rotor) along with a stationary support structure
    - Rotating part of the structure to be modeled must be axisymmetric
  - *Rotating reference frame:*
    - The structure has no stationary parts and the entire structure is rotating
    - Consider only the Coriolis force

Rotordynamics - reference frames

Stationary Reference Frame	***Rotating Reference Frame***
Not applicable in static analysis	Applicable in static analysis
Can generate Campbell plots for computing rotor critical speeds.	Campbell plots are not applicable for computing rotor critical speeds.
Structure must be axisymmetric about spin axis.	Structure need not be axisymmetric about spin axis.
Rotating structure can be part of a stationary structure.	Rotating structure must be the only part of an analysis model (ex: gas turbine engine rotor).
Supports more than one rotating structure spinning at different rotational speeds about different axes of rotation (ex: a multi-spool gas turbine engine).	Supports only a single rotating structure (ex: a single-spool gas turbine engine).

↑

Our focus in this presentation

Applicable ANSYS element types

	Stationary Reference Frame	Rotating Reference Frame
Rel. 10.0	BEAM4, PIPE16, MASS21 BEAM188, BEAM189	SHELL181, PLANE182, PLANE183, SOLID185 SOLID186, SOLID187, BEAM188, BEAM189, SOLSH190, MASS21
Rel. 11.0	SOLID185, SOLID186, SOLID187, SOLID45, SOLID95	
Rel. 12.0	SHELL63 SHELL181, SHELL281 SOLID272, SOLID273 PIPE288, PIPE289	

Rotating damping

- **Considered if the rotating structure has:**
  - **structural damping (MP, DAMP or BETAD)**
  - **or a localized rotating viscous damper (bearing)**
- **The damping forces can induce unstable vibrations.**
- **The rotating damping effect is activated along with the Coriolis effect (CORIOLIS command).**

Damper	COMBI214
Beam	BEAM4, PIPE16 BEAM188, BEAM189
Solid	SOLID45, SOLID95 SOLID185, SOLID186, SOLID187
General axisymmetric	SOLID272, SOLID273 (new in V 12.0)

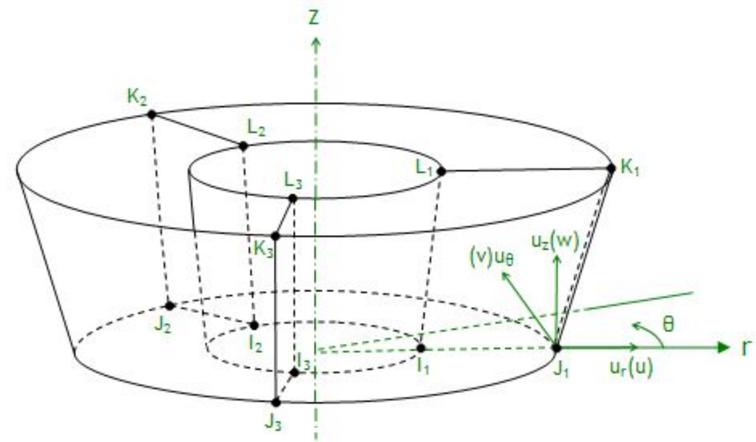
Elements supporting rotating damping

General axisymmetric element

In v12.0, the new SOLID272 (4nodes) and SOLID273 (8nodes) generalized axisymmetric elements:

- **are computationally efficient when compared to 3D solid**
- **support 3D non axisymmetric loading**

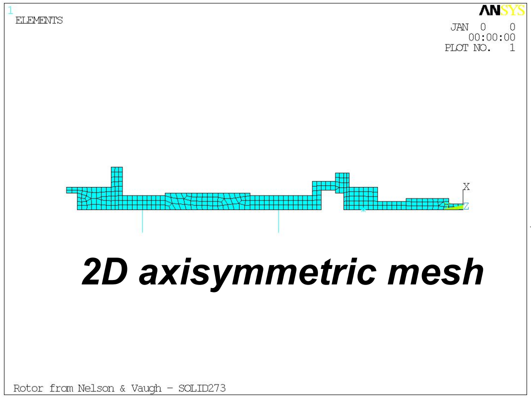
Example of mesh for SOLID272 element with 3 circumferential nodes.
Only (I1 J1 K1 L1) are input while all others nodes are



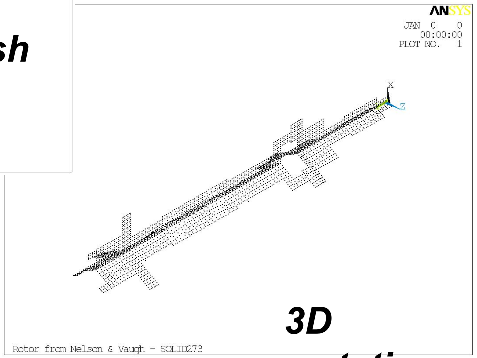
Generalized axisymmetric element

- **Allow a very fast setup of axisymmetric 3D parts:**
  - **Slice an axisymmetric 3D CAD geometry to get planar model**
  - **Mesh with 272/273 elements**
  - **No need to calculate equivalent beam sections**
  - **Can be combined with full 3D models, including contact**

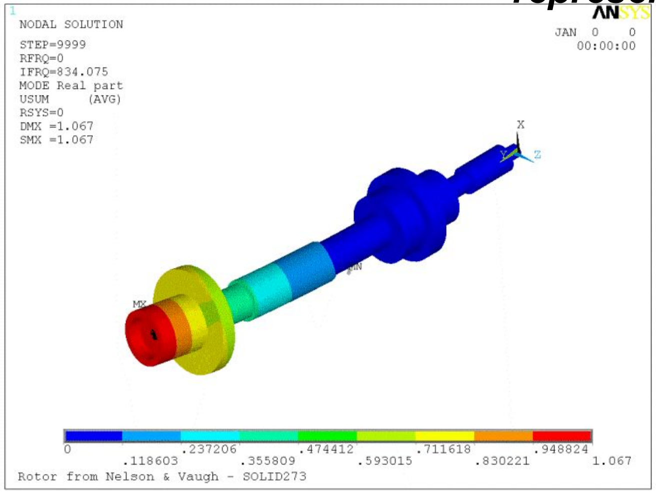
- **Support Gyroscopic effect in the stationary reference frame**



2D axisymmetric mesh



3D representation



3D results (not necessarily axisymmetric)

Typical Rotor – Bearing System

Bearing support coefficients

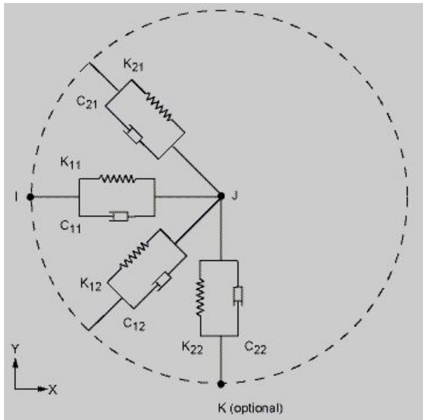
$$\begin{bmatrix} C_{xx} & C_{xy} \\ C_{yx} & C_{yy} \end{bmatrix} \begin{Bmatrix} \dot{u}_x \\ \dot{u}_y \end{Bmatrix} + \begin{bmatrix} K_{xx} & K_{xy} \\ K_{yx} & K_{yy} \end{bmatrix} \begin{Bmatrix} u_x \\ u_y \end{Bmatrix} = \begin{Bmatrix} F_x \\ F_y \end{Bmatrix}$$

Bearing coefficients may be function of rotational speed:

$$C(\omega) \qquad K(\omega)$$

Bearings

- **2D spring/damper with cross-coupling terms:**
  - **Real constants are stiffness and damping coefficients and can vary with spin velocity w**
- **Bearing element choice depends on:**
  - **Shape (1D, 2D, 3D)**
  - **Cross terms**
  - **Nonlinearities**



	Description	Stiffness and Damping cross terms	Nonlinear stiffness and damping characteristics
COMBIN14	Uniaxial spring/damper	No	No
COMBI214	2-D spring/damper	Unsymmetric	Function of the rotational velocity
MATRIX27	General stiffness or damping matrix	Unsymmetric	No
MPC184	Multipoint constraint element	Symmetric for linear characteristics - None for nonlinear characteristics	Function of the displacement

Coriolis / Gyroscopic effect

CORIOLIS, ***Option***, **--, --,** ***RefFrame***, ***RotDamp***
Applies the Coriolis effect to a rotating structure.

Option **Flag to activate or deactivate the Coriolis effect:**
- **1 (ON or YES) — Activate. This value is the default.**
- **0 (OFF or NO) — Deactivate.**

RefFrame **Flag to activate or deactivate a stationary reference frame.**
- **1 (ON or YES) — Activate.**
- **0 (OFF or NO) — Deactivate. This value is the default.**

RotDamp **Flag to activate or deactivate rotating damping effect.**
- **1 (ON or YES) — Activate.**
- **0 (OFF or NO) — Deactivate. This value is the default**

Eigensolver	Input	Usages	Applicable Matrices	Extraction Technique
QR Damped	**MODOPT**, QRDAMP	– Brake squeal and rotordynamics eigenproblems – Able to extract complex eigenvalues resulting from damping in the system (ALPHAD, BETAD, etc.) – Performance is similar to Block Lanczos – Good for up to, say 1 million DOF's extracting, say less than 100 modes.	K, C, M (non-symmetric **except** M)	Block lanczos and QR algorithm for the modal subspace matrices
Damped	**MODOPT**, DAMP	– Rotordynamics eigenproblems – Noise Vibration Harshness (NVH) problems with structural acoustics coupling and damping – Optimal performance up to about 200K DOF's, extracting, say 100 modes – Doesn't support modal superposition transient or harmonic analysis	K, C, M (non-symmetric)	A subspace method based on Variational Technology (VT) algorithm

Specify rotational velocity: ω

OMEGA, ***OMEGX***, ***OMEGY***, ***OMEGZ***, ***KSPIN***

Rotational velocity of the structure.
SOLUTION: inertia

activate *KSPIN* for gyroscopic effect in rotating reference frame (by default for dynamic analyses)

CMOMEGA, ***CM_NAME***, ***OMEGAX***, ***OMEGAY***, ***OMEGAZ***, ***X1***, ***Y1***, ***Z1***, ***X2***, ***Y2***, ***Z2***, ***KSPIN***

Rotational velocity -element component about a user-defined rotational axis.
SOLUTION: inertia

RSTMAC, file1, Lstep1, Sbstep1, file2, Lstep2, Sbstep2, TolerN, MacLim, Cname, KeyPrint

Filei **First Jobname (DB and RST files)**
Lstepi **Load step number in file1.rst**
Sbstepi **Substep number (or All) in file1.rst**

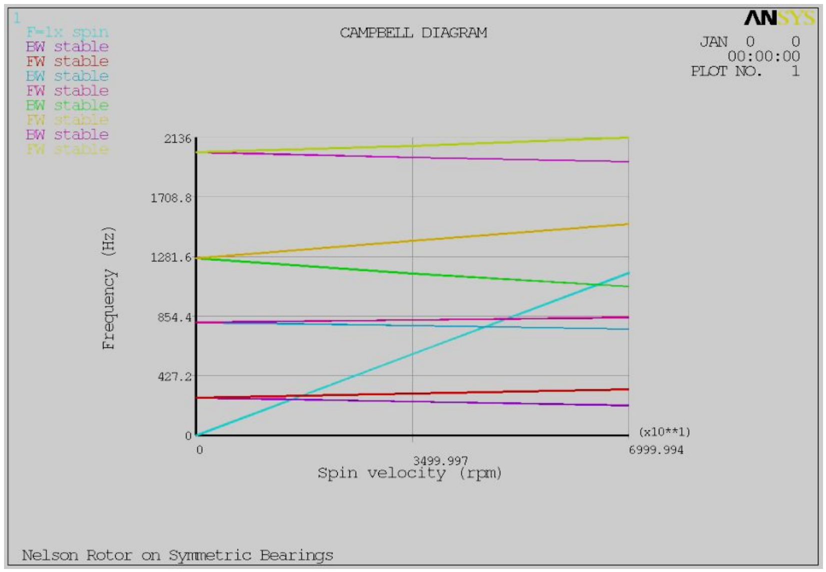
TolerN **Tolerance for node matching**
MacLim **Smallest acceptable value of Modal Assurance Criterion for solution matching**
Cname **Name of the component based on nodes (file1.db)**
KeyPrint **Printout options**

```
*********************************** MATCHED SOLUTIONS ***********************************
 Substep in         Substep in          MAC value      Frequency            Frequency
   tbeam.rst          tsolid.rst                     difference (Hz)         error (%)
       1                  1                1.000      -0.11E-01               0.2
       2                  2                1.000       0.46E-02               0.1
       3                  3                1.000      -0.26E-01               0.2
       4                  4                1.000      -0.27E-01               0.1
       5                  5                1.000      -0.41E-01               0.1
       6                  6                1.000      -0.13E+00               0.2
       7                  7                1.000      -0.11E+00               0.2
       8                  8                1.000      -0.82E-01               0.1
       9                  9                1.000       0.11E+00               0.1
      10                 10                1.000       0.96E+00               0.6
```

Campbell diagram

- Variation of the rotor natural frequency with respect to rotor speed ω
- In modal analysis perform multiple load steps at different angular velocities ω
- Campbell commands
  - **CAMPB**: support Campbell for prestressed structures (/SOLU)
  - **PLCAMP**: display Campbell diagram (/POST1)
  - **PRCAMP**: print frequencies and critical speeds (/POST1)

Campbell diagram



PLCAMP, ***Option*, *SLOPE*, *UNIT*, *FREQB*, *Cname*, *STABVAL***

Option
- ***Flag to activate or deactivate sorting***

SLOPE
- ***The slope of the line which represents the number of excitations per revolution of the rotor.***

UNIT
- ***Specifies the unit of measurement for rotational angular velocities***

FREQB
- ***The beginning, or lower end, of the frequency range of interest.***

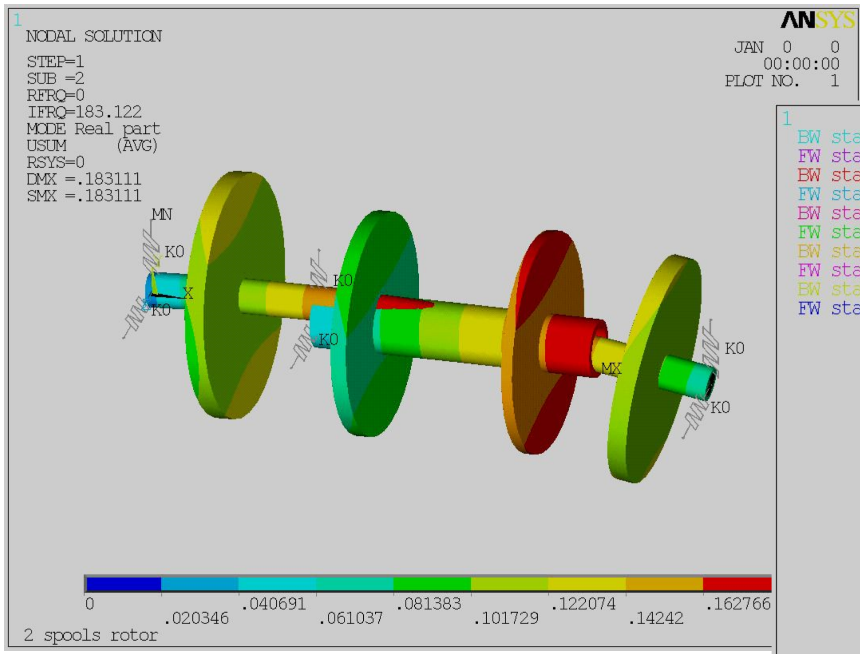
Cname
- ***The rotating component name***

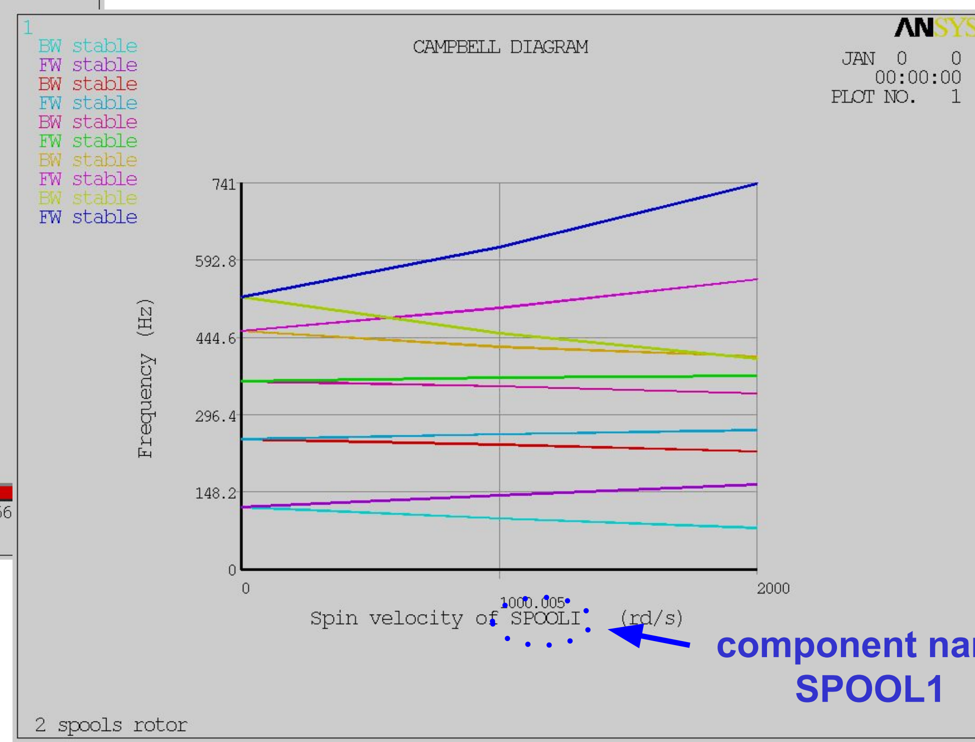
STABVAL
- ***Plot the real part of the eigenvalue (Hz)***

Rotordynamics – multi-spool rotors

More than 1 spool and / or non-rotating parts, use components (CM) and component rotational velocities (CMOMEGA).

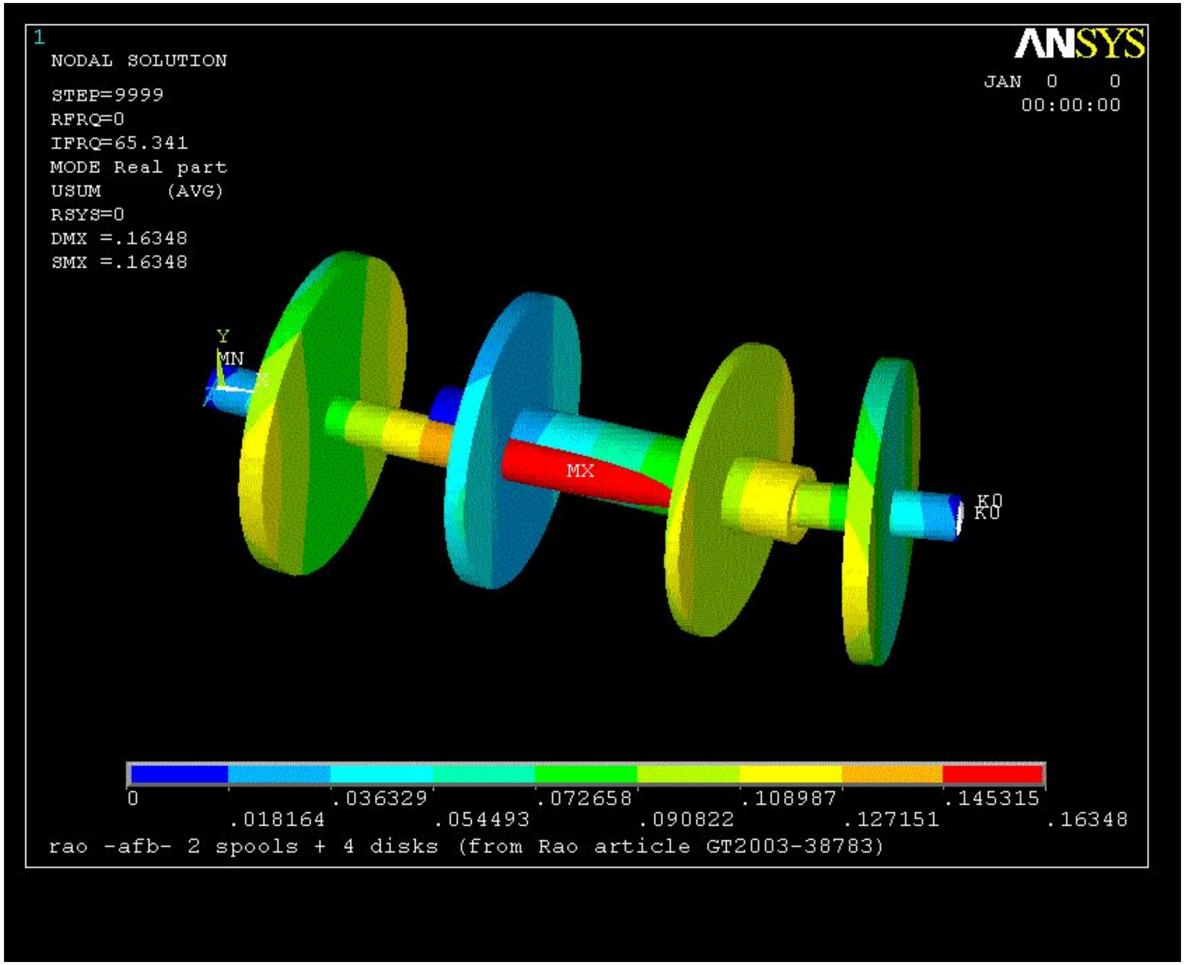
PLCAMP, ***Option, SLOPE, UNIT, FREQB, Cname***





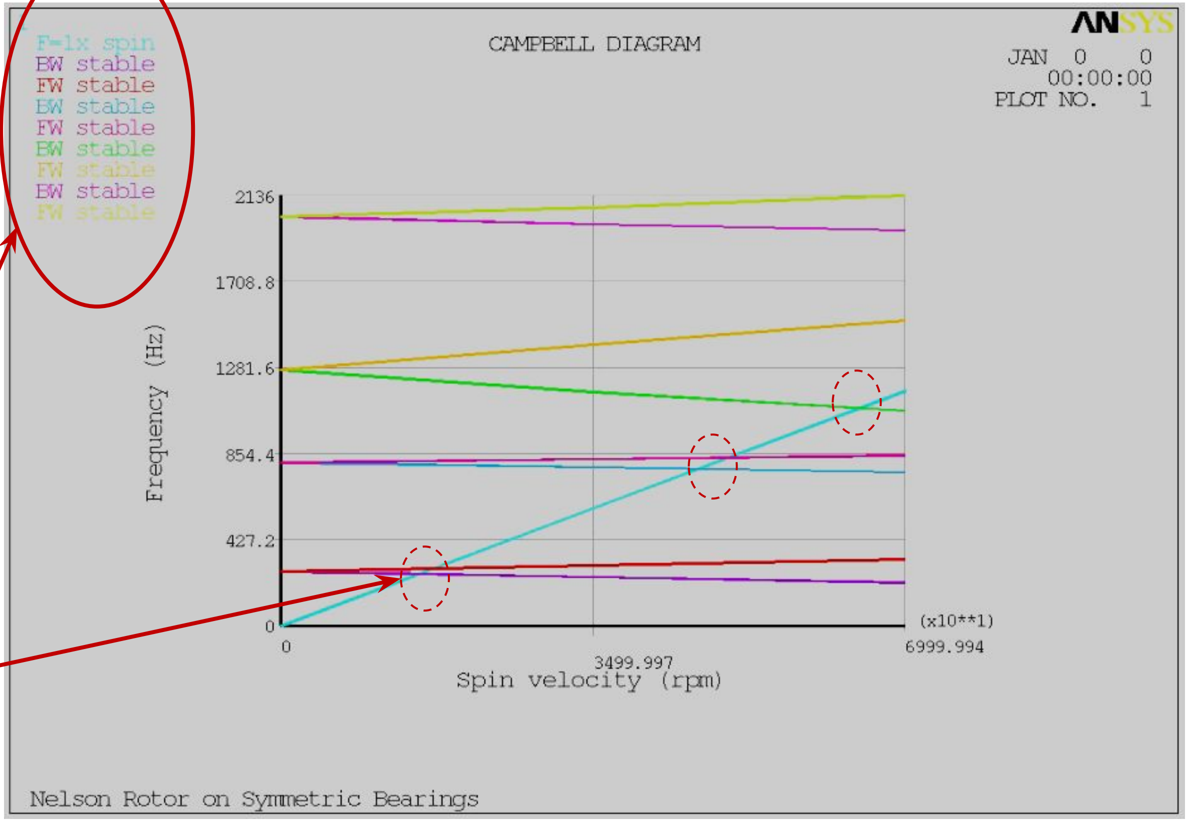
component name SPOOL1

Whirl animation (ANHARM command)



Campbell diagrams & whirl

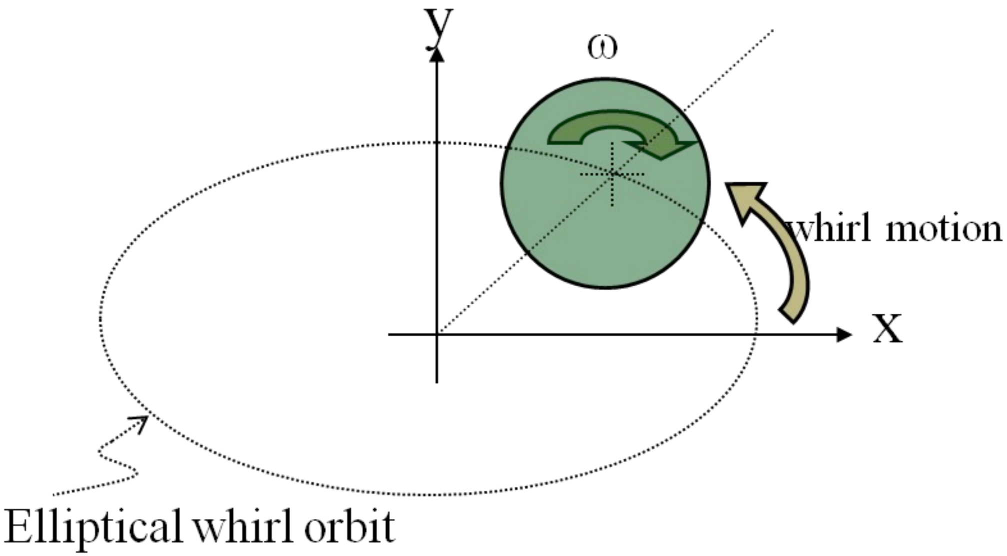
- Variation of the rotor natural frequencies with respect to rotor speed ω
- In modal analysis perform multiple load steps at different angular velocities ω
- As frequencies split with increasing spin velocity, ANSYS identifies:
  - forward (FW) and backward (BW) whirl
  - stable / unstable operation
  - critical speeds
- Also available for multispool models



