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***DyRoBeS© Manual* | Rotor**

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Other Topics

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Introduction

DyRoBeS©_Rotor is a powerful rotor dynamics program based on Finite Element Analysis (FEA). This program has been developed for the analysis of free and forced vibrations (Lateral, Torsional, and Axial) of multi-shaft and multi-branch flexible rotor-bearing-support systems. The acronym, ***DyRoBeS***©, denotes Dynamics of Rotor Bearing Systems.

The program contains extensive modeling, analysis, and post-processing capabilities. This Windows based software is very user friendly and easy to use. The operation is entirely consistent with the industrial standard operation in the window environment. Context Sensitive Help can be obtained at any time by pressing <F1> key. The help files are provided in two formats, hlp and chm. Some are provided in pdf format. The contents in the hlp and chm files are identical. The author likes hlp format, however, the latest default format by Microsoft is chm format. If you have a new computer with latest Windows operation system and can not open hlp help file, you may download the WinHlp program from Microsoft web site. Or you may simply click on the chm files to open the help file.

The lateral vibration of the discretized system is described by two translational (x, y) and two rotational (θ_x, θ_y) coordinates at each finite element station, i.e. 4 degrees-of-freedom (DOF) at each shaft station. The motion of a flexible support is described by two translational displacements (x, y). The analyses for lateral vibration contain:

Static Deflection and Bearing/Constraint Reactions

Critical Speed Analysis

Critical Speed Map Analysis

Whirl Speed and Stability Analysis

Steady State Synchronous Response Analysis – Linear System

Steady State Synchronous Response Analysis – NonLinear System

Time Transient Analysis (Time Domain)

Steady State Harmonic Excitation Response Analysis

Steady Maneuver Load Analysis

Time Transient Analysis (Frequency Domain)

Catenary (Gravity Sag) Analysis

For torsional vibration, the motion of each finite element station is described by a rotational displacement (θ_z) about the spinning axis. The systems can be continuous, discrete, or the combination of continuous and discrete elements. The analyses for torsional vibration are:

Natural Frequencies and Modes Calculation

Steady State Forced Response Analysis

Transient Analysis (Time Dependent Excitations)

Startup Transient Analysis (Speed Dependent Excitations)

For forced response analyses including the transient analysis, the modal damping can be specified if the direct damping is not readily available. Details on the damping model are explained in the Modal Damping.

For axial vibration, the motion of each finite element station is described by a translational displacement (z) along the spinning axis. Similar to the torsional model, the systems can be continuous, discrete, or the combination of continuous and discrete elements. The analyses for axial vibration are:

Natural Frequencies and Modes Calculation

Steady State Forced Response Analysis

Again, for forced response analysis, the modal damping can be specified if the direct damping is not readily available. Details on the damping model are explained in the Modal Damping.

In Ver 17, a new feature is implemented, that is the lateral, torsional, and axial vibration can be coupled together for a geared system. That is, there are 6 degrees of freedom for every finite element station. For more details, click the following link:

Coupled Lateral-Torsional-Axial Analysis

Several design Tools are also provided in this program. They are: Aerodynamic Cross Coupling Calculation, Rolling Element Bearing Stiffness Calculation, Liquid Annular Seal Dynamic Coefficients, Gas Laby Seal Dynamic Coefficients, Squeeze Film Damper Stiffness and Damping Calculation, Mass/Inertia Properties Calculation, Rotor Orbit Analysis, Balancing Calculation, and Auto-Balancing Analysis

A number of Examples are included in the software package to demonstrate the program's features and capabilities. The user is encouraged to go through the examples and verify the results.

See also **DyRoBeS©_BePerf**.

Getting Started

DyRoBeS is a Windows-Based program. Its operation is entirely consistent with the industrial standard operation in the Windows environment. There are many ways to invoke a Windows-Based program. Several commonly used methods to start **DyRoBeS-Rotor** are described below:

1. Click **DyRoBeS-Rotor** from the Start Menu - Program - **DyRoBeS** folder,

2. Double click the **Rotor.exe** file from **Windows Explorer**,

3. Double click the **DyRoBeS-Rotor** icon if you have created the shortcut,

4. Double click the data file (*.ROT) created by **DyRoBeS-Rotor**,

This method is useful if you want to open an existing data file. However, you may need to

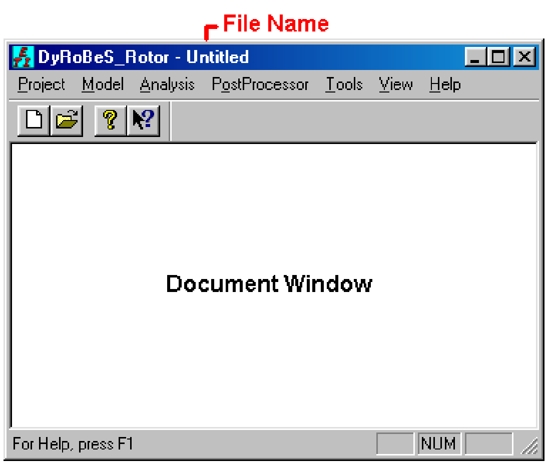
change the file properties and select the current and latest Rotor.exe program to be the

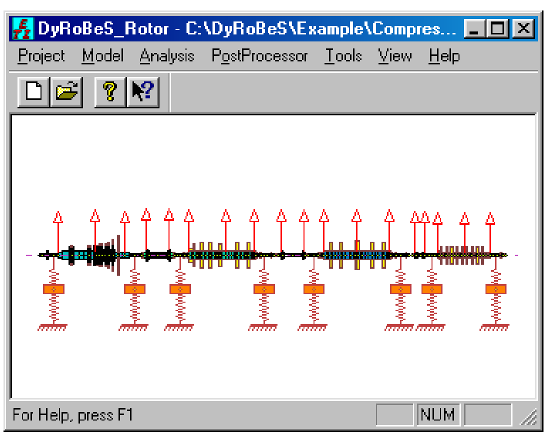
default open program.

When you start **DyRoBeS-Rotor** by using methods 1, 2, and 3, the mainframe window, as shown below, appears on the screen. The default file name **Untitled** is shown in the filename header and the document window is empty. Now, you are ready to open an existing file or create a new file.

When you start **DyRoBeS-Rotor** by double clicking the data file (method 4), the document window shows the rotor model and the filename is shown in the filename header, as shown below. Now, you can make changes, perform another analysis, or review the results.

You can resize the mainframe window as you wish. To resize the window, simply drag the window boundaries or click on the resize button.



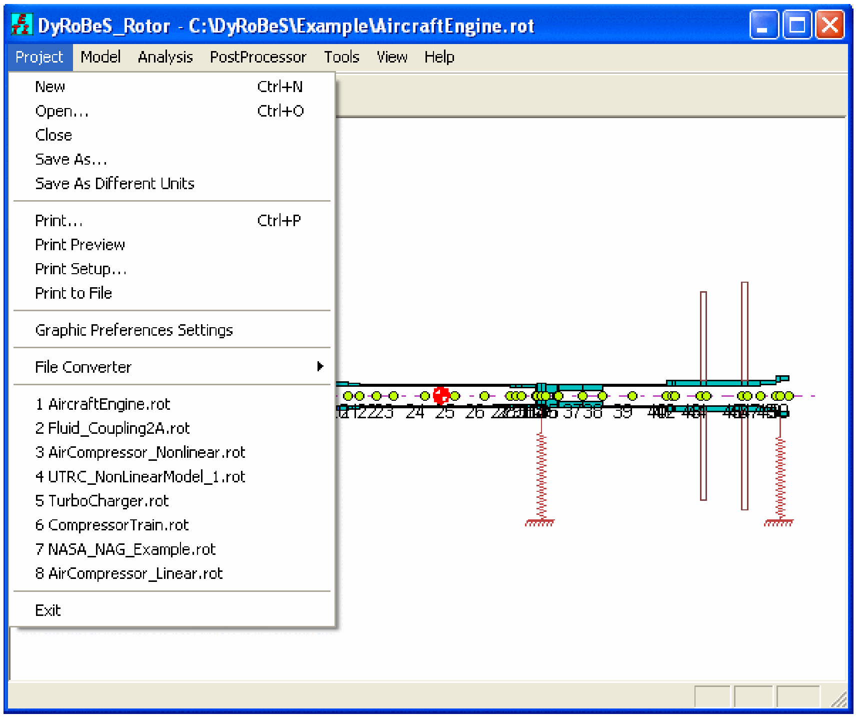


The help files are provided in two formats, hlp and chm. Some are provided in pdf format. The contents in the hlp and chm files are identical. The author likes hlp format, however, the latest default format by Microsoft is chm format. If you have a new computer with latest Windows operation system and can not open hlp help file, you may download the WinHlp program from Microsoft web site. Or you may simply click on the chm files to open the help file.

Many examples are provided under \Example subdirectory. Users are encouraged to go through these examples.

Project

A **Project** is also called a **File** or a **Document**, which contains the rotor bearing system data, run time parameters, and all the related postprocessor files. All the options under the Project, as shown in the following Figure, are self-explanatory. You can start with a **New** file, **Open** an existing file, **Close** the current file, **Save** the file **As** a separate filename, or **Save** the file **As** a separate filename with **different unit** system. Eight most recently opened files are listed in the **recent-file-list** for quick selection. The filename and pathname follow the standard convention for the Windows environment. The data file has the extension **.rot**.

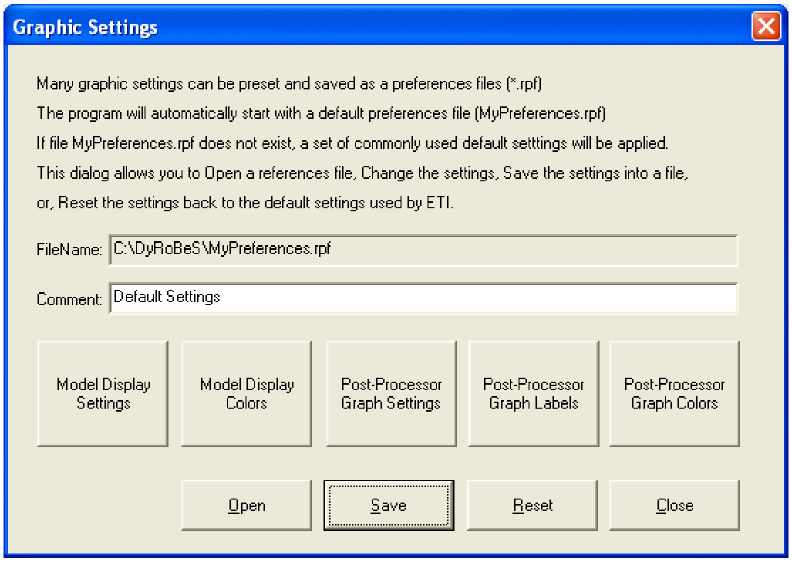


Once an existing file is opened/selected or a new file is created, the associated rotor bearing system model will be graphically displayed in the document window. You can **Print** the document window to the printer or to a bmp or gif file.

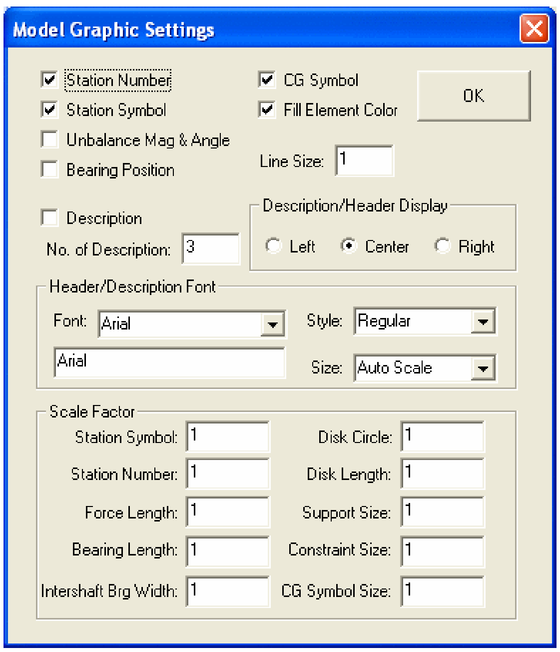
Save As Different Units option allows you to convert the current data file into different unit system with another specified file name without overwriting the original file.

File Converter option allows you to convert the old DOS ***DyRoBeS*** files into the new *Windows* version format. The data file for the DOS version does not have file extension, but the new *Windows* version data files have the file extension .rot. This file extension will be added automatically for you. If you have other data format, contact ***DyRoBeS*** developer to add a converter for you.

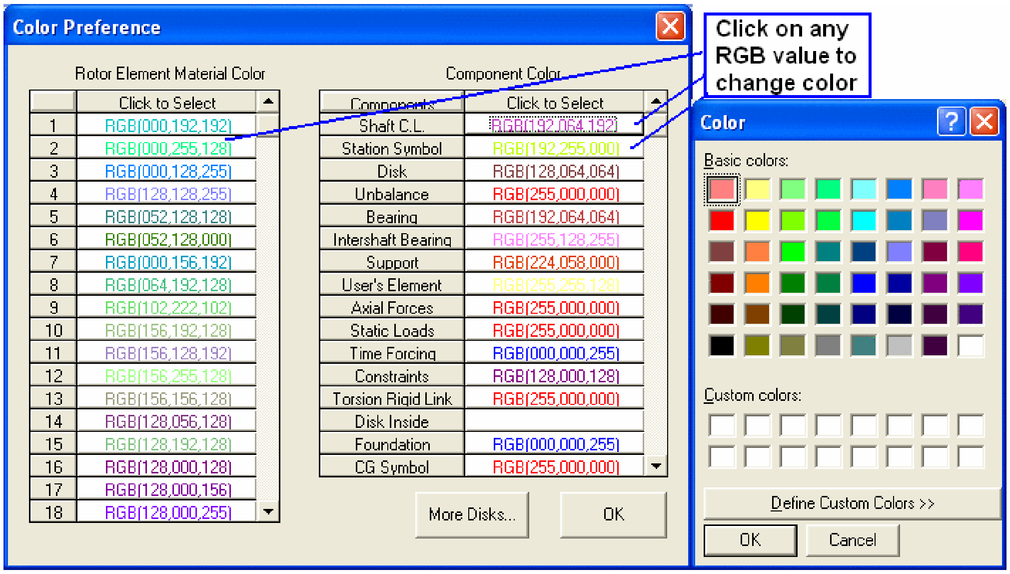
Graphic Preferences Settings allows you to set your own preferences settings for many graphic features and plot labels. You can save these settings into a preference file, such as one for screen, one for printer, or one for each different rotor-bearing data file. To change color for a specific setting, simply click the RGB color value to open the Color Dialog Box for selection, as illustrated in the following figure. The startup preferences file named **MyPreferences.rpf** will be automatically opened and applied when ***DyRoBeS-Rotor*** is activated. This will be your own default startup file. If MyPreferences.rpf file does not exist, the default settings by ETI will be applied. You can also restore the ETI defaults by clicking the Reset button. There are five preference settings input buttons shown in the screen.



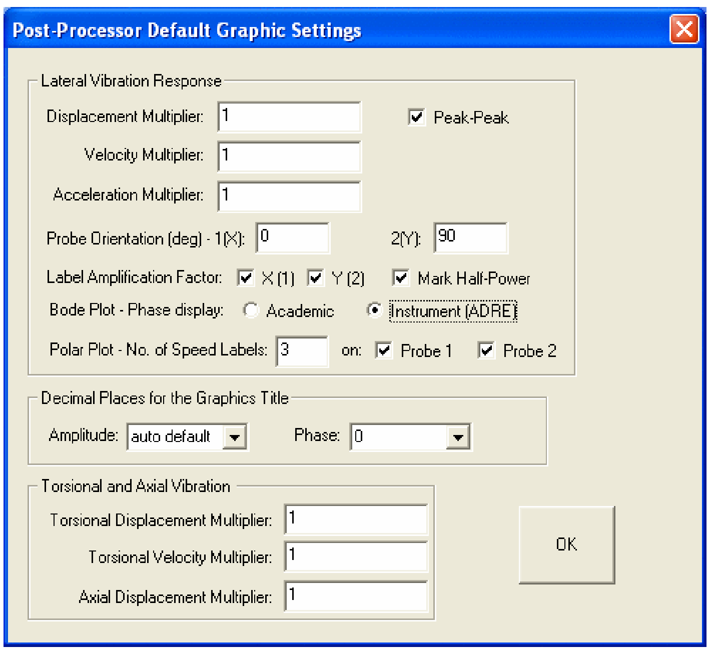
Model Display Settings



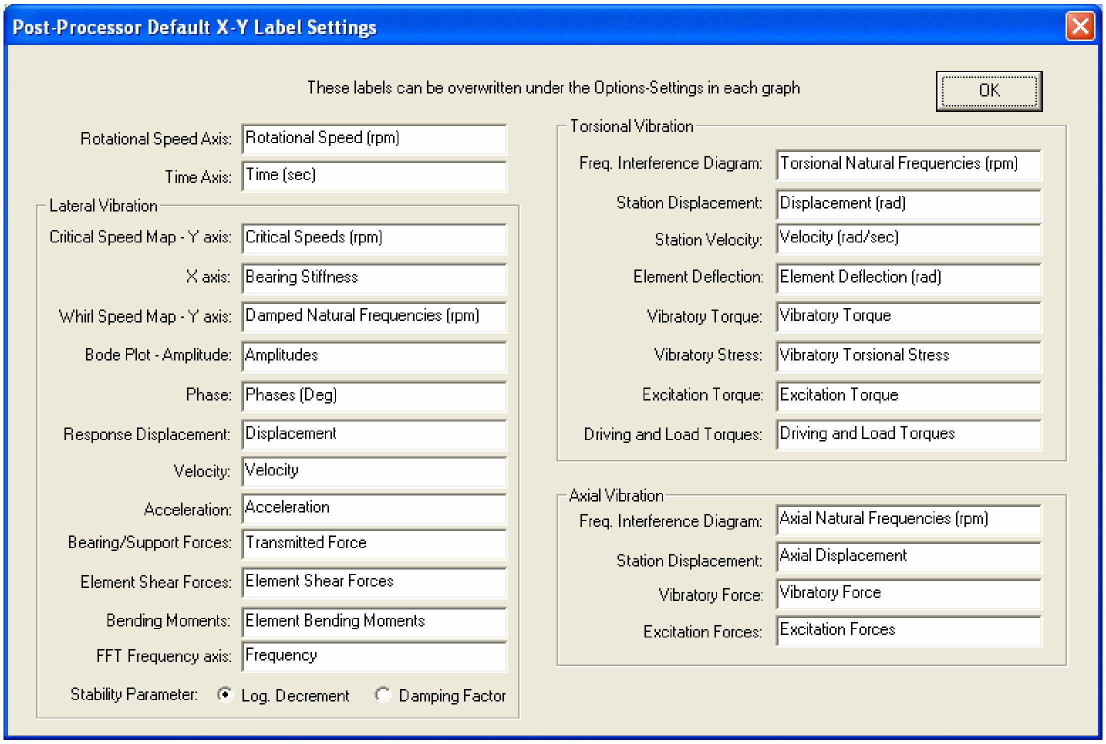
Model Display Colors



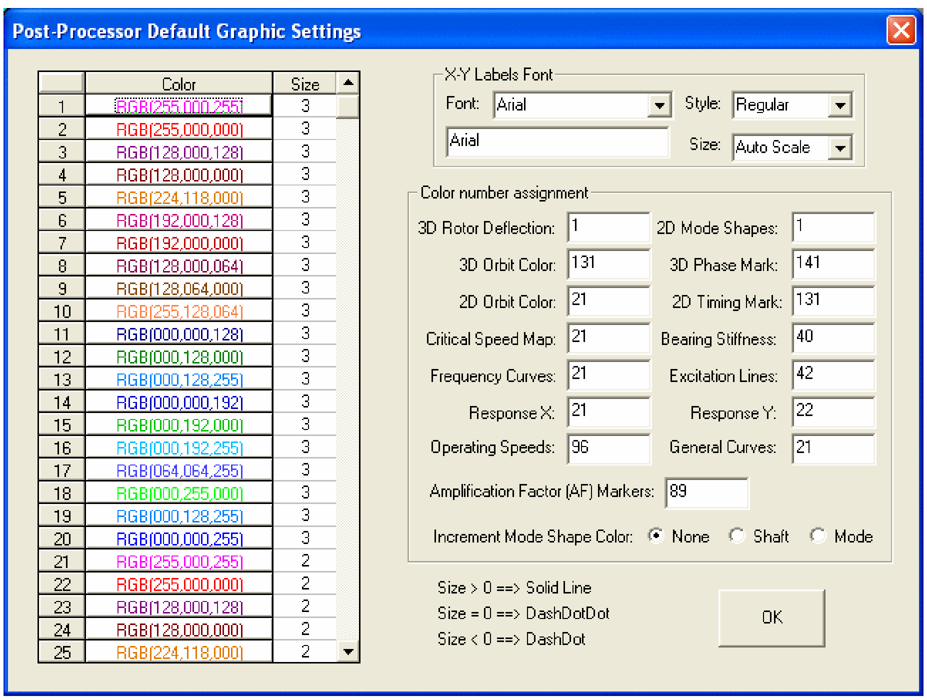
Post-Processor Graph Settings



Post-Processor Graph Labels



Post-Processor Graph Colors



See also File Extension.

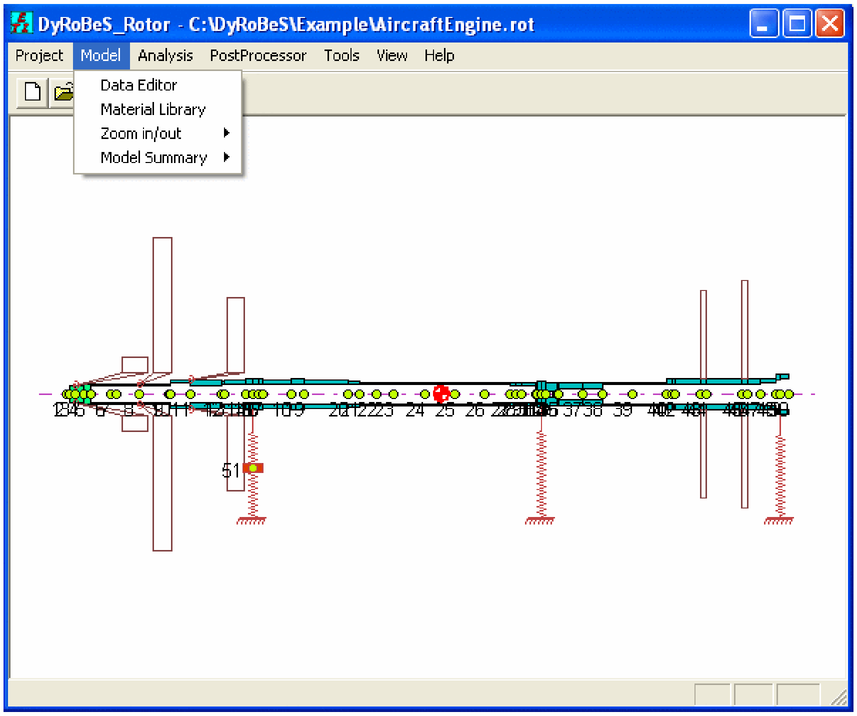
Description of File Name Extension

After the analysis is performed, additional files will be created for the post processing. The description of the file extension is listed below for reference.

Analysis Name	ASCII TEXT	Binary (Post-Process)
Input File	ROT	
Lateral Analysis		
Model Summary	OU0	None
Static Deflection& Bearing Loads	OU1	ST, STX
Critical Speed Analysis	OU2	CS, CSE, CSX
Critical Speed Map	OU3	CSM
Whirl Speed/Stability Analysis	OU4	WS, WSM
Steady Synchronous Response Linear Systems	OU5	UR, URF
Steady Synchronous Response Non-Linear Systems	OU6	UR, URF
Time Transient Analysis	OU7	TR, TRF, FCS
Steady State Harmonic Excitation Including non-synchronous Linear Systems	OU8	VR, VRF
Steady Maneuver Load Analysis	OU9	SM, SMX
Torsional Analysis		
Model Summary	OT0	None
Natural Frequency Analysis	OT1	OTA, OTX
Steady Forced Response	OT2	OTB, OTY
Time Transient – Time Dependent Excitations	OT3	OTC, OTZ
Startup Transient – Speed Dependent Excitations	OT4	OTD, OTZ
Axial Analysis		
Model Summary	OA0	
Natural Frequency Analysis	OA1	OAA, OAX
Steady Forced Response	OA2	OAB, OAY

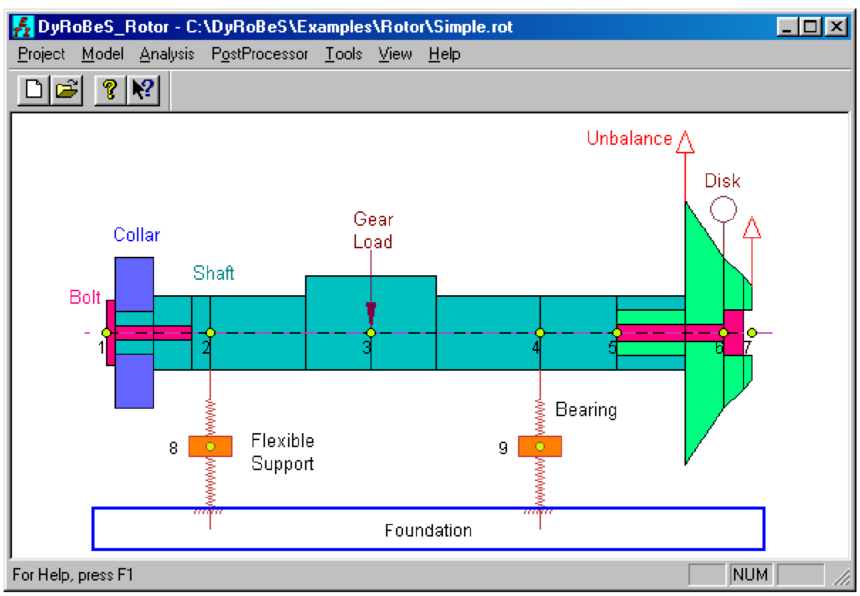
Model

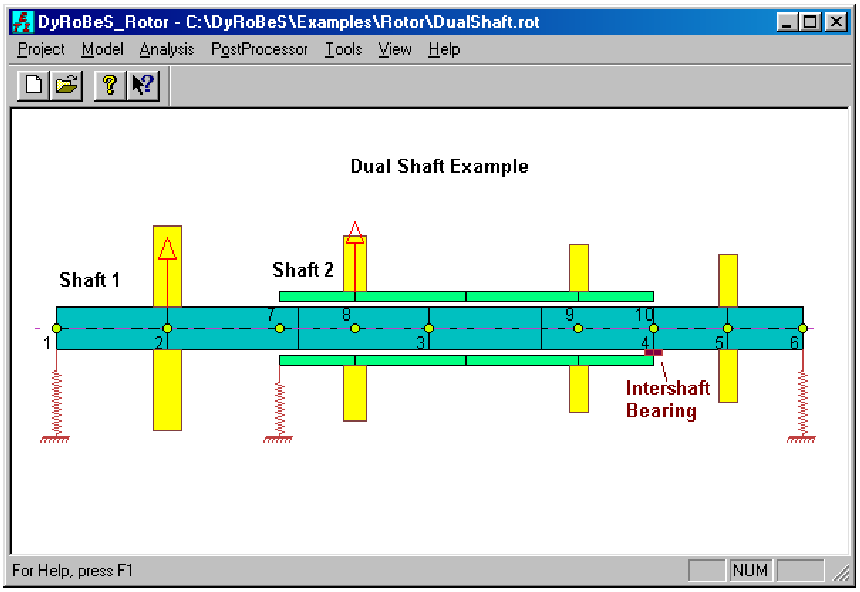
Under the Model menu, you can build a new rotor bearing system model or modify/edit your existing model by clicking the Data Editor. The usage of the Data Editor will be explained in Modeling and Data Editor session. You can also create and modify your Material Library. Zoom in/out allows you to visualize the model in detail. The Model Summary summarizes the system parameters and tabulates the related input data in a very organized ASCII format (text) that allows you to verify your model.

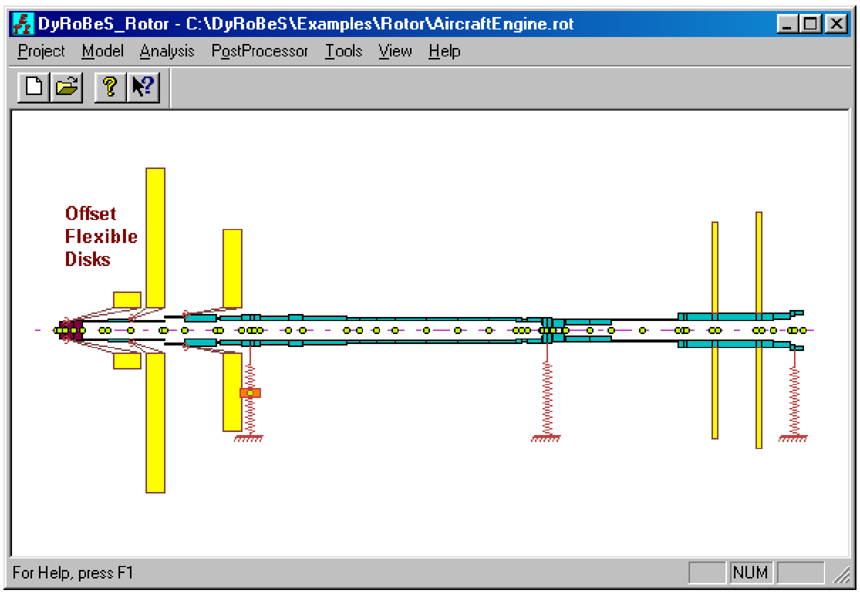


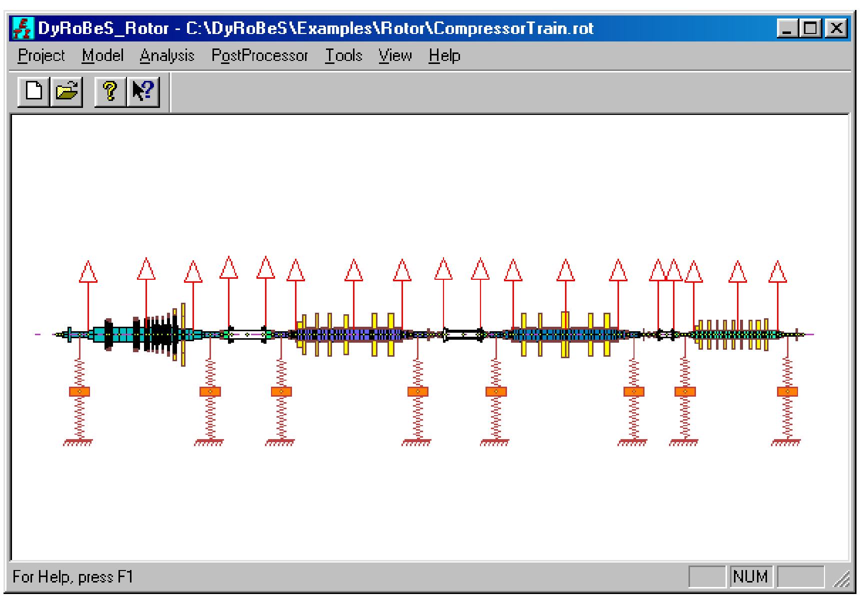
Typical Rotor Configurations

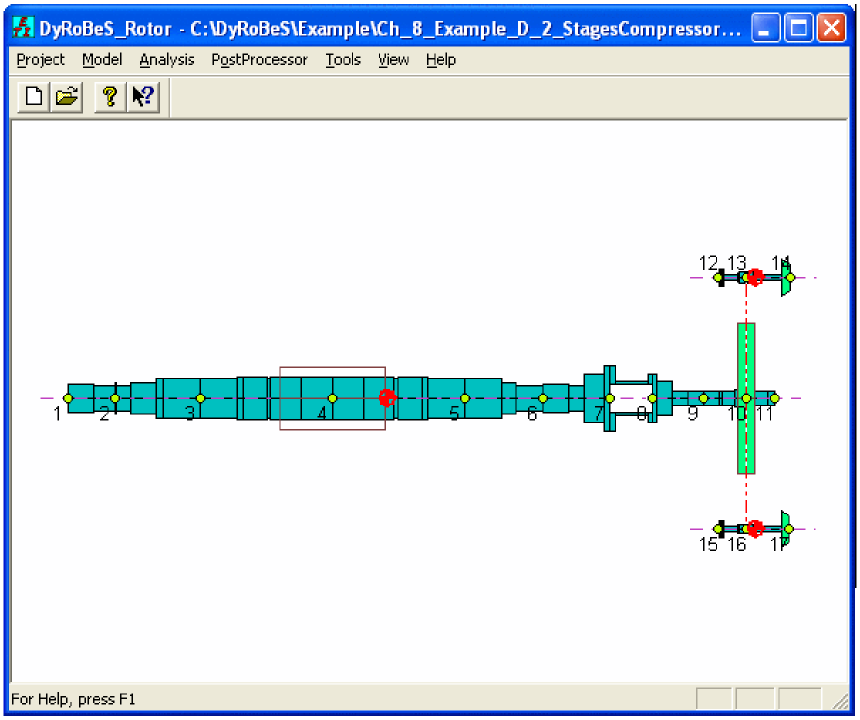
Followings are some typical rotor configurations:




DyRoBeS_Rotor - C:\DyRoBeS\Examples\Rotor\DualShaft.rot
Project Model Analysis PostProcessor Tools View Help
Dual Shaft Example
Shaft 1
Shaft 2
1
2
3
4
5
6
7
8
9
10
Intershaft Bearing
For Help, press F1
NUM

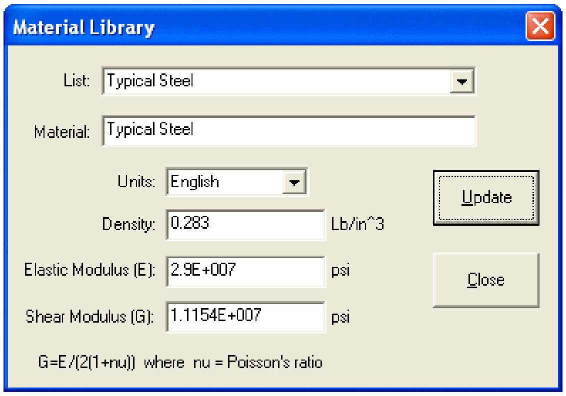

DyRoBeS_Rotor - C:\DyRoBeS\Examples\Rotor\AircraftEngine.rot
Project Model Analysis PostProcessor Tools View Help
Offset Flexible Disks
For Help, press F1
NUM


DyRoBeS_Rotor - C:\DyRoBeS\Examples\Rotor\CompressorTrain.rot
Project Model Analysis PostProcessor Tools View Help
For Help, press F1
NUM


DyRoBeS_Rotor - C:\DyRoBeS\Example\Ch_8_Example_D_2_StagesCompressor...
Project Model Analysis PostProcessor Tools View Help
12 13 14
1 2 3 4 5 6 7 8 9 10 11
15 16 17
For Help, press F1

Material Library

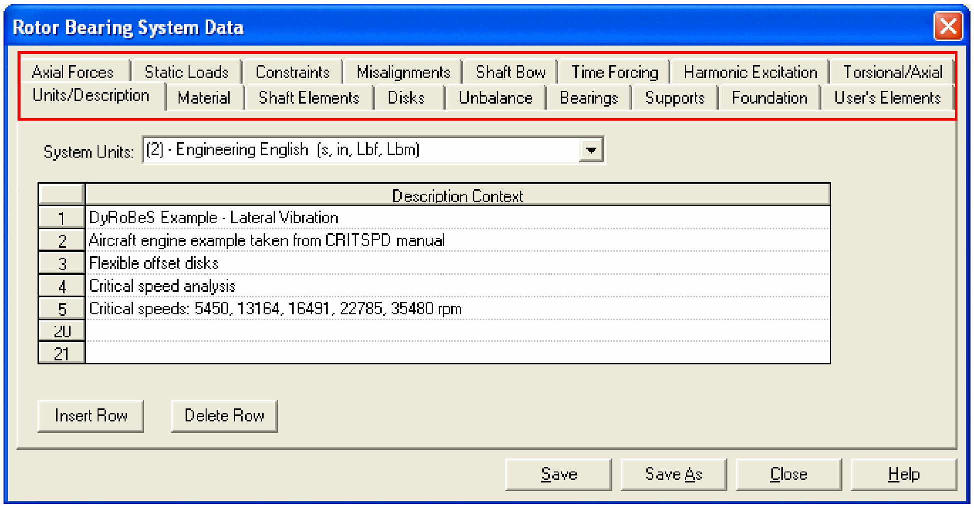
Use this option to add and modify your materials. Use the ***List*** to select a material, then enter the material data. Click ***Update*** to save the material data into the material library.



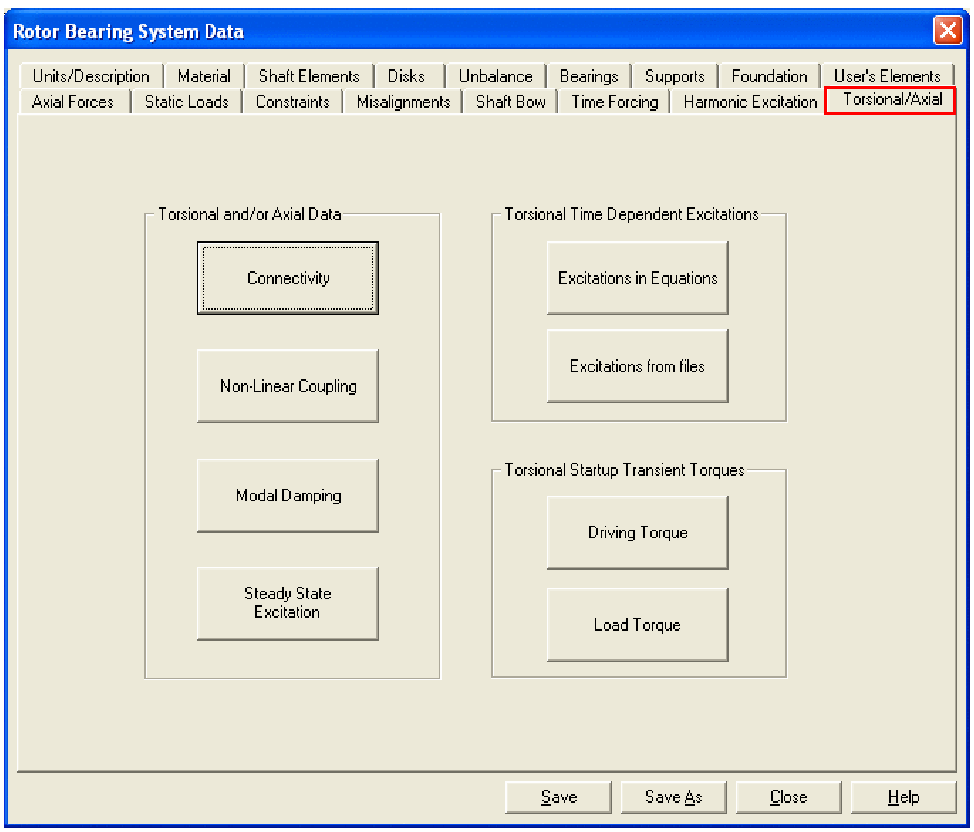
Modeling/Data Editor

Before building the model, it is important to know the coordinate system. **Z-axis is the spinning axis in DyRoBeS.** The lateral vibration of the discretized system is described by two translational (x, y) and two rotational (θ_x, θ_y) coordinates at each finite element station. For torsional vibration, the motion of each finite element station is described by a rotational displacement (θ_z) about the spinning axis. For axial vibration, the motion of each finite element station is described by a translational displacement (z) along the spinning axis.

A number of folders, as shown below, with different tabs are provided in the Data Editor and you can enter your rotor bearing system data accordingly. The description for each folder will be explained in the following sessions.



There are more tabs (buttons) under the Torsional/Axial Tab as shown below.



Since the same data file can be used in the lateral, torsional, and axial vibrations, data in the irrelevant folder will not be used in a particular analysis. The usage of the data folder is listed below for reference.

Data Utilization for Lateral, Torsional, and Axial Analysis

Folder	Lateral	Torsional	Axial
Units/Description	Yes	Yes	Yes
Material	Yes	Yes	Yes
Shaft Elements	Yes	Yes	Yes
Disks	Yes	Yes	Yes
Unbalance	Yes	No	No
Bearings	Yes	No	No
Supports	Yes	No	No
Foundation	Yes	No	No
User's Elements	Yes	No	No
Axial Forces	Yes	No	No
Static Loads	Yes	No	No
Constraints	Yes	No	No
Misalignments	Yes	No	No
Shaft Bow	Yes	No	No
Time Forcing Functions	Yes	No	No
Harmonic Excitations	Yes	No	No
Torsional/Axial Tab			
Connectivity	No	Yes	Yes
Non-Linear Coupling	No	Yes	No
Modal Damping	No	Yes	Yes
Steady State Excitation	No	Yes	Yes
Excitations in Equations	No	Yes	No
Excitations from Data Files	No	Yes	No
Startup - Driving Torque	No	Yes	No
Startup - Load Torque	No	Yes	No

It should be noted that the same model will be used in lateral, torsional, and axial analyses, even though some of the inputs are not required for certain analyses and these entries will be ignored.

Since saving data is just a click on the **Save** button, it is a good practice to save your data from time to time to prevent any data loss. If you wish to save a different model without changing the original data, use **Save As** to save the data into another file,

See also:

Data Editor

Unit Systems

Description

Material

Shaft Elements

Rigid/Flexible Disks

Unbalance

Bearings and Seals

0-Linear constant bearing

1-Speed dependent bearing

2-Bearing coefficients from external data file

3-Pseudo bearing

4-Squeeze film damper

5-Plain journal bearing

6-Generalized non-linear isotropic bearing

7-Active magnetic bearing (Linear Model)

8-Active magnetic bearing (Nonlinear Model)

9-Floating ring bearing

Unit Systems

The data input is not restricted to a specific system of units. In addition to the consistent units of user's choice, four sets of system of units have been introduced in this version:

Unit = 0 => Consistent Units of User's Choice

Unit = 1 => Consistent English Units (sec, in, Lbf, Lbf-s^2/in)

Unit = 2 => Engineering English Units (sec, in, Lbf, Lbm)

Unit = 3 => Consistent SI Units (sec, m, N, kg)

Unit = 4 => Engineering Metric Units (sec, mm, N, kg)

For Unit=0, consistent units, it is essential that all of the input parameters be expresses in a consistent set of units of the user's choice. For Unit=1 and 3, these are two sets of consistent units, however, their units will be displayed on the input screen and in the output text files. For Unit=2 and 4, these are two sets of units that are commonly used in the engineering field. Since they are not consistent units, their input units are explicitly specified on the input screen. Users must pay attention to their units.

Units Systems	Consistent English Unit = 1	Engineering English Unit = 2	Consistent SI Unit = 3	Engineering Metric Unit = 4	
Basic Quantities					
Time	Second (s)	Second (s)	Second (s)	Second (s)	
Length	in	in	m	**mm**	
Force	Lbf	Lbf	Newton (N)	Newton (N)	
Mass	Lbf-s^2/in	Lbm	kg = N-s^2/m	**kg**	
Inputs					
Material Properties					
Density	Lbf-s^2/in^4	**Lbm/in^3**	kg/m^3	**kg/m^3**	
Modulus	Lbf/in^2 (psi)	Lbf/in^2 (psi)	N/m^2 (Pa)	N/mm^2 (MPa)	
Shaft Elements					
Length, Diameter	in	in	m	mm	
Disks					
Mass	Lbf-s^2/in	Lbm	kg	**kg**	
Inertia	Lbf-s^2-in	**Lbm-in^2**	kg-m^2	**kg-m^2**	
Skew Angle	degree	degree	degree	degree	
Unbalance					
Imbalance (*me*)	Lbf-s^2	**oz-in**	kg-m	**kg-mm**	
Angle	degree	degree	degree	degree	
Flexible Supports					
Mass	Lbf-s^2/in	Lbm	kg	**kg**	
Damping	Lbf-s/in	Lbf-s/in	N-s/m	N-s/mm	
Stiffness	Lbf/in	Lbf/in	N/m	N/mm	
Forces/Moments					
Forces	Lbf	Lbf	N	N	
Moments/Torque	Lbf-in	Lbf-in	N-m	N-mm	
Misalignment/Bow					
Deflection: x, y	in	in	m	mm	
Theta: x, y	degree	degree	degree	degree	
Time Forcing					
Force or Moment	Lbf Lbf-in	Lbf Lbf-in	N N-m	N N-mm	
Time Constant	1/s	1/s	1/s	1/s	
Excitation Freq.	rpm	rpm	rpm	rpm	
ϕ - Phase	degree	degree	degree	degree	
Bearings					
Stiffness – Kt	Lbf/in	Lbf/in	N/m	N/mm	
Damping – Ct	Lbf-s/in	Lbf-s/in	N-s/m	N-s/mm	
Stiffness – Kr	Lbf-in/rad	Lbf-in/rad	N-m/rad	N-mm/rad	
Damping – Cr	Lbf-in-s/rad	Lbf-in-s/rad	N-m-s/rad	N-mm-s/rad	
Length	in	in	m	mm	
Lubricant Viscosity	Reyn (Lbf-s/in^2)	Reyn (Lbf-s/in^2)	Pascal-second (N-s/m^2)	CentiPoise =1.0E03 * Pa-s	
Lubricant Density	Lbf-s^2/in^4	**Lbm/in^3**	kg/m^3	**kg/m^3**	
Pressure Drop Across seal	psi	psi	Pa	Bar =1.0E-05 * Pa	

Linear PID + Low Pass Filter Active Magnetic Bearings					
Proportional Gain	Lbf/in	Lbf/in	N/m	N/mm	
Integral Gain	Lbf/(in-s)	Lbf/(in-s)	N/(m-s)	N/(mm-s)	
Derivative Gain	Lbf-s/in	Lbf-s/in	N-s/m	N-s/mm	
Cut-Off Freq.	Hz	Hz	Hz	Hz	
NonLinear Transient Active Magnetic Bearings					
Proportional Gain	A/in	A/in	A/m	A/mm	
Integral Gain	A/(in-s)	A/(in-s)	A/(m-s)	A/(mm-s)	
Derivative Gain	A-s/in	A-s/in	A-s/m	A-s/mm	
Force Constant	Lbf-in^2/A^2	Lbf-in^2/A^2	N-m^2/A^2	N-mm^2/A^2	
Air Nominal Gap	in	in	m	mm	
Current	A	A	A	A	
Torsional					
Stiffness – K	Lbf-in/rad	Lbf-in/rad	N-m/rad	N-mm/rad	
Damping – C	Lbf-in-s/rad	Lbf-in-s/rad	N-m-s/rad	N-mm-s/rad	
Axial					
Stiffness – K	Lbf/in	Lbf/in	N/m	N/mm	
Damping – C	Lbf-s/in	Lbf-s/in	N-s/m	N-s/mm	
Gravity - g_o	386.088 in/s^2	386.088 in/s^2	9.8066 m/ s^2	9806.6 mm/s^2	
Outputs					
Displacements	in	in	m	mm	
Velocity	in/s	in/s	m/s	mm/s	
Acceleration	in/s^2	in/s^2	m/s^2	mm/s^2	
Rotations	rad	rad	rad	rad	
Force	Lbf	Lbf	N	N	
Moments/Torque	Lbf-in	Lbf-in	N-m	N-mm	

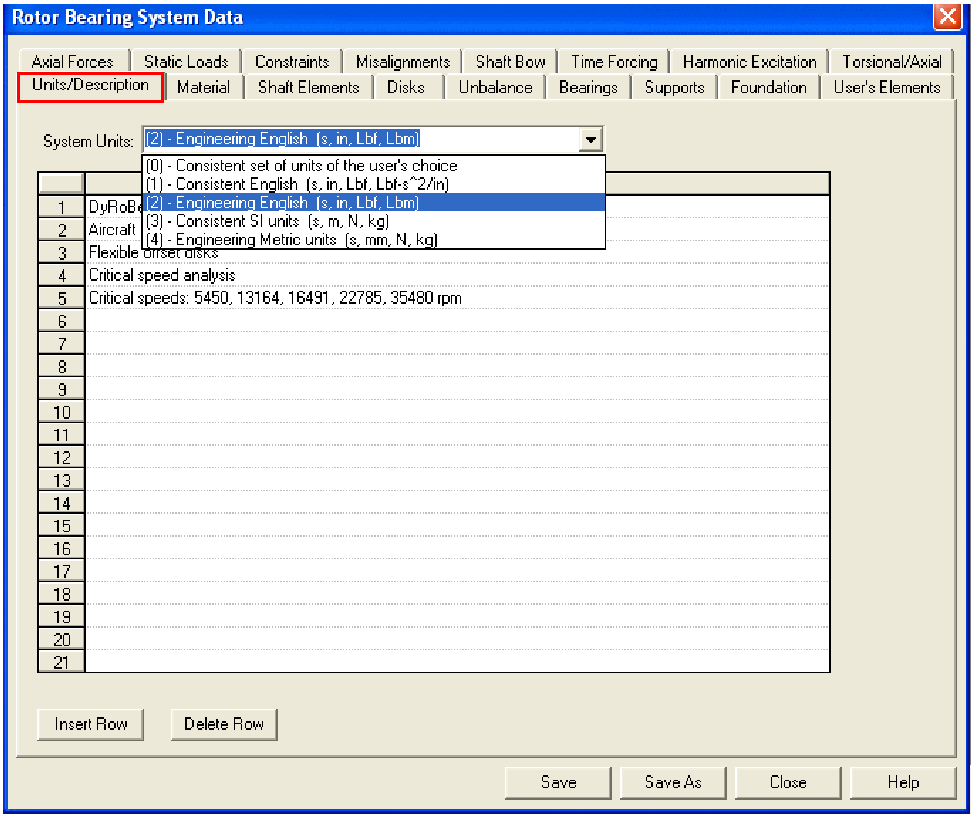
The conversions between English and Metric units are list below for quick reference.