Rotor whirl

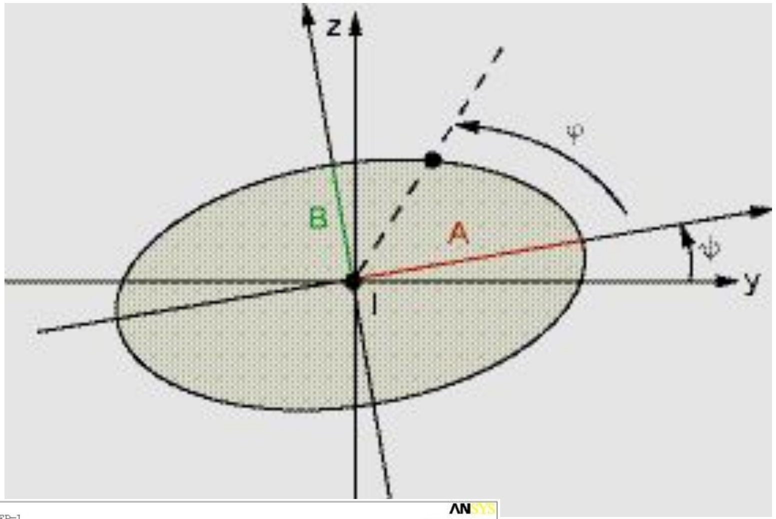
- ***Forward whirl***: when ω and the whirl motion are rotating in the same direction

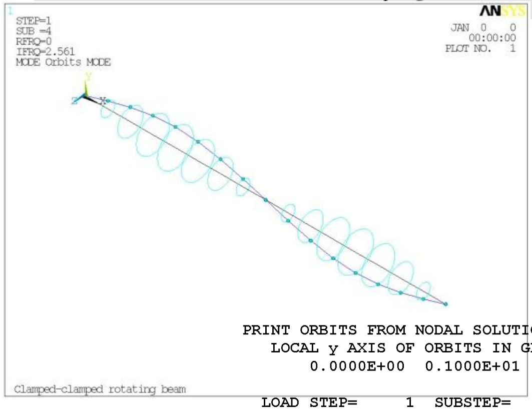
- ***Backward whirl***: when ω and the whirl motion are rotating in opposite directions



Orbit plots

- In a plane perpendicular to the spin axis, the orbit of a node is an ellipse
- It is defined by three characteristics: semi axes A, B and phase ψ in a local coordinate system (x, y, z) where x is the rotation axis
- Angle ϕ is the initial position of the node with respect to the major semi-axis A.
- Orbit plots are available for beam models





Plot orbit: PLORB

Print orbit: PRORB

```
PRINT ORBITS FROM NODAL SOLUTION
   LOCAL y AXIS OF ORBITS IN GLOBAL COORDINATES
       0.0000E+00  0.1000E+01  0.0000E+00

 LOAD STEP=      1  SUBSTEP=      4
  RFRQ=     0.0000      IFRQ=     2.5606      LOAD CASE=    0
ORBIT

   NODE       A           B          PSI         PHI         ymax        zmax
      1    0.0000      0.0000      0.0000      0.0000      0.0000      0.0000
      2    0.0000      0.0000      0.0000      0.0000      0.0000      0.0000
      3   0.38232     0.38232      0.0000      0.0000     0.38232     0.38232
      4   0.70711     0.70711      0.0000      0.0000     0.70711     0.70711
      5   0.92301     0.92301      0.0000      0.0000     0.92301     0.92301
```

Rotordynamics – forced response

Possible excitations caused by rotation velocity ω are:

- **Unbalance (ω)**
- **Coupling misalignment (2* ω)**
- **Blade, vane, nozzle, diffusers (s* ω)**
- **Aerodynamic excitations as in centrifugal compressors (0.5* ω)**

Ansys command for *synchronous and asynchronous* forces

SYNCHRO, ***ratio, cname***

- ***ratio***
  - The ratio between the frequency of excitation, f, and the frequency of the rotational velocity of the structure.
- ***Cname***
  - The name of the rotating component on which to apply the harmonic excitation.

Note: The **SYNCHRO** command is valid only for full harmonic analysis (**HROPT**,*Method* = FULL)

$\omega = 2\pi f / ratio$ *where, f* = excitation frequency (defined in **HARFRQ**)

The rotational velocity, ω, is applied along the *direction cosines* of the rotation axis (specified via an **OMEGA** or **CMOMEGA** command)

Unbalance response

How to input unbalance forces?

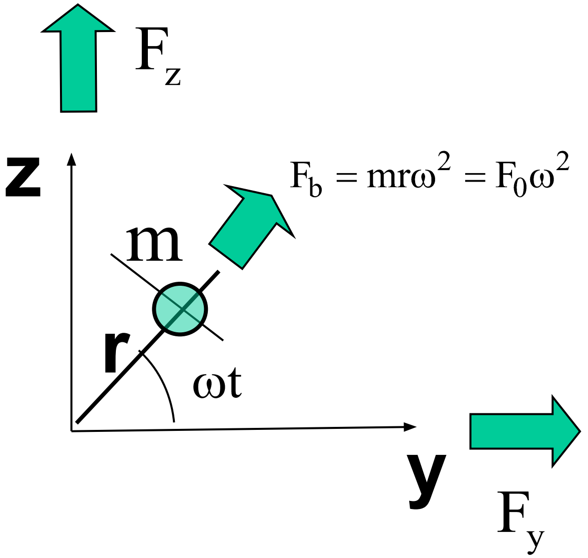
$$F_y = F_b \cos \omega t = F_b e^{j\omega t}$$

$$F_z = F_b \sin \omega t = F_b \cos(\omega t - \pi / 2)$$

$$\Rightarrow F_z = -jF_b e^{j\omega t}$$

! Example of input file

```
/prep7
…
F0=m*r
F, node, fy, F0
F, node, fz, , - F0
```



Rotordynamics – unbalance response

! Input unbalance forces

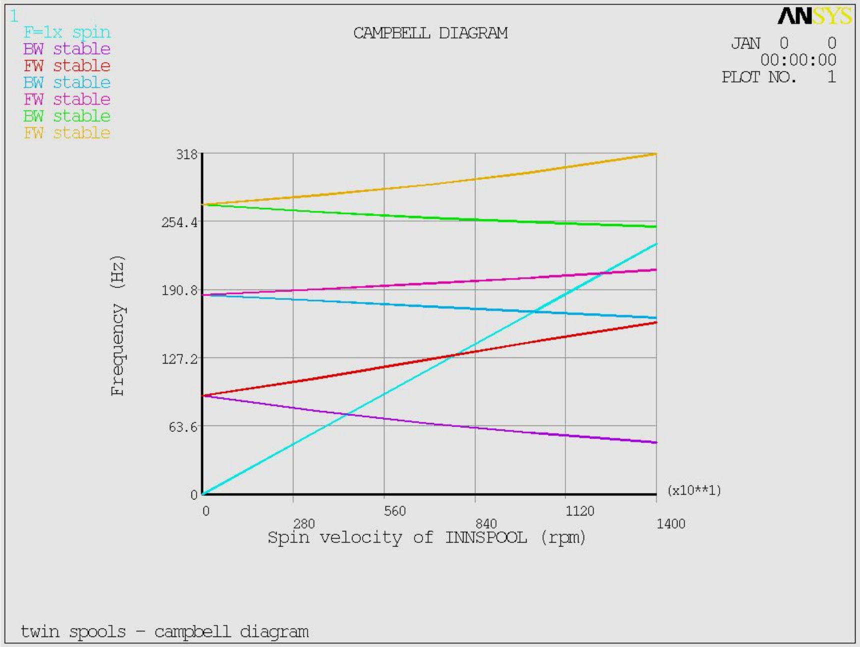
f0 = 70e-6

F, 7, FY, f0

F, 7, FZ, , -f0

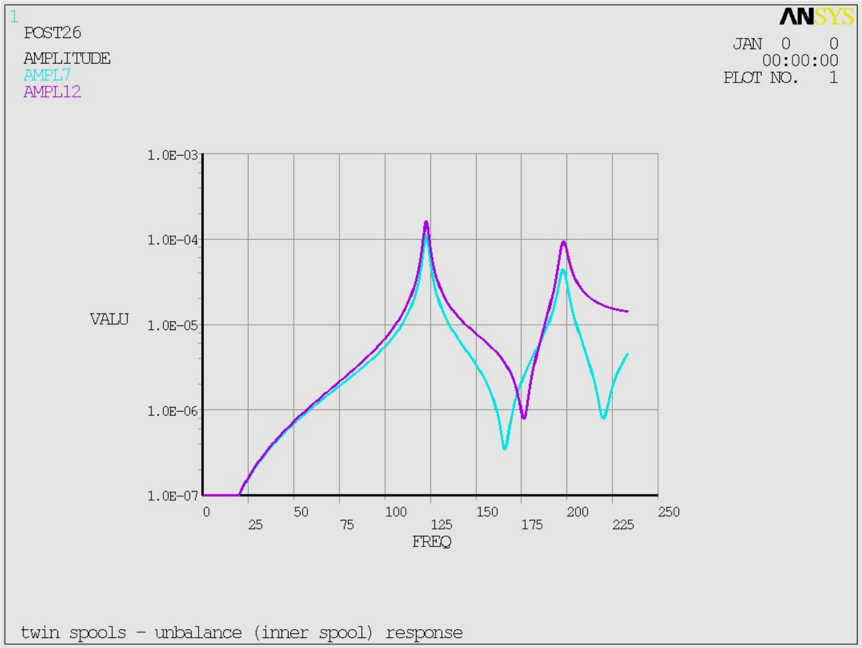
! Campbell plot of inner spool

plcamp, ,1.0, rpm, , innSpool



antype, harmic

synchro, , innSpool



- Self-excited vibrations in a rotating structure cause an increase of the vibration amplitude over time such as shown below.

- Such instabilities, if unchecked, can result in equipment damage.

- The most common sources of instability are:
  - Bearing characteristics
  - Internal rotating damping (material damping)
  - Contact between rotating and static parts

- Instabilities can be identified by performing a transient analysis or a modal analysis (complex frequencies)

For problems involving spinning structures with gyroscopic effects, and/or damped structural eigenfrequencies, the eigensolutions obtained with the Damped Method and QR Damped Method are complex. The eigenvalues $\bar{\lambda}_i$ are given by:

$$\bar{\lambda}_i = \sigma_i \pm j\omega_i$$

where:

$\bar{\lambda}_i$ = complex eigenvalue

σ_i = real part of the eigenvalue

ω_i = imaginary part of the eigenvalue (damped circular frequency)

$j = \sqrt{-1}$

The dynamic response of the system is given by:

$$\{u_i\} = \{\phi_i\}e^{\bar{\lambda}_i t}$$

where:

t = time

The ith eigenvalue is stable if σ_i is negative and unstable if σ_i is positive.

Modal damping ratio

The modal damping ratio is given by:

$$\alpha_i = \frac{-\sigma_i}{|\lambda_i|} = \frac{-\sigma_i}{\sqrt{\sigma_i^2 + \omega_i^2}} \quad (15\text{-}214)$$

where:

α_i = modal damping ratio of the ith eigenvalue

It is the ratio of the actual damping to the critical damping.

Logarithmic decrement

The logarithmic decrement represents the logarithm of the ratio of two consecutive peaks in the dynamic response ((Equation 15-213)). It can be expressed as:

$$\delta_i = \ln\left(\frac{u_i(t+T_i)}{u_i(t)}\right) = 2\pi\frac{\sigma_i}{\omega_i} \quad (15\text{-}215)$$

where:

δ_i = logarithmic decrement of the ith eigenvalue

T_i = damped period of the ith eigenvalue defined by:

$$T_i = \frac{2\pi}{\omega_i} \quad (15\text{-}216)$$

L O A D   S T E P   O P T I O N S

LOAD STEP NUMBER.   2

INERTIA LOADS	X	Y	Z
OMEGA.	3141.6	0.0000	0.0000

***** DAMPED FREQUENCIES FROM REDUCED DAMPED EIGENSOLVER *****

MODE	COMPLEX FREQUENCY (HERTZ)			MODAL DAMPING RATIO
1	-27.142724	203.90118	j	0.13195307
	-27.142724	-203.90118	j	0.13195307
2	-0.18391233	272.56561	j	0.67474502E-03
	-0.18391233	-272.56561	j	0.67474502E-03

L O A D   S T E P   O P T I O N S

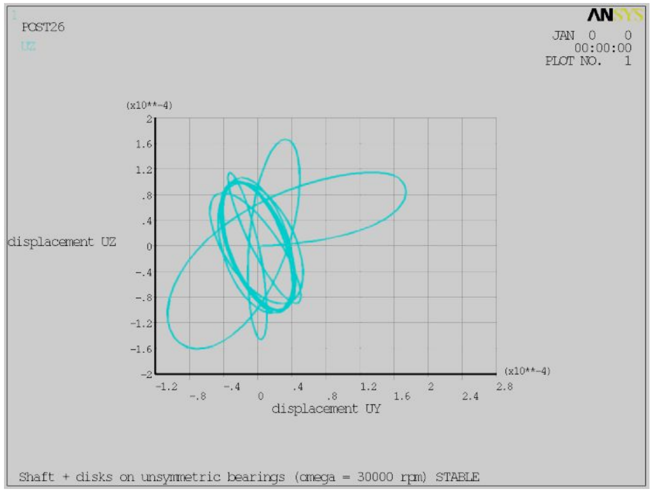
LOAD STEP NUMBER.   3

INERTIA LOADS	X	Y	Z
OMEGA.	6283.2	0.0000	0.0000

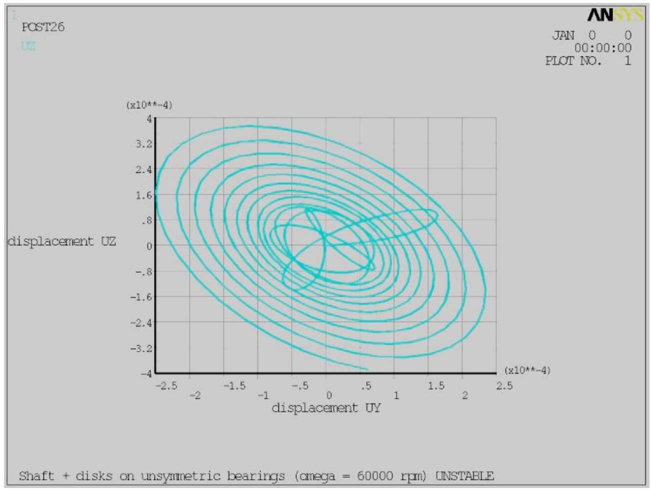
***** DAMPED FREQUENCIES FROM REDUCED DAMPED EIGENSOLVER *****

MODE	COMPLEX FREQUENCY (HERTZ)			MODAL DAMPING RATIO
1	-30.277781	186.52468	j	0.16022861
	-30.277781	-186.52468	j	0.16022861
2	6.0020412	289.58296	j	0.20722049E-01
	6.0020412	-289.58296	j	0.20722049E-01

Stable at 30,000 rpm (3141.6 rad/sec)

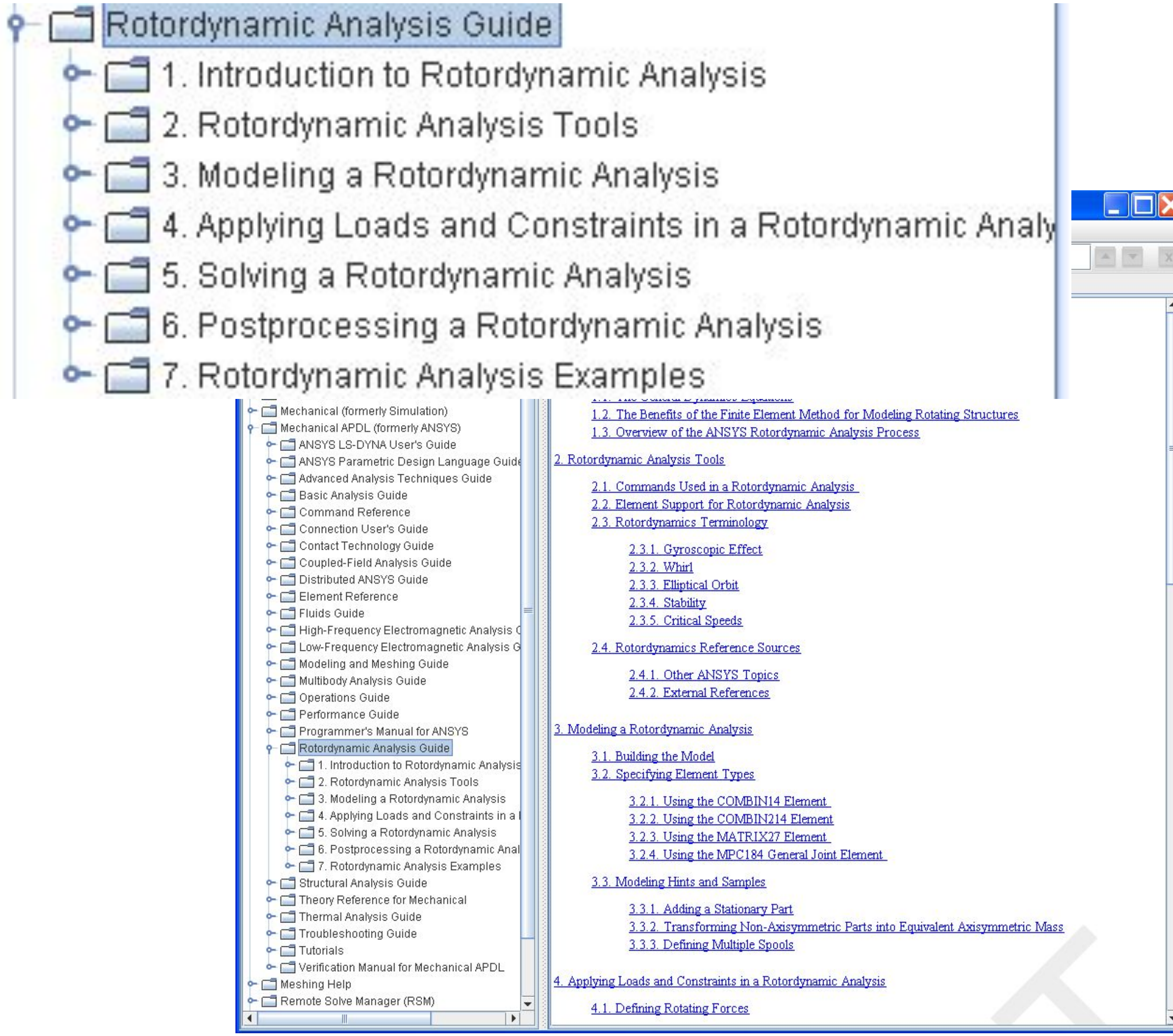


Unstable at 60,000 rpm (6283.2 rad/sec)

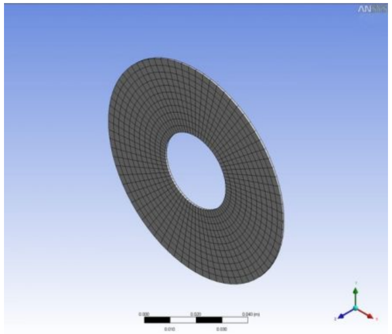


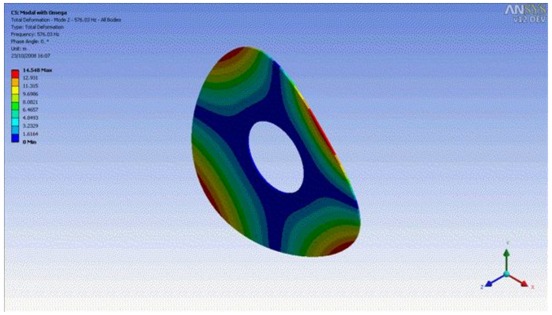
Rotordynamics analysis guide

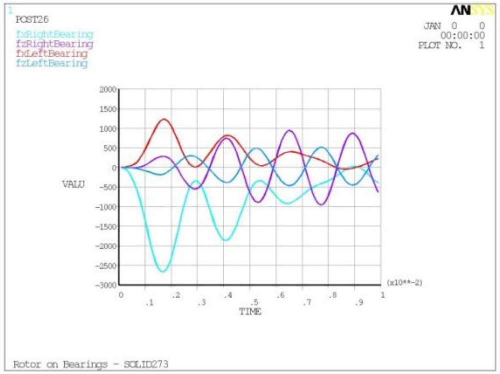
- **New at release 12.0**
- **Provides a detailed description of capabilities**
- **Provides guidelines for rotordynamics model setup**



Sample models available





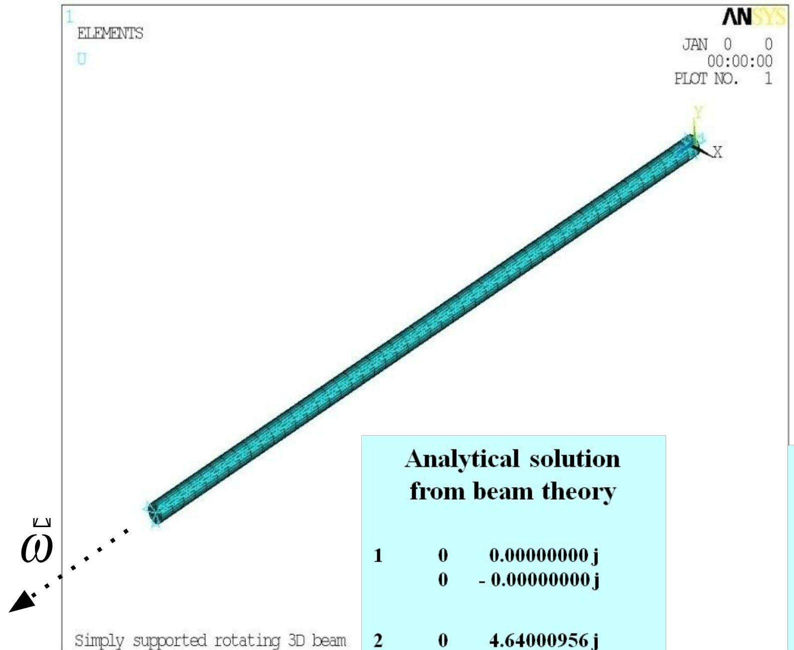


Some examples

Validation examples

Generic validation model

- **Modal analysis of a 3D beam (solid elements), ω=30000 rpm**
- **Excellent agreement between simulation and theory**
- **Ref:** ***Gerhard Sauer & Michael Wolf, 'FEA of Gyroscopic effects,' Finite Elements in Analysis & Design, 5, (1989), 131-140***



Analytical solution from beam theory

1	0	0.00000000 j
	0	- 0.00000000 j
2	0	4.64000956 j
	0	- 4.64000956 j
3	0	8.32109166 j
	0	- 8.32109166 j
4	0	18.5600383
	0	- 18.5600383
5	0	33.2843666 j
	0	- 33.2843666 j
6	0	41.7600861 j
	0	41.7600861 j
7	0	74.889824 j
	0	- 74.889824 j
8	0	74.2401530 j
	0	-74.2401530 j

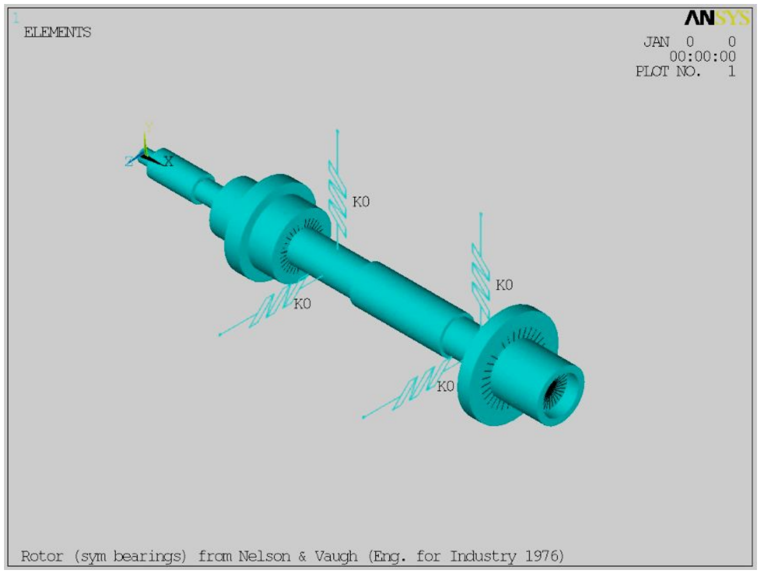
less than 0.5% error

Finite element solution (SOLID185)

1	-0.62751987E-08	0.27924146E-03j
	-0.62751987E-08	-0.27924146E-03j
2	0.0000000	4.6316102 j
	0.0000000	-4.6316102 j
3	0.0000000	8.2842343 j
	0.0000000	-8.2842343 j
4	0.0000000	18.515548 j
	0.0000000	-18.515548 j
5	0.0000000	33.062286 j
	0.0000000	-33.062286 j
6	0.0000000	41.619417 j
	0.0000000	-41.619417 j
7	0.0000000	73.890203 j
	0.0000000	-73.890203 j
8	0.0000000	74.113637 j
	0.0000000	-74.113637 j

Nelson rotor (beams & bearings)