Unit	English	Metric (SI)	Conversions (* = multiply)
Time	second (s)	second (s)	
Length	in	Meter (m)	m = 0.025400 * in
	in	mm	mm = 25.4 * in
Force	Lbf	Newton (N) 1N = 1kg * 1m/s^2	N = 4.448222 * Lbf N = 9.8066 * kgf kgf = 0.4535924 * Lbf
Moment	Lbf-in	N-m	N-m = 0.1129846 * Lbf-in
Mass	Lbf-s^2/in	kg = N-s^2/m	kg = 0.4535924 * Lbm kg = 175.1266 * Lbf-s^2/in Lbm = 386.088 * Lbf-s^2/in Lbm = 2.20462 * kg
Density	Lbf-s^2/in^4	kg/m^3	kg/m^3 = 2.767990E+04 * Lbm/in^3 kg/m^3 = 1.068688E+07 * Lbf-s^2/in^4
			g/cm^3 = 1 * g /cc g/cm^3 = 2.767990E+01 * Lbm/in^3 Lbm/in^3 = 0.0361273 * g/cm^3
Inertia	Lbf-s^2-in	kg-m^2	kg-m^2 = 0.1129846 * Lbf-s^2-in
Moduli	Lbf/in^2 (psi)	N/m^2 (Pa) kN/ m^2 (kPa)	Pa = 6.894757E+03 * psi kPa = 6.894757 * psi
Lateral Kt	Lbf/in	N/m	N/m = 175.1266 * Lbf/in N/mm = 0.1751266 * Lbf/in
Lateral Ct	Lbf-s/in	N-s/m	N-s/m = 175.1266 * Lbf-s/in N-s/mm = 0.1751266 * Lbf-s/in
Torsional K	Lbf-in/rad	N-m/rad	N-m/rad = 0.1129846 * Lbf-in/rad N-mm/rad = 112.9846 * Lbf-in/rad
Gravity - g	386.088 in/s^2	9.8066 m/s^2	
Temperature	°F	°C	°C = (°F-32) * 5/9
Viscosity	Reyn (Lbf-s/in^2)	centiPoise	cP = 6.894757E06 * Reyn
			cP = 1.0E03 * Pa-s cP = g/cc * cSt (mm^2/s)
Flow rate	gpm (gal/min)	m^3/hour	m^3/hour = 0.2271 * gpm liter/min = 3.785412 * gpm liter/min = 16.66667 * m^3/hour
Power	hp	kWatt	kWatt = 0.7457 * hp
Unbalance			oz-in = 0.03527 * g-in g-in = 28.35 * oz-in

Unit/Description

When creating a rotor model, the first thing needs to do is to select the unit system. For details on unit systems, click Unit System.

The descriptions are used to describe the system to be modeled or the analysis to be performed. The purpose of using the description headers is to provide explanation for you or another person who might need to see or modify this data file. It is highly recommended, as a good engineering practice, to have adequate descriptions to document the model. These descriptions can also be displayed on the graphical output. Number of description lines shown in the model display is specified in the **Graphic Preferences Settings** under the Project menu.



Material Properties

The various shaft element materials used in the model are entered in this data entry form. Different colors are used to represent different materials in the system configuration plot. The colors are defined in the **Graphic Preferences Settings** under the Project menu. The material property number is used in the Shaft Elements data entry. Failure to specify a material will result in an error message. The material properties of steel are listed for reference:

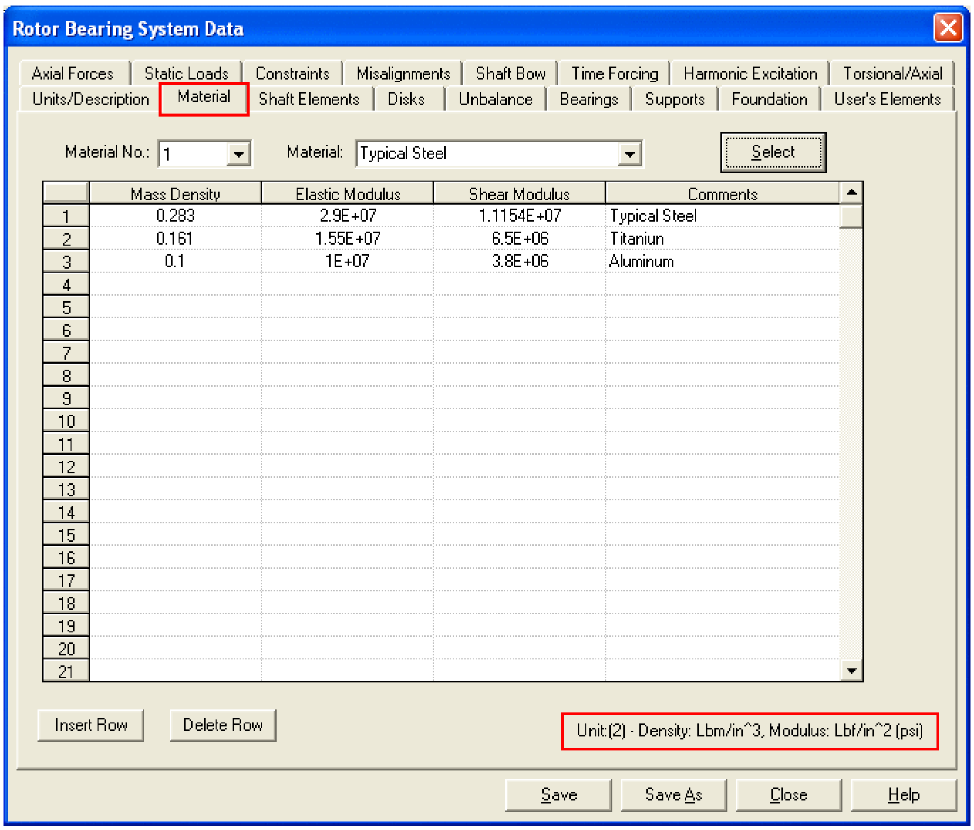
Mass Density (ρ) = 7.329E-04 Lbf-s^2/in^4 = 7832 kg/m^3 = 0.283 Lbm/in^3

Elastic Modulus (E) = 29.0E+06 Lbf/in^2 = 20.0E+10 N/m^2 (Pascal)

Shear Modulus (G) = E / 2.6

You can type in the material properties for each material number or you can utilize the material library to fill in the material properties. If you want to use the library, specify the material number first, pick the material, then click ***Select*** to select the material. In the right bottom of the input screen, the proper units for the specified unit system are displayed for reference.

See also Unit System, Material Library

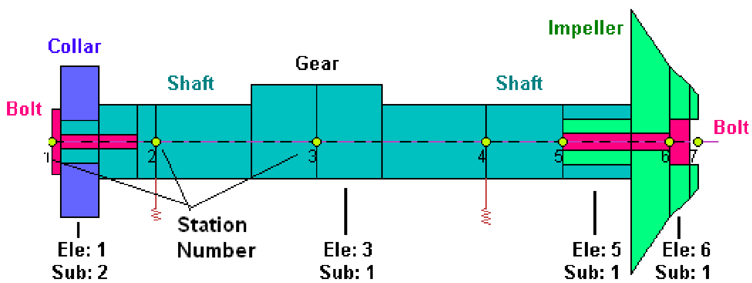


Shaft Elements

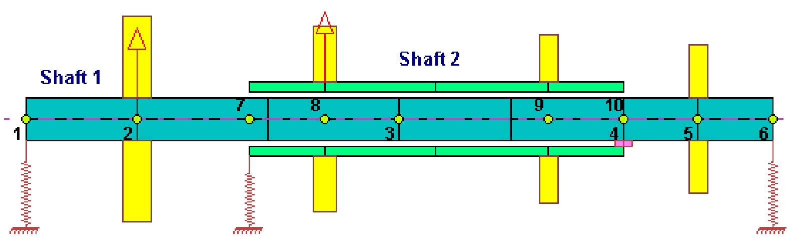
The shafts of the rotor system are numbered consecutively from 1 to *Ns* and the finite element stations of the model are numbered consecutively starting with 1 at the left end of shaft 1 and continuing to the last station at the right end of shaft *Ns*. The shafts are made up of Elements with the numbering for each shaft starts at the left end. **Stations are located at the ends of the Elements.** The term Station is commonly used in the rotordynamics instead of the term Node which is generally used in finite element literature because of the alternate meaning that Node has in the vibration mode shapes. Three types of elements are included in ***DyRoBeS*©** and they are: Cylindrical Element, Conical (Tapered) Element, and User's Supplied Element.

Element *i* is located immediately to the right of station *i*. Each element may possess several subelements (starting with 1 at the left of the element) thereby allowing for reasonable flexibility in modeling systems with several geometric discontinuities. Each subelement may possess several levels or layers (starting with zero). The use of subelement is strongly encouraged in the modeling. This will save tremendous computational time with little loss of accuracy in the result. However, when using subelements, it should be kept in mind that Disks and Bearings can only be placed at the ends of the elements (stations).

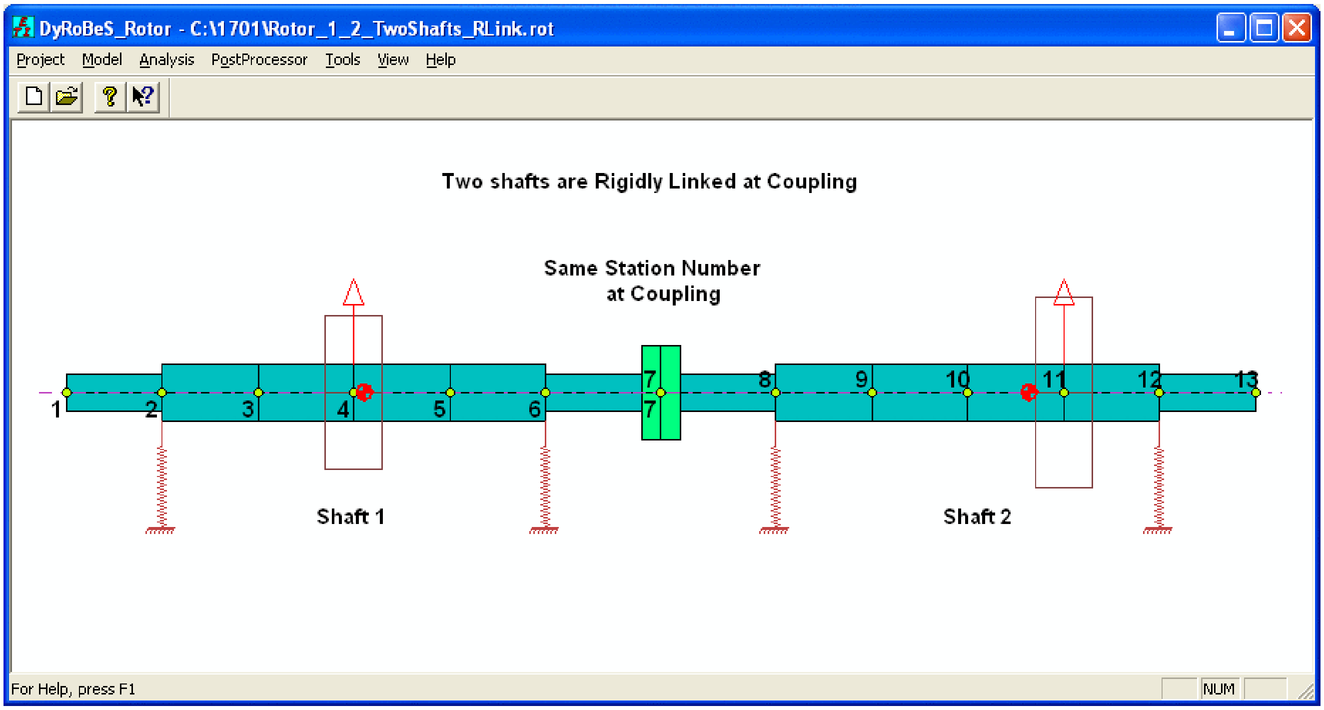
Note that an element does not exist for the last station of each shaft. For multi-shaft systems, the station numbers are consecutive and the element numbers are not continuous in general. However, in Ver 17. it allows a rigid link between two shafts with the same rotor station and rotor speeds. That means, the starting station of the second shaft has the same station number as the last station of the first shaft. This provides some modeling flexibility for larger rotor system with multiple shafts connected by couplings, such as large turbine-generator sets.



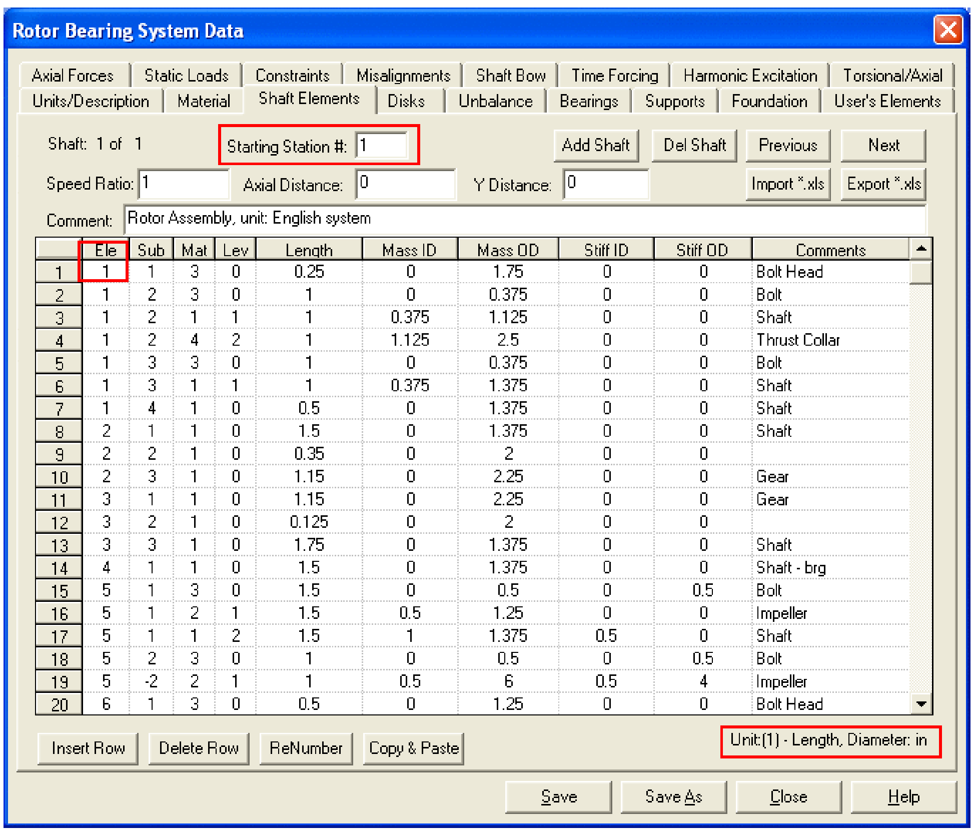
Dual rotor system, two shafts are connected by bearings.



Two-shafts system, two shafts are Rigidly Linked at coupling station.

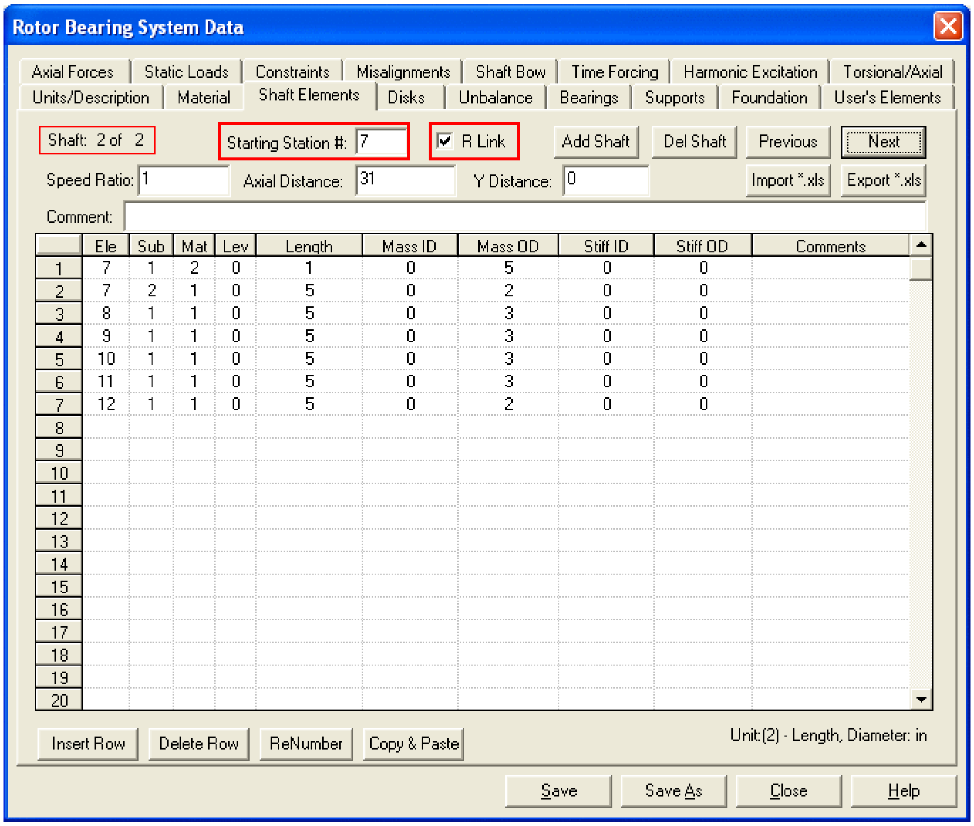


The Shaft Elements input page is shown below:



	Ele	Sub	Mat	Lev	Length	Mass ID	Mass OD	Stiff ID	Stiff OD	Comments
1	1	1	3	0	0.25	0	1.75	0	0	Bolt Head
2	1	2	3	0	1	0	0.375	0	0	Bolt
3	1	2	1	1	1	0.375	1.125	0	0	Shaft
4	1	2	4	2	1	1.125	2.5	0	0	Thrust Collar
5	1	3	3	0	1	0	0.375	0	0	Bolt
6	1	3	1	1	1	0.375	1.375	0	0	Shaft
7	1	4	1	0	0.5	0	1.375	0	0	Shaft
8	2	1	1	0	1.5	0	1.375	0	0	Shaft
9	2	2	1	0	0.35	0	2	0	0	
10	2	3	1	0	1.15	0	2.25	0	0	Gear
11	3	1	1	0	1.15	0	2.25	0	0	Gear
12	3	2	1	0	0.125	0	2	0	0	
13	3	3	1	0	1.75	0	1.375	0	0	Shaft
14	4	1	1	0	1.5	0	1.375	0	0	Shaft - brg
15	5	1	3	0	1.5	0	0.5	0	0.5	Bolt
16	5	1	2	1	1.5	0.5	1.25	0	0	Impeller
17	5	1	1	2	1.5	1	1.375	0.5	0	Shaft
18	5	2	3	0	1	0	0.5	0	0.5	Bolt
19	5	-2	2	1	1	0.5	6	0.5	4	Impeller
20	6	1	3	0	0.5	0	1.25	0	0	Bolt Head

Following shows the second shaft is rigidly linked to the first shaft. This input screen is not related to the previous screen input.



The control buttons ***<Add Shaft>***, ***<Del Shaft>***, ***<Previous>***, and ***<Next>*** allow you to add a new shaft data, delete an existing shaft, and switch the shaft data for multi-shaft systems. The ***<Import *.xls>*** and ***<Export *.xls>*** buttons allow you to import and export the shaft data from and to the MS Excel file. You can manipulate the shaft data in MS Excel and then Import into DyRoBeS. The ***<Insert Row>*** and ***<Delete Row>*** allow you to insert a new row or delete a row. **<ReNumber>** allows you to re-number the element and subelement numbers. A new element number is always started with a subelement number 1. **<Copy & Paste>** allows you to copy and paste row data. Standard *Windows* commands ***<Ctrl+C>*** and ***<Ctrl+V>*** can be used to copy and paste a single data field. In the right bottom of the tab, the proper units for the specified unit system are displayed for reference.

The following data fields are explained:

1.Starting Station #: The starting station (also element) number of the current shaft.

Starting Station = 1 for shaft 1. This number is updated automatically and is used to remind the users of the starting element number of that particular shaft. In general, if two shafts are connected by bearings and not rigidly linked together, then the second shaft starting station number (for example 7) is the first shaft last station number (for example 6) plus 1 (6+1=7) in the dual rotor system example shown above. In this dual rotor system, the station number is consecutive (from 6 to 7), but the element number will be not continuous (from 5 to 7). If two rotors are rigidly linked together at coupling with the same rotating speeds as shown in the above figure, then the second shaft starting station number will be the same as the last station number of the first shaft (for example 7 in the above figure). In this case, the station number repeats once (for example 7) and the element number are continuous.

1a.R Link: If the current shaft is Rigidly Linked to the previous shaft, then check this box to avoid the checking features provided in previous version.

2.Speed Ratio: This field is used to calculate the speeds of the multi-shaft systems.

Speed Ratio = 1 for shaft 1. Or you can simply input the shaft rotational speeds,

the program will calculate the ratios for you. Use zero (0) for non-rotating structure.

Positive speed ratio indicates the rotor rotates CCW in the positive Z direction and

negative indicates CW rotation in the negative Z direction for multiple-shafts system.

3.Axial Distance: The distance measured from the station 1 of shaft 1 to the starting

station of the current shaft. It is used for geometric configuration plot only and

it does not affect the numerical results. Distance = 0 for shaft 1.

4.Y Distance: Again, this value is used for geometric plot only. Zeros indicate that the

shafts are concentric, non-zeros are used in the Torsional and Axial multi-branch systems to represent vertical offset caused by gears, etc..

5.Comment: Comments on the shaft.

6.Ele: Element number. The Element number in the first row must be equal to the

Starting Station #.

7.Sub: Subelement number. Each element can contain a max of 20 subelements. Positive value represents a cylindrical element and **negative** value indicates that this subelement is a conical (tapered) element.

8.Lev: Level for the subelement. Each subelement can have a max of 10 levels (layers). Level zero (0) is the core data.

9.Mat: Material number for this entry. Mat = 0 for User Supplied Subelement.

10.Length: Subelement length. For the same subelement number with different levels, the subelement length should be the same.

11.Mass ID: Mass inner diameter (For conical element: left end inner diameter).

12.Mass OD: Mass outer diameter (For conical element: left end outer diameter).

13.Stiff ID: Stiffness inner diameter (For conical element: right end inner diameter).

14.Stiff OD: Stiffness outer diameter (For conical element: right end outer diameter).

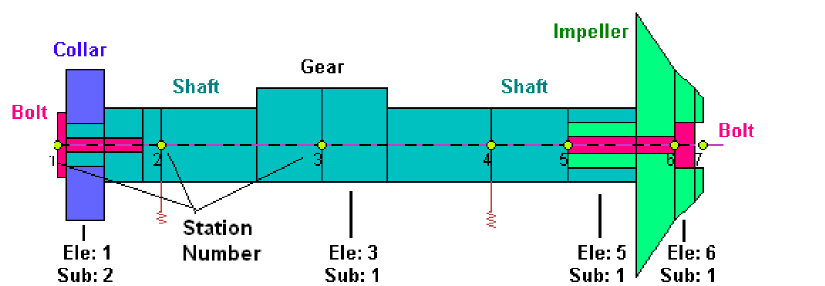
15.Comments: Description for this entry.

For a standard cylindrical element, the Mass ID and OD are used for kinetic energy (mass, gyroscopic matrices) calculation and the Stiffness ID and OD are used for potential energy (stiffness matrix) calculation. If Stiffness OD = 0, then the Stiffness ID and OD will be reassigned to be equal to the Mass ID and OD. This option can save data entry time if you decide to use the same model for the kinetic energy and potential energy calculation. In the configuration plot, the upper half represents the Mass model and the lower half represents the Stiffness model.

However, for a conical element, the Mass ID and OD are the LEFT end inner and outer diameters and the Stiffness ID and OD are the RIGHT end inner and outer diameters. The mass model and stiffness model are using the same geometry.

Example 1: Single shaft system

A single shaft model and the associated shaft data are shown below:

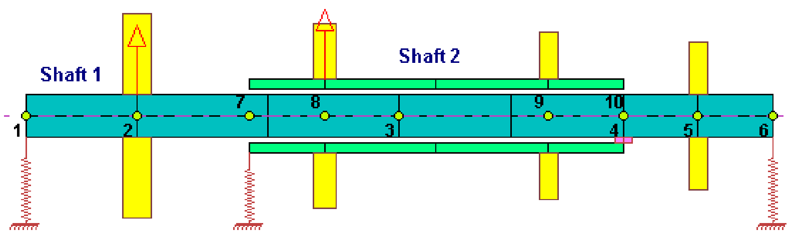


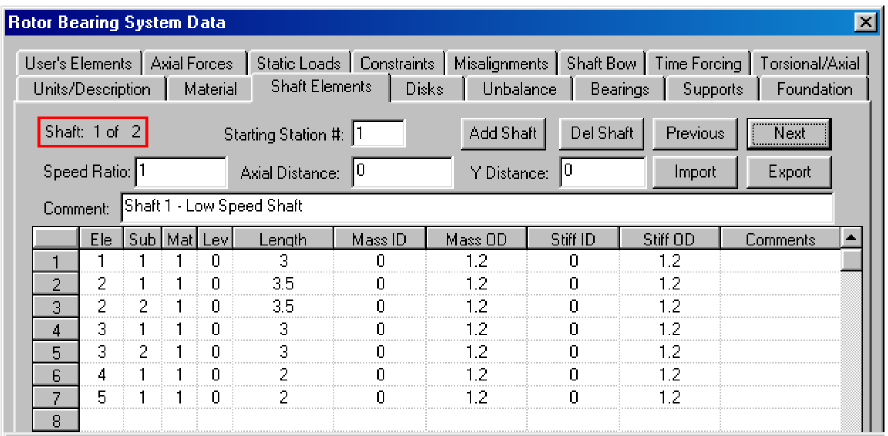
	Ele	Sub	Mat	Lev	Length	Mass ID	Mass OD	Stiff ID	Stiff OD	Comments
1	1	1	3	0	0.25	0	1.75	0	0	Bolt Head
2	1	2	3	0	1	0	0.375	0	0	Bolt
3	1	2	1	1	1	0.375	1.125	0	0	Shaft
4	1	2	4	2	1	1.125	4	0	0	Thrust Collar
5	1	3	3	0	1	0	0.375	0	0	Bolt
6	1	3	1	1	1	0.375	2	0	0	Shaft
7	1	4	1	0	0.5	0	2	0	0	Shaft
8	2	1	1	0	2.5	0	2	0	0	Shaft
9	2	2	1	0	1.7	0	3	0	2	Gear
10	3	1	1	0	1.7	0	3	0	2	Gear
11	3	2	1	0	2.75	0	2	0	2	Shaft
12	4	1	1	0	2	0	2	0	0	Shaft - brg
13	5	1	3	0	1.8	0	0.5	0	0.5	Bolt
14	5	1	2	1	1.8	0.5	1.25	0	0	Impeller
15	5	1	1	2	1.8	1.25	2	0.5	0	Shaft
16	5	2	3	0	1	0	0.5	0	0.5	Bolt
17	5	-2	2	1	1	0.5	7	0.5	4	Impeller
18	6	1	3	0	0.5	0	1.25	0	0	Bolt Head
19	6	-1	2	1	0.5	1.25	4	1.25	3	Impeller
20	6	-2	2	0	0.25	1.25	3	1.25	2.5	Impeller
21										

The model shown above has six (6) elements (7 stations). The substations between the major stations are not numbered. The different colors in the shaft elements represent the different material properties. A thrust collar in the left hand side and an impeller in the right hand side are attached to the shaft through bolts. Element 1 Subelement 2, and Element 5 Subelement 1 have three levels. Element 1 Subelement 3, Element 5 Subelement 2, and Element 6 Subelement 1 have two levels. As can be seen from the figure, the bearings are located at the major stations 2 and 4. A bearing cannot be applied at a substation. It is also observed that the rotor (gear portion) is not symmetric about the shaft centerline. The profile above the shaft centerline represents the rotor mass distribution, whereas the distribution below the centerline represents the rotor stiffness distribution.

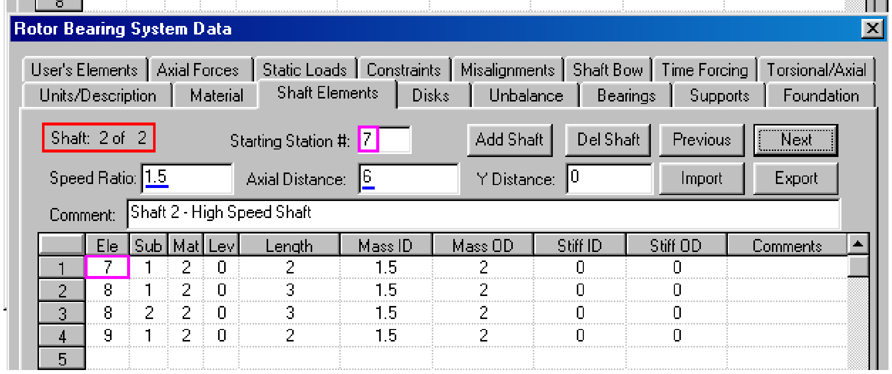
Example 2: Dual shafts system

A dual shafts model and the associated shaft data are shown below:





	Ele	Sub	Mat	Lev	Length	Mass ID	Mass OD	Stiff ID	Stiff OD	Comments
1	1	1	1	0	3	0	1.2	0	1.2	
2	2	1	1	0	3.5	0	1.2	0	1.2	
3	2	2	1	0	3.5	0	1.2	0	1.2	
4	3	1	1	0	3	0	1.2	0	1.2	
5	3	2	1	0	3	0	1.2	0	1.2	
6	4	1	1	0	2	0	1.2	0	1.2	
7	5	1	1	0	2	0	1.2	0	1.2	
8										

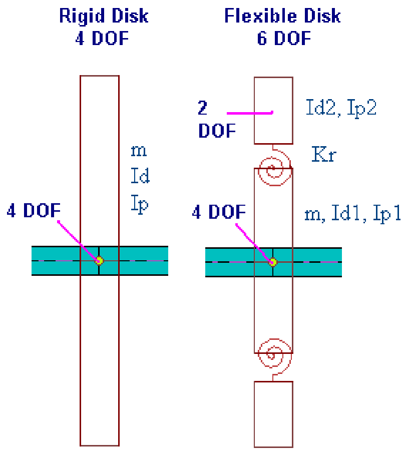


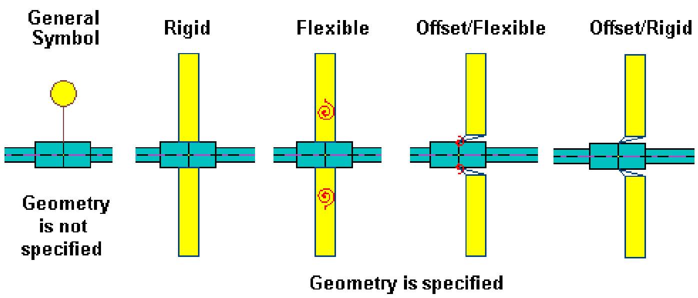
	Ele	Sub	Mat	Lev	Length	Mass ID	Mass OD	Stiff ID	Stiff OD	Comments
1	7	1	2	0	2	1.5	2	0	0	
2	8	1	2	0	3	1.5	2	0	0	
3	8	2	2	0	3	1.5	2	0	0	
4	9	1	2	0	2	1.5	2	0	0	
5										

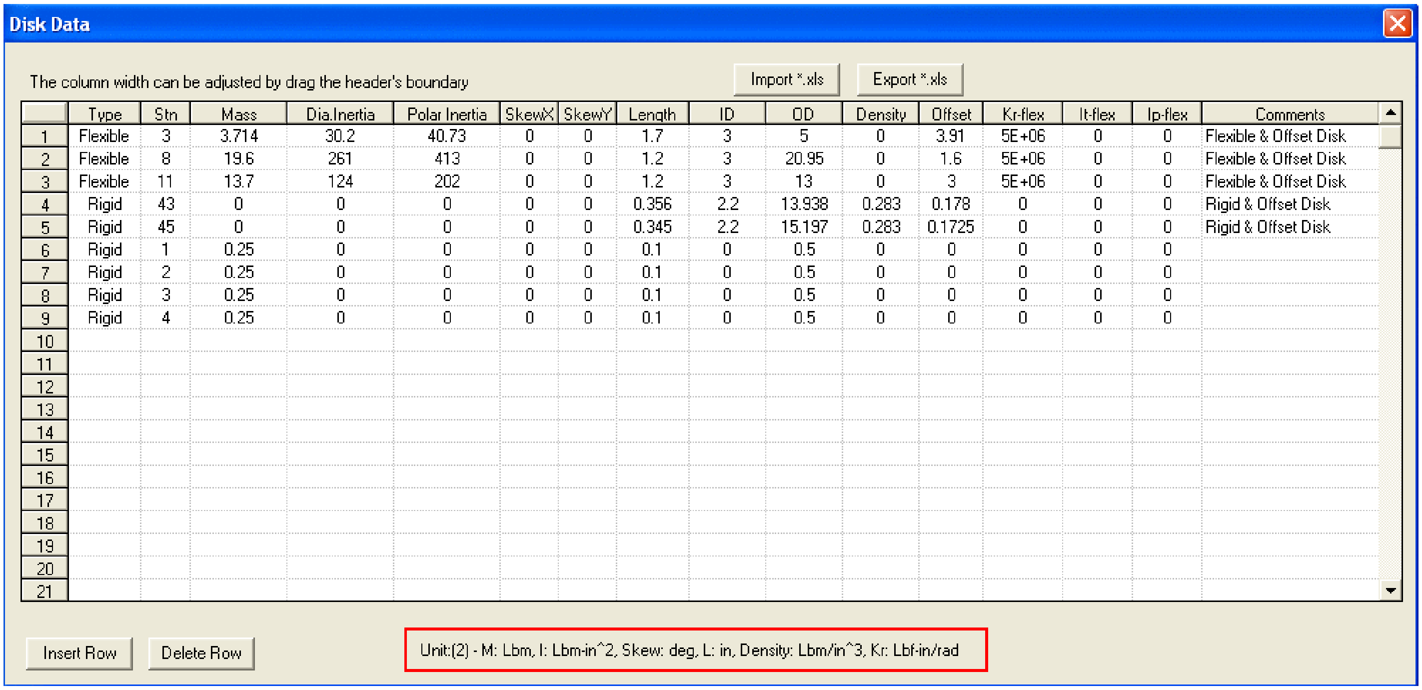
The shaft 1 starts from station 1 to station 6 and shaft 2 is from station 7 to station 10. To add the second shaft data, click the <**Add Shaft**> button. Note that an element does not exist for the last station of each shaft. The shaft 2 speed is 1.5 times the shaft 1 speed. Note that a positive speed ratio indicates shaft CCW rotation in the positive Z direction and negative indicates shaft CW rotation in the negative Z direction.

Rigid/Flexible Disks

Impellers, fans, turbine blades, collars, coupling hubs, etc, can all be modeled as disks. A concentrated disk can be located at any finite element station. Disks cannot be placed at a substation, and must be placed at stations. Multiple disks are allowed at the same station. For a rigid disk, there are four-degrees-of-freedom (dof), 2 translational and 2 rotational displacements, describing the motion of the disk. These dofs of a disk are the same as the dofs of the finite element station of the rotor (shaft) where the disk is attached to. However, for a flexible disk, two additional rotational dofs are introduced, that is, there is a total of 6 dofs for each flexible disk. The diametral and polar moment of inertias for the inner and outer disks are typically carefully adjusted to match the first disk diametral resonant frequency. Since the outer disk has only 2 rotational dofs and no translational dofs, it only possesses the moments of inertias. All the mass is lumped into the inner disk. The flexible disk option can be important for large overhung rotors, such as large gas turbines where the disk flexibility must be taken into consideration. Depending on the attachment method, the disk can also be offset from the attached station. The symbols used to represent the disk are shown below:







	Type	Stn	Mass	Dia.Inertia	Polar Inertia	SkewX	SkewY	Length	ID	OD	Density	Offset	Kr-flex	It-flex	Ip-flex	Comments
1	Flexible	3	3.714	30.2	40.73	0	0	1.7	3	5	0	3.91	5E+06	0	0	Flexible & Offset Disk
2	Flexible	8	19.6	261	413	0	0	1.2	3	20.95	0	1.6	5E+06	0	0	Flexible & Offset Disk
3	Flexible	11	13.7	124	202	0	0	1.2	3	13	0	3	5E+06	0	0	Flexible & Offset Disk
4	Rigid	43	0	0	0	0	0	0.356	2.2	13.938	0.283	0.178	0	0	0	Rigid & Offset Disk
5	Rigid	45	0	0	0	0	0	0.345	2.2	15.197	0.283	0.1725	0	0	0	Rigid & Offset Disk
6	Rigid	1	0.25	0	0	0	0	0.1	0	0.5	0	0	0	0	0	
7	Rigid	2	0.25	0	0	0	0	0.1	0	0.5	0	0	0	0	0	
8	Rigid	3	0.25	0	0	0	0	0.1	0	0.5	0	0	0	0	0	
9	Rigid	4	0.25	0	0	0	0	0.1	0	0.5	0	0	0	0	0	
10																
11																
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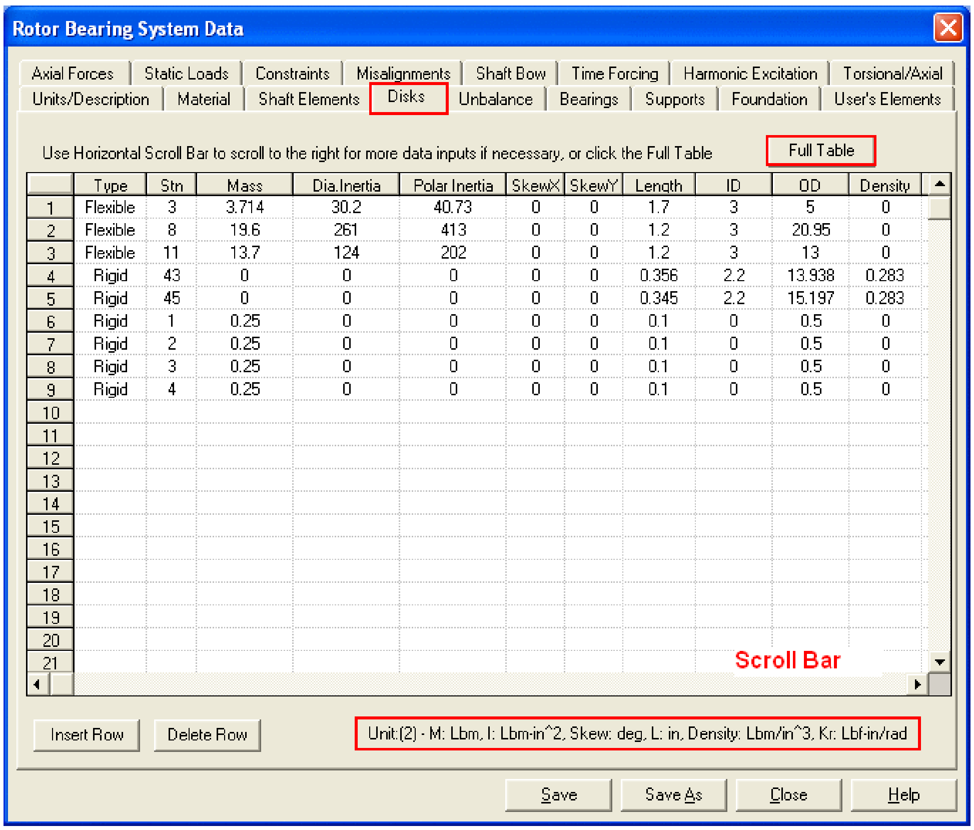

	Type	Stn	Mass	Dia.Inertia	Polar Inertia	SkewX	SkewY	Length	ID	OD	Density
1	Flexible	3	3.714	30.2	40.73	0	0	1.7	3	5	0
2	Flexible	8	19.6	261	413	0	0	1.2	3	20.95	0
3	Flexible	11	13.7	124	202	0	0	1.2	3	13	0
4	Rigid	43	0	0	0	0	0	0.356	2.2	13.938	0.283
5	Rigid	45	0	0	0	0	0	0.345	2.2	15.197	0.283
6	Rigid	1	0.25	0	0	0	0	0.1	0	0.5	0
7	Rigid	2	0.25	0	0	0	0	0.1	0	0.5	0
8	Rigid	3	0.25	0	0	0	0	0.1	0	0.5	0
9	Rigid	4	0.25	0	0	0	0	0.1	0	0.5	0
10											
11											
12											
13											
14											
15											
16											
17											
18											
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20											
21											

When entering the disks tab (input page), a full table is shown as above figure. You can enter all the data here for a full view. After this full table is closed, a small screen appears, which has the same size as other input pages. You can still edit the disk data using scroll bar in this smaller view. The control buttons ***<Import *.xls>*** and ***<Export *.xls>*** allow you to import and export the disk data from and to the MS Excel file. You can manipulate the disk data in MS Excel and then Import into DyRoBeS. The ***<Insert Row>*** and ***<Delete Row>*** allow you to insert a new row or delete a row. **<Full Table>** allows you to expand the table into a full input page.

The input data for disks are explained below:

1. Type: **R**igid or **F**lexible. Enter **R** or **F**.

2. Stn: Station number where the disk is located.

3. Mass: Disk mass.

4. Dia. Inertia: Diametral (transverse) moment of inertia. For a flexible disk, it is the inner disk's inertia.

5. Polar Inertia: Polar moment of inertia. For a flexible disk, it is the inner disk's inertia.

6. Skew x: Disk skew angle about x-axis in degree.

7. Skew y: Disk skew angle about y-axis in degree.

8. Length: Disk axial length.

9. ID: Disk ID.

10. OD: Disk OD. These values are used to graphically display the disk and may be used for the additional mass properties if the Disk **Density** is not zero.

If **Length** and **OD** are zero, then the disk is represented by a circle and a straight line connected to the station. If the **Length** and **OD** are not zero, the disk is plotted based on the given **Length**, **ID** and **OD**.

11. Density: Disk density. For a non-zero density entered, additional mass properties calculated based on the disk geometry (**Length**, **ID**, and **OD**) will be added into the total mass properties.

Note: for flexible disks, these additional mass properties will be added into the inner disk properties.

12. Offset: Non-zero value for the offset disk. Positive value if the disk is offset to the right and negative value if the disk is offset to the left. This value is required only when the disk is offset from the connecting rotor (shaft) station.

13. Rotational K: Rotational stiffness used to connect the inner and outer disks. Required only when the disk is flexible.

14. Id (outer): Diametral (transverse) moment of inertia of the outer disk. Used only when the disk is flexible

15. Ip (outer): Polar moment of inertia of the outer disk. Used only when the disk is flexible

16. Comment: Disk description.

The disk skew angles due to assembly are entered in this folder. However, the total disk skew angles are the combination of the skew caused by assembly and the skew caused by Shaft Bow. For flexible disks, the skew angles are applied on both inner and outer disks. A convenient tool for the calculation of mass properties of a homogeneous solid is provided under Tools menu. You can use this tool to calculate the mass properties of a cylindrical or tapered disk. In the right bottom of the screen, the proper units for the specified unit system are displayed for reference.

For details of using flexible disk and offset disk, see book "Practical Rotordynamics and Fluid Film Bearing Design", by W. J. Chen.

See also Mass/Inertia Properties Calculation

Unbalance

The mass unbalances of a rotating assembly are usually determined by using the multi-plane balancing machines. These mass unbalances are discrete and located at different planes with a magnitude of (*mass* * *eccentricity*). The unbalance planes may be located at the ends of each subelement in this program. These unbalance forces are assumed to be discrete and independent. Multiple unbalances are allowed at the same location.

For a mass unbalanced disk at station ***i*** with an unbalance **me**, this unbalance can be placed at the left end of element *i* with a magnitude of *me*. However, precautions must be taken for the unbalanced disk located at the right end (last station) of each shaft. Since there is no element number corresponding to the last station number on a shaft (explained in the Shaft Elements), this unbalance can be placed at the right end of the last subelement.

1. Ele: Element number.

2. Sub: Subelement number.

3. Type: Two types of synchronous unbalance excitations are allowed.

0 – Typical mass unbalance force, the mass unbalance force is (me x speed^2).

The data input for the amplitude is **me (**mass x eccentricity) without speed component.

1 – Constant magnitude synchronous excitation, this can be due to many other sources, such

as magnet force in the motor, etc.. The amplitude of this synchronous excitation is a

constant and independent from rotor speed. The data input for the amplitude **is Fo**.

4. Left Unb Amp.: Left end unbalance amplitude (*me* or Fo).

5. Left Ang: Left end unbalance phase angle measured from X-axis (degree).

6. Right Unb Amp: Right end unbalance amplitude (*me* or Fo).

7. Right Ang: Right end unbalance phase angle.

8. Comment: Description.

The unbalance force is represented by an arrow vector in the mass model. In the right bottom of the screen, the proper units for the specified unit system are displayed for reference. Some commonly used unit conversions for type 0 are listed for reference. For type 1, the unit for amplitude is force (Lbf or Newton).