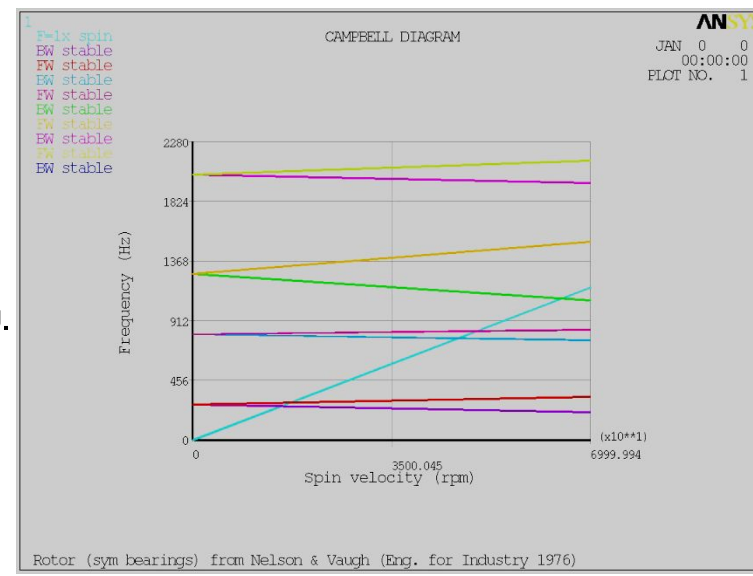
ANSYS Training Manual



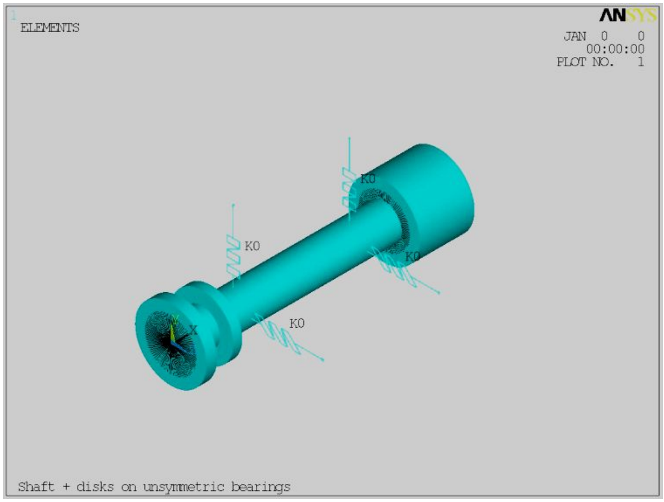
Damped Natural Frequencies (Hz)						
	Whirl		0 rpm		70,000 rpm	
F (Hz)	Ansys	[1]	Ansys	[1]	Ansys	[1]
1	BW	BW	271.2	271.1	214.5	213.6
2	FW	FW	271.2	271.1	329.8	330.6
3	BW	BW	808.8	806.4	762.4	760.0
4	FW	FW	808.8	806.4	844.9	842.6
5	BW	BW	1272.0	1273.0	1068.7	1066.5
6	FW	FW	1272.0	1273.0	1516.2	1522.0

Critical speeds (rpm)	
Ansys	[1]
15,494	15,470
17,146	17,159
46,729	46,612
50,114	49,983
64,924	64,752
95,747	96,457

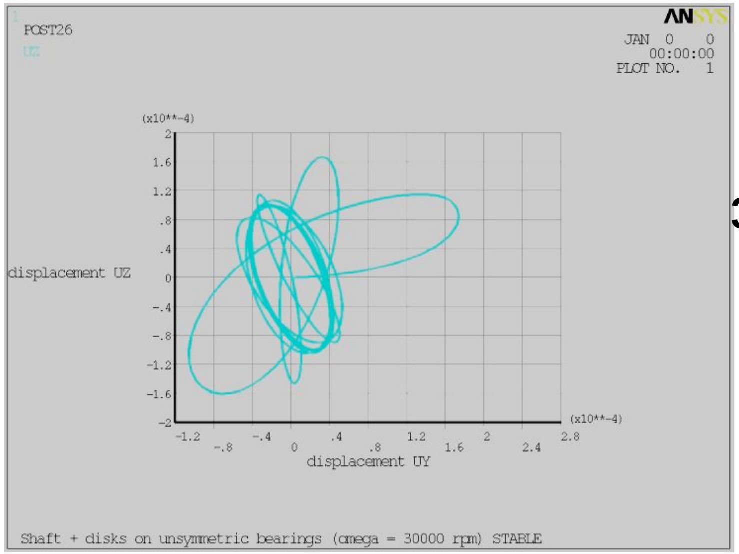
Ref. [1]: 'Dynamics of rotor-bearing systems using finite elements,' J. of Eng. for Ind., May 1976



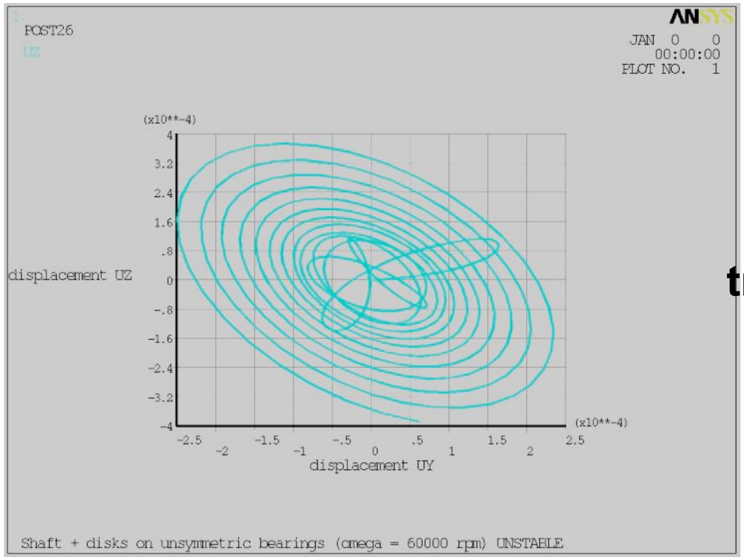
Instability analysis – transient analysis



Rotor with unsymmetrical bearings

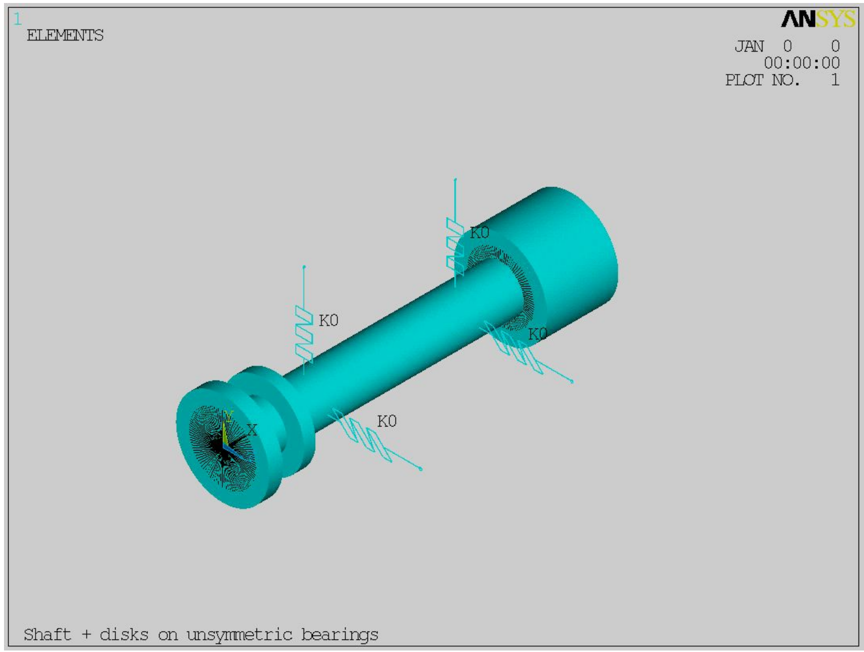


30,000 rpm; closed trajectory: stable



60,000 rpm; open trajectory: unstable

Instability analysis – modal analysis



LOAD STEP OPTIONS

LOAD STEP NUMBER. 2

INERTIA LOADS	X	Y	Z
OMEGA.	3141.6	0.0000	0.0000

30,000 rpm

***** DAMPED FREQUENCIES FROM REDUCED DAMPED EIGENSOLVER *****

MODE	COMPLEX FREQUENCY (HERTZ)			MODAL DAMPING RATIO
1	-27.142724	203.90118	j	0.13195307
	-27.142724	-203.90118	j	0.13195307
2	-0.18391233	272.56561	j	0.67474502E-03
	-0.18391233	-272.56561	j	0.67474502E-03

All complex frequencies' real parts are negative: stable

LOAD STEP OPTIONS

LOAD STEP NUMBER. 3

INERTIA LOADS	X	Y	Z
OMEGA.	6283.2	0.0000	0.0000

60,000 rpm

***** DAMPED FREQUENCIES FROM REDUCED DAMPED EIGENSOLVER *****

MODE	COMPLEX FREQUENCY (HERTZ)			MODAL DAMPING RATIO
1	-30.277781	186.52468	j	0.16022861
	-30.277781	-186.52468	j	0.16022861
2	6.0020412	289.58296	j	0.20722049E-01
	6.0020412	-289.58296	j	0.20722049E-01

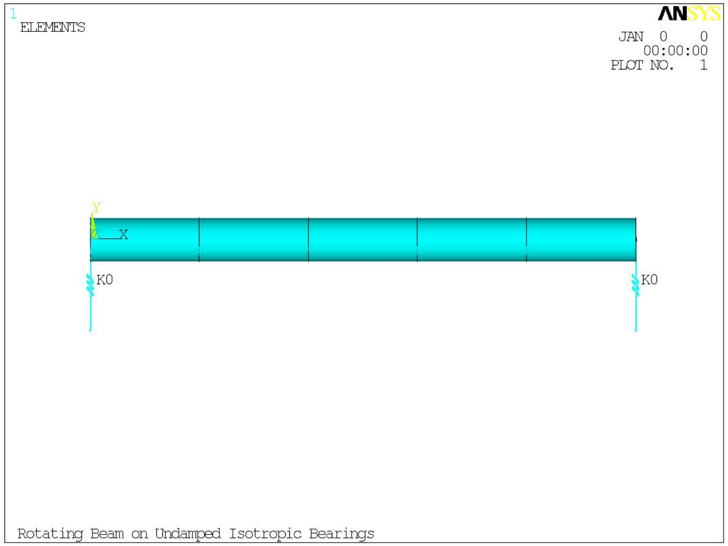
One complex frequency has a positive real part: unstable

Results obtained from a modal analysis with QRDAMP solver

Effect of rotating damping

Rotating damping example

- **Comparison of the dynamics of a simple model with and without rotating damping effect activated:**
  - **Rotating beam**
  - **Isotropic bearings**
  - **Proportional damping**

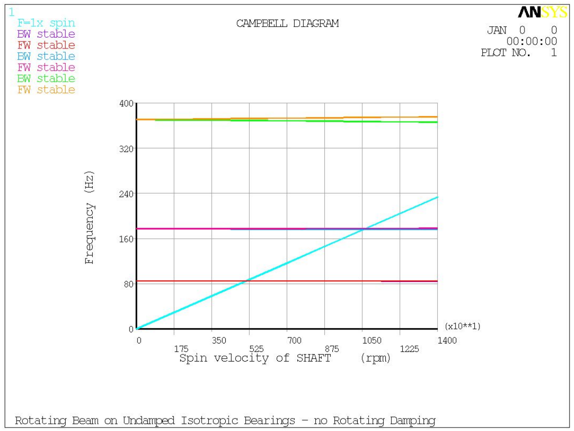


- ***Ref: ANSYS VM 261***
- ***E.S. Zorzi, H.D. Nelson, "Finite element simulation of rotor-bearing systems with internal damping," ASME Journal of Engineering for Power, Vol. 99, 1976, pg 71-76.***

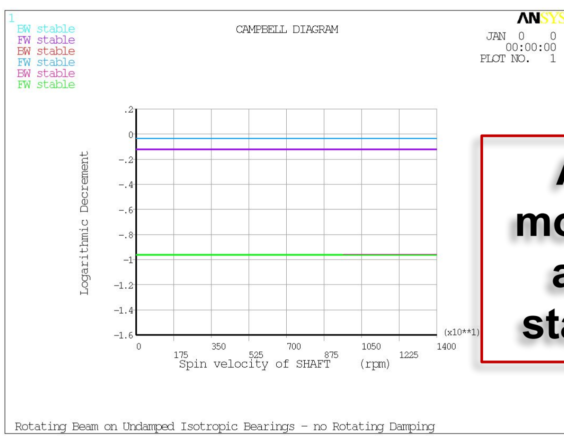
No damping

With damping

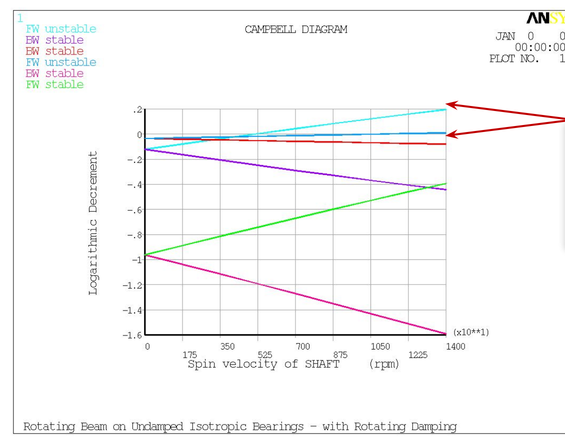
Frequencies



Stability

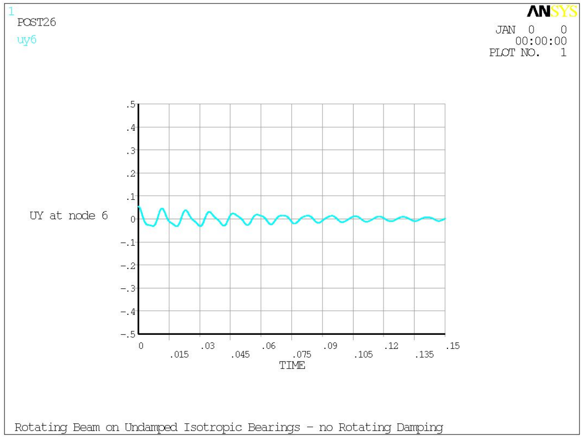


All modes are stable

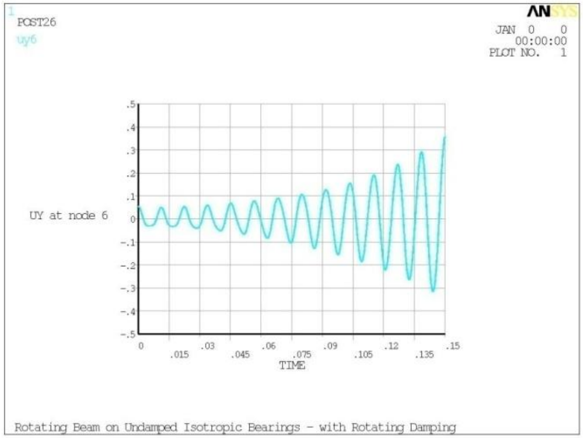


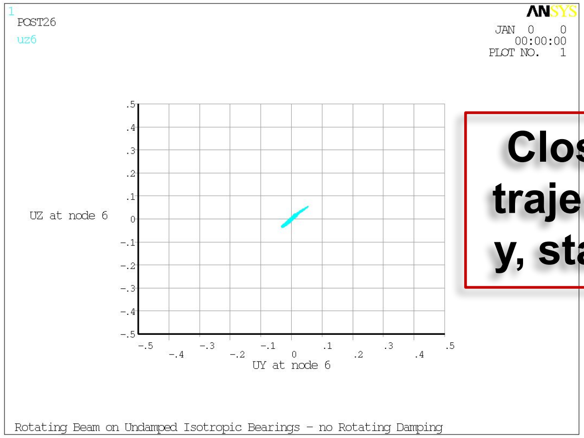
Instable modes

No damping

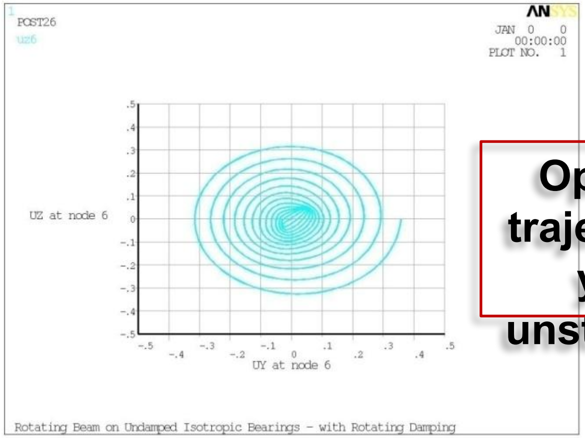


With damping



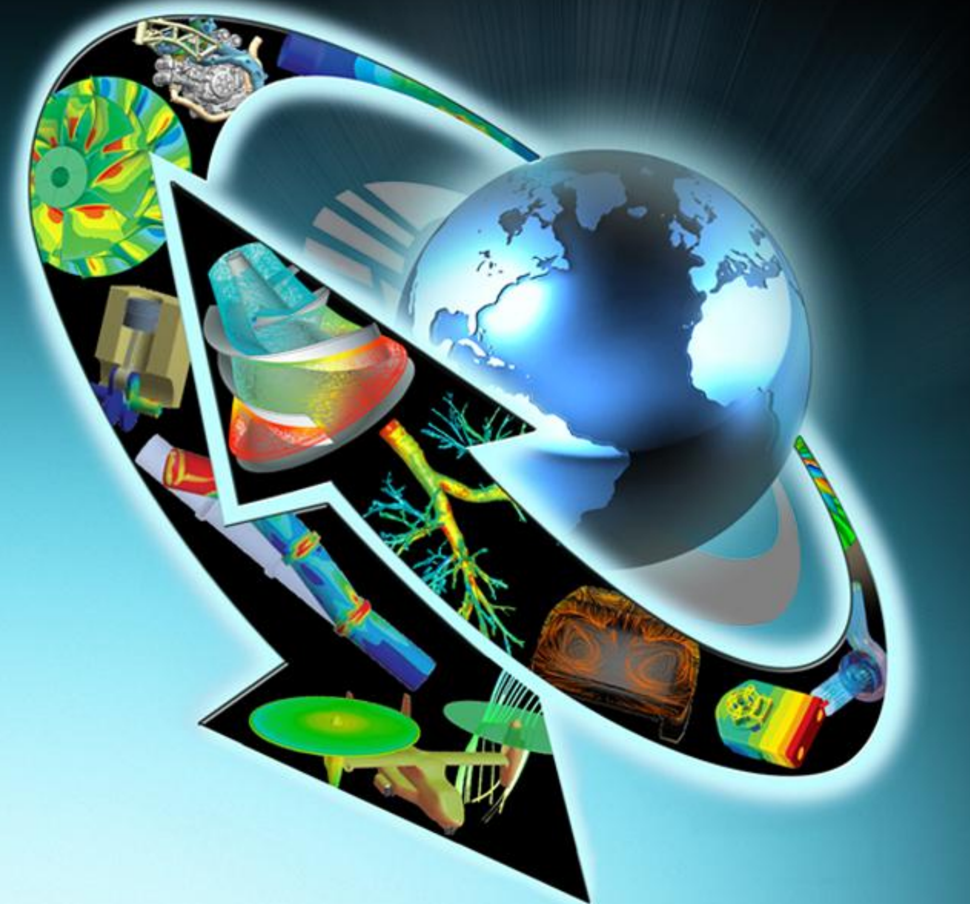


Closed trajector y, stable

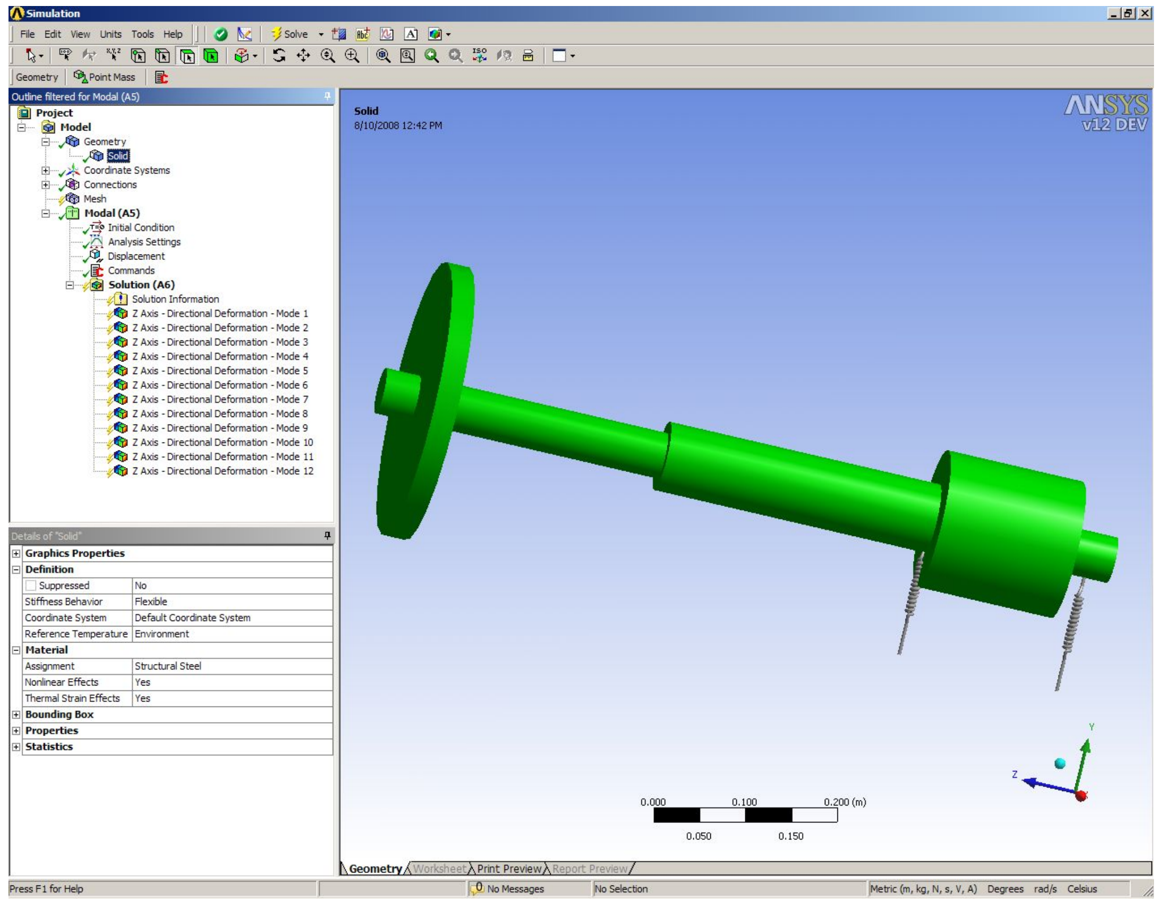


Open trajector y, unstable

Rotordynamics with ANSYS Workbench



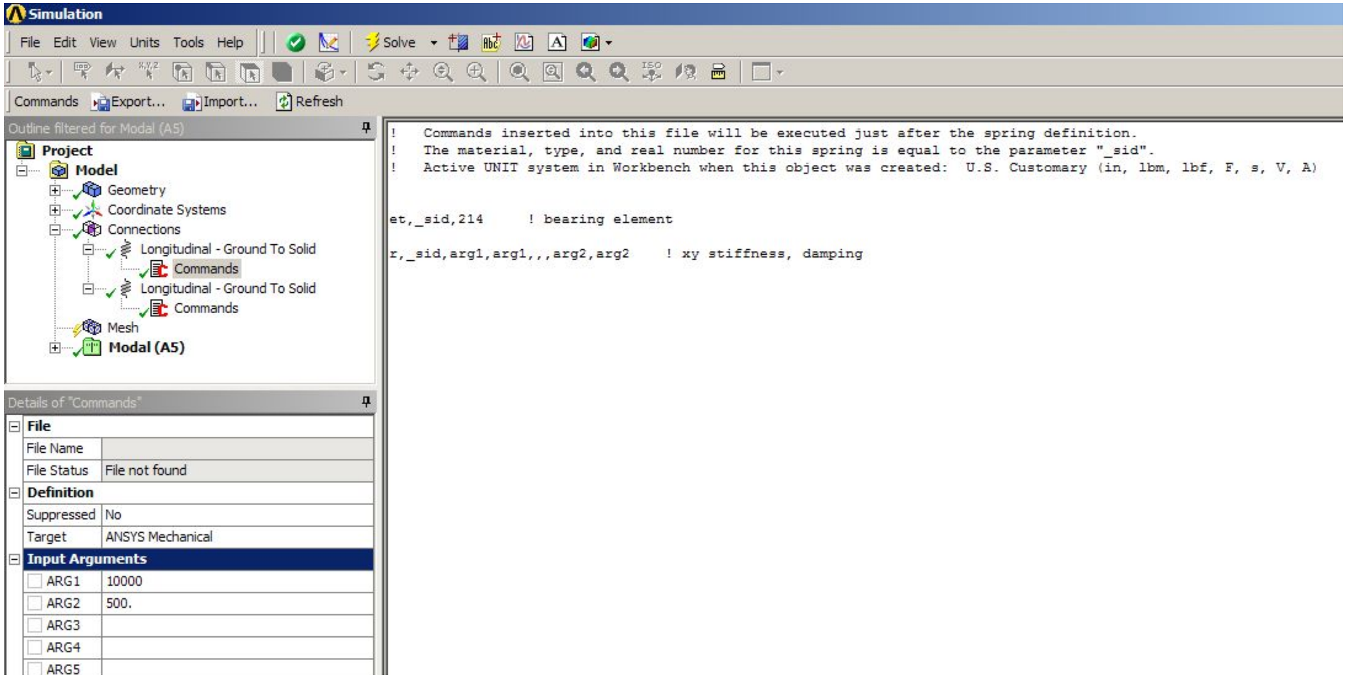
- The database contains a generic steel rotor created in ANSYS DesignModeler to which two “Springs to Ground” have been added.