1 g-in = 0.03527 oz-in

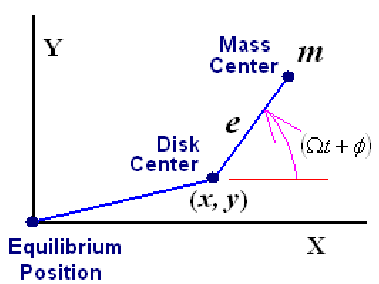
1 oz-in = 28.35 g-in

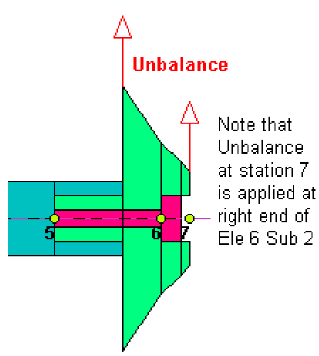
Unbalance Force in X direction

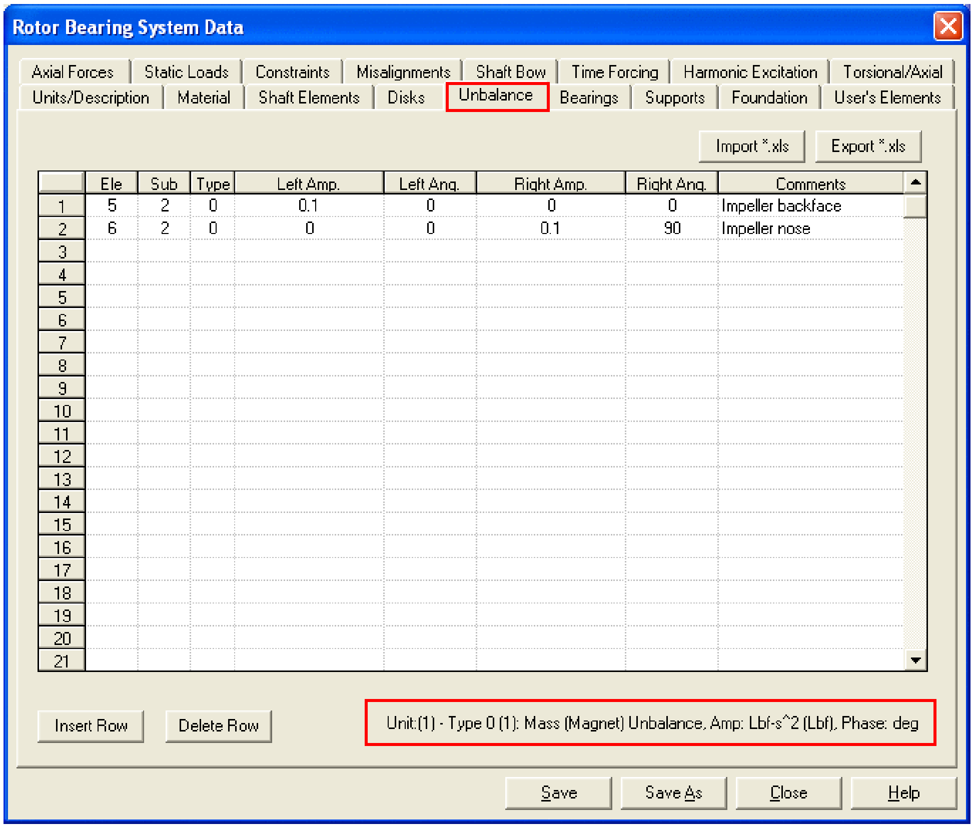
$$me\Omega^2 \cos(\Omega t + \phi) = me\Omega^2(\cos\phi \cdot \cos\Omega t - \sin\phi \cdot \sin\Omega t)$$

Unbalance Force in Y direction

$$me\Omega^2 \sin(\Omega t + \phi) = me\Omega^2(\sin\phi \cdot \cos\Omega t + \cos\phi \cdot \sin\Omega t)$$

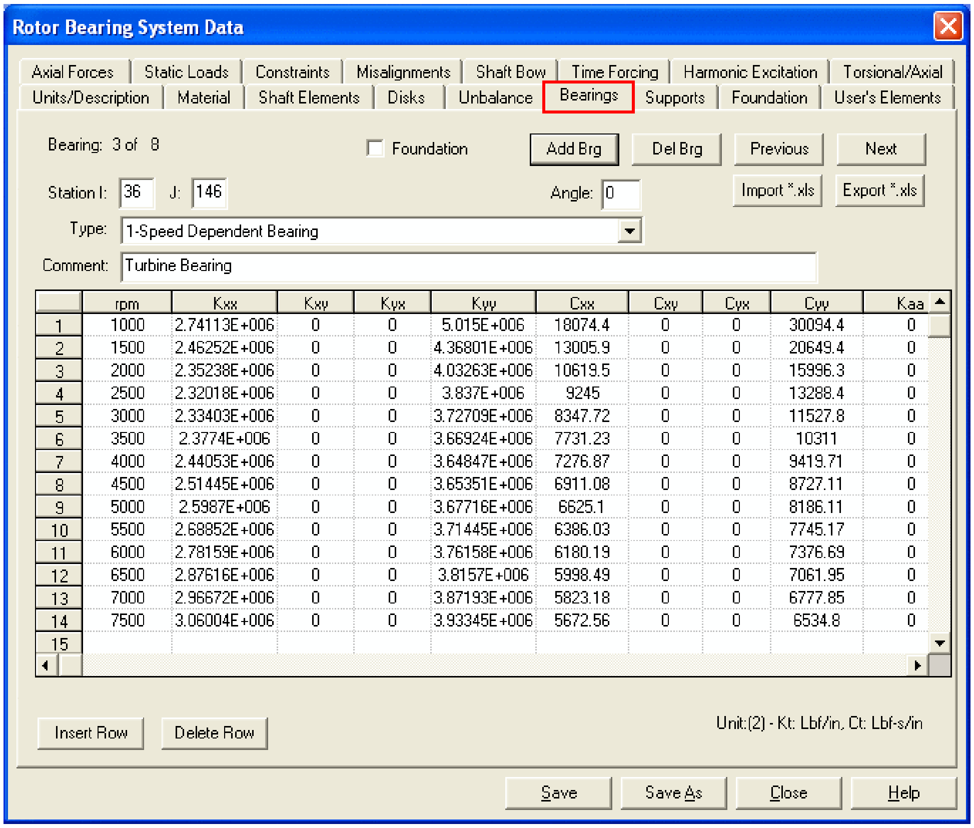





Rotor Bearing System Data
Axial Forces
Static Loads
Constraints
Misalignments
Shaft Bow
Time Forcing
Harmonic Excitation
Torsional/Axial
Units/Description
Material
Shaft Elements
Disks
Unbalance
Bearings
Supports
Foundation
User's Elements
Import *.xls
Export *.xls
Ele Sub Type Left Amp. Left Ang. Right Amp. Right Ang. Comments
1 5 2 0 0.1 0 0 0 Impeller backface
2 6 2 0 0 0 0.1 90 Impeller nose
Insert Row
Delete Row
Unit:(1) - Type 0 (1): Mass (Magnet) Unbalance, Amp: Lbf-s^2 (Lbf), Phase: deg
Save
Save As
Close
Help

Bearings

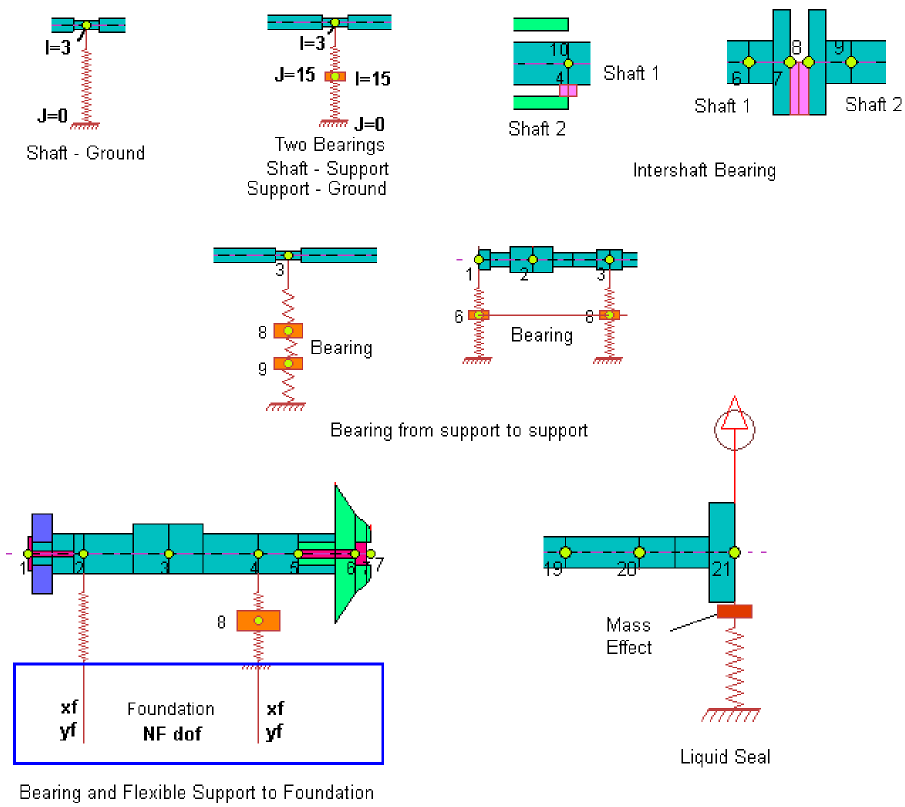
All the bearing/damper/seals/support forces acting on, or interacting between, the rotating assemblies and non-rotating structures fall into this category. Bearing is an interconnection component, which connects two finite element stations (station I and station J). The bearing can be of any type, such as linear/nonlinear, real/pseudo, fluid film, rolling elements, or active magnetic bearings, and aerodynamic forces, seals, fluid coupling, rubbing, etc. Nonlinear bearings can only be used in the time transient analysis, with the exception of squeeze film dampers and generalized non-linear isotropic bearing. Squeeze film dampers (Type-4) and non-linear isotropic bearings (Type-6) can be analyzed in both the steady state synchronous response analysis and time transient analysis. Different types of bearings or even same type bearing can be at the same finite element station depending upon the modeling technique. Numerous predefined bearing types are provided in the program. If you like to define your own bearing type, please contact the program developer for implementation.



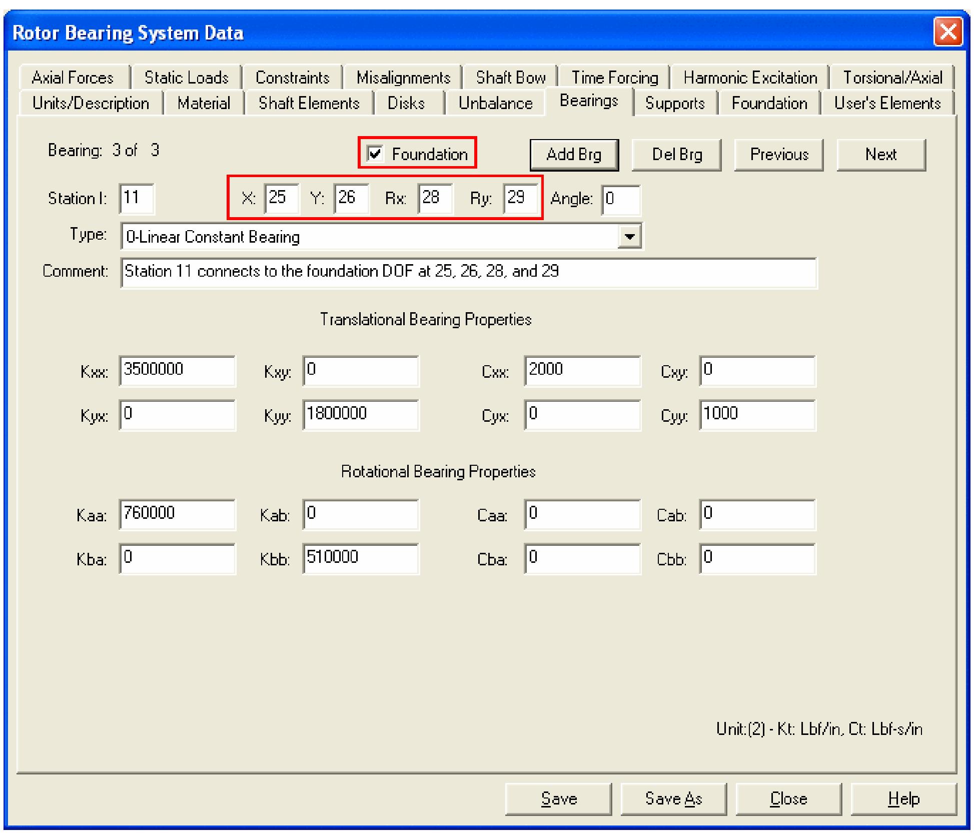
Similar to the Shaft Elements input, the control buttons ***<Add Brg>***, ***<Del Brg>***, ***<Previous>***, and ***<Next>*** allow you to add a new bearing data, delete an existing bearing, and switch the bearing data for systems with multiple bearings. The ***<Import *.xls>*** and ***<Export *.xls>*** buttons, shown only when bearing type 1 is selected, allow you to import and export the speed dependent bearing data from and to the MS Excel file.

Some common input data are explained below.

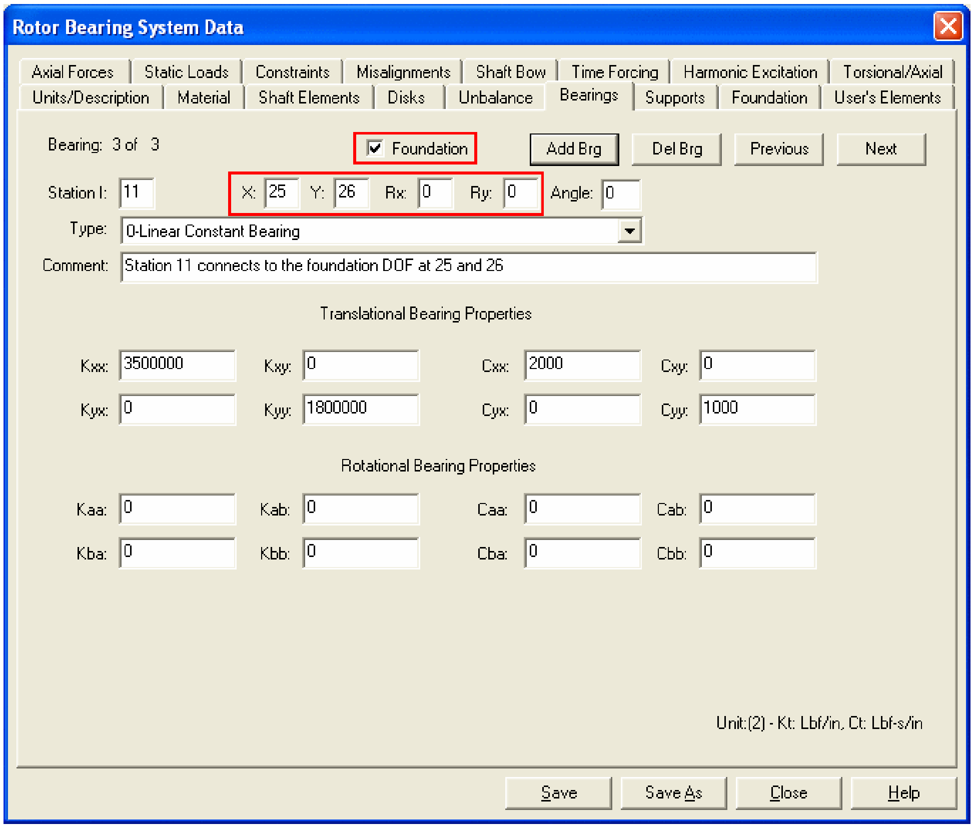
Station I and **Station J**: A bearing connects two stations. Station I cannot be zero. Station J = 0 if it is connected to the rigid ground. J can also be a connecting station number of an intershaft bearing in a multi-shaft system or the flexible bearing support station number for a flexible support system. Several configurations for the bearing connection are shown below.



Foundation: If the foundation box is checked, this bearing is connected to the foundation. The foundation data is provided under Foundation tab. If Foundation box is checked, additional inputs (X, Y, Rx, Ry) appear in the screen. The X, Y, Rx, and Ry are the degrees-of-freedom in the foundation. The bearing connects the degrees-of-freedom of the station I to the (X, Y, Rx, Ry) coordinates in the foundation, as shown below:



Similar to the Station I and J inputs, if the inputs for (X, Y, Rx, Ry) are zero, then this DOF is constrained and connected to the ground.



See Foundation for more detail.

Angle: Degree measured from the global fixed coordinate system to the local coordinate system where the bearing coefficients are defined. The angle is measured in the counterclockwise direction. The dynamic characteristics of the bearings are usually defined (specified) in the local coordinate system, therefore, a coordinate transformation is performed by the program to convert the bearing stiffnesses and dampings in the local coordinate system to the stiffnesses and dampings in the global coordinate system. In ***DyRoBeS***, it is recommended that this transformation be performed in ***BePerf*** for computational efficiency, not in ***Rotor*** here.

Comment: Description of this bearing.

Types: Currently, the following types of bearings are supported in the program and they are explained in the following sessions.

0-Linear constant bearing

1-Speed dependent bearing

2-Bearing coefficients from external data file

3-Pseudo bearing

4-Squeeze film damper

5-Plain journal bearing

6-Generalized non-linear isotropic bearing

7-Active magnetic bearing (Linear Model)

8-Active magnetic bearing (Nonlinear Model)

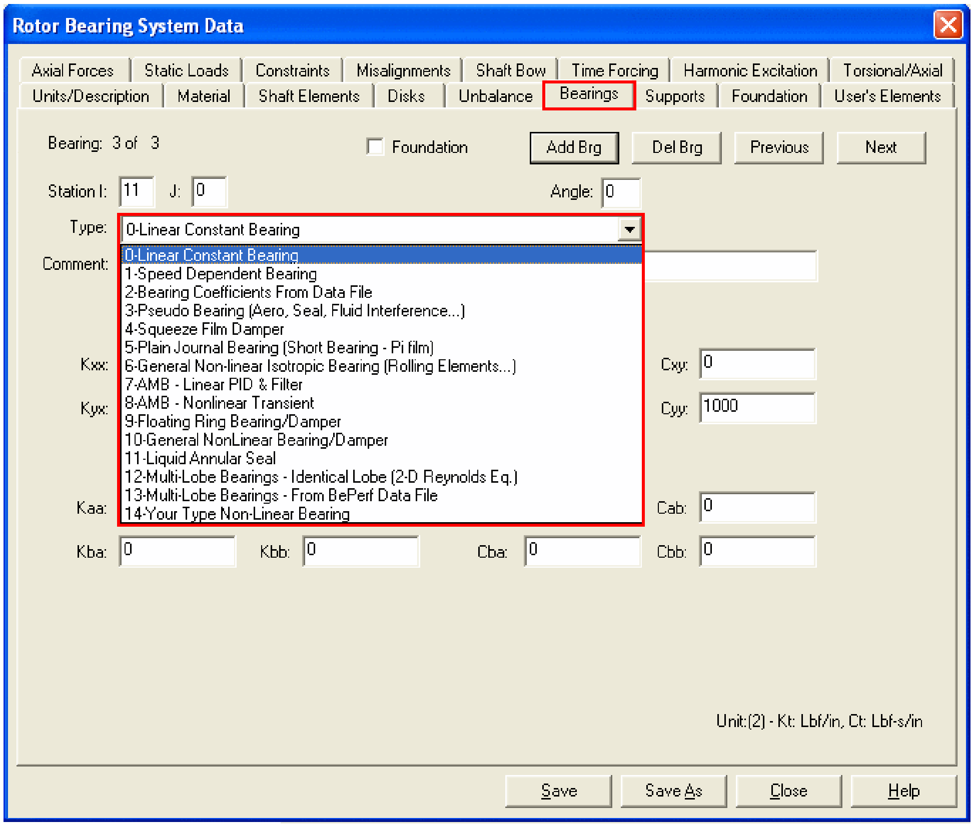
9-Floating ring bearing

10-General non-linear polynomial bearing

11-Liquid annular seal

12-Multi-Lobe hydrodynamic bearing (input data in Rotor)

13-Multi-Lobe hydrodynamic bearing (input data in BePerf)



For every different type bearing, the input screen is different. Only the related inputs for the selected type bearing are shown in the screen. Bearing inputs are explained below:

Linear Bearing

This option is used to specify linear, speed independent bearing coefficients. The linear bearing forces are of the form:

$$\begin{Bmatrix}F_x\\F_y\\M_x\\M_y\end{Bmatrix} = -\begin{bmatrix}K_{xx}&K_{xy}&0&0\\K_{yx}&K_{yy}&0&0\\0&0&K_{\alpha\alpha}&K_{\alpha\beta}\\0&0&K_{\beta\alpha}&K_{\beta\beta}\end{bmatrix}\begin{Bmatrix}x\\y\\\theta_x\\\theta_y\end{Bmatrix} - \begin{bmatrix}C_{xx}&C_{xy}&0&0\\C_{yx}&C_{yy}&0&0\\0&0&C_{\alpha\alpha}&C_{\alpha\beta}\\0&0&C_{\beta\alpha}&C_{\beta\beta}\end{bmatrix}\begin{Bmatrix}\dot{x}\\\dot{y}\\\dot{\theta}_x\\\dot{\theta}_y\end{Bmatrix}$$

or in matrix form:

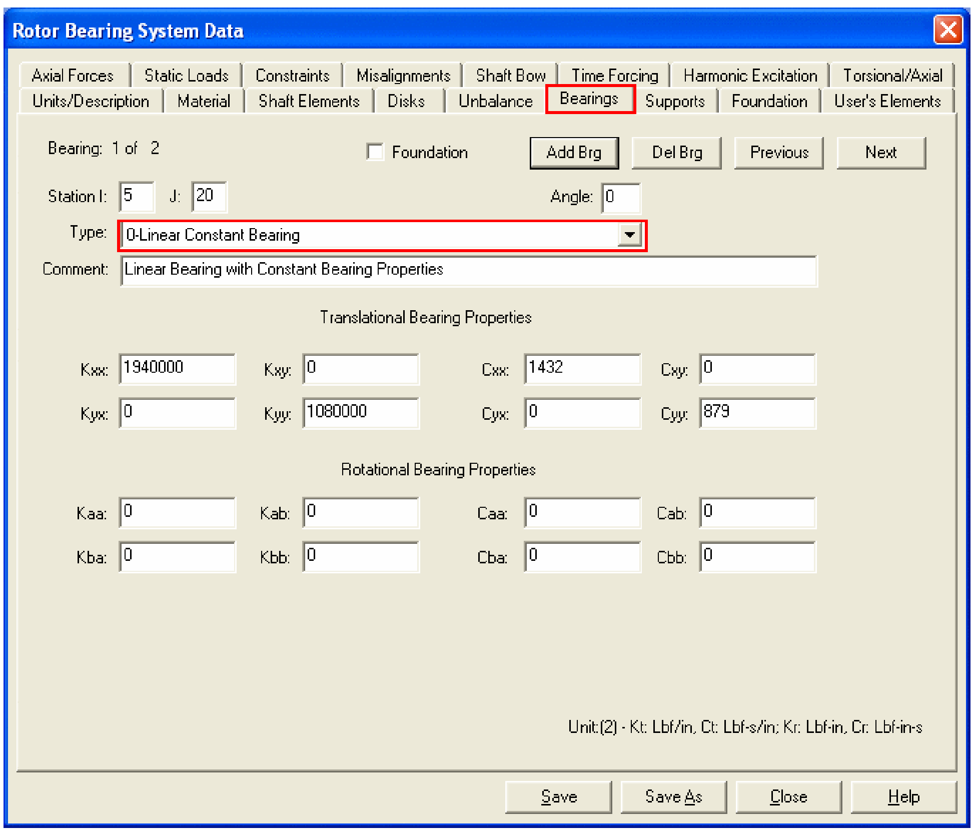
$$\mathbf{Q} = -\mathbf{K}\mathbf{q} - \mathbf{C}\dot{\mathbf{q}}$$

where

Kxx	Kxy	Kyx	Kyy	Translational stiffnesses
Cxx	Cxy	Cyx	Cyy	Translational dampings
Kαα	Kαβ	Kβα	Kββ	Rotational stiffnesses
Cαα	Cαβ	Cβα	Cββ	Rotational dampings

For bearings connecting station I and J, the bearing model becomes:

$$\begin{Bmatrix}\mathbf{Q}_i\\\mathbf{Q}_j\end{Bmatrix} = -\begin{bmatrix}\mathbf{K}&-\mathbf{K}\\-\mathbf{K}&\mathbf{K}\end{bmatrix}\begin{Bmatrix}\mathbf{q}_i\\\mathbf{q}_j\end{Bmatrix} - \begin{bmatrix}\mathbf{C}&-\mathbf{C}\\-\mathbf{C}&\mathbf{C}\end{bmatrix}\begin{Bmatrix}\dot{\mathbf{q}}_i\\\dot{\mathbf{q}}_j\end{Bmatrix}$$



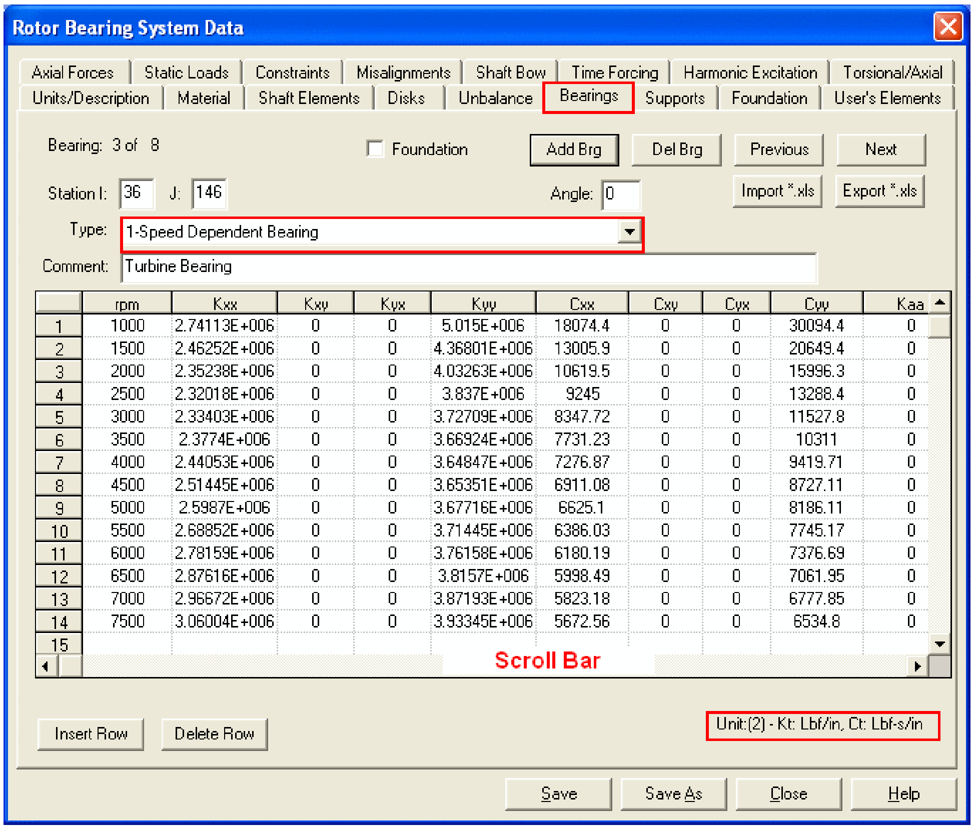
See also Bearings.