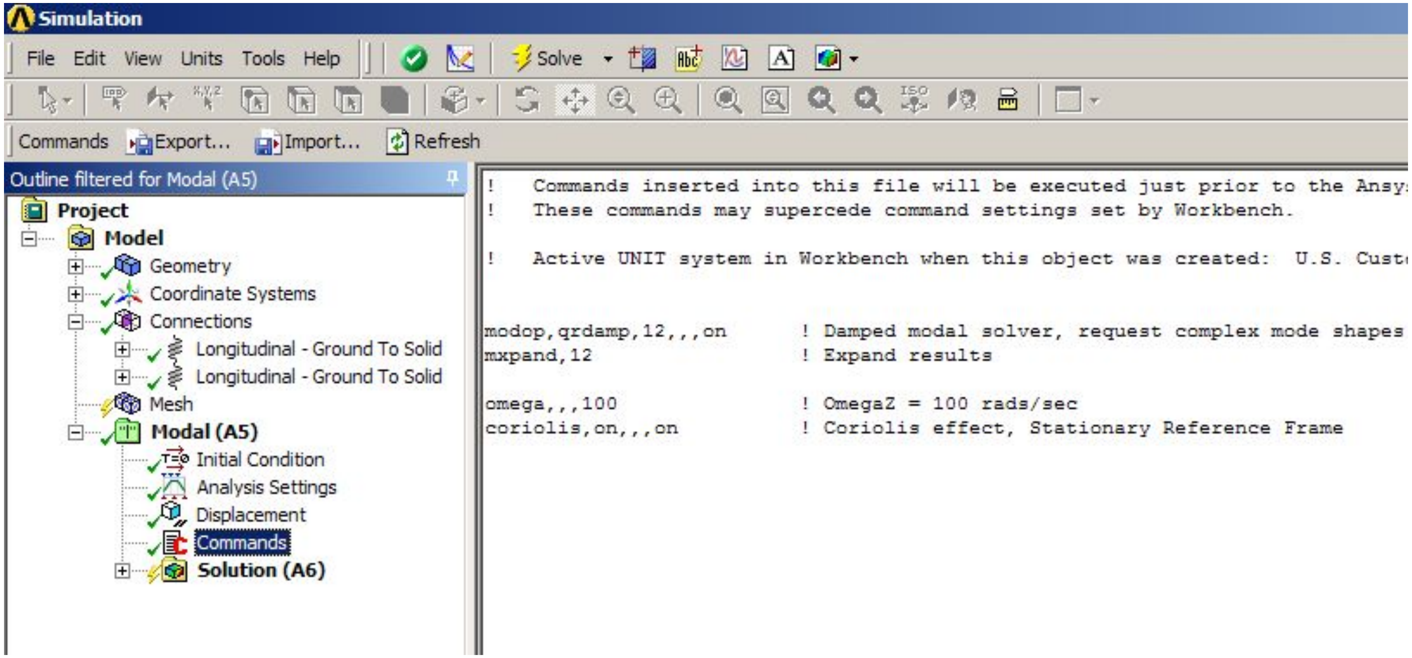
Bearing definition

- **The standard Simulation springs are changed to bearing elements utilizing the parameter, _sid to change the spring element types to 214.**

- **The stiffness and damping values are defined with the input argument values shown in the Details window.**



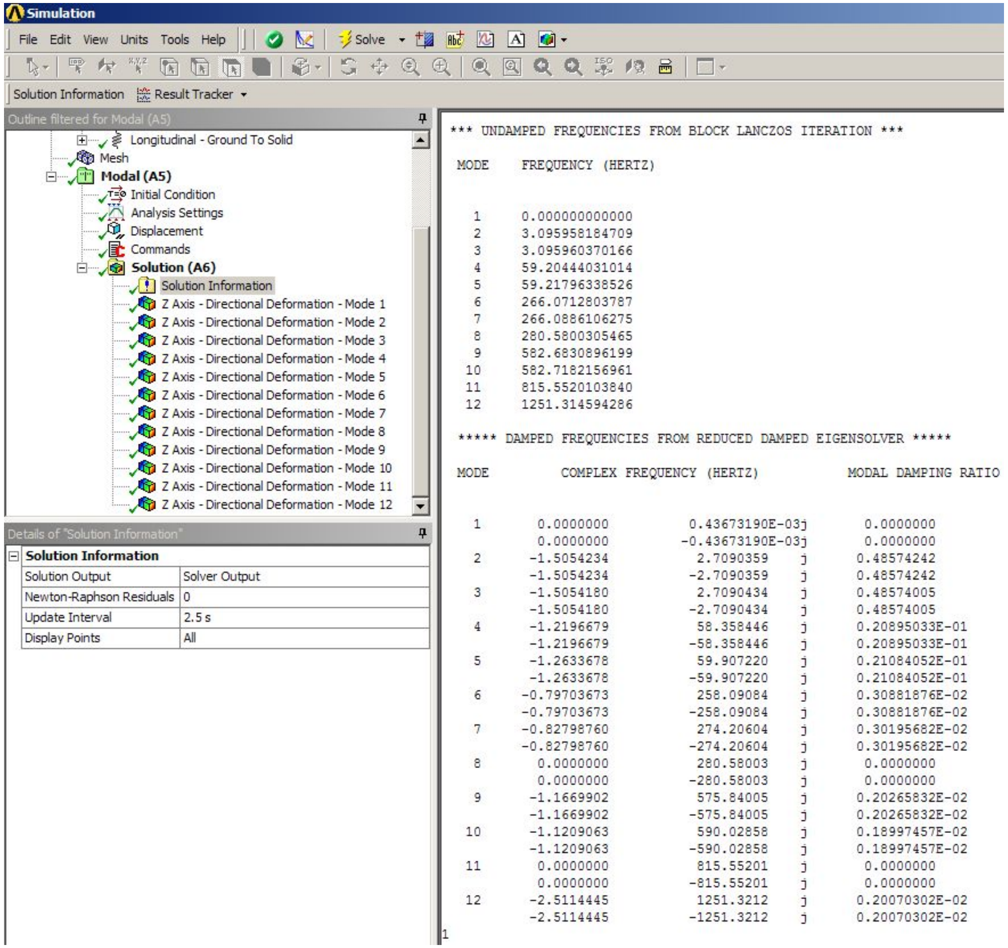
Solution settings for modal analysis



- **A commands object inserted into the analysis branch switches the default modal solver to QRDAMP and requests complex mode shapes.**
- **A spin rate of 100 radians per sec. is specified about the z axis and coriolis effects in the stationary reference frame are requested.**

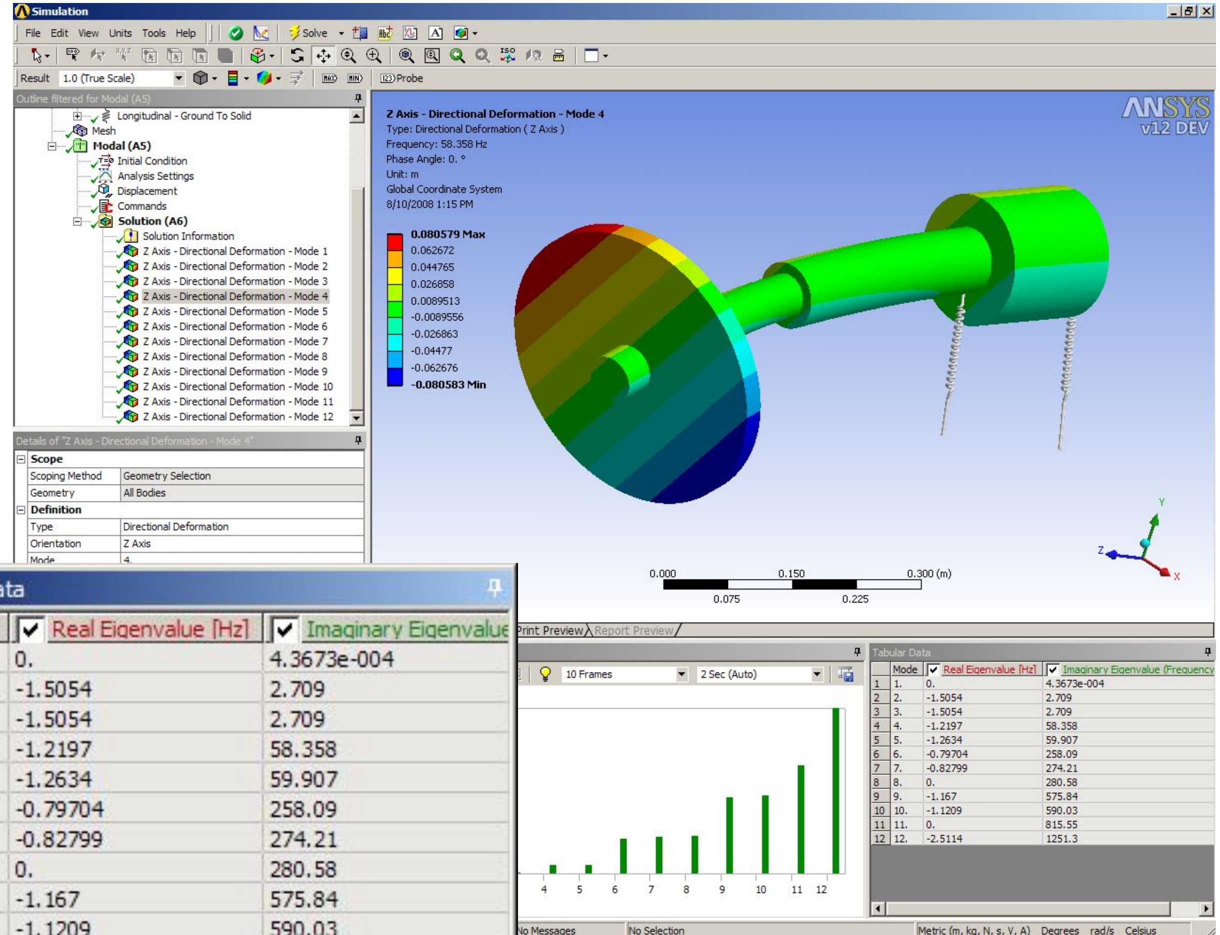
Solution information

- **While the solution is running, the solution output can be monitored.**
- **The output shown is the undamped and damped frequencies.**
- **The real component of the complex frequency is the stability number, the exponent in the expression for damped free vibration.**
- **A negative number indicates the mode is stable.**



- Complex modal results are shown in the tabular view of the results.

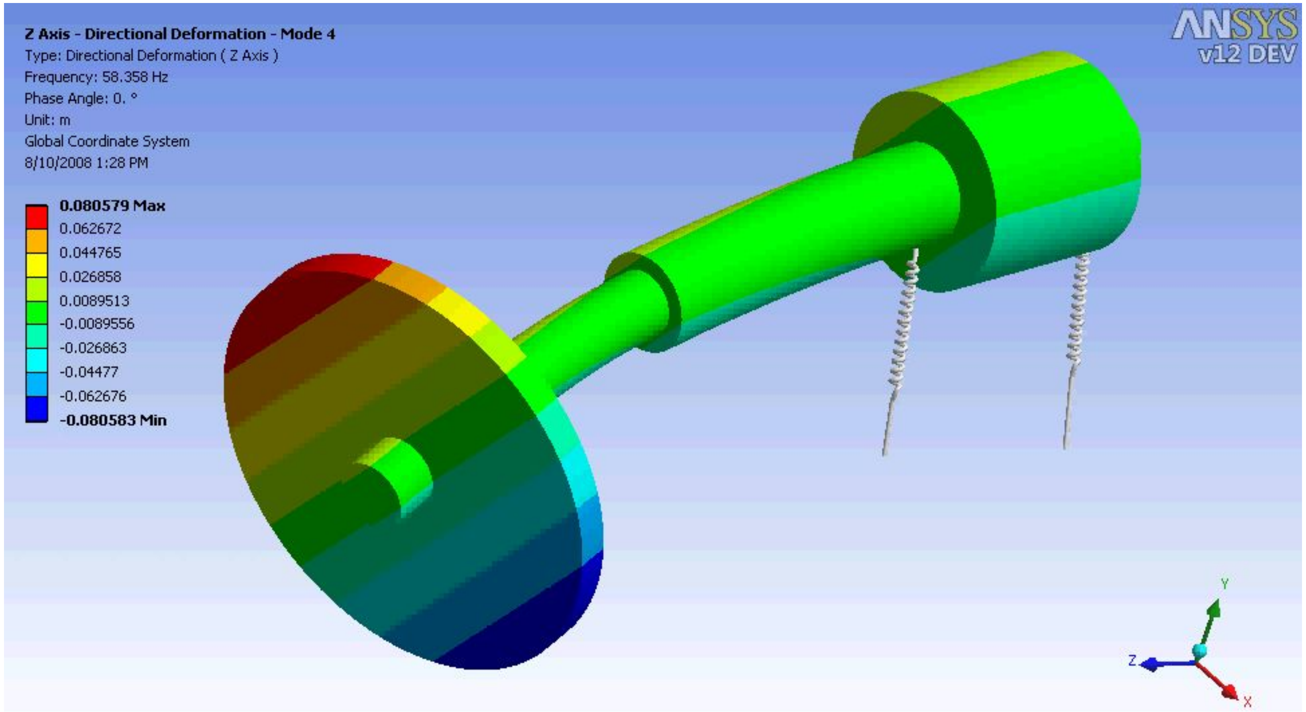
- Complex eigenshapes can be animated.



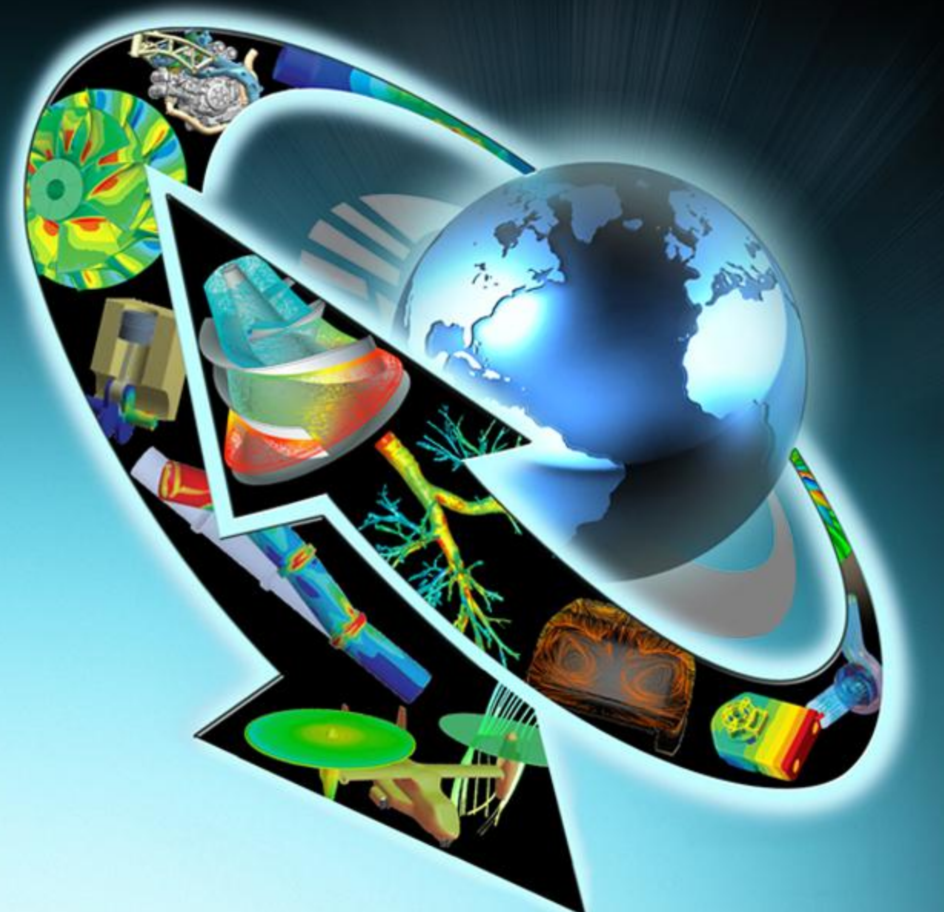
Tabular Data

	Mode	Real Eigenvalue [Hz]	Imaginary Eigenvalue
1	1.	0.	4.3673e-004
2	2.	-1.5054	2.709
3	3.	-1.5054	2.709
4	4.	-1.2197	58.358
5	5.	-1.2634	59.907
6	6.	-0.79704	258.09
7	7.	-0.82799	274.21
8	8.	0.	280.58
9	9.	-1.167	575.84
10	10.	-1.1209	590.03
11	11.	0.	815.55
12	12.	-2.5114	1251.3

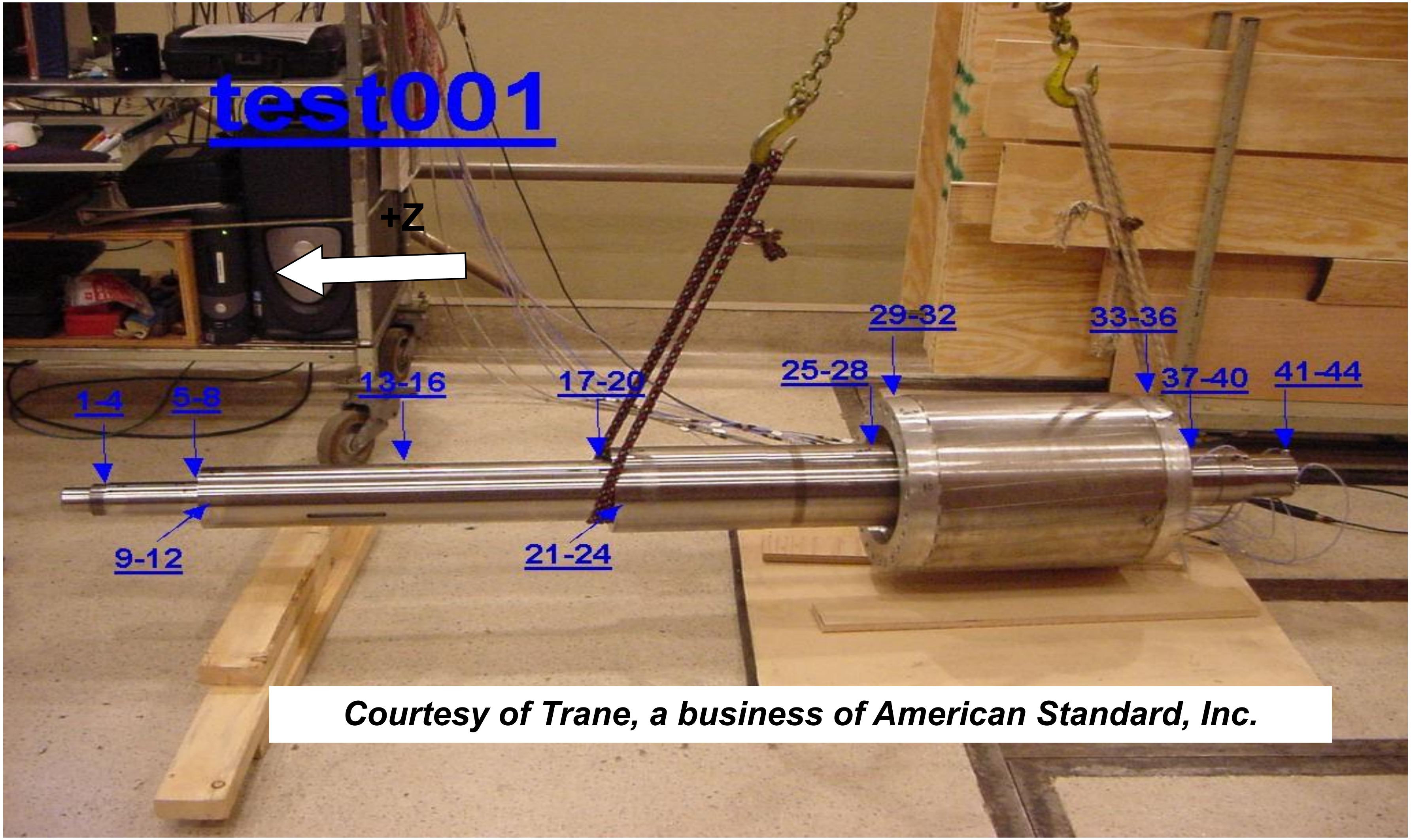
Animated modal shape



Compressor model
Solid model & casing simulation

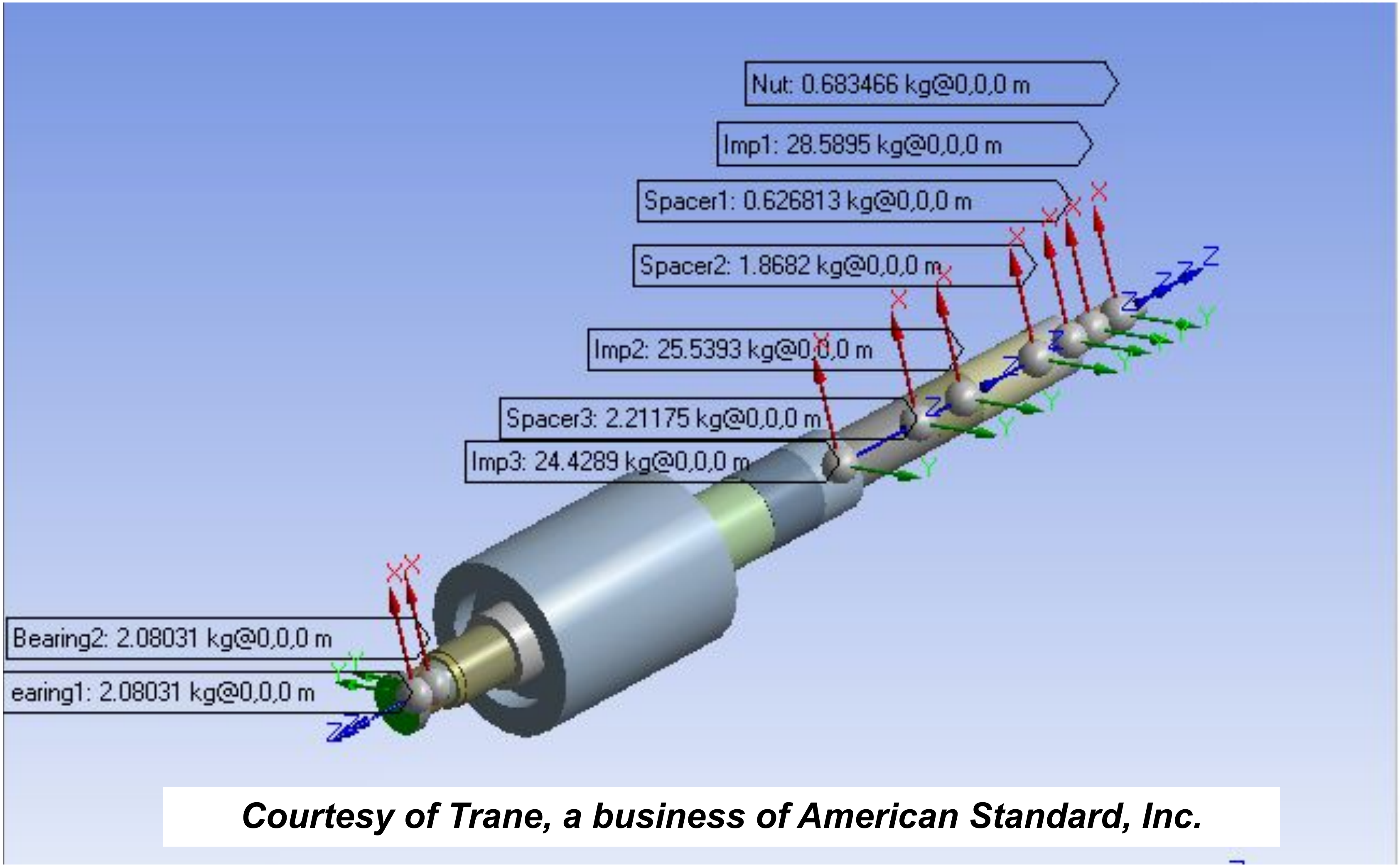


Compressor: free-free testing apparatus used for initial model calibration



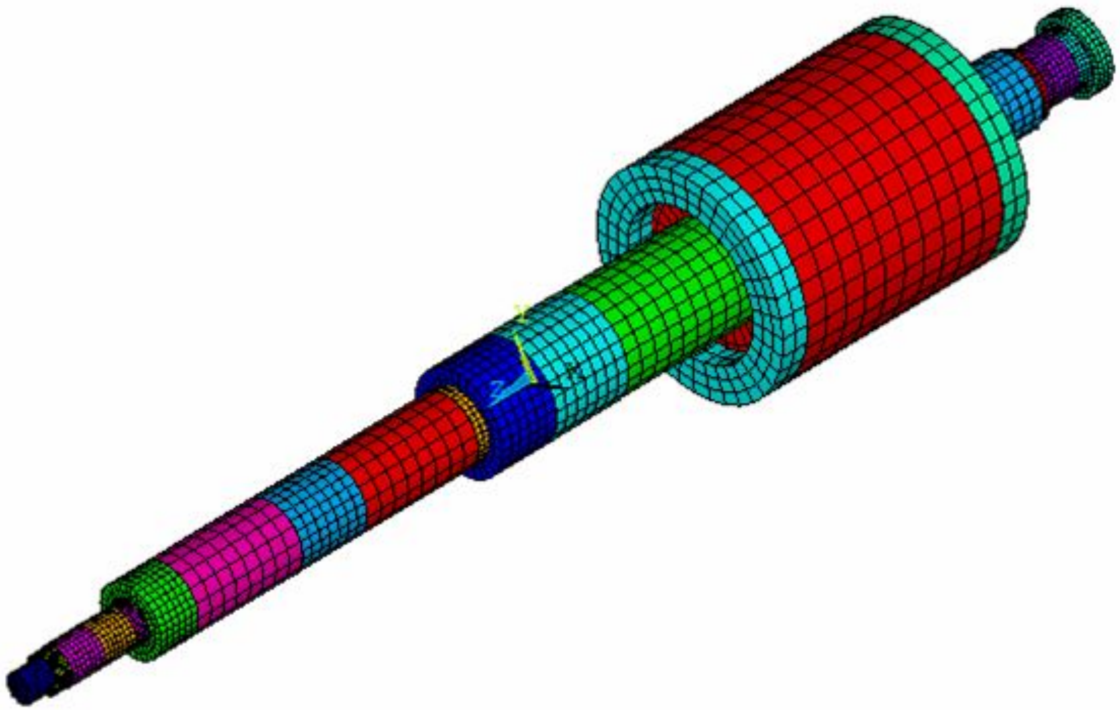
Courtesy of Trane, a business of American Standard, Inc.

Compressor: location of lumped representation of impellers and bearings

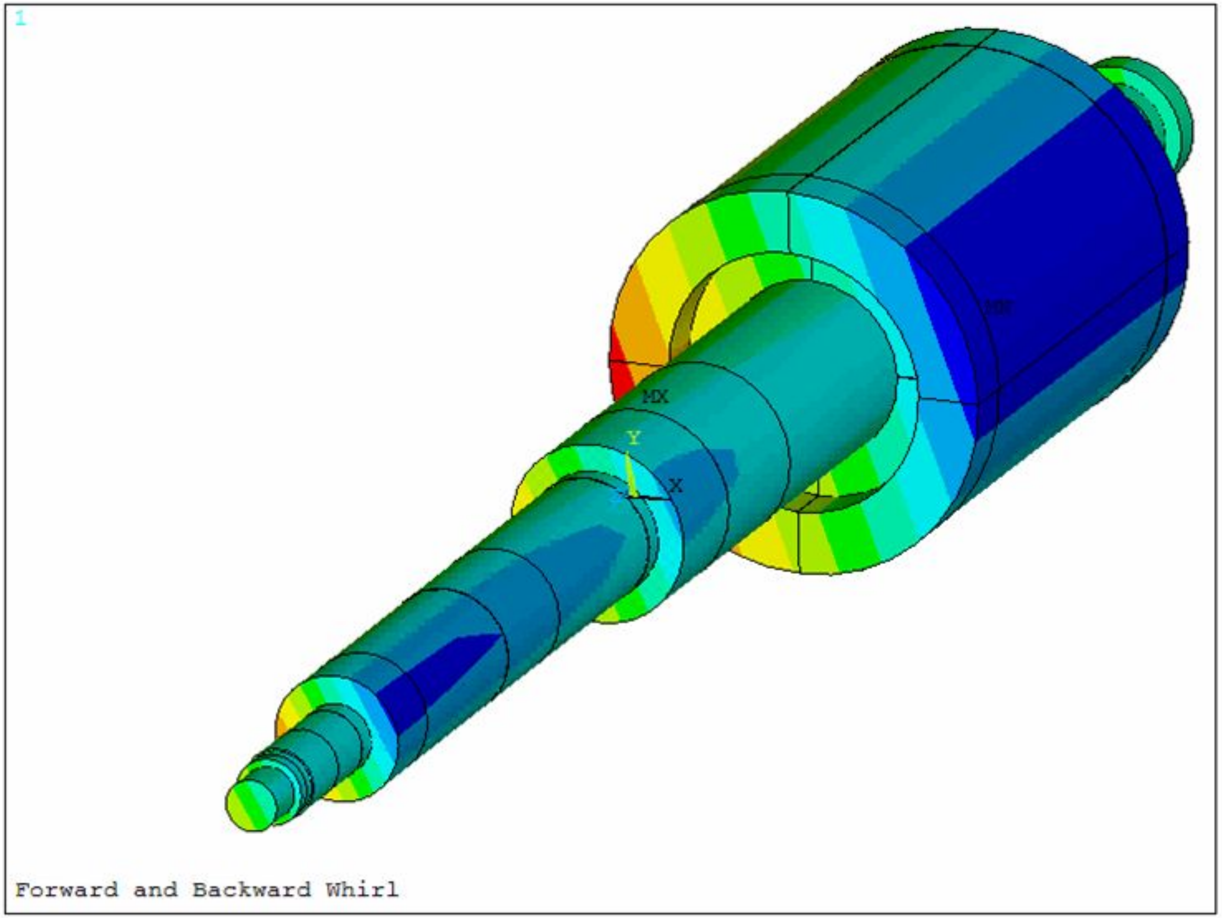


Courtesy of Trane, a business of American Standard, Inc.

Compressor: SOLID185 mesh of shaft

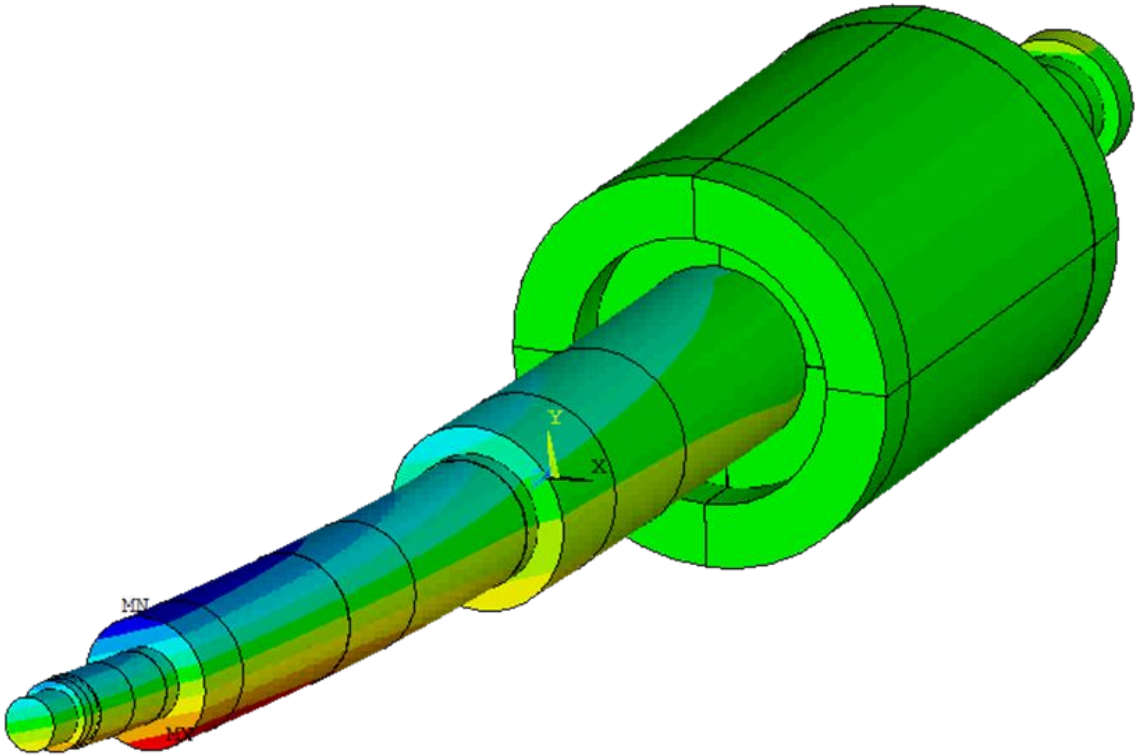


Compressor: forward whirl mode



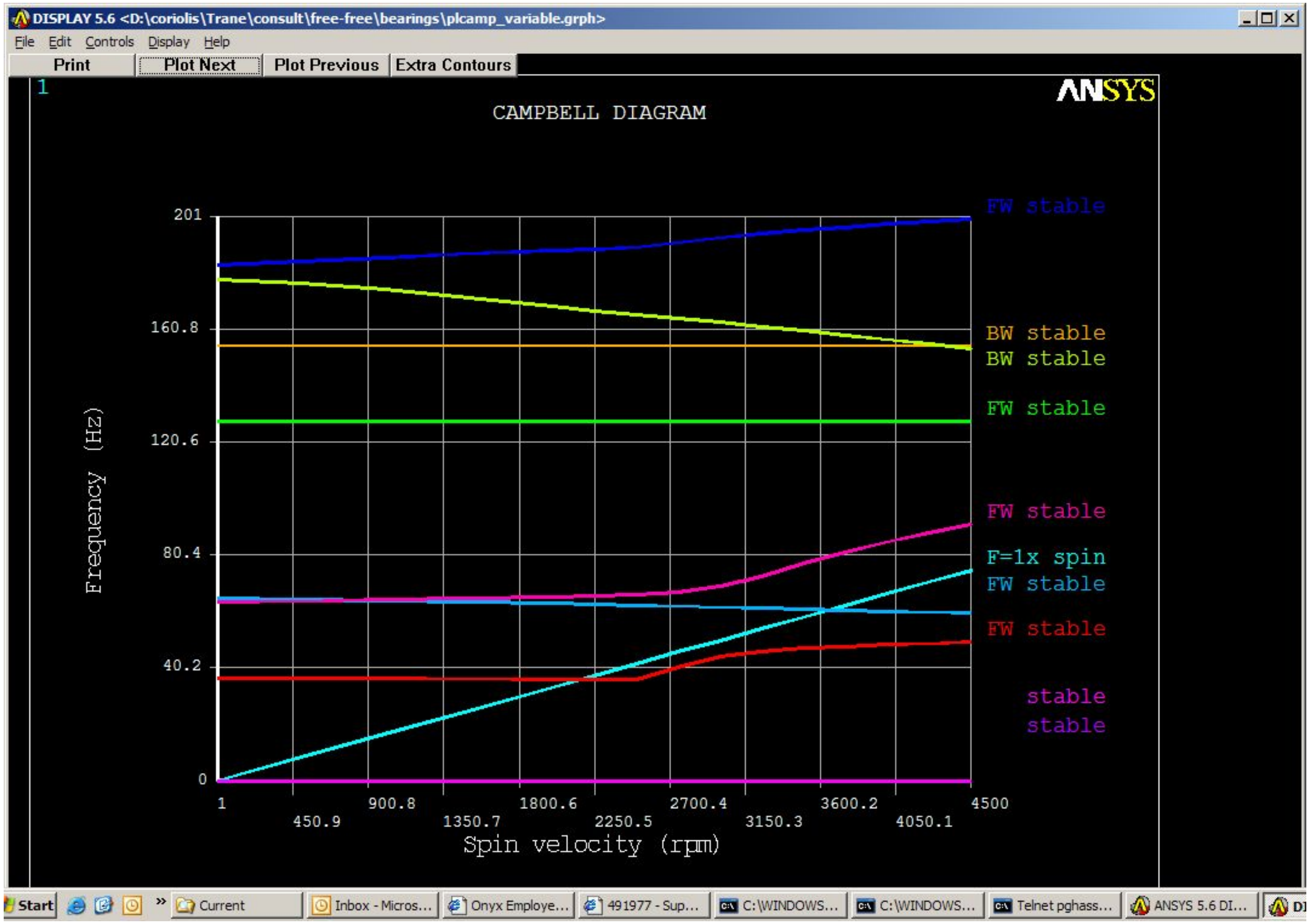
Courtesy of Trane, a business of American Standard, Inc.

Compressor: backward whirl mode

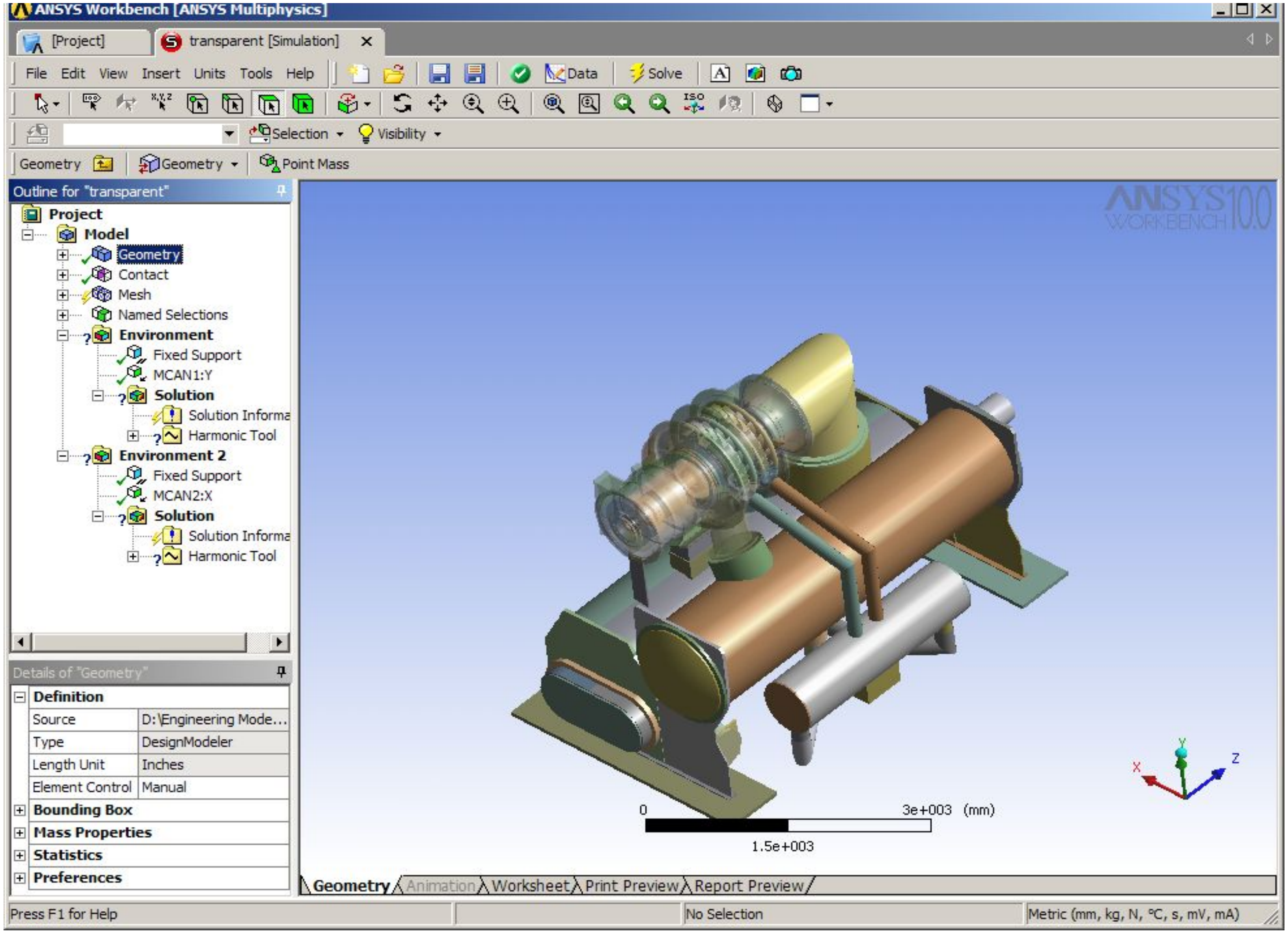


Courtesy of Trane, a business of American Standard, Inc.

Compressor: Campbell diagram with variable bearings

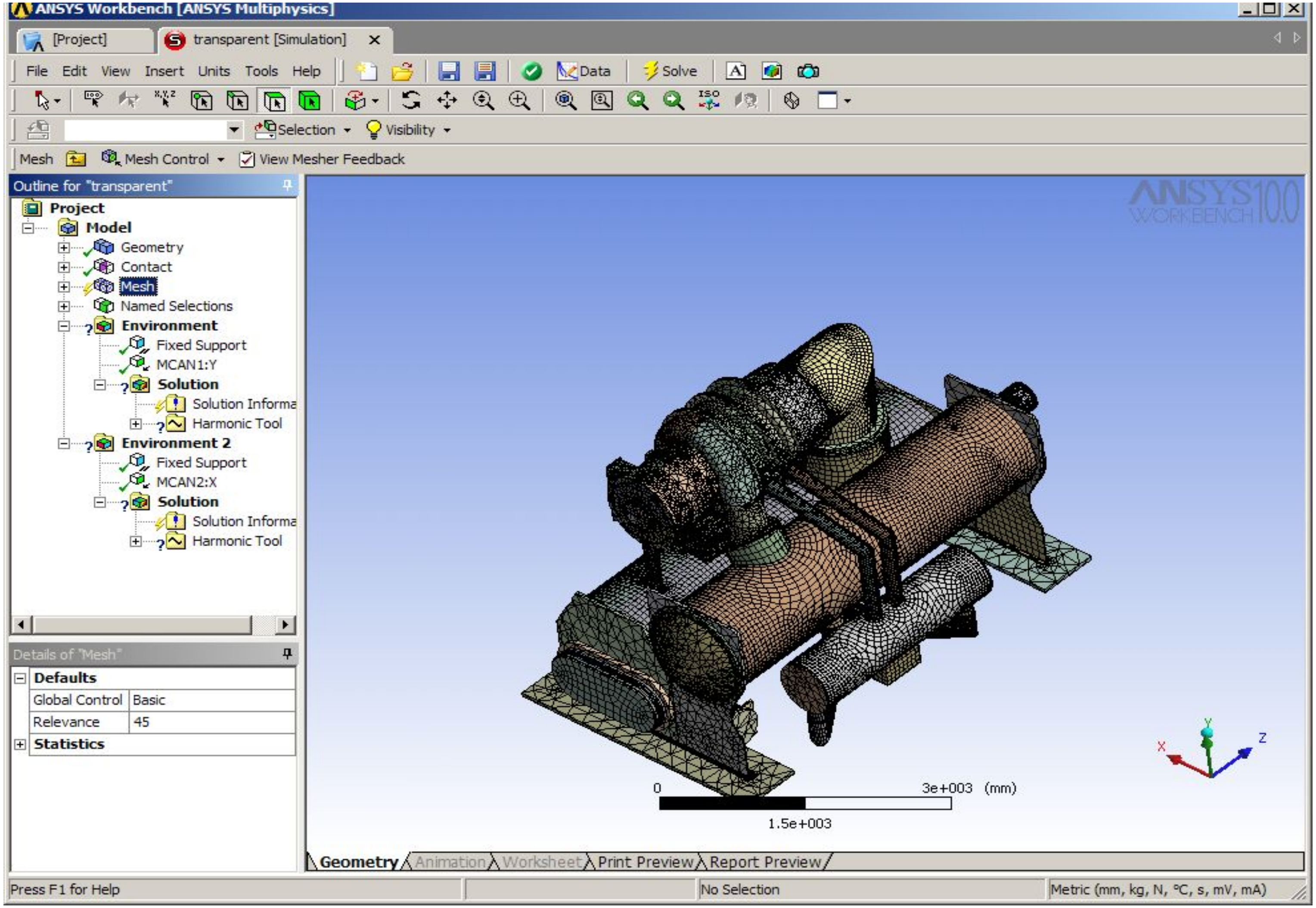


Solid model of rotor with chiller assembly



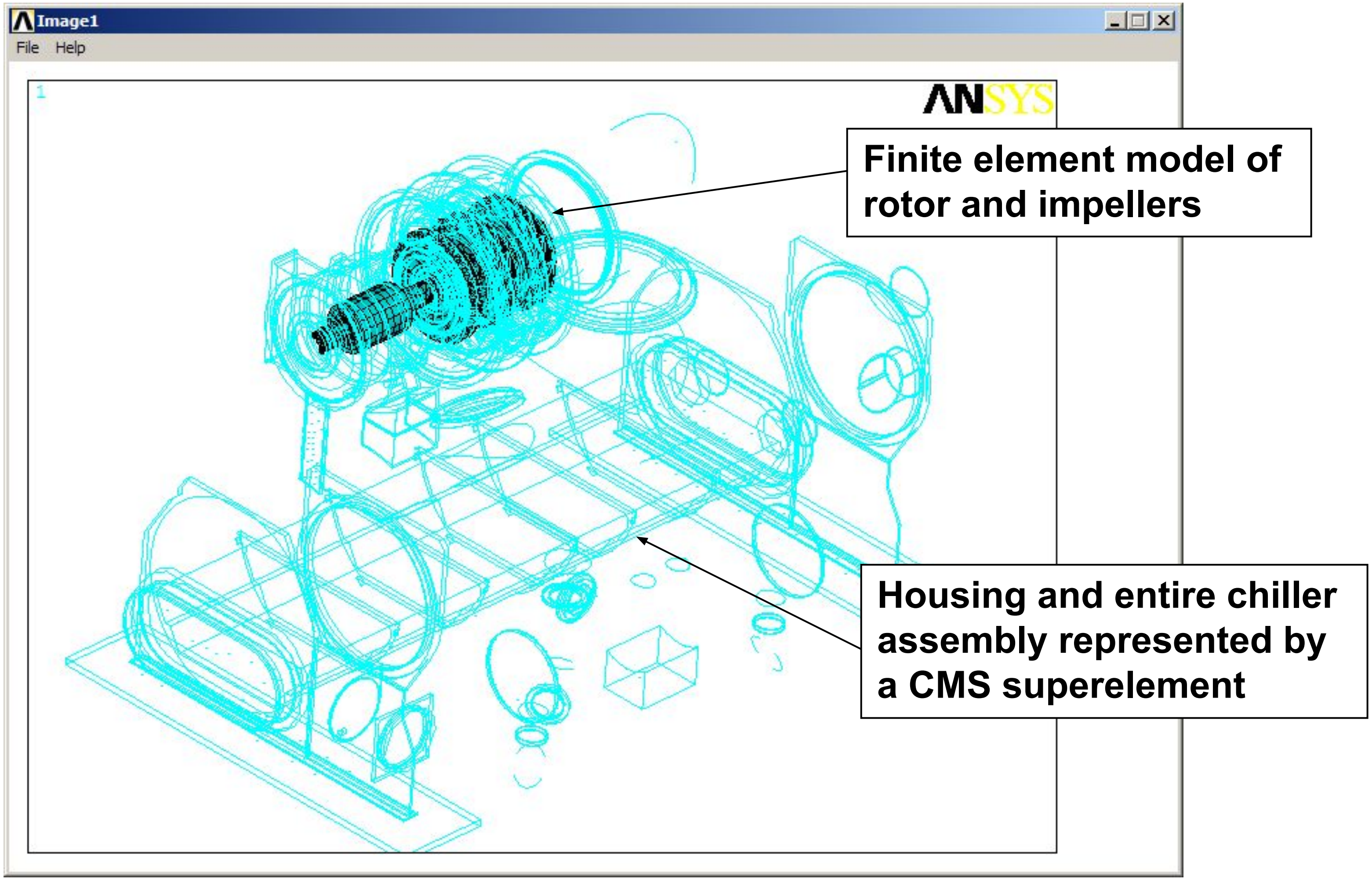
Courtesy of Trane, a business of American Standard, Inc.

Meshed rotor and chiller assembly

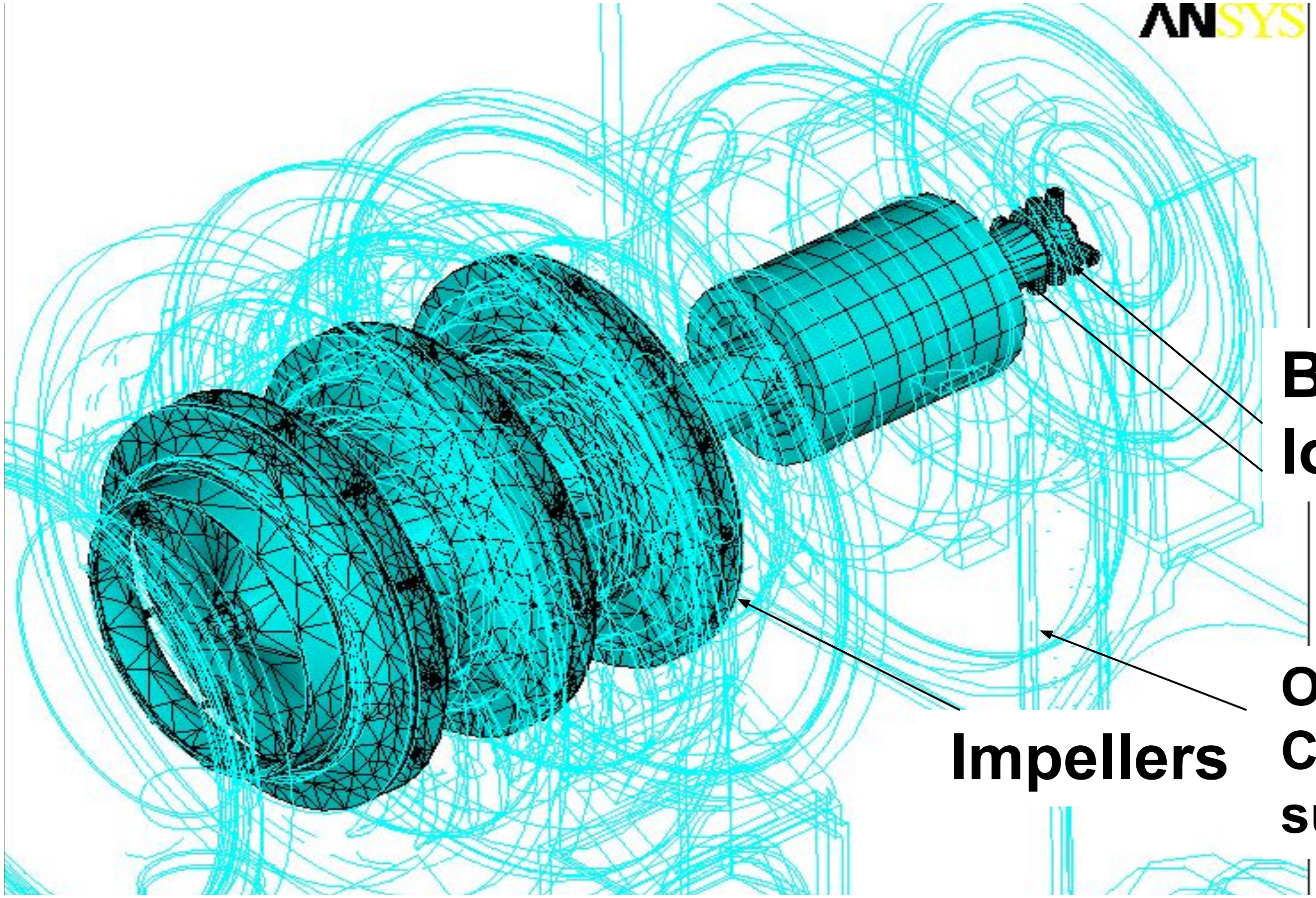


Courtesy of Trane, a business of American Standard, Inc.