Speed (Frequency) Dependent Bearing

This option allows you to specify speed dependent linearized bearing coefficients. These coefficients are usually obtained from the bearing performance programs or experiments. A total of twenty-four (24) bearing dynamic coefficients may be provided for each shaft rotational speed. **The rotational speed of shaft 1 is used for the case of multi-shaft systems**. If the number of data (speed) points is greater than or equal to 3, a spline function is used in the program to interpolate the coefficients for the shaft speeds requested in the analysis that are not given in the data points. If the number of data (speed) points is equal to 2, a linear function is used to obtain the coefficients for the shaft speeds that are not given in the data points. If there is only one point in the data, then it will be treated as a linear constant bearing. The use of spline function allows a small number of needed data points. Since the spline function is used in the interpolation, the data must be entered in the increasing order according to the speed value. Also, the extrapolation of the data outside the data range can be very unpredictable. Caution must be taken when extrapolating the data. If the bearing/support properties are frequency dependent, for synchronous vibration, these frequency dependent coefficients may be entered as speed dependent coefficients.

The 24 bearing (or equivalent) coefficients are translational stiffness, damping, and mass coefficients, and rotational stiffness, damping, and inertia coefficients. Therefore, the data format is shown as follows:

rpm, Kxx,Kxy,Kyx,Kyy,Cxx,Cxy,Cyx,Cyy,Mxx,Mxy,Myx,Myy, Kaa,Kab,Kba,Kbb,Caa,Cab,Cba,Cbb,Maa,Mab,Mba,Mbb



	rpm	Kxx	Kxy	Kyx	Kyy	Cxx	Cxy	Cyx	Cyy	Kaa
1	1000	2.74113E+006	0	0	5.015E+006	18074.4	0	0	30094.4	0
2	1500	2.46252E+006	0	0	4.36801E+006	13005.9	0	0	20649.4	0
3	2000	2.35238E+006	0	0	4.03263E+006	10619.5	0	0	15996.3	0
4	2500	2.32018E+006	0	0	3.837E+006	9245	0	0	13288.4	0
5	3000	2.33403E+006	0	0	3.72709E+006	8347.72	0	0	11527.8	0
6	3500	2.3774E+006	0	0	3.66924E+006	7731.23	0	0	10311	0
7	4000	2.44053E+006	0	0	3.64847E+006	7276.87	0	0	9419.71	0
8	4500	2.51445E+006	0	0	3.65351E+006	6911.08	0	0	8727.11	0
9	5000	2.5987E+006	0	0	3.67716E+006	6625.1	0	0	8186.11	0
10	5500	2.68852E+006	0	0	3.71445E+006	6386.03	0	0	7745.17	0
11	6000	2.78159E+006	0	0	3.76158E+006	6180.19	0	0	7376.69	0
12	6500	2.87616E+006	0	0	3.8157E+006	5998.49	0	0	7061.95	0
13	7000	2.96672E+006	0	0	3.87193E+006	5823.18	0	0	6777.85	0
14	7500	3.06004E+006	0	0	3.93345E+006	5672.56	0	0	6534.8	0
15										

See also Bearings and Linear bearing

Bearing Coefficients from External Data File

This option allows the analysis program to import bearing coefficients from an external data file. The data filename must include the full path. It is recommended to use **Browse** button to select the file. The external bearing data file must be in the following ASCII format:

First Line

N, unit, Nc, kRot, 0, 0, 0 comments

N: number of speed points

Unit: unit number, from 0 to 4. See Unit System

Nc: number of brg coefficients, Nc, either 0, 8, 12, or 24

Note: Nc = 8, only translational K and C are considered.

Nc = 12, Translational K, C, and M are considered.

Nc = 24, than all the translational and rotational K, C, and M are considered.

Nc = 0, it will be re-assigned to be 8

KRot = 0 (default) indicates that the bearing coefficients are obtained by assuming the shaft rotation is CCW. Almost all the bearing coefficients are calculated in this way.

KRot = 1 indicates that the bearing coefficients are calculated by assuming the shaft rotation is CW. This option is used for the geared system where different direction of shaft rotation occurs. However, there is a Check option in the Lateral-Torsional-Axial vibration to convert these coefficients. Therefore, kRot == 0 should be used in almost all the cases. It is used mainly for program verification purposes.

repeat *N* times with the following data

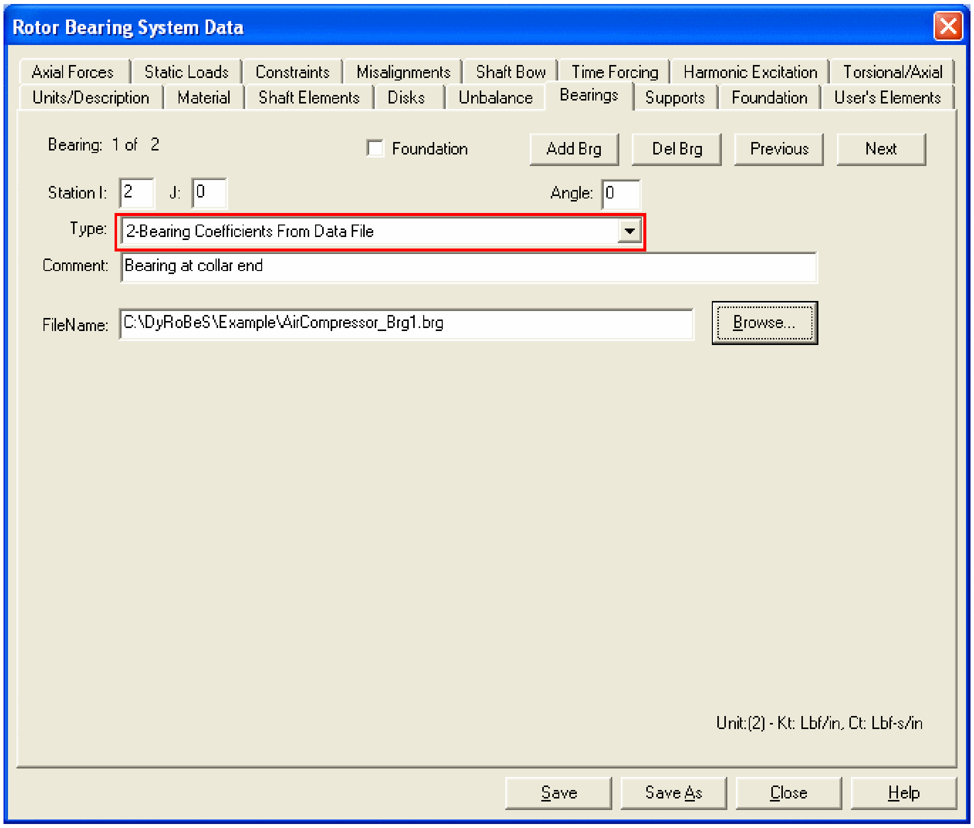
rpm, Kxx,Kxy,Kyx,Kyy,Cxx,Cxy,Cyx,Cyy,Mxx,Mxy,Myx,Myy, Kaa,Kab,Kba,Kbb,Caa,Cab,Cba,Cbb,Maa,Mab,Mba,Mbb,

The rotational speed of shaft 1 is used for the case of multi-shaft systems. Again, If the number of data (speed) points is greater than or equal to 3, a spline function is used in the program to interpolate the coefficients for the shaft speeds requested in the analysis that are not given in the data points. If the number of data (speed) points is equal to 2, a linear function is used to obtain the coefficients for the shaft speeds that are not given in the data points. If there is only one point in the data, then it will be treated as a linear constant bearing. The use of spline function allows a small number of needed data points. Since the spline function is used in the interpolation, the data must be entered in the increasing order according to the speed value. Also, the extrapolation of the data outside the data range can be very unpredictable. Caution must be taken when extrapolating the data.

Example:

```
 9  2  8  0 0 0 0
 5000  .1832E+06  .1552E+06  -.1603E+06  .1885E+06  672   3    3   689
10000  .3628E+06  .3035E+06  -.3301E+06  .3920E+06  665   3    3   703
15000  .5316E+06  .4536E+06  -.4974E+06  .5726E+06  660   6    6   706
20000  .6935E+06  .6038E+06  -.6649E+06  .7387E+06  657   8    8   707
25000  .8557E+06  .7483E+06  -.8416E+06  .9170E+06  654   8    8   712
30000  .1021E+07  .8852E+06  -.1030E+07  .1117E+07  652   4    4   721
35000  .1190E+07  .1013E+07  -.1233E+07  .1342E+07  650  -5   -5   732
40000  .1364E+07  .1132E+07  -.1450E+07  .1596E+07  647  -10  -10  745
45000  .1544E+07  .1242E+07  -.1682E+07  .1879E+07  645  -15  -15  759
```

In the above example, only the translational stiffness and damping are considered, therefore, only 8 coefficients are given in the data file.

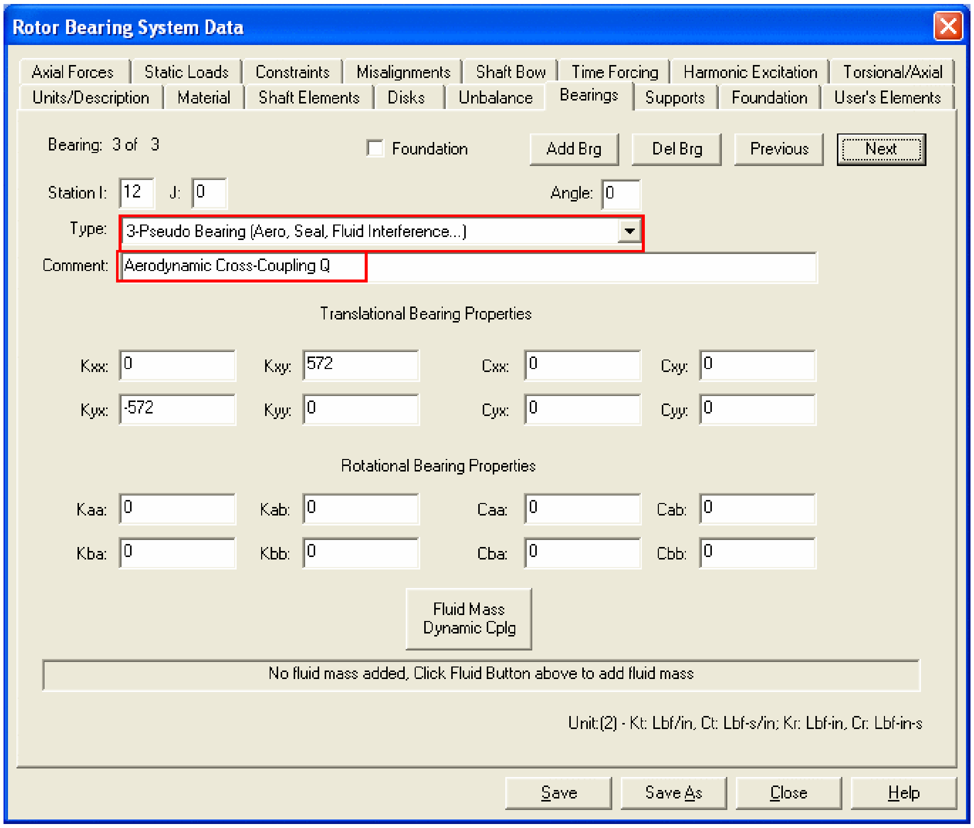


See also Bearings and Linear bearing

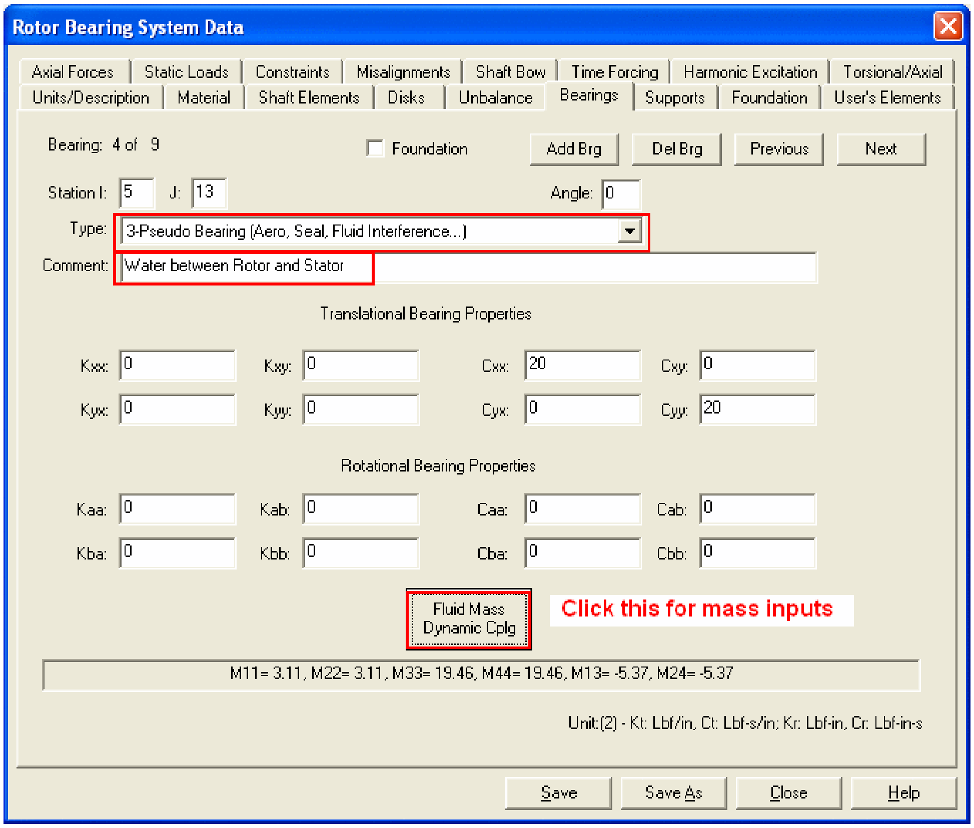
Pseudo Bearing

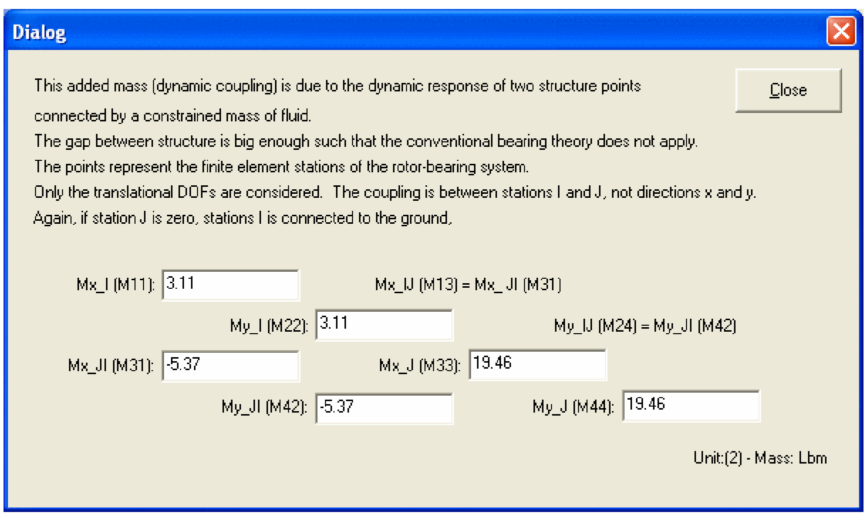
This option allows you to input the **bearing like** forces in the form of stiffnesses, dampings, and masses. The bearing coefficients are entered in the same format as linear bearing option and are treated in the same way as the linear bearings. Fluid Element (Dynamic Coupling) between two finite elements can be considered in this option. Note the fluid element inertia forces acting on the stations are different from the conventional bearing inertia force. They are not action and reaction forces as those of the conventional bearing forces. For more information on the fluid element, click Fluid Elements.

Example: Aerodynamic Cross-Coupling Q



Example: Structure Vibration in Fluid





See also Bearings, Linear bearing, and Liquid Seals.

Squeeze Film Damper

Squeeze film dampers can be modeled with or without centering springs. The centering spring is assumed to be isotropic if exists. For the nonlinear centering spring, a generalized non-linear isotropic bearing can be used in parallel or in series with the squeeze film damper. In the steady state synchronous response analysis, centered circular orbits are assumed. The general motion of a plain fluid film journal bearing or a squeeze film damper is governed by the Reynolds equation, which is derived from the Navier-Stokes equation. The fluid film forces acting on the journal are determined by application of boundary conditions and integration of pressure distribution. The general incompressible laminar Reynolds equation is given by:

$$\frac{1}{R^2}\frac{\partial}{\partial\theta}\left(\frac{h^3}{\mu}\frac{\partial P}{\partial\theta}\right)+\frac{\partial}{\partial Z}\left(\frac{h^3}{\mu}\frac{\partial P}{\partial Z}\right)=6\left(\omega_b+\omega_j-2\dot{\phi}\right)\frac{\partial h}{\partial\theta}+12\frac{\partial h}{\partial t}$$

where

ω_j = journal rotational speed

ω_b = bearing rotational speed

$\dot{\phi}$ = precession of the line of centers (whirl speed)

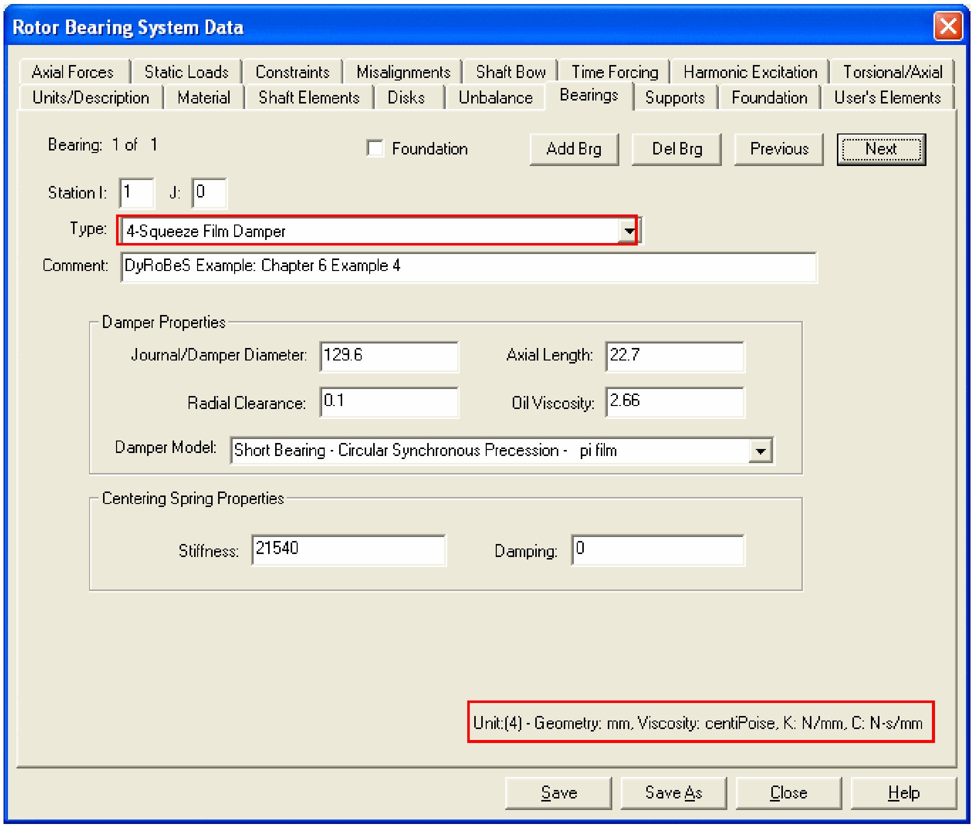
For a squeeze film damper, the damper is free to precess, but not rotate. Furthermore, the last term vanishes at steady state condition. For the transient analysis, short bearing and pi film assumptions are utilized. For the steady state response, the following table summarizes the equivalent stiffness and damping for the cases of circular synchronous motion about the origin and pure radial motion with no precession for the conditions of cavitation (pi film) and no cavitation (2pi film).

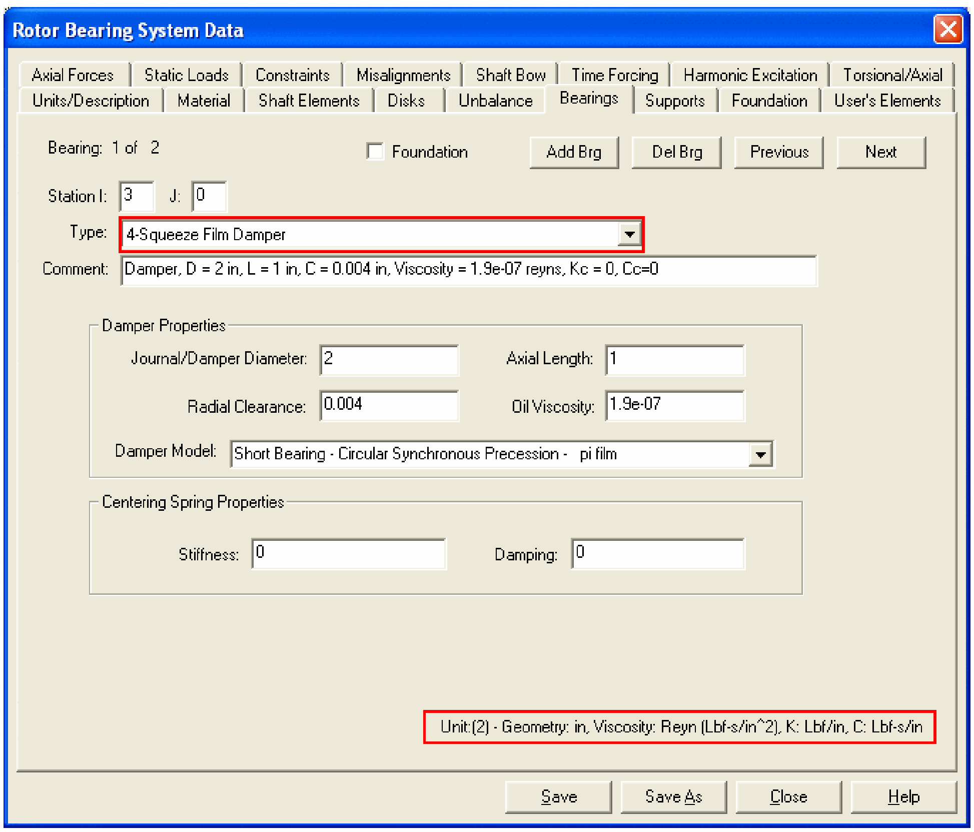
Bearing	Film	Motion	Stiffness	Damping
Short Bearing	π film	Circular Synchronous Precession	$\frac{2\mu R L^3 \varepsilon \omega}{C^3(1-\varepsilon^2)^2}$	$\frac{\mu R L^3 \pi}{2C^3(1-\varepsilon^2)^{3/2}}$
	2π film		0	$\frac{\mu R L^3 \pi}{C^3(1-\varepsilon^2)^{3/2}}$
	π film	Pure Radial Squeeze Motion	0	$\frac{\mu R L^3 \pi (2\varepsilon^2+1)}{2C^3(1-\varepsilon^2)^{5/2}}$
	2π film		0	$\frac{\mu R L^3 \pi (2\varepsilon^2+1)}{C^3(1-\varepsilon^2)^{5/2}}$
Long Bearing	π film	Circular Synchronous Precession	$\frac{24\mu R^3 L \varepsilon \omega}{C^3(2+\varepsilon^2)(1-\varepsilon^2)}$	$\frac{12\mu R^3 L\pi}{C^3(2+\varepsilon^2)(1-\varepsilon^2)^{1/2}}$
	2π film		0	$\frac{24\mu R^3 L\pi}{C^3(2+\varepsilon^2)(1-\varepsilon^2)^{1/2}}$

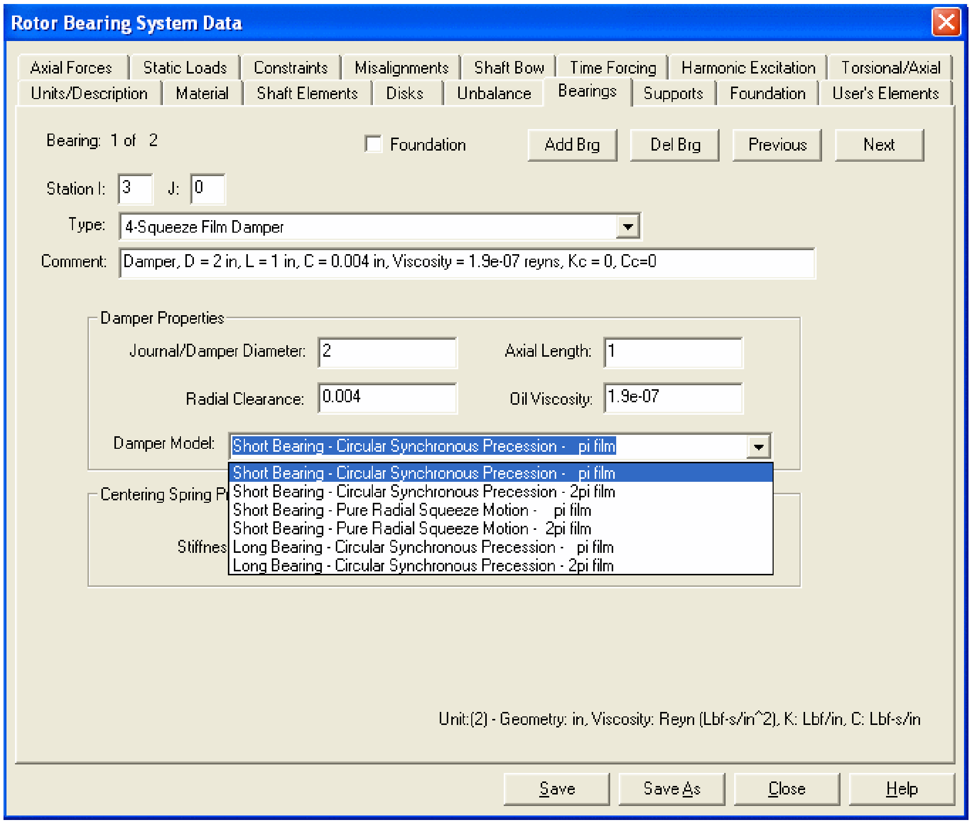
Where R = damper radius
L = damper axial length
C = radial clearance
ω = whirl speed
μ = oil viscosity
ε = eccentricity ratio

Note that for the circular synchronous motion, the equivalent stiffness term is a highly nonlinear function of eccentricity and may lead to a nonlinear **jump phenomenon** under high rotor unbalance. Caution must be taken while designing the damper, since it can significantly either improve or degrade the dynamic characteristics of the rotor system.