Analysis model – supporting structure represented by CMS superelement

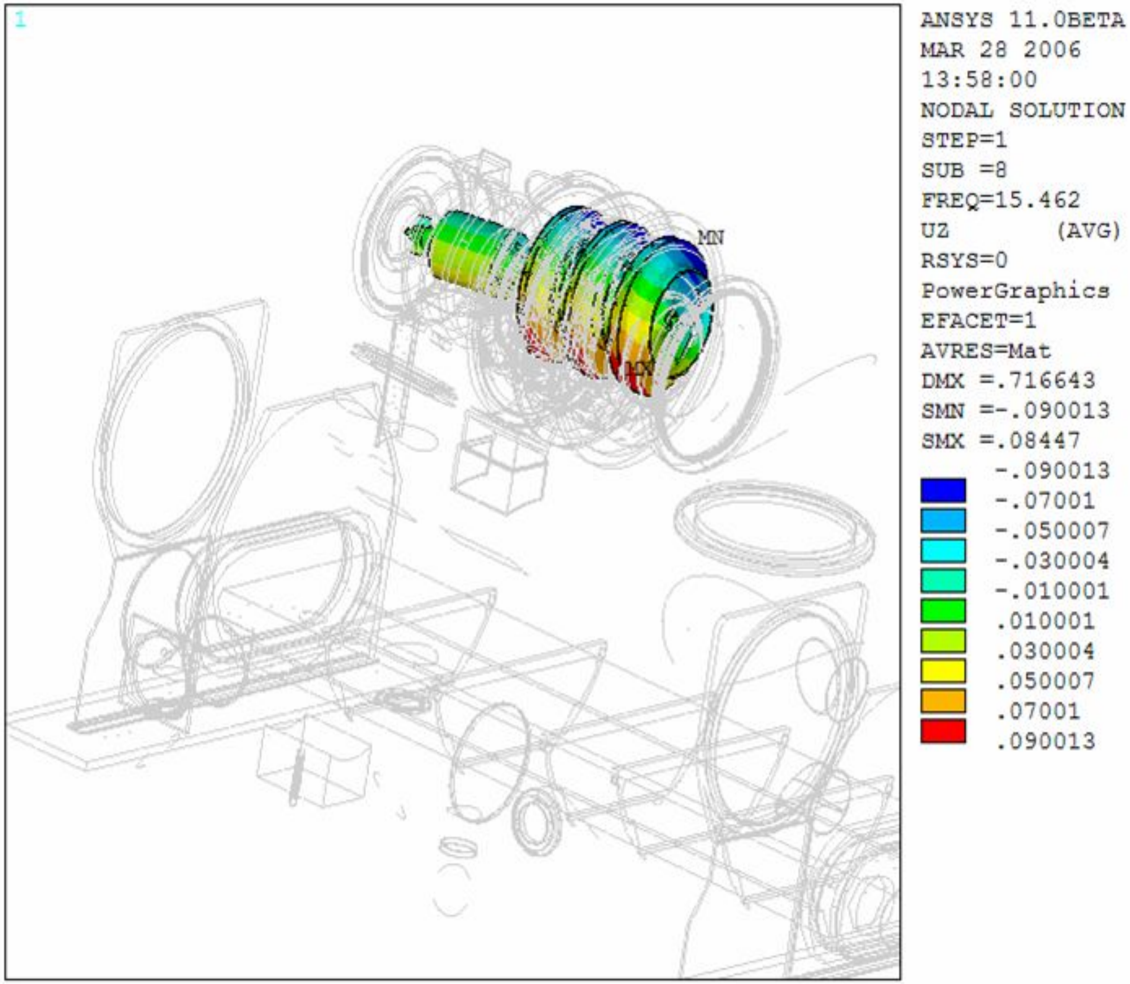


Courtesy of Trane, a business of American Standard, Inc.



Courtesy of Trane, a business of American Standard, Inc.

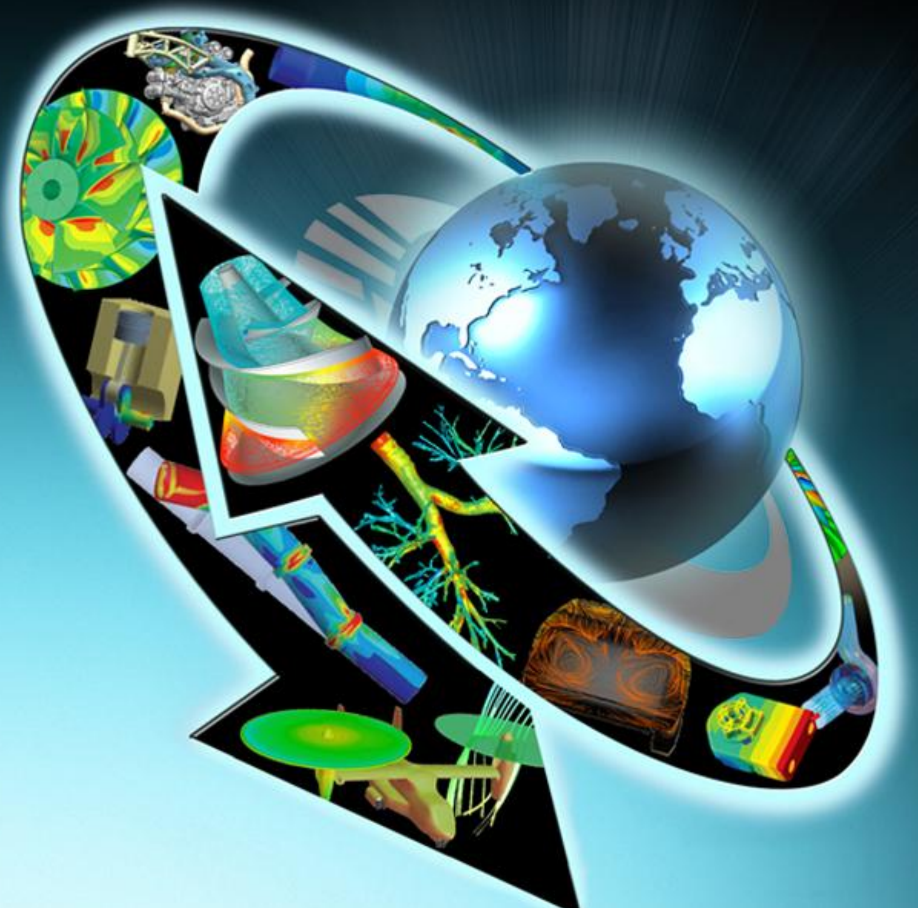
Typical mode animation



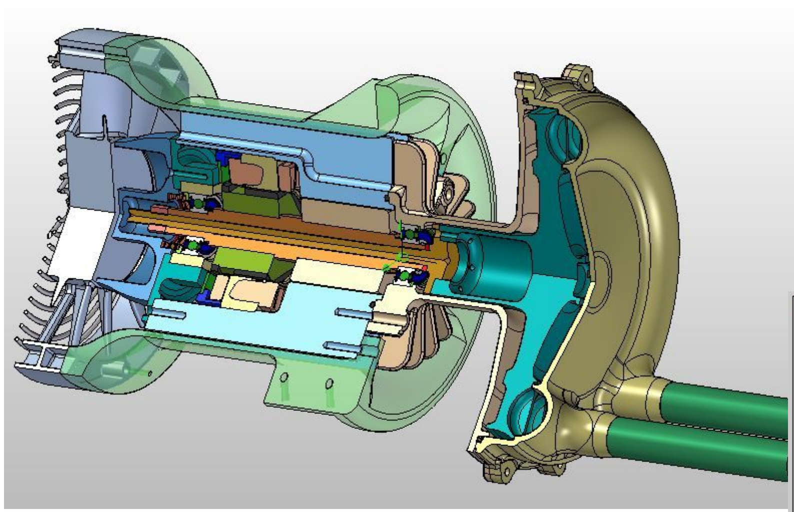
Courtesy of Trane, a business of American Standard, Inc.

Blower shaft model

Transient startup & effect of prestress



Blower shaft - model

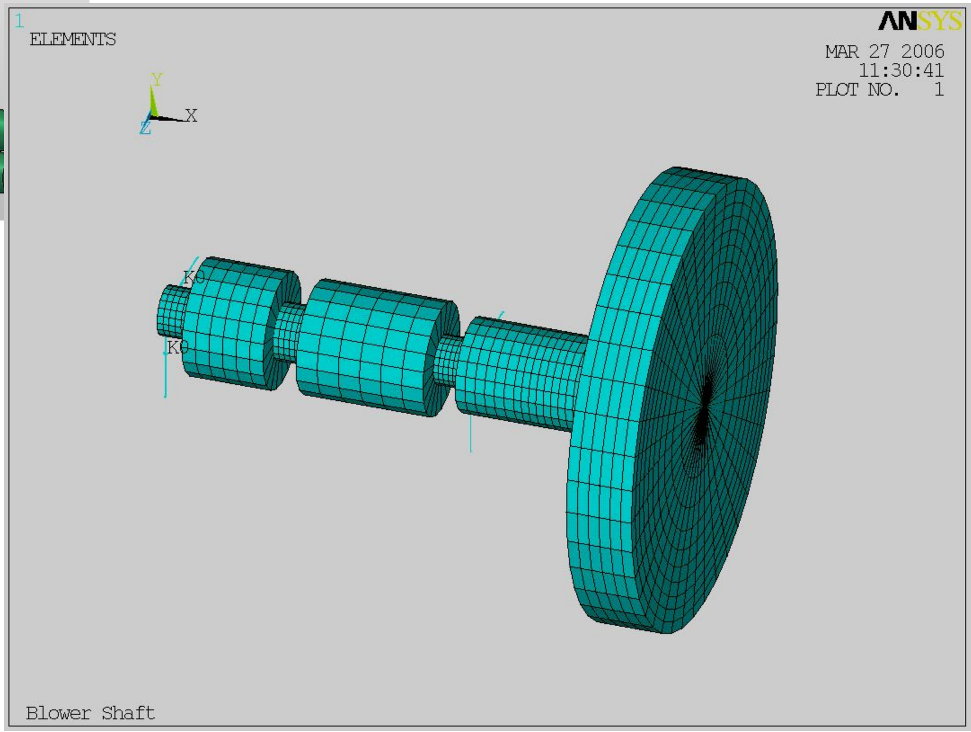


Impeller to pump hot hydrogen rich mix of gas and liquid into solid oxyde fluid cell

Spin 10,000 rpm

ANSYS model of rotating part

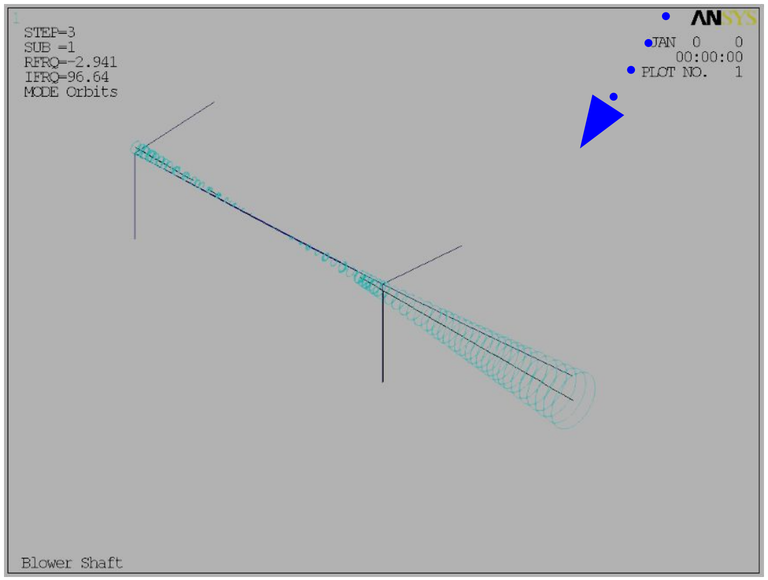
99 beam elements & 2 bearing elements

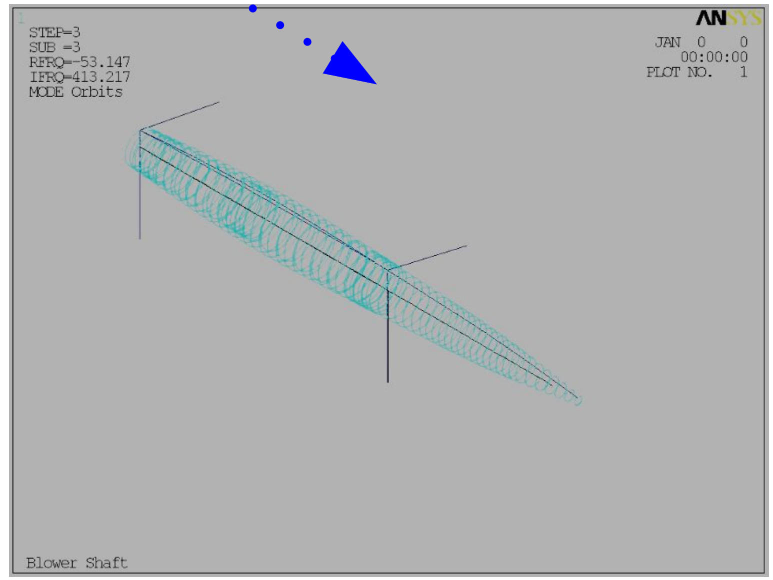


Frequencies and corresponding mode shapes orbits

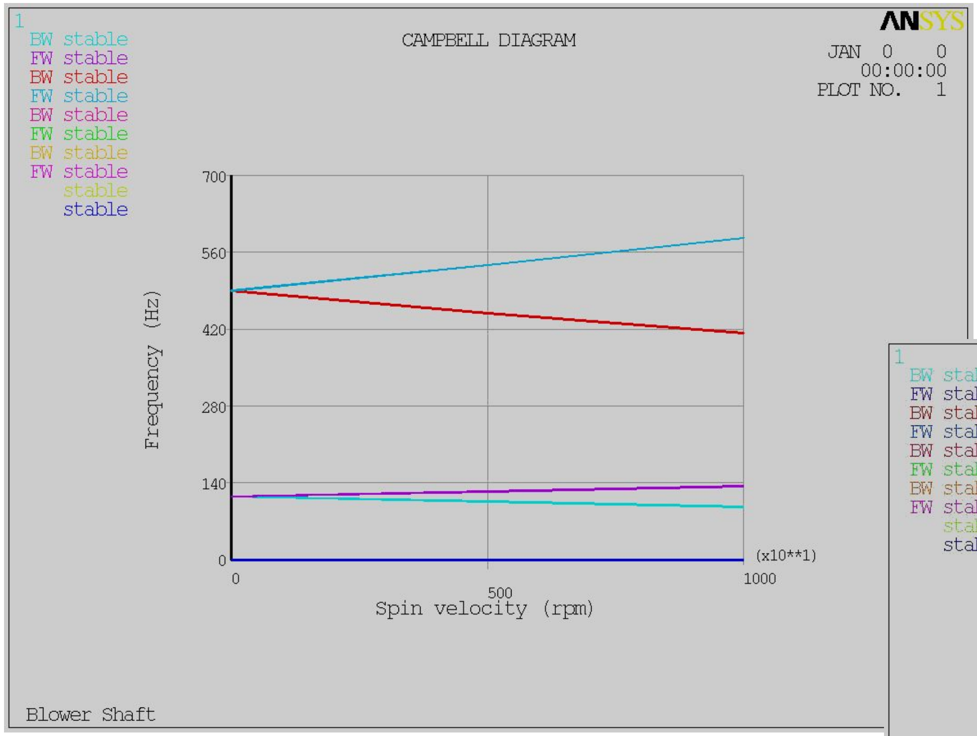
***** FREQUENCIES (Hz) FROM CAMPBELL (sorting on) *****

Spin(rpm)		0.000	5000.000	10000.000
1.00xSpin		0.000	83.333	166.667
1	BW	115.552	105.999	96.640
2	FW	115.552	124.949	133.875
3	BW	490.534	448.773	413.217
4	FW	490.534	537.184	586.075



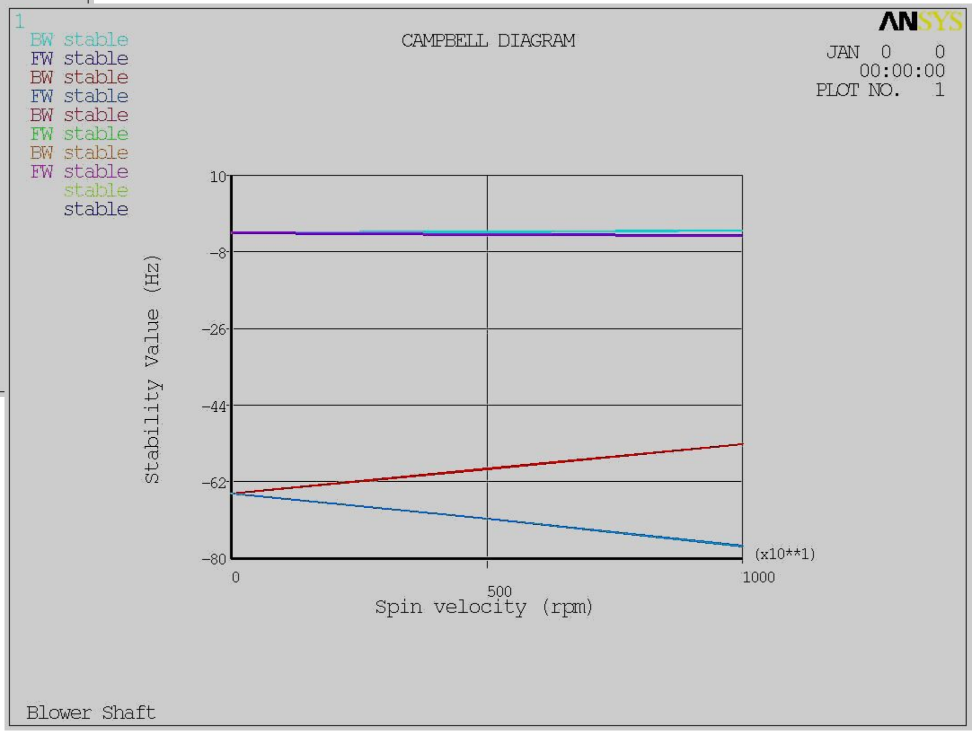


Blower shaft – modal analysis



Frequencies

Stability

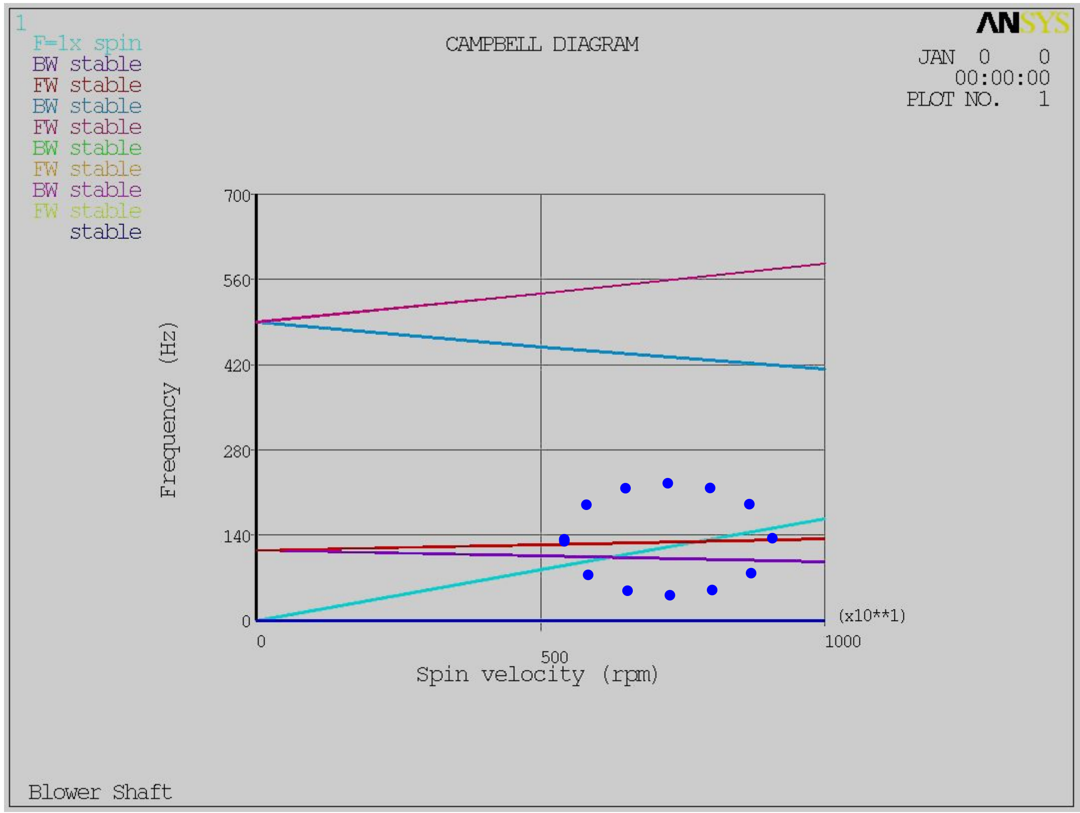


Blower shaft – critical speed

***** CRITICAL SPEEDS (rpm) FROM CAMPBELL (sorting on) *****

Slope of line : 1.000

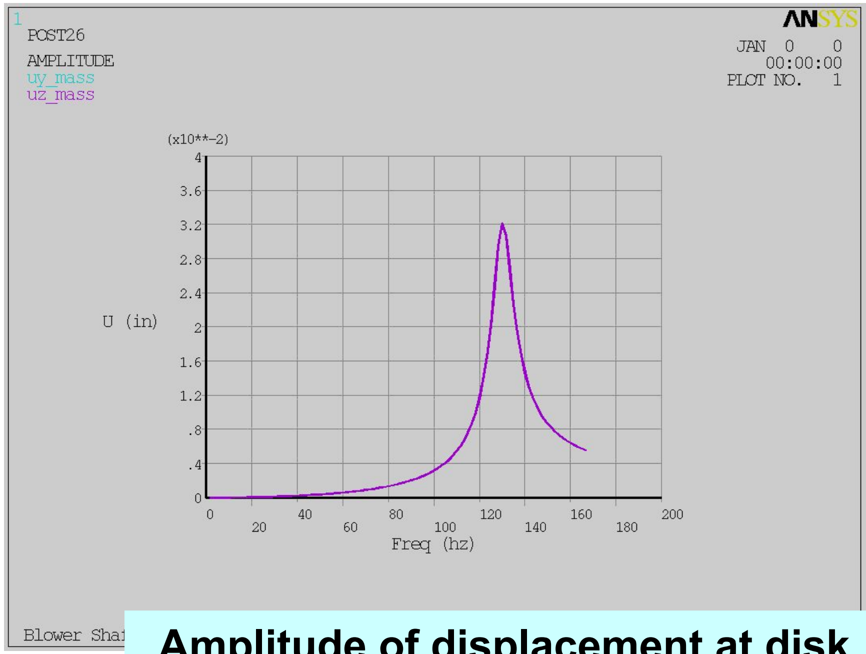
1	6222.614
2	7796.469
3	none
4	none



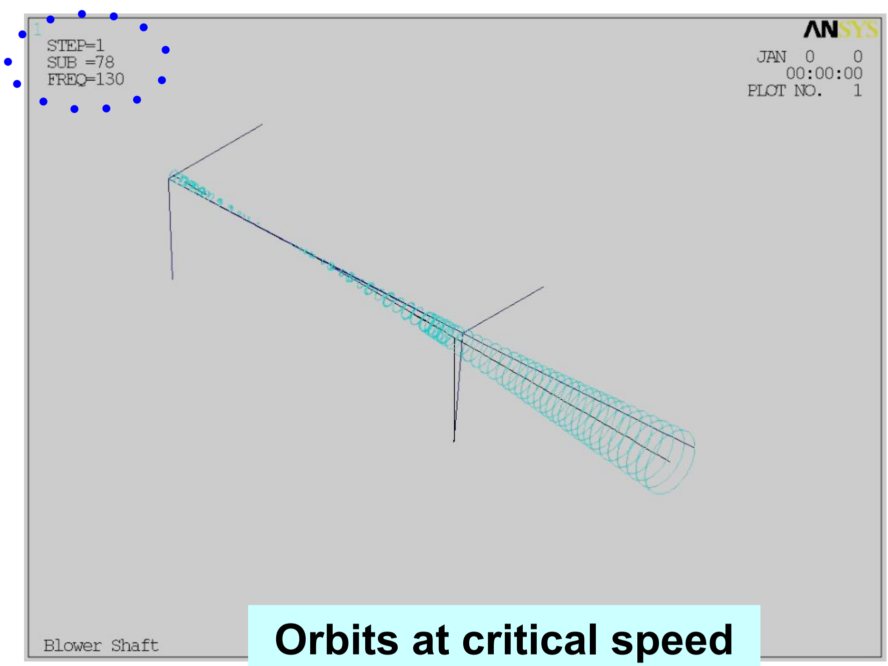
Blower shaft – unbalance response

Harmonic response to disk unbalance

- Disk eccentricity is .002”
- Disk mass is .0276 lbf-s2/in.
- Sweep frequencies 0-10000 rpm

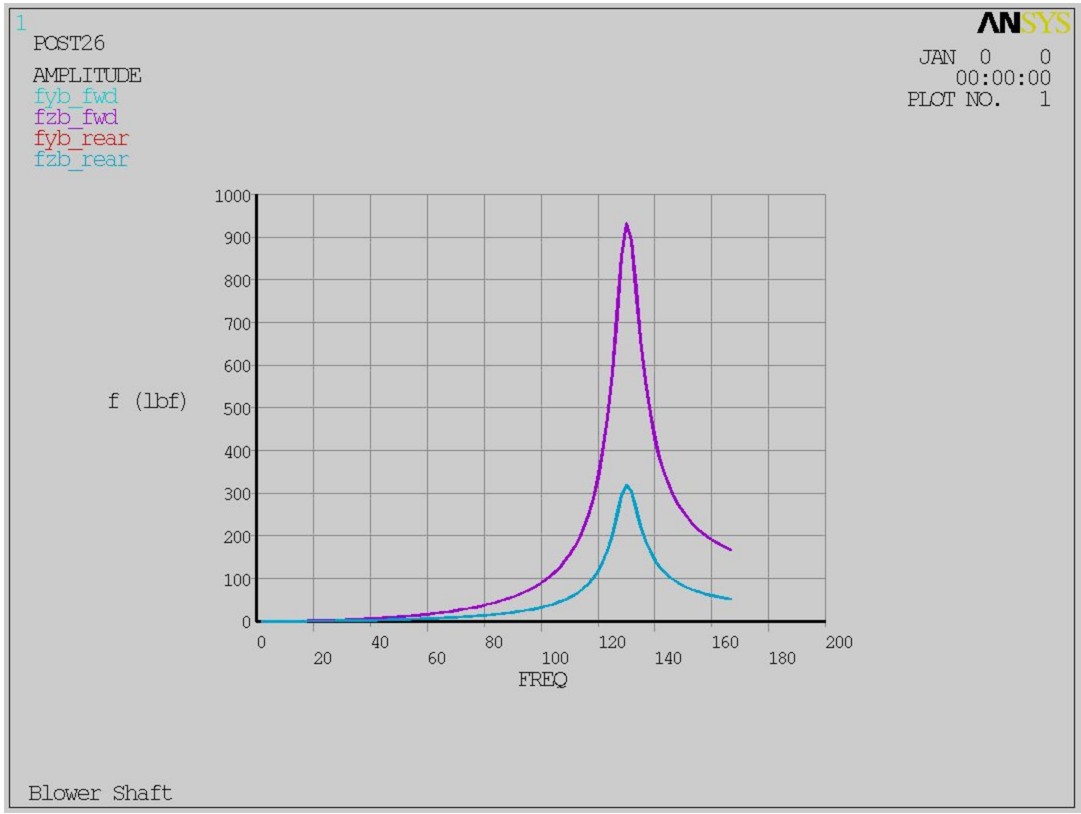


Amplitude of displacement at disk



Orbits at critical speed

Bearings reactions

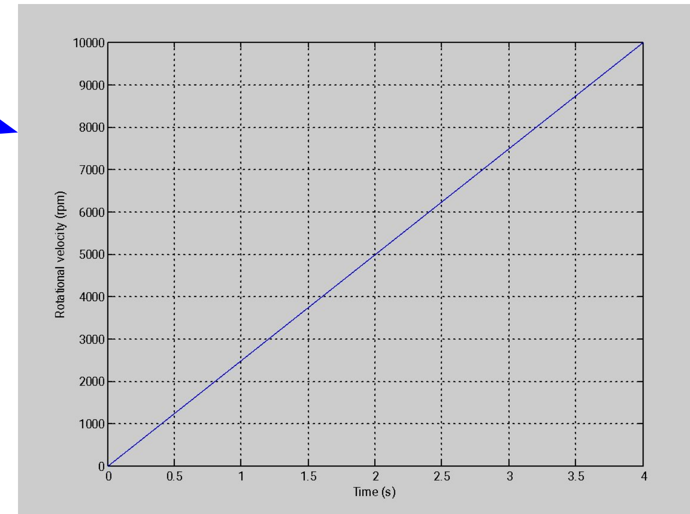


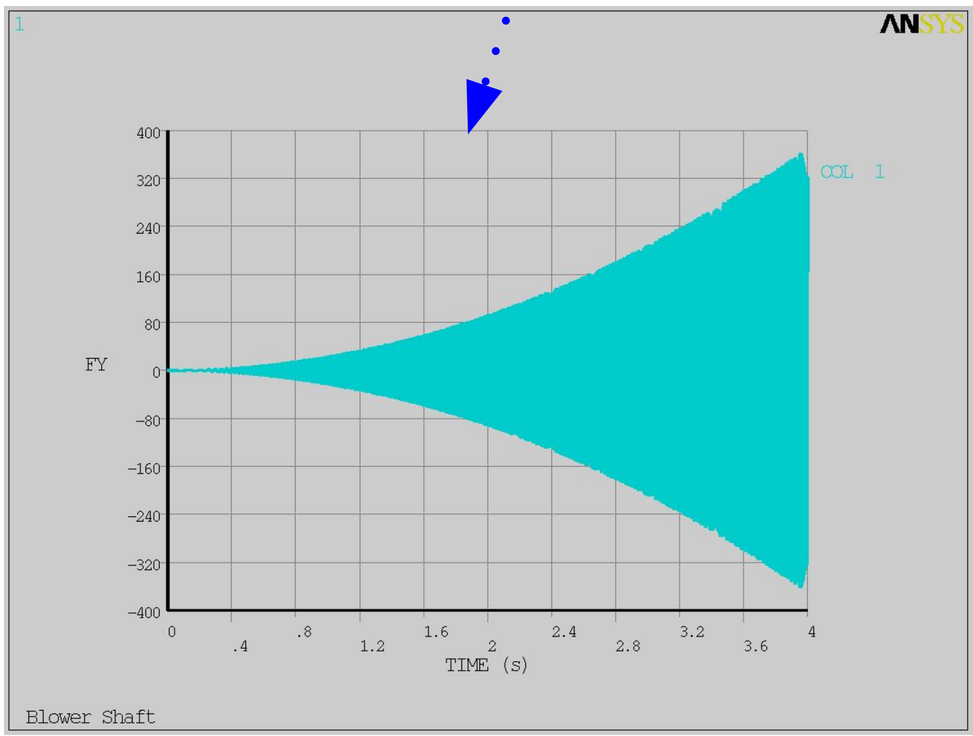
Forward bearing is more loaded than rear one as first mode is a disk mode.

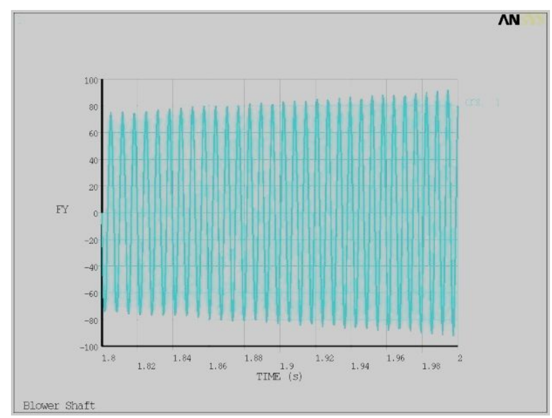
Blower shaft – start up

Transient analysis

- Ramped rotational velocity over 4 seconds
- Unbalance transient forces FY and FZ at disk



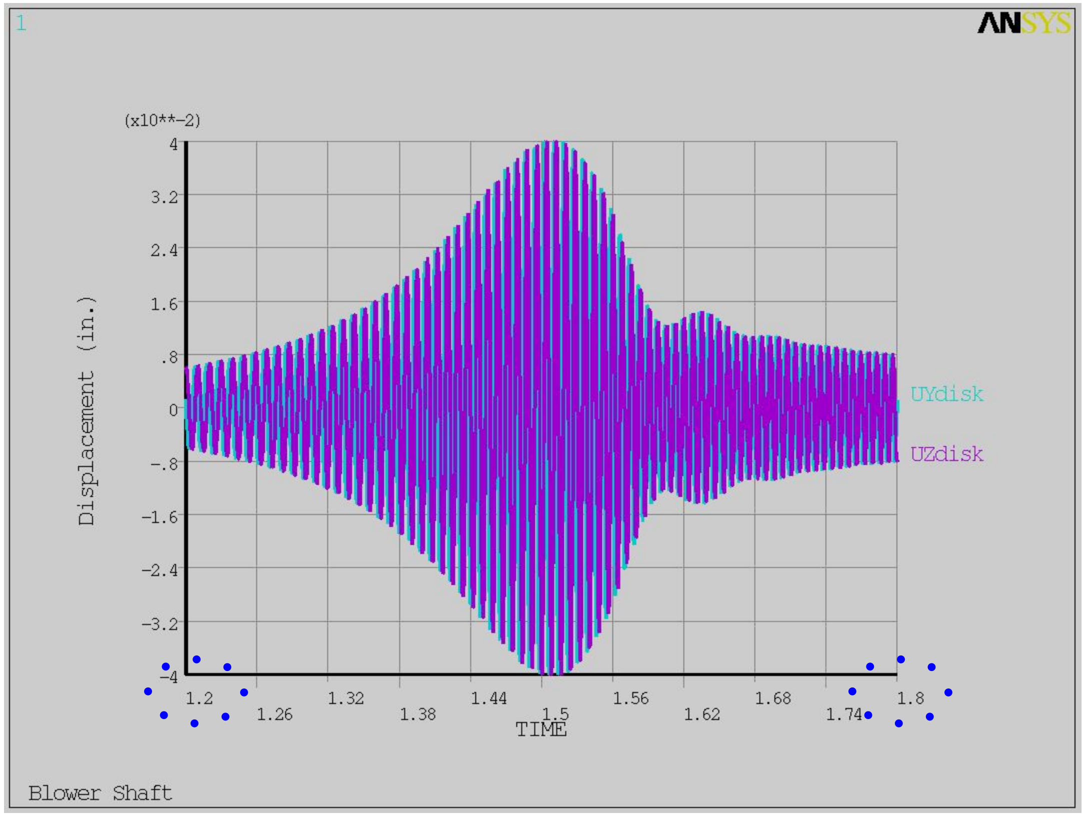




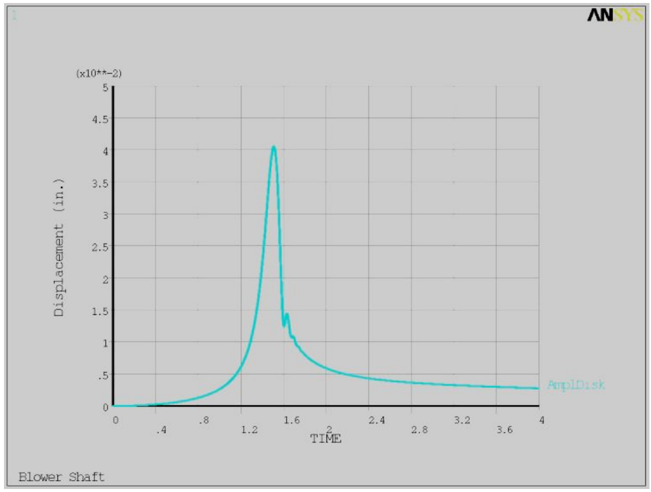
Zoom of transient force

Blower shaft – start up

Displacement U_Y and U_Z at disk zoom on critical speed passage



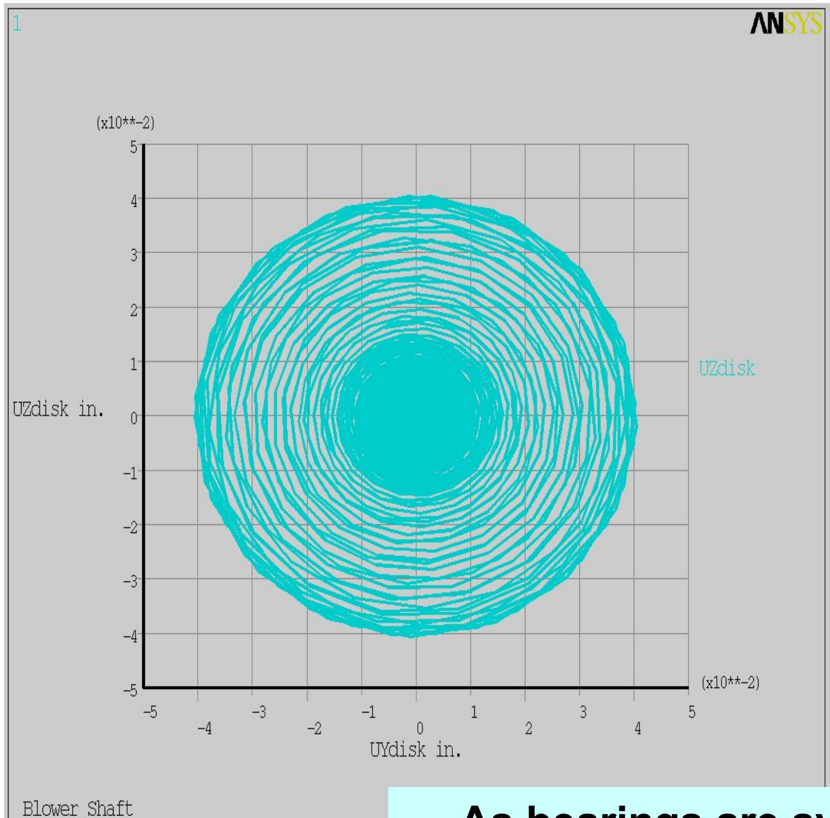
Amplitude of displacement at disk



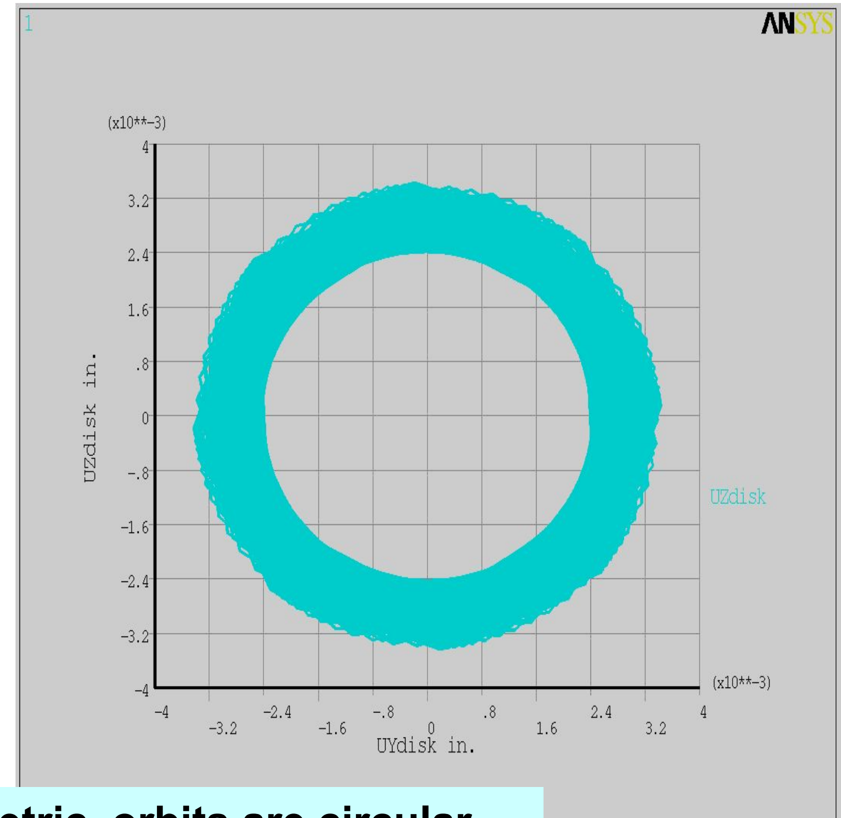
Blower shaft – start up

Transient orbits

0 to 4 seconds



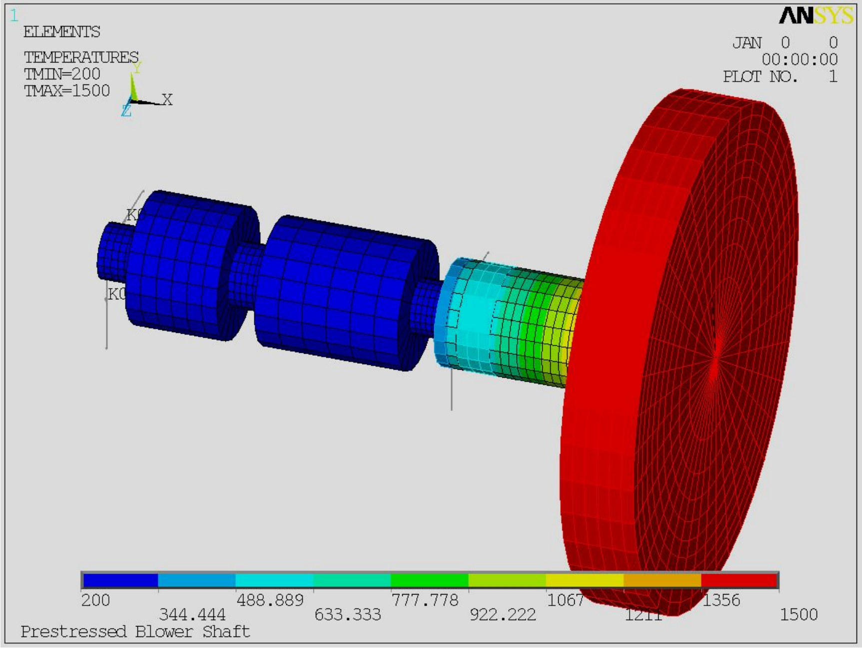
3 to 4 seconds

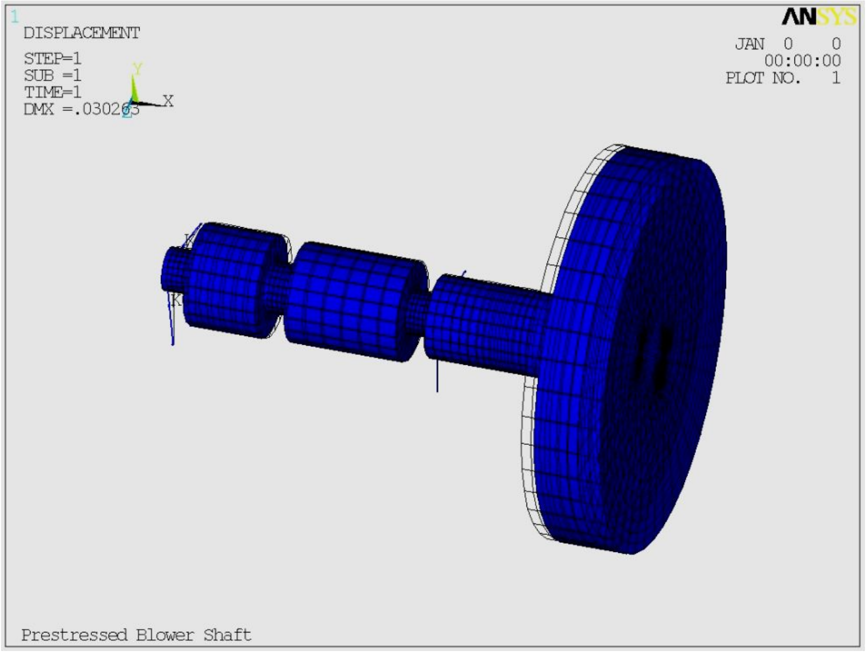


As bearings are symmetric, orbits are circular

Include prestress due to thermal loading:

Thermal body load up to 1500 deg F

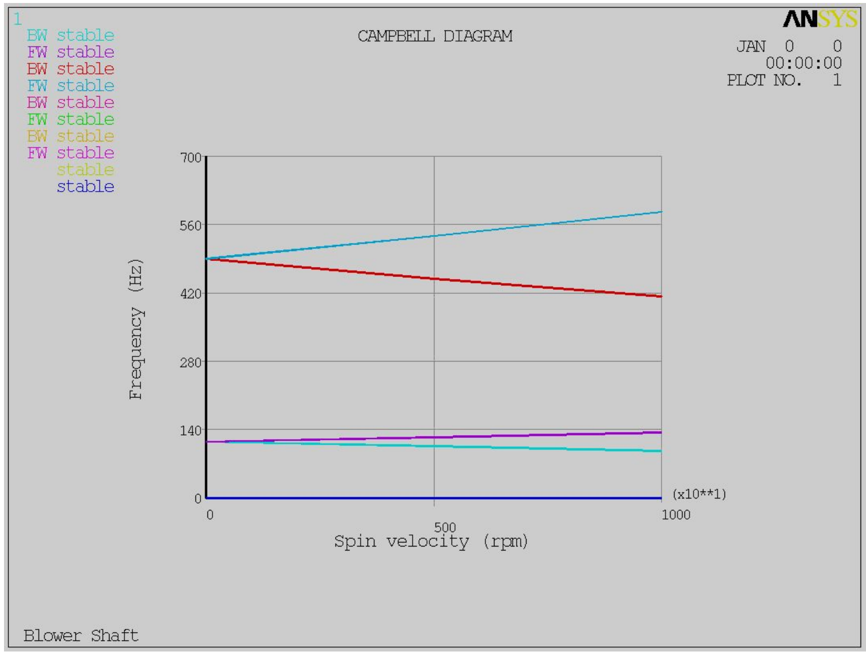




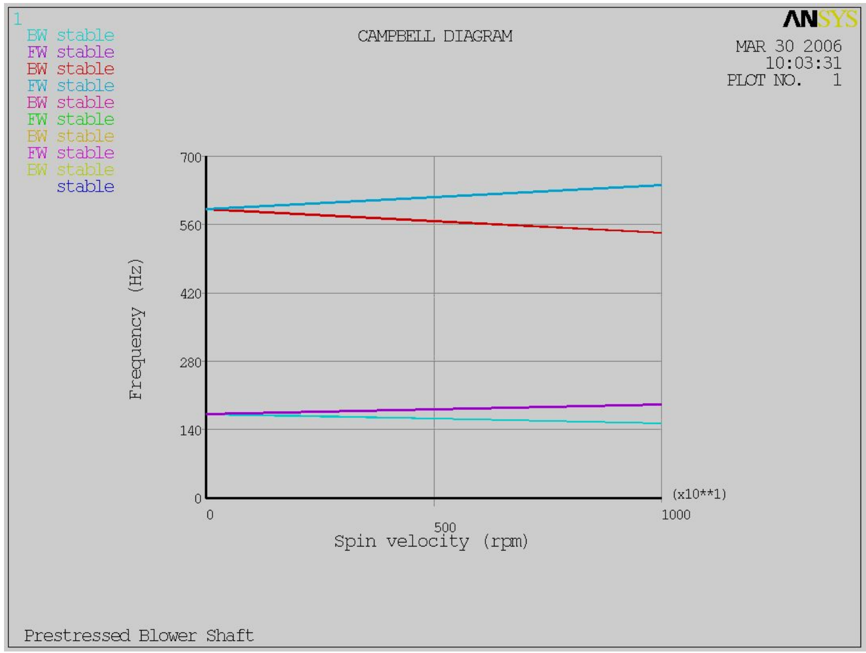
Resulting static displacements

Blower shaft – Campbell diagram comparison

No prestress



With thermal prestress

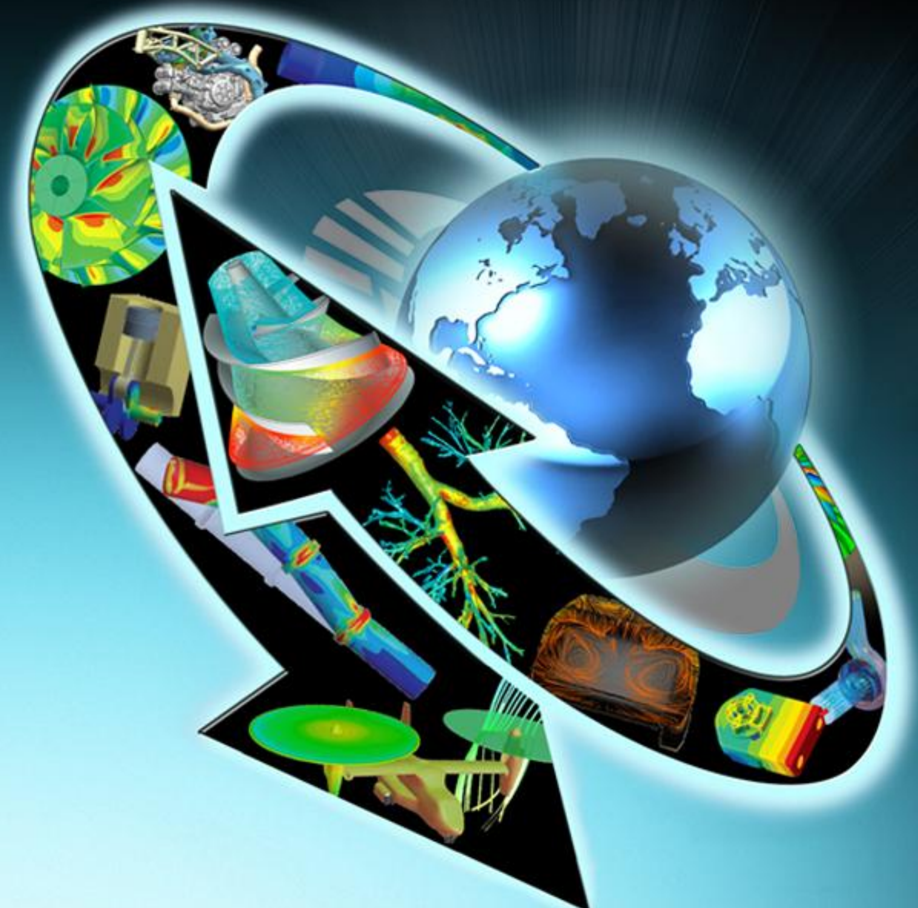


Demo's Agenda

- 3D model
- Point mass by user
- Automatic Rigid Body

- B.C. / Remote displacement
- Bearing (Combi214)
- Joint (Cylindrical, Spherical, BUSHING)
  - Relative to ground / to stator

Rotordynamics with ANSYS Workbench
A workflow example

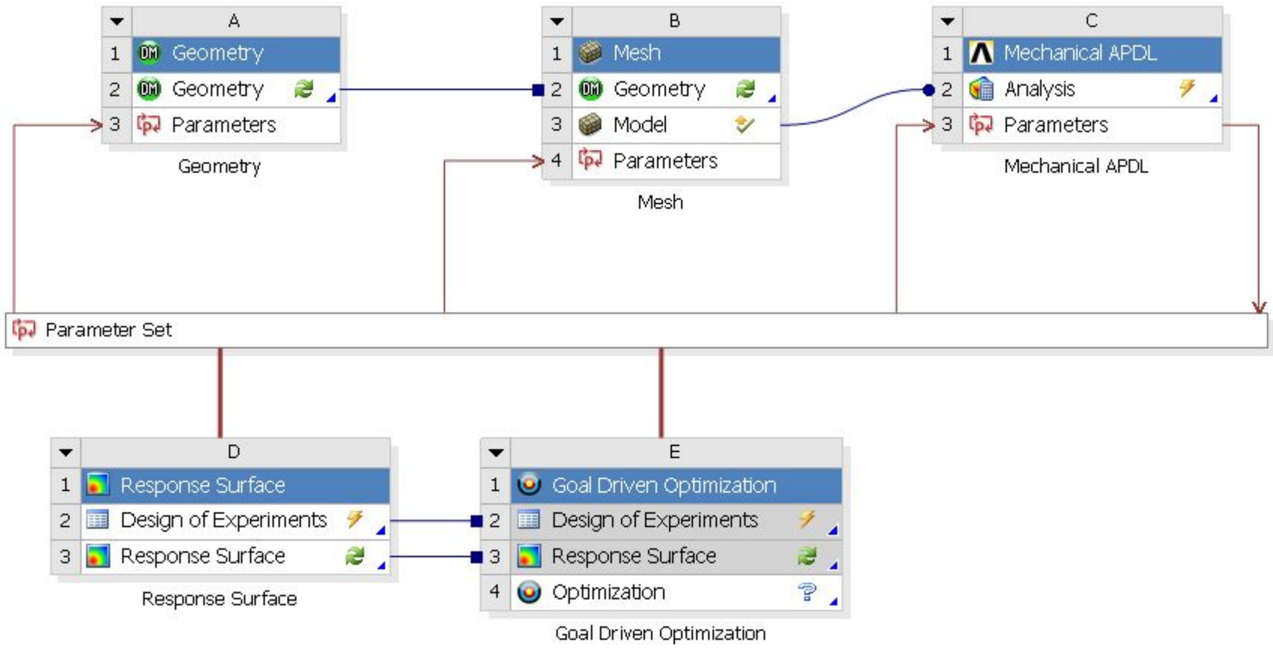


Storyboard

- The geometry is provided in form of a Parasolid file
- Part of the shaft must be reparametrized to allow for diameter variations
- A disk must be added to the geometry
- Simulation will be performed using the generalized axisymmetric elements, mixing WB features and APDL scripting
- Design analysis will be made with variations of bearings properties and geometry

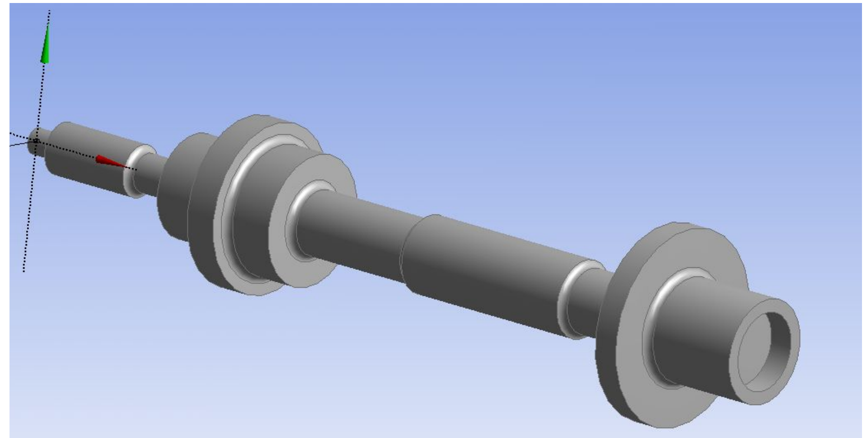
Project view

- **Upper part of the schematics defines the simulation process (geometry to mesh to simulation)**
- **Parameters of the model are gathered in one location (geometry, bearing stiffness)**
- **Lower part of the schematics contains the design exploration tools**



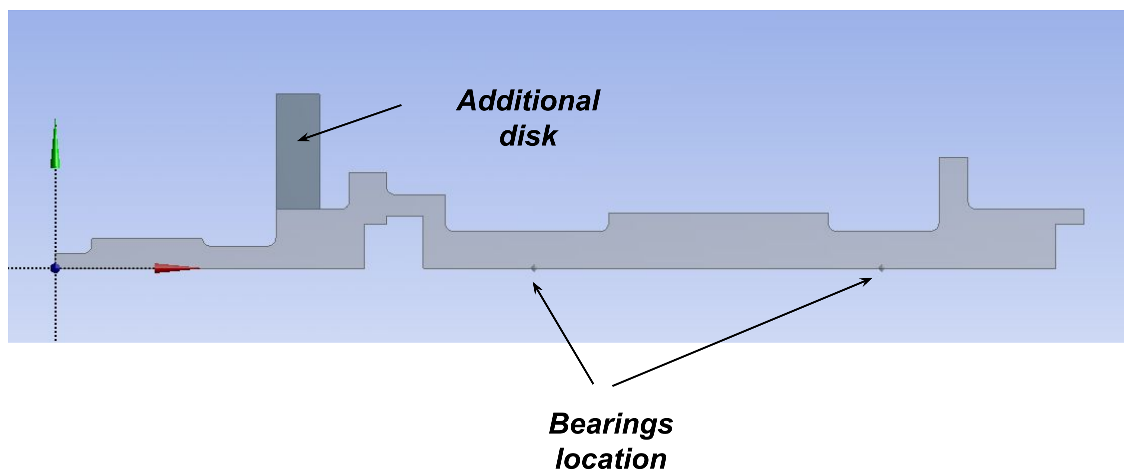
Geometry setup

- Geometry is imported in Design Modeler
- A part of the shaft is redesigned with parametric dimensions
- Model is sliced to be used with axisymmetric elements
- Bearing locations are defined
- A disc is added to the geometry

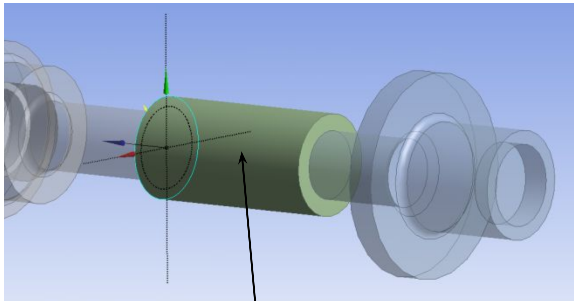


Initial 3D geometry

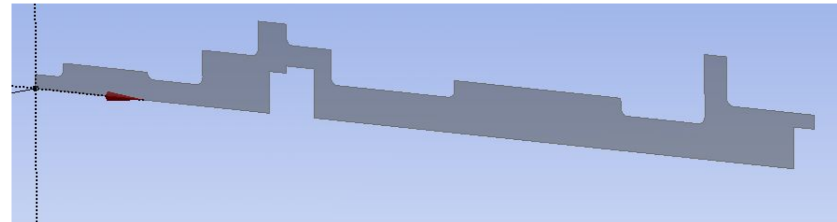
Final axisymmetric model



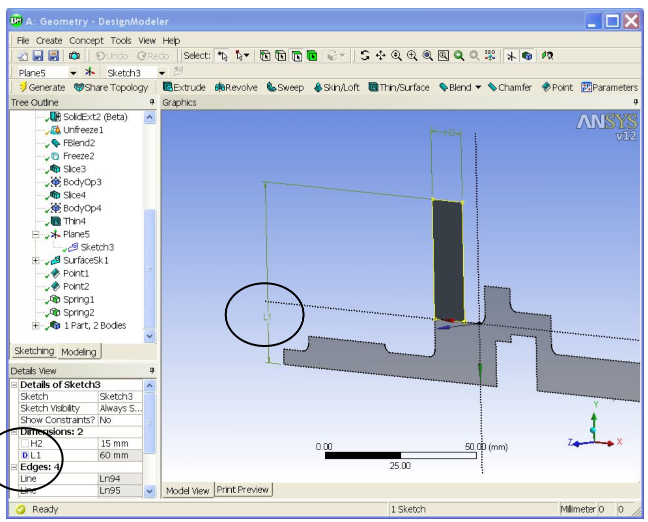
Geometry details



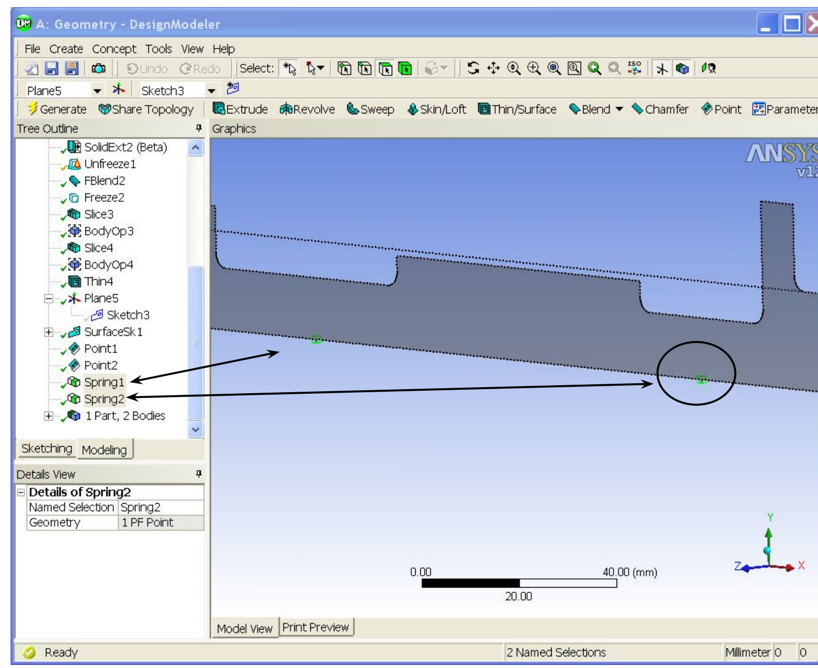
Part of the original shaft is removed and recreated with parametric radius



3D Model sliced to create axisymmetric model

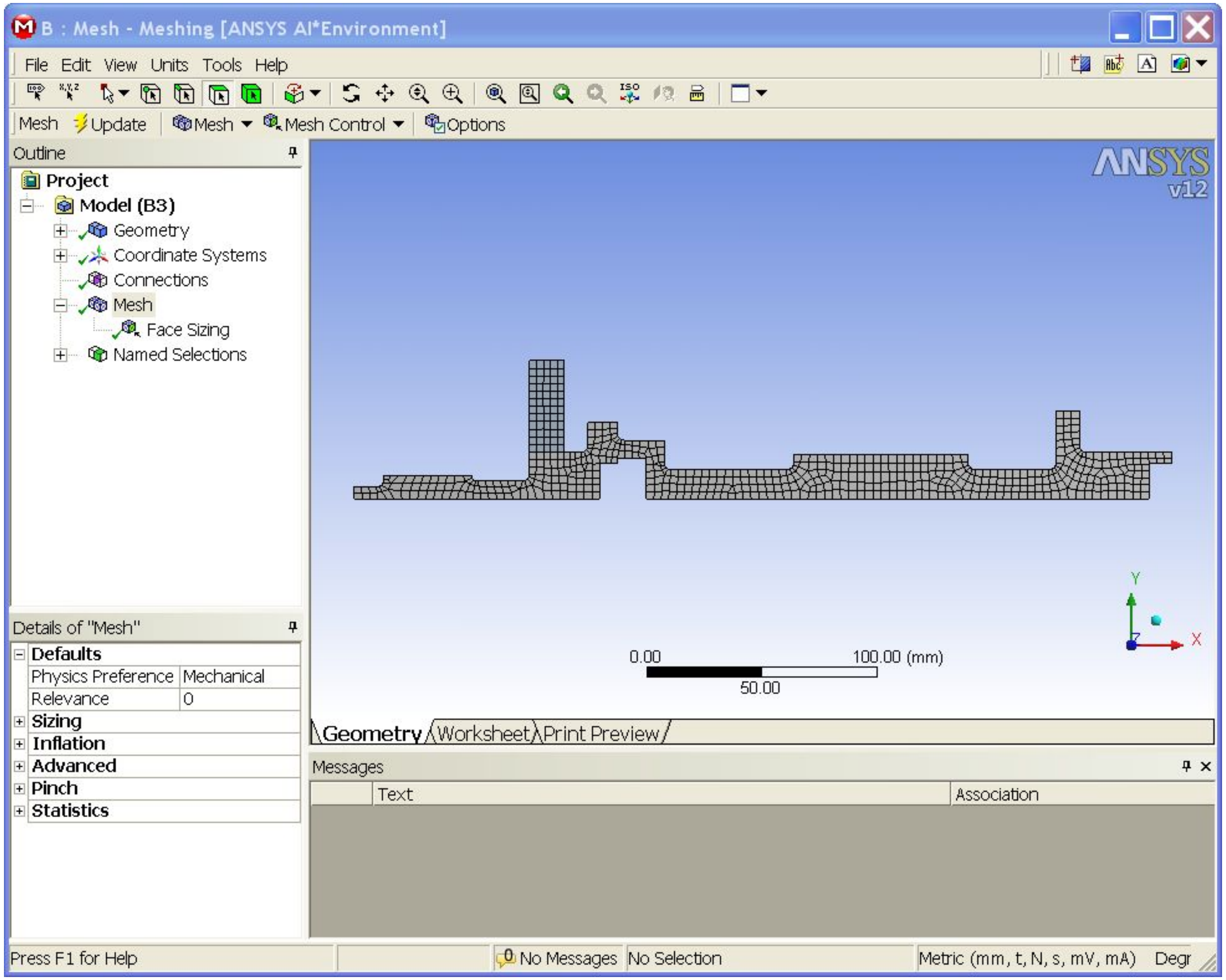


Additional disk created with parameters (the outer diameter will be used for design analysis)

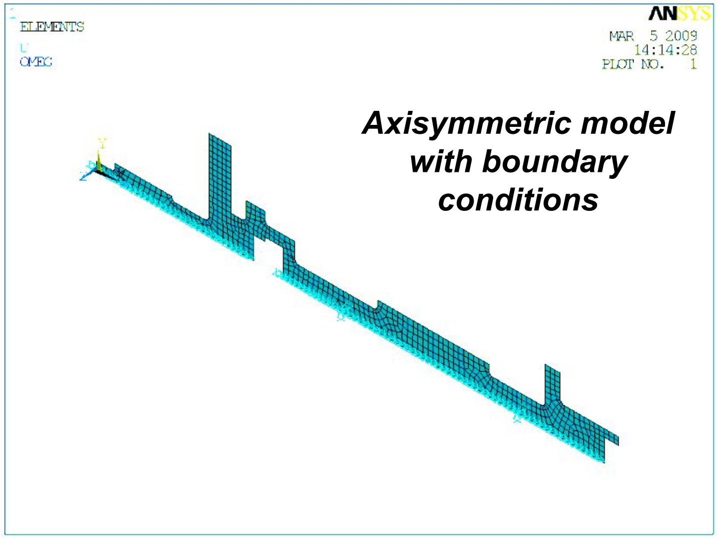


Bearing locations and named selections are created (named selections will be transferred as node components for the simulation)

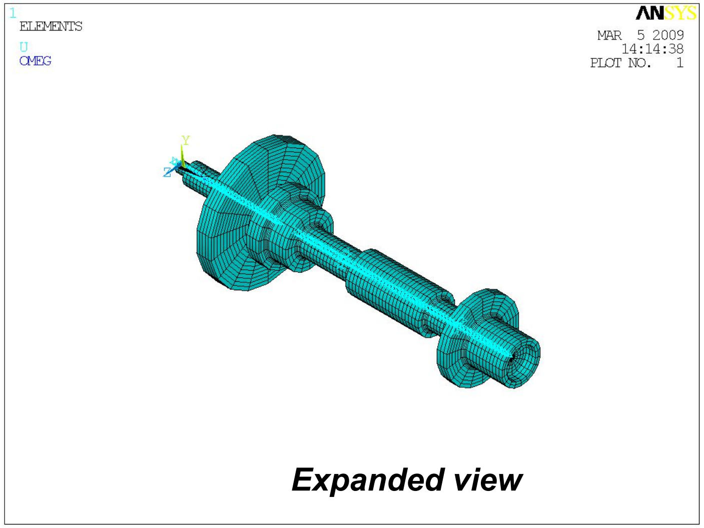
- **The model is meshed using the WB meshing tools**



- **Simulation is performed using an APDL script that defines:**
  - **Element types**
  - **Bearings**
  - **Boundary conditions**
  - **Solutions settings (Qrdamp solver…)**
  - **Post-processing (Campbell plots and extraction of critical speeds)**



Axisymmetric model with boundary conditions



Expanded view

Rotordynsetup.inp

```
/prep7

MP,EX,1,2.078e+5
MP,DENS,1,7806e-12
MP,NUXY,1,0.33

bestif=4.837e4

nspin=10
maxspin=50000
*DIM,SPIN,,nspin
*do,i,1,nspin
SPIN(i) = (i-1)*50000/(nspin-1)
*enddo

! Change element type to 272 axisymm

esel,s,enam,,200
et,1,272,,3
SECT,1,AXIS
SECDATA,1,0,0,0,1,0,0

emodif,all,type,1
emodif,all,sect,1
emodif,all,mat,1

NAXIS
ALLSEL,ALL

/COM, create springs and fix ends
et,100,combi214,,1
r,100,bestif
type,100
real,100
*get,nmax,node,0,count
cmsel,s,spring1
n0=ndnext(0)
n,nmax+1,nx(n0),ny(n0)
e,n0,nmax+1
d,n0,all
d,nmax+1,all
alls
*get,nmax,node,0,count
cmsel,s,spring2
n0=ndnext(0)
n,nmax+1,nx(n0),ny(n0)
e,n0,nmax+1
d,nmax+1,all
d,n0,all
alls

```

Mesh transferred as mesh200 elements, converted to solid272

Spring1 component comes from named selection

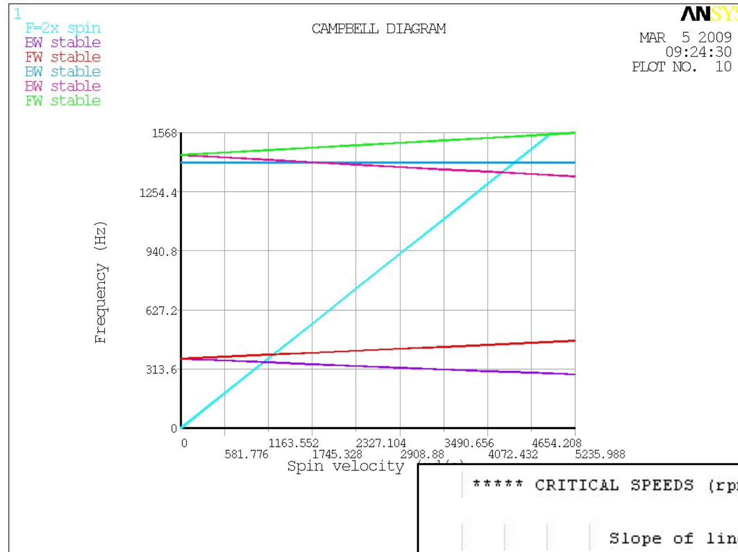
Rotordynsetup.inp


```
emodif,all,mat,1

NAXIS
ALLSEL,ALL

/COM, create springs and fix ends
et,100,combi214,,1
r,100,bestif
type,100
real,100
*get,nmax,node,0,count
cmsel,s,spring1
n0=ndnext(0)
n,nmax+1,nx(n0),ny(n0)
e,n0,nmax+1
d,n0,all
d,nmax+1,all
alls
*get,nmax,node,0,count
cmsel,s,spring2
n0=ndnext(0)
n,nmax+1,nx(n0),ny(n0)
e,n0,nmax+1
d,nmax+1,all
d,n0,all
alls
```


```

/COM, SUPPRESSING AXIAL MOTION IN THE SHAFT
NSEL,S,LOC,Y,0
NSEL,R,LOC,Z,0
D,ALL,UX,0
NSEL,ALL
FINI

/COM, PERFORMING CAMPBELL ANALYSIS USING QRDAMP EIGEN SOLVER
/SOLU
ANTYPE,MODAL
MODOPT,DAMP,10,1.0,,                                   ! COMPUTE COMPLEX EIGEN M
MXPAND,10,,,YES                                   ! EXPAND ALL THE MODES WITH STR
CORIOLIS,ON,,,ON                                      ! CORIOLIS ON IN A STATIONA
RATIO = 4*ATAN(1)/30                                  ! CONVERT RPM INTO RADIANS,

*DO,I,1,nspin
OMEGA,SPIN(I)*RATIO                               ! SOLVE FOR DIFFERENT ROTATIONA
SOLVE
*ENDDO

FINI

/POST1
```

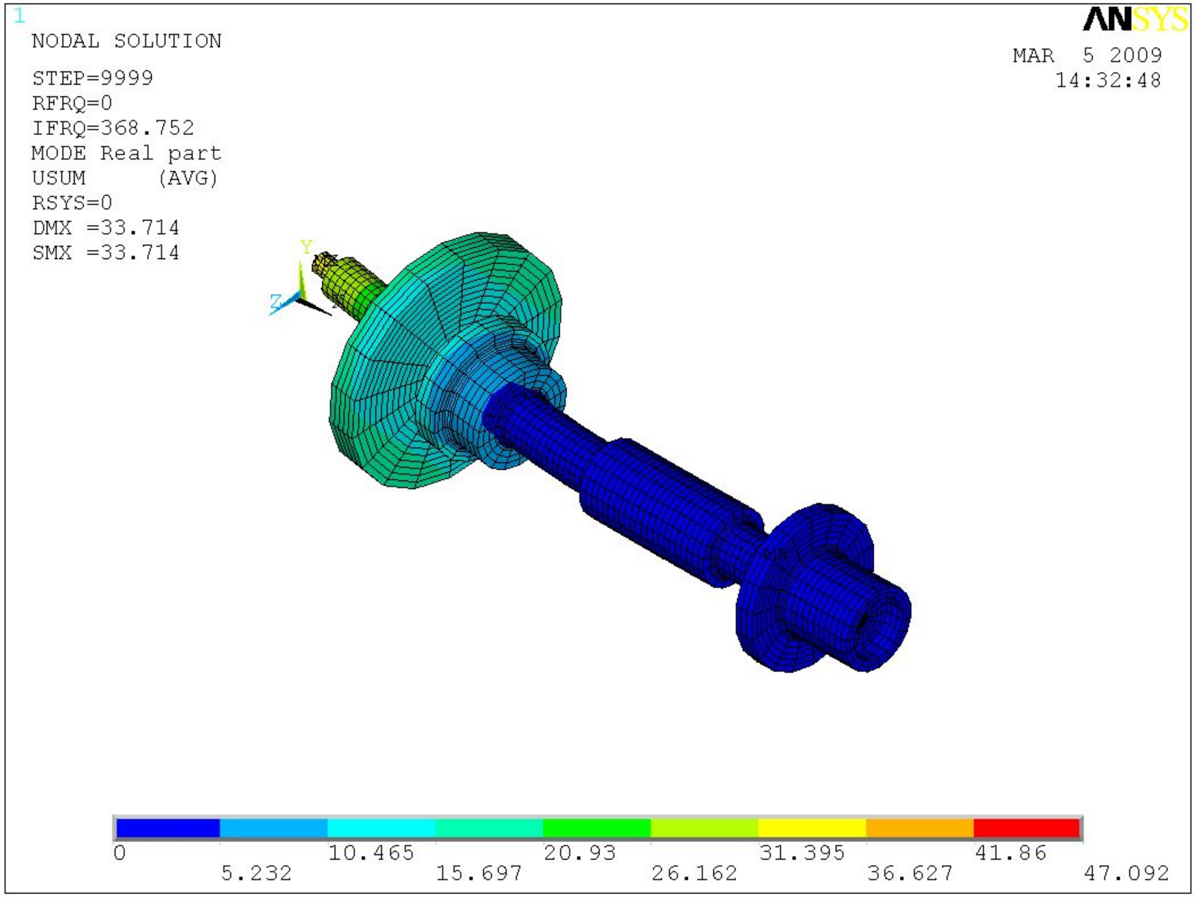
Simulation results

- **The APDL scripts can create plots and animations**
- **The results can also be analyzed within the Mechanical APDL interface**
- **Results are extracted using *get commands and exposed as WB parameters (showing the performance of the design)**




```
***** CRITICAL SPEEDS (rpm) FROM CAMPBELL (sorting on) *****

        Slope of line :      4.000

              1          5384.796
              2          5688.545
              3         21155.597
              4         21020.742
              5         22531.352

PLOT CAMPBELL DIAGRAM
   Sorting : ON
   Slope of line :    4.000
   X axis unit : rpm
```

Mode animation (expanded view)



- **The model has 2 geometry parameters (disc and shaft radius) as well as a stiffness parameters (bearings stiffness)**

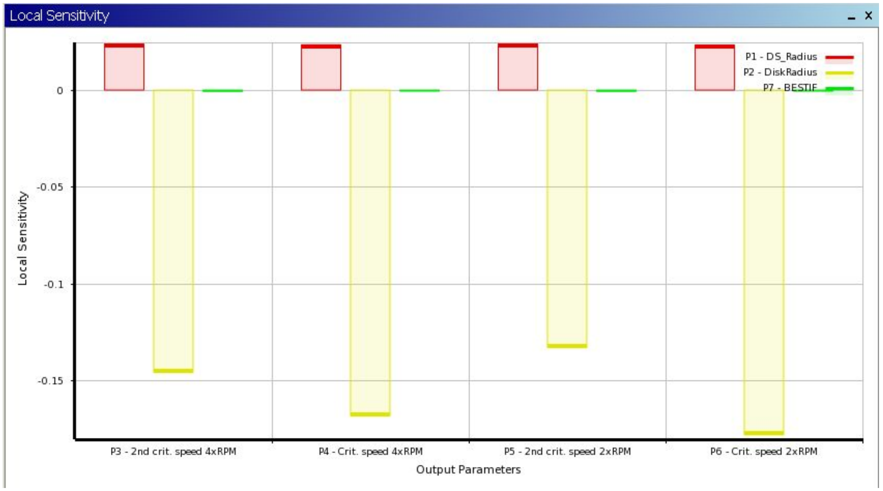
- **4 output parameters are investigated: first and second critical speeds at 2xRPM and 4xRPM (obtained from theCampbell diagrams and `*get` commands)**

Outline of All Parameters

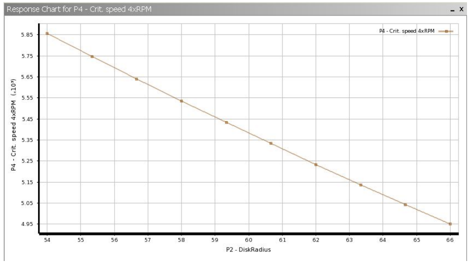
	A	B	C
1	ID	Parameter Name	Value
2	⊟ Input Parameters		
3	P1	DS_Radius	19
4	P2	DiskRadius	60
5	P7	BESTIF	48370
*	New input parameter	New name	New expression
7	⊟ Output Parameters		
8	P3	2nd crit. speed 4xRPM	5688.5
9	P4	Crit. speed 4xRPM	5384.8
10	P5	2nd crit. speed 2xRPM	11715
11	P6	Crit. speed 2xRPM	10496

Sample results

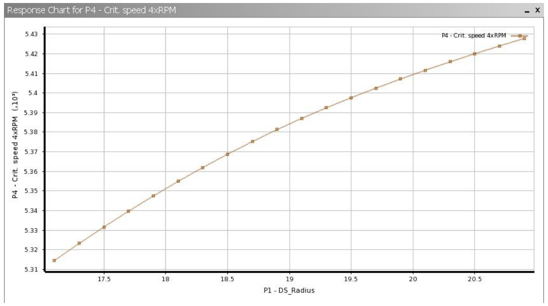
- **A response surface of the model is created using a Design of Experiments**
- **Curves, surfaces and sensitivity plots are created and the design can be investigated**
- **Optimization tools are also available**



Sensitivity plots: the bearing stiffness has no influence on the first and second critical speeds, the disc radius is the key parameter



Evolution of critical speed with shaft and disc radius



Optimization

- **A multi-objective optimization is described and possible candidates are found (usually, there are multiple acceptable configurations)**
- **Trade-off plots give an indication about the achievable performance**

Table of Schematic E4: Optimization

	A	B	C	D	E	F	G	H
1		P1 - DS_Radius	P2 - DiskRadius	P7 - BESTIF	P3 - 2nd crit. speed 4xRPM	P4 - Crit. speed 4xRPM	P5 - 2nd crit. speed 2xRPM	P6 - Crit. speed 2xRPM
2	Optimization Study							
3	Objective	No Objective	No Objective	No Objective	No Objective	Seek Target	Maximize	No Objective
4	Target Value					5000		
5	Importance	Default	Default	Default	Default	Higher	Default	Default
6	GDO Sample Set 1							
7	Candidate A	20.746	65.901	51743	5344.9	4999.2	11083	9694.5
8	Candidate B	20.381	65.76	53017	5346.3	5002.2	11085	9701.5
9	Candidate C	20.868	65.807	47497	5353	5007.9	11099	9712