The data fields are self-explanatory. In the right bottom of the screen, the proper units for the specified unit system are displayed for reference.


Rotor Bearing System Data
Axial Forces
Static Loads
Constraints
Misalignments
Shaft Bow
Time Forcing
Harmonic Excitation
Torsional/Axial
Units/Description
Material
Shaft Elements
Disks
Unbalance
Bearings
Supports
Foundation
User's Elements
Bearing: 1 of 1
Foundation
Add Brg
Del Brg
Previous
Next
Station I: 1
J: 0
Type: 4-Squeeze Film Damper
Comment: DyRoBeS Example: Chapter 6 Example 4
Damper Properties
Journal/Damper Diameter: 129.6
Axial Length: 22.7
Radial Clearance: 0.1
Oil Viscosity: 2.66
Damper Model: Short Bearing - Circular Synchronous Precession - pi film
Centering Spring Properties
Stiffness: 21540
Damping: 0
Unit:(4) - Geometry: mm, Viscosity: centiPoise, K: N/mm, C: N-s/mm
Save
Save As
Close
Help


Rotor Bearing System Data
Axial Forces
Static Loads
Constraints
Misalignments
Shaft Bow
Time Forcing
Harmonic Excitation
Torsional/Axial
Units/Description
Material
Shaft Elements
Disks
Unbalance
Bearings
Supports
Foundation
User's Elements
Bearing: 1 of 2
Foundation
Add Brg
Del Brg
Previous
Next
Station I: 3
J: 0
Type: 4-Squeeze Film Damper
Comment: Damper, D = 2 in, L = 1 in, C = 0.004 in, Viscosity = 1.9e-07 reyns, Kc = 0, Cc=0
Damper Properties
Journal/Damper Diameter: 2
Axial Length: 1
Radial Clearance: 0.004
Oil Viscosity: 1.9e-07
Damper Model: Short Bearing - Circular Synchronous Precession - pi film
Centering Spring Properties
Stiffness: 0
Damping: 0
Unit:(2) - Geometry: in, Viscosity: Reyn (Lbf-s/in^2), K: Lbf/in, C: Lbf-s/in
Save
Save As
Close
Help



See also Bearings.

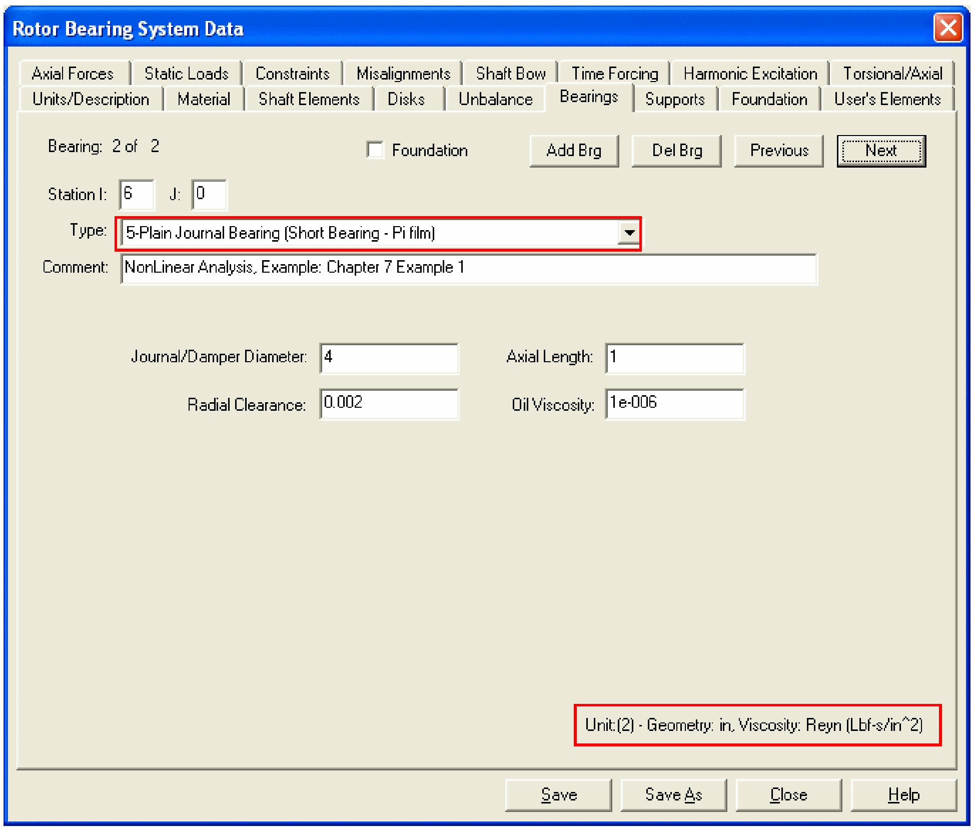
Plain Journal Bearing

A plain journal bearing model based on pi film short bearing theory is provided. The incompressible laminar Reynolds equation for the short bearing theory is:

$$\frac{\partial}{\partial Z}\left(\frac{h^3}{\mu}\frac{\partial P}{\partial Z}\right) = 6\ \omega_j \frac{\partial h}{\partial \theta} + 12\frac{\partial h}{\partial t}$$

The data fields are self-explanatory. In the right bottom of the screen, the proper units for the specified unit system are displayed for reference.

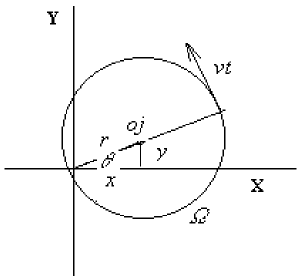
For a complete 2 dimensional Reynolds Equation solver, use Type 12-Multi-Lobe hydrodynamic bearing (input data in Rotor) or Type 13-Multi-Lobe hydrodynamic bearing (input data in BePerf).



See also Bearings, Multi-Lobe Hydrodynamic Bearing

Generalized Non-Linear Isotropic Bearings

Two types of the generalized non-linear isotropic bearing are provided in this bearing option. This option replaces the previous version for the rolling element bearing with clearance. The journal motion is shown below:



(x, y) are shaft center displacements and $r = \sqrt{x^2 + y^2}$. The tangential velocity of the shaft at the contact point

$$v_t = R\Omega + (-\dot{x}\sin\theta + \dot{y}\cos\theta) = R\Omega + \left(\frac{-\dot{x}y + \dot{y}x}{r}\right)$$

where R is the shaft radius and Ω is the rotor speed.

$(-F_r)$ is the radial restoring force acting on the shaft center due to radial displacement, $(-F_t)$ is the tangential force due to Coulomb Friction:

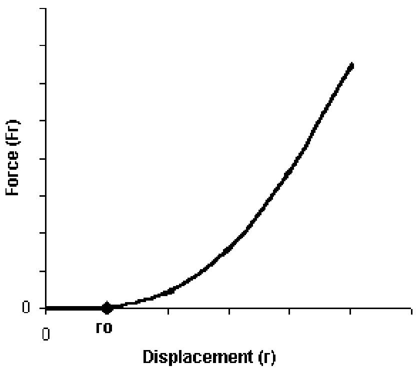
$$-F_t = \mu(-F_r)\, sign(v_t)$$

The total forces action on the shaft due to displacement (r), friction (μ), and linear damping (C) are:

$$F_x = (-F_r)\cos\theta - (-F_t)\sin\theta - C\dot{x}$$
$$F_y = (-F_r)\sin\theta + (-F_t)\cos\theta - C\dot{y}$$

Two types of equations are provided for the radial force and they are:

Case 1: Continuous Force-Displacement Curve



When $r < r_0$ (deadband or gap)

$$F_r = 0,\ F_t = 0,\ F_x = 0,\ F_y = 0$$

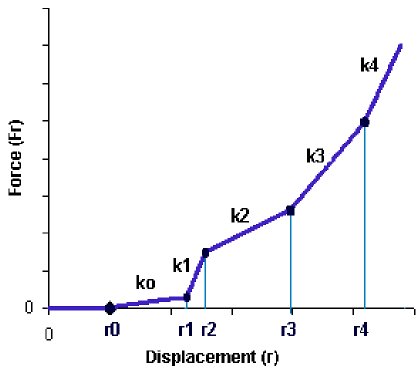
No forces are acting on the shaft when shaft vibration is smaller than the clearance.

When $r_0 \le r$

$$F_r = k_0(r-r_0)^{a_0} + k_1(r-r_0)^{a_1} + k_2(r-r_0)^{a_2} + k_3(r-r_0)^{a_3} + k_4(r-r_0)^{a_4}$$
$$F_t = \mu\, F_r\, sign(v_t)$$

Note that r_o can be zero if no gap or deadband exists.

Case 2: Piecewise linear curves



where k_i is the slope from r_i to r_{i+1}

When $r < r_0$ (deadband or gap)

$$F_r = 0,\ F_t = 0,\ F_x = 0,\ F_y = 0$$

No forces are acting on the shaft when shaft vibration is smaller than the clearance.

When $r_0 \le r < r_1$

$$F_r = k_0\,(r - r_0)$$
$$F_t = \mu_0\, F_r$$

When $r_1 \le r < r_2$

$$F_r = k_0\,(r_1 - r_0) + k_1\,(r - r_1)$$
$$F_t = \mu_1\, F_r$$

When $r_2 \le r < r_3$

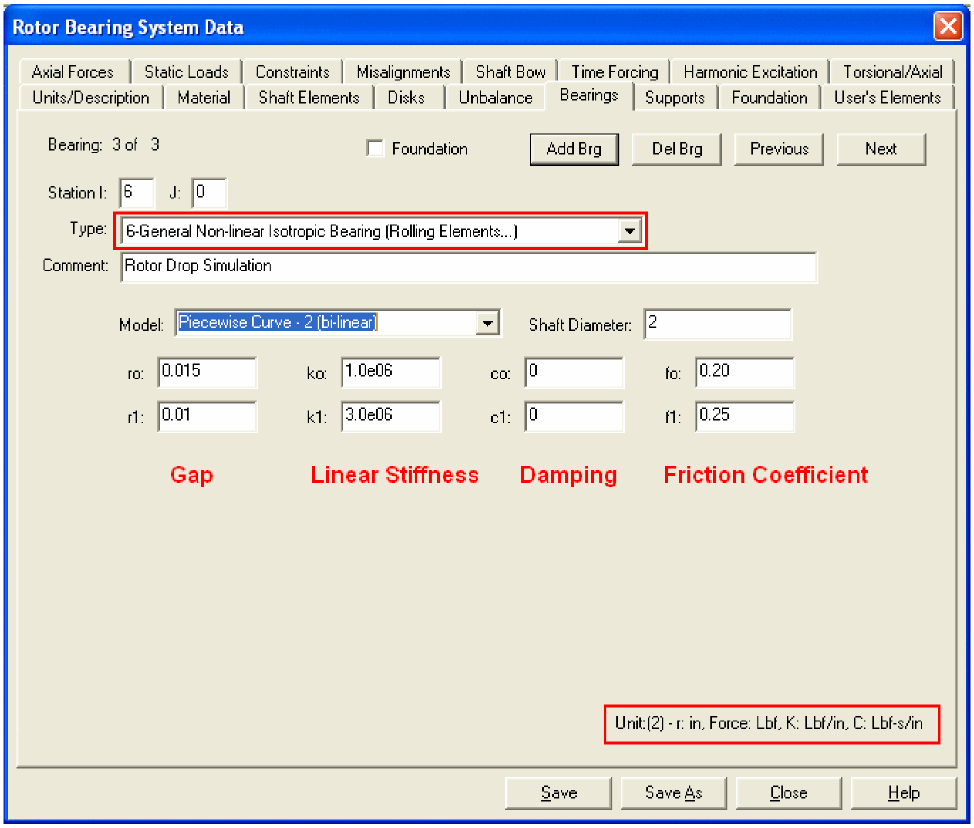
$$F_r = k_0\,(r_1 - r_0) + k_1\,(r_2 - r_1) + k_2\,(r - r_2)$$
$$F_t = \mu_2\, F_r$$

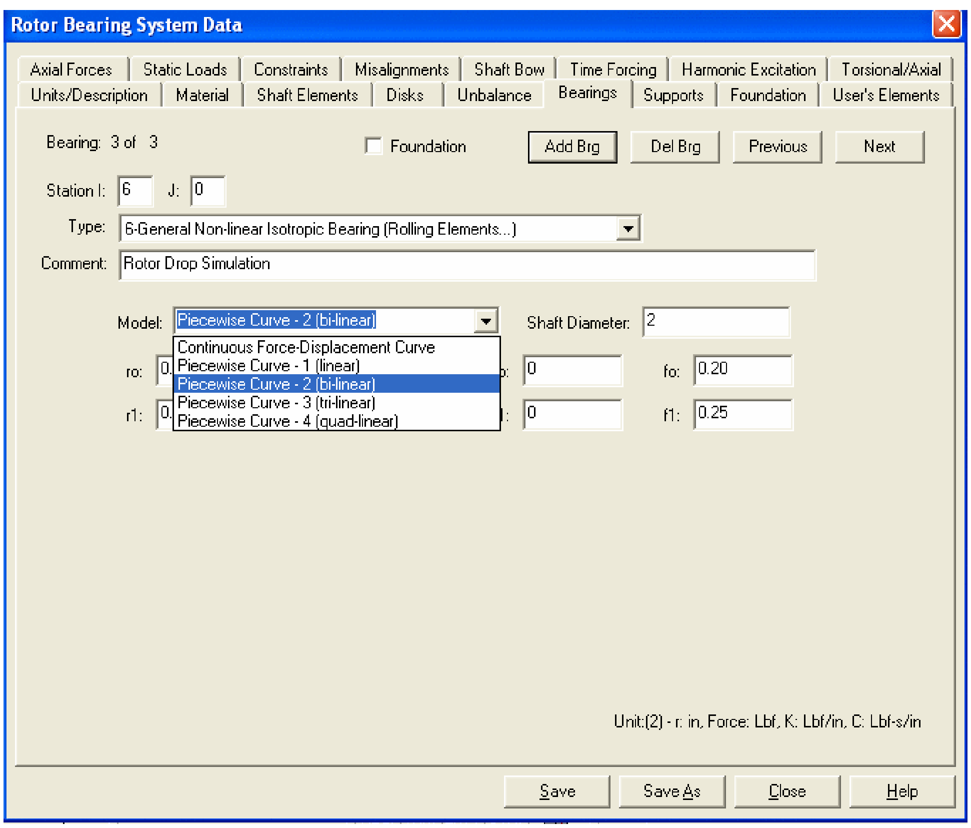
When $r_3 \le r < r_4$

$$F_r = k_0\,(r_1 - r_0) + k_1\,(r_2 - r_1) + k_2\,(r_3 - r_2) + k_3\,(r - r_3)$$
$$F_t = \mu_3\, F_r$$

and so on.

A linear rolling element bearing with clearance (deadband) can also be modeled in this case.





See also Bearings.

Active Magnetic Bearing

Two options are used to model the active magnetic bearing. The linear Proportional-Integral-Derivative controller with low pass filter is used in the steady state analysis (Stability and Forced Response Analyses). The nonlinear active magnetic bearing requires more input data and is used in the non-linear transient analysis. For both options, the sensor stations may be different from the bearing stations (sensor non-collocation) and the model may be different for the two bearing axes.

Active Magnetic Bearing 1 - Linear PID controller with low pass filter

This bearing is modeled as a PID controller in series with a unity gain, first order low pass filter (generally used to model the amplifier). Two additional degrees of freedom will be added to each of the x and y equations to model the controller states for each bearing. However, these two additional degrees of freedom will not be available for displaying in the post-processor. The output of the PID controller at each axis is:

$$C_p x_s + C_i \int x_s dt + C_d \dot{x}_s$$

Where x_s is the displacement at sensor location. The control force at each direction in the S-domain is:

$$F = \left(C_p + C_i \frac{1}{s} + C_d s \right) \left(\frac{2\pi f_c}{s + 2\pi f_c} \right) x_s$$

Active Magnetic Bearing 2 - Non-Linear Transient Analysis

This bearing is a standard PID controlled active magnetic bearing with sensor non-collocation, gap non-linearity and current saturation effects for the transient analysis only. The control current is determined from the following expression:

$$i_c = C_p x_s + C_i \int x_s dt + C_d \dot{x}_s$$

The currents supplied to the magnetic bearing are determined from the following:

$$i_1 = i_{b,p} - i_c$$

$$i_2 = i_{b,n} + i_c$$

$$\text{if } \; i < 0, \quad i = 0; \qquad \text{if } \; i > i_{\text{limit}}, \quad i = i_{\text{limit}}$$

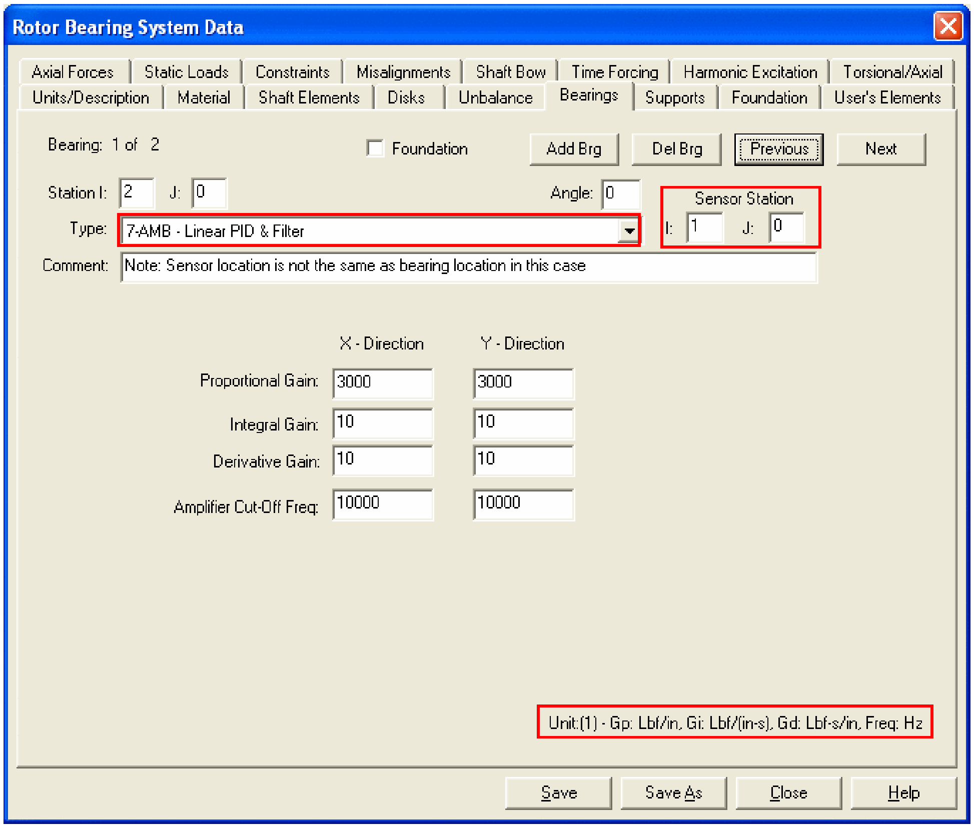
The force in the magnetic bearing is:

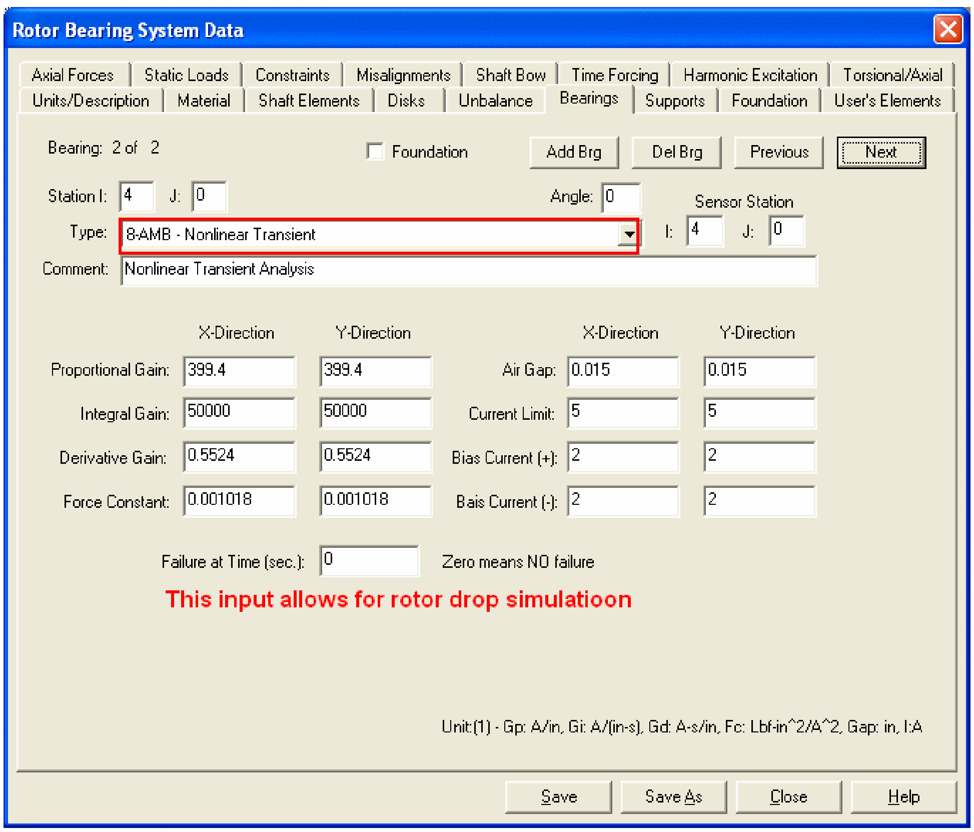
$$F = F_c \cdot \left[\left(\frac{i_1}{h_1} \right)^2 - \left(\frac{i_2}{h_2} \right)^2 \right]$$

where

$$h_1 = gap - x_b$$

$$h_2 = gap + x_b$$

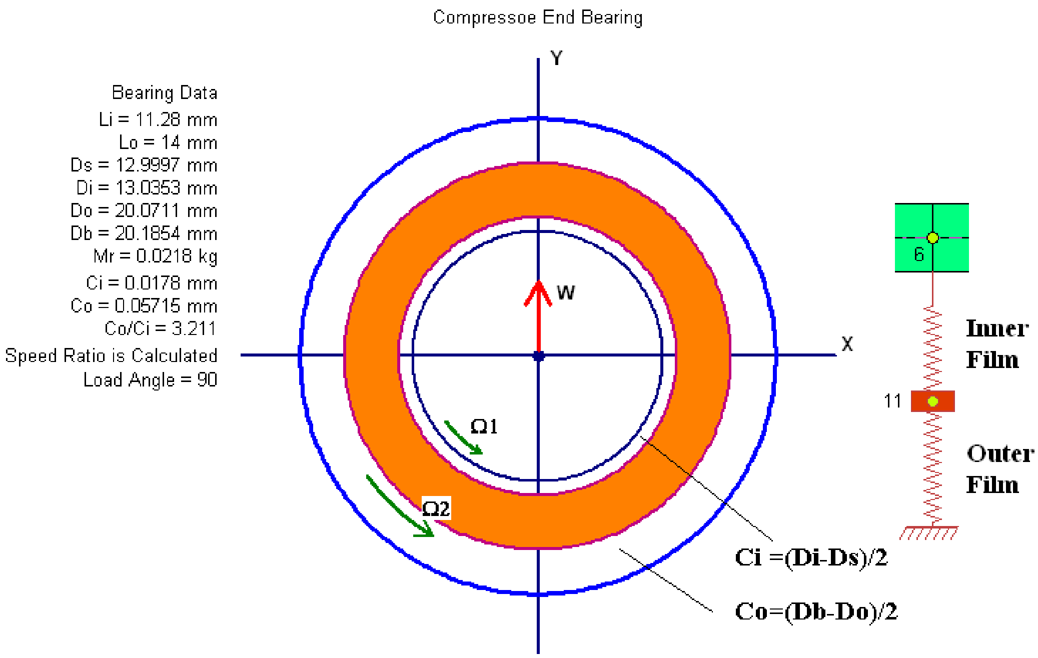


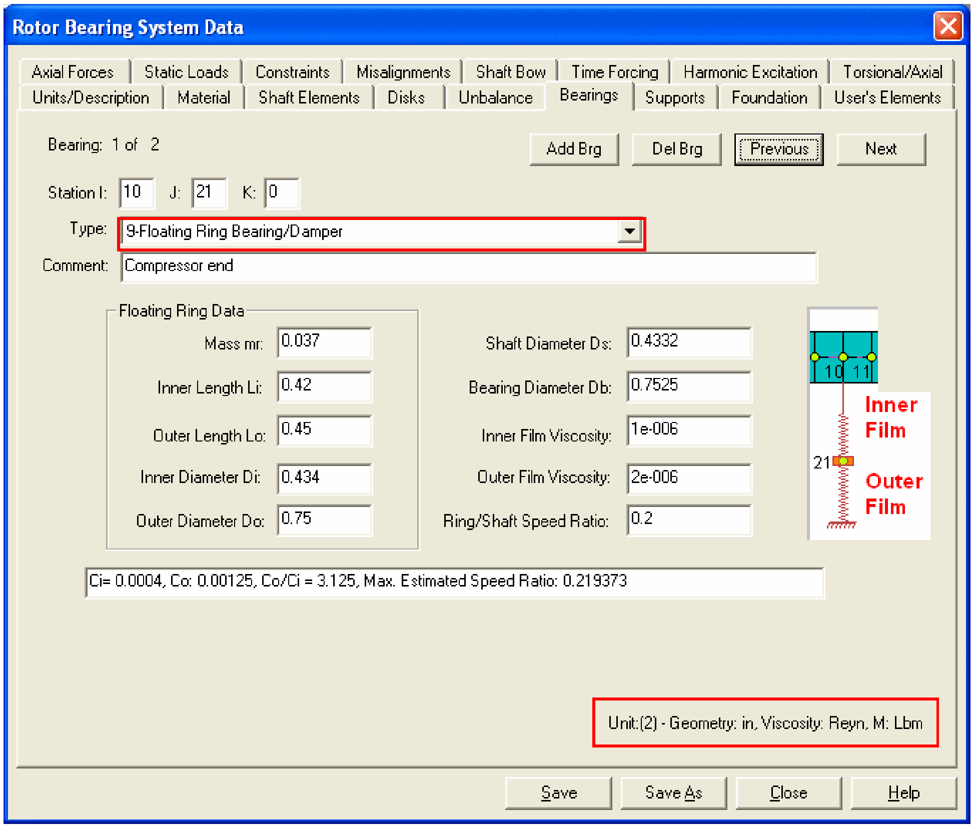


See also Bearings

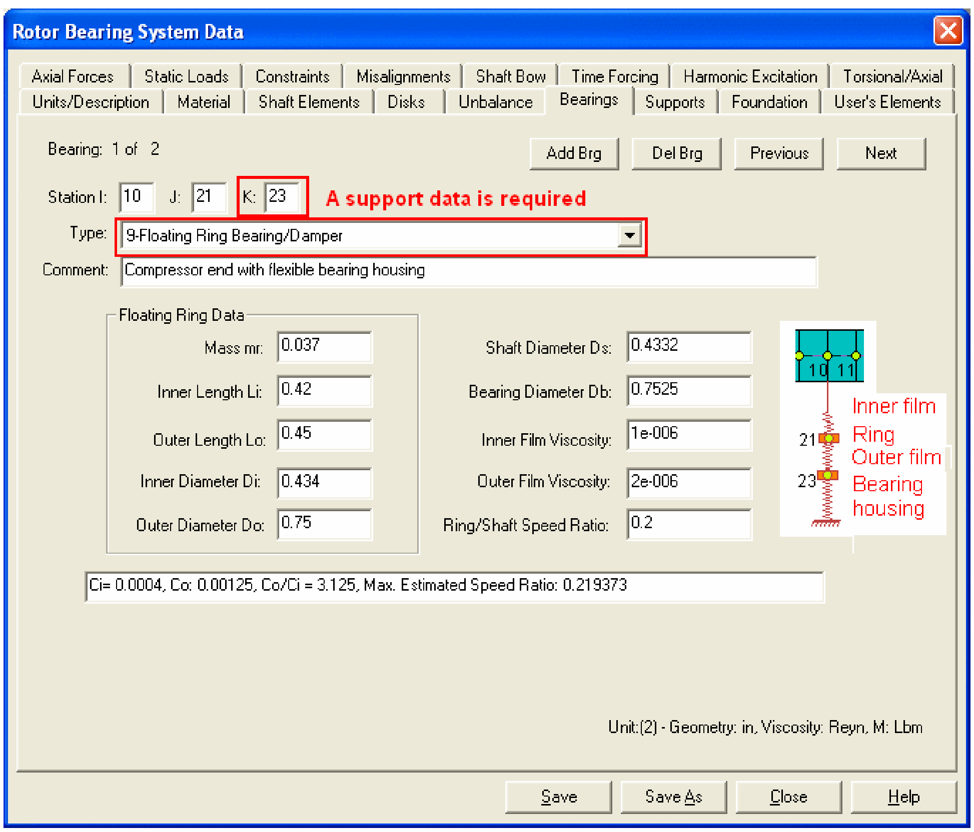
Floating Ring Bearing

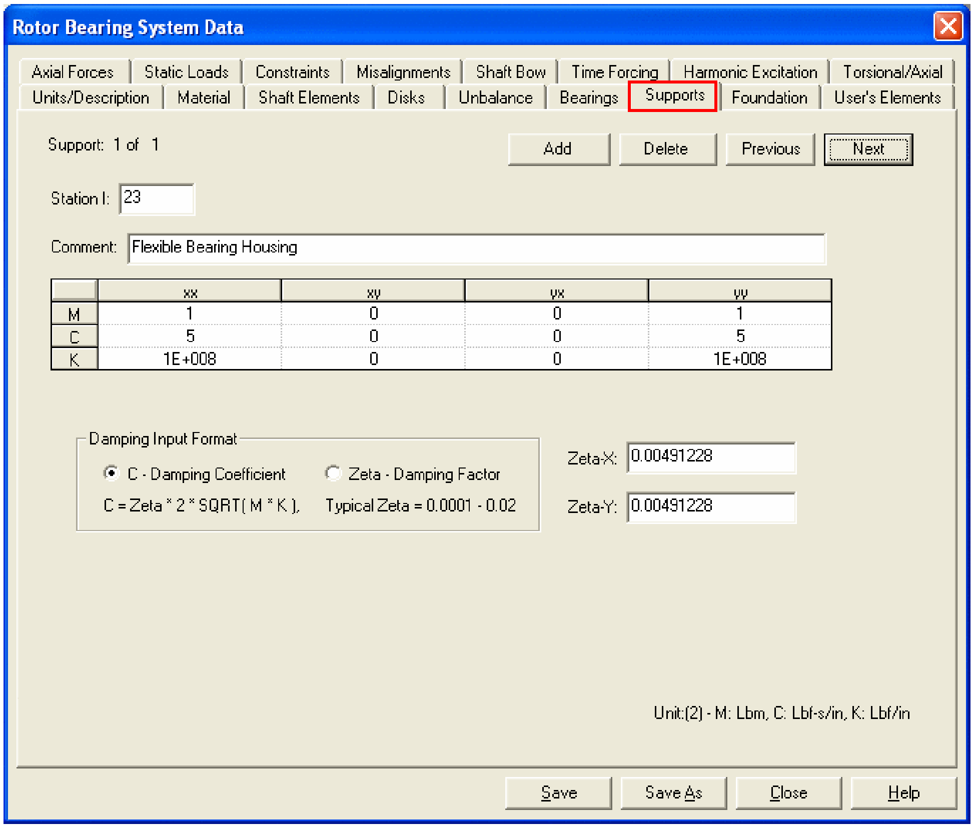
Floating Ring Bearing can be treated as two fluid film bearings in series. The inner film has two rotating surface (shaft and ring). The outer film bearing has only one rotating surface (ring). IN rotordynamics, additional two degrees of freedom are introduced for each floating ring bearing due to its ring mass. The ring mass station is a support station, which is automatically created when floating ring bearing is selected. The ring mass cannot be zero to ensure the mass matrix being positive definite. The Station I is the station at the rotor, Station J is the ring mass station, and Station K is the support station if a flexible support exists under the floating ring bearing. If station K=0, the floating ring bearing is connected to the ground. If station K is not zero, then a support must be created by using the Flexible Supports tab. The non-zero Ring/Shaft Speed Ratio is for the conventional floating ring bearing that the ring rotates with a fraction of the rotor speed. A zero Ring/Shaft Speed Ratio can be used for the floating ring damper where the ring is constrained from rotation with anti-rotational pins. Note that the floating ring bearing is treated as a nonlinear bearing if bearing Type 9 is used. If you want to performance Linear Analysis, use BePerf to create the linearized bearing coefficients and then import them using bearing Type 2. Advanced Speed Dependent Variables Button allows for speed dependent viscosities, speed ratios, and clearances. Note that clearances vary with the temperature, but temperature is dependent upon the rotor speed.

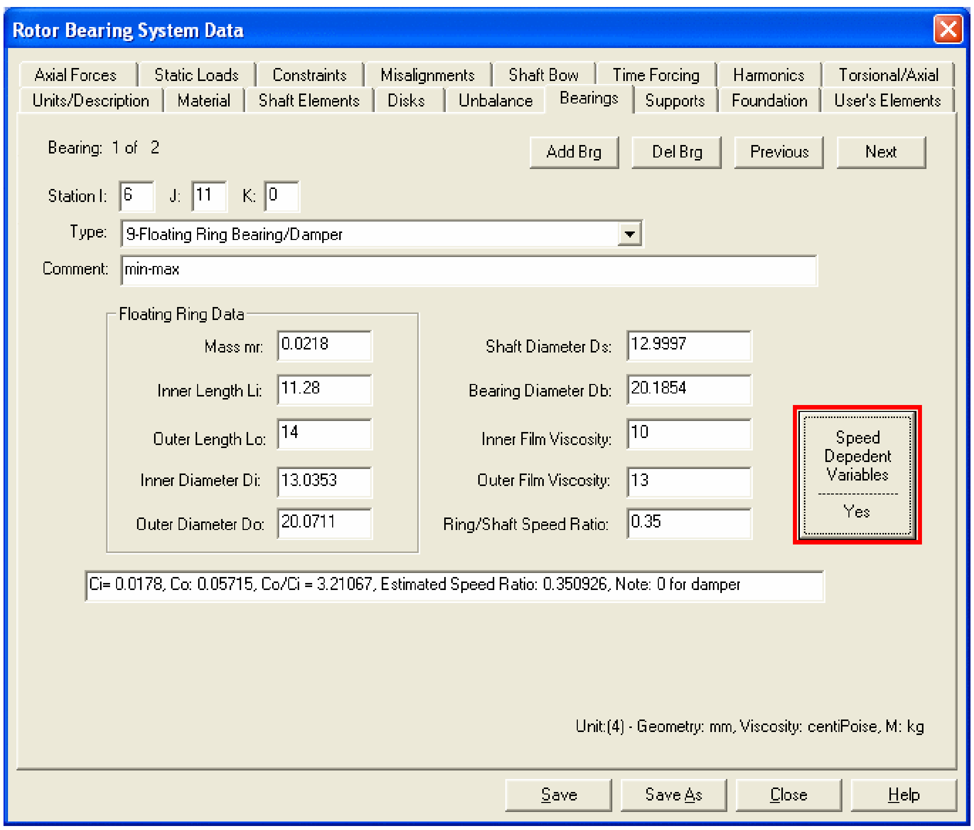




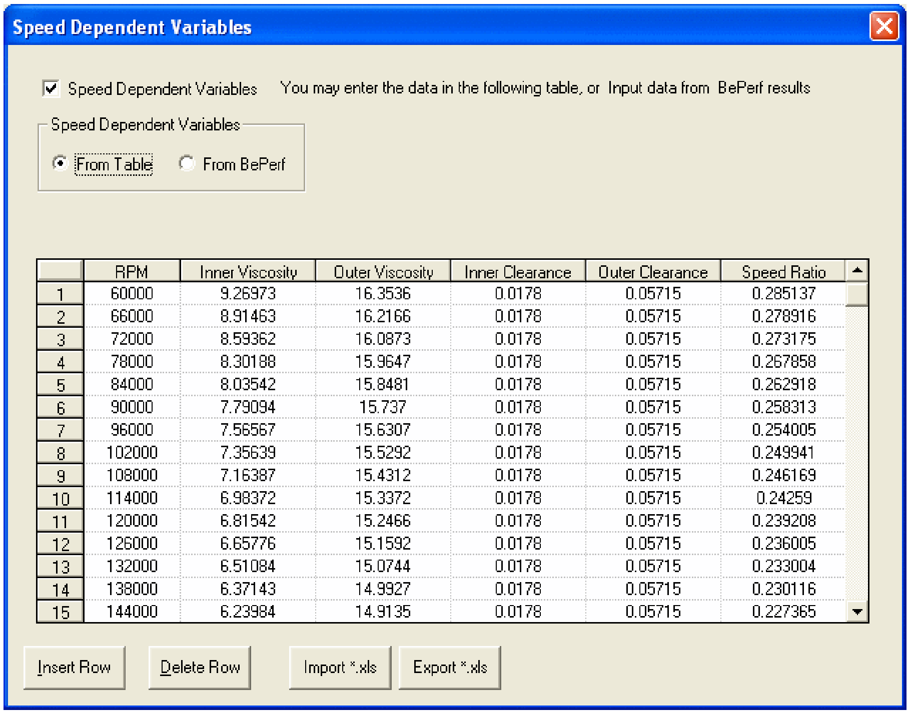
If bearing housing is flexible, a support data must be entered for this flexible housing.



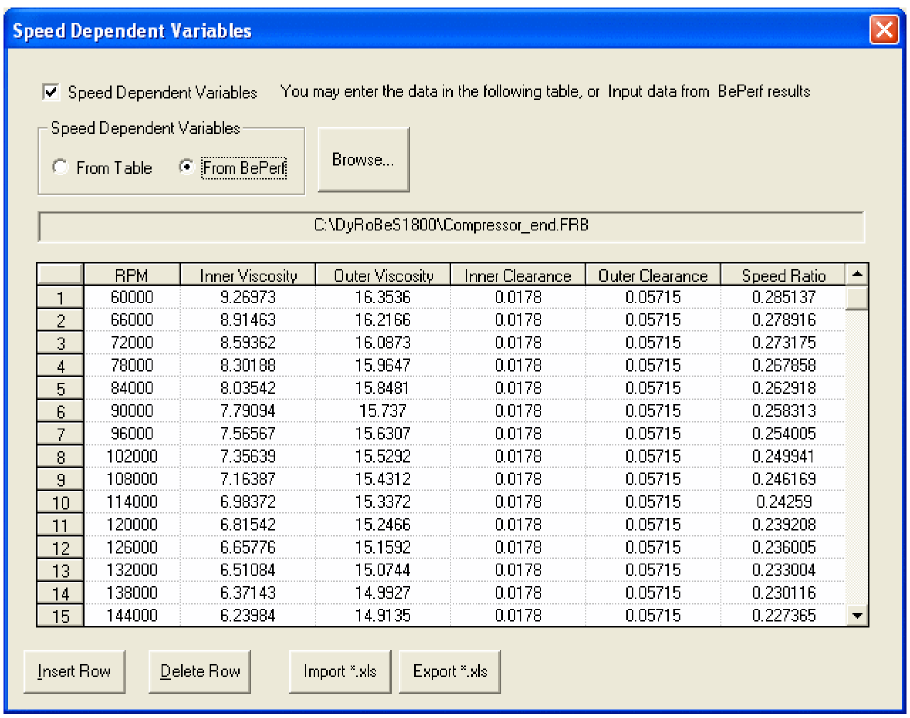




In the Speed Dependent Variables data page, make sure CHECK the Speed Dependent Variables box if you want to use the speed dependent variables. There are two ways to enter the speed dependent variables. One is to enter the data into the table directly, the other is to read the data from the BePerf's results. If you select From BePerf, then you need to click the Browse button to select the BePerf data file. The program will read the variables from BePerf outputs and fill in the table if the outputs are available, otherwise, the program will prompt you to create the outputs from BePerf. Select Yes, to create the outputs file and fill the table. You may also enter he speed dependent variables manually, this allows for the variations in the clearances too.



	RPM	Inner Viscosity	Outer Viscosity	Inner Clearance	Outer Clearance	Speed Ratio
1	60000	9.26973	16.3536	0.0178	0.05715	0.285137
2	66000	8.91463	16.2166	0.0178	0.05715	0.278916
3	72000	8.59362	16.0873	0.0178	0.05715	0.273175
4	78000	8.30188	15.9647	0.0178	0.05715	0.267858
5	84000	8.03542	15.8481	0.0178	0.05715	0.262918
6	90000	7.79094	15.737	0.0178	0.05715	0.258313
7	96000	7.56567	15.6307	0.0178	0.05715	0.254005
8	102000	7.35639	15.5292	0.0178	0.05715	0.249941
9	108000	7.16387	15.4312	0.0178	0.05715	0.246169
10	114000	6.98372	15.3372	0.0178	0.05715	0.24259
11	120000	6.81542	15.2466	0.0178	0.05715	0.239208
12	126000	6.65776	15.1592	0.0178	0.05715	0.236005
13	132000	6.51084	15.0744	0.0178	0.05715	0.233004
14	138000	6.37143	14.9927	0.0178	0.05715	0.230116
15	144000	6.23984	14.9135	0.0178	0.05715	0.227365



	RPM	Inner Viscosity	Outer Viscosity	Inner Clearance	Outer Clearance	Speed Ratio
1	60000	9.26973	16.3536	0.0178	0.05715	0.285137
2	66000	8.91463	16.2166	0.0178	0.05715	0.278916
3	72000	8.59362	16.0873	0.0178	0.05715	0.273175
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5	84000	8.03542	15.8481	0.0178	0.05715	0.262918
6	90000	7.79094	15.737	0.0178	0.05715	0.258313
7	96000	7.56567	15.6307	0.0178	0.05715	0.254005
8	102000	7.35639	15.5292	0.0178	0.05715	0.249941
9	108000	7.16387	15.4312	0.0178	0.05715	0.246169
10	114000	6.98372	15.3372	0.0178	0.05715	0.24259
11	120000	6.81542	15.2466	0.0178	0.05715	0.239208
12	126000	6.65776	15.1592	0.0178	0.05715	0.236005
13	132000	6.51084	15.0744	0.0178	0.05715	0.233004
14	138000	6.37143	14.9927	0.0178	0.05715	0.230116
15	144000	6.23984	14.9135	0.0178	0.05715	0.227365

See also Bearings and Flexible Supports.

General Non-Linear Polynomial Bearing

This is a general non-linear bearing. The bearing characteristics is modeled as a polynomial as shown below:

Forces acting on the rotor: X = Xi - Xj (relative displacement), Vx = X dot (Relative Velocity) ...etc.

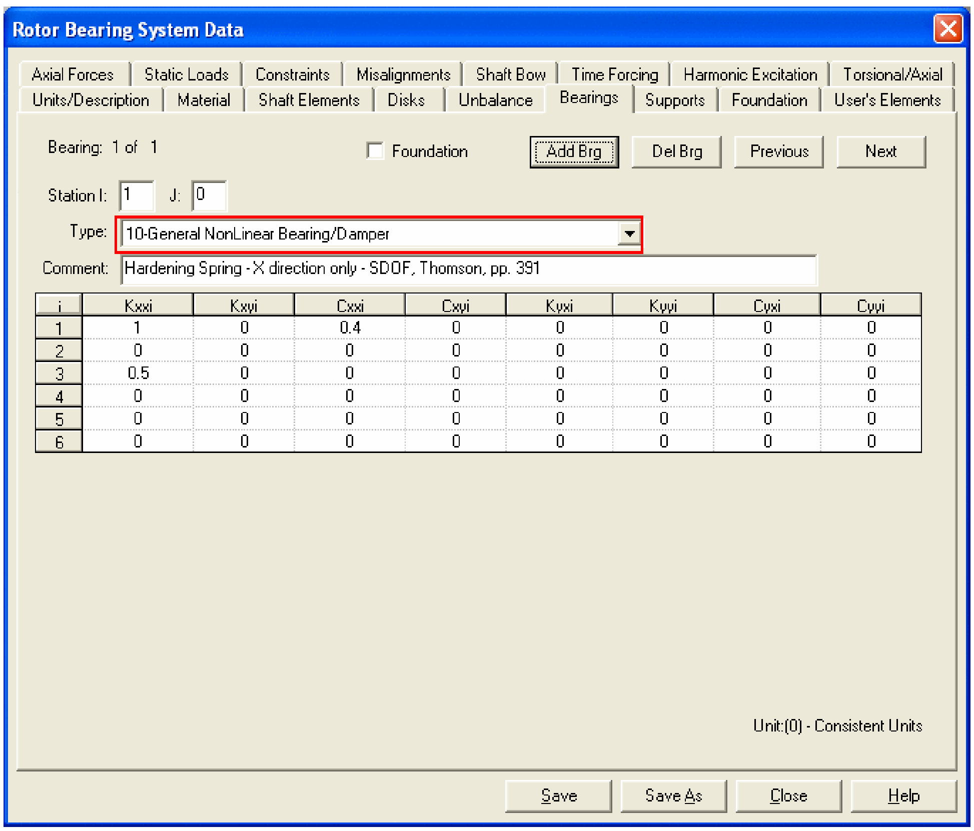
Fx = - (Kxx1 X + Kxx2 X^2 + Kxx3 X^3 + ...+ Kxxn X^n + Kxy1 Y + Kxy2 Y^2 + Kxy3 Y^3 +...+ Kxyn Y^n
+ Cxx1 Vx + Cxx2 Vx^2 + Cxx3 Vx^3 + ...+ Cxxn Vx^n + Cxy1 Vy + Cxy2 Vy^2 + Cxy3 Vy^3 +...+ Cxyn Vy^n)

Fy = - (Kyx1 X + Kyx2 X^2 + Kyx3 X^3 + ...+ Kyxn X^n + Kyy1 Y + Kyy2 Y^2 + Kyy3 Y^3 +...+ Kyyn Y^n
+ Cyx1 Vx + Cyx2 Vx^2 + Cyx3 Vx^3 + ...+ Cyxn Vx^n + Cyy1 Vy + Cyy2 Vy^2 + Cyy3 Vy^3 +...+ Cyyn Vy^n)

For a nonlinear equation:

$$\ddot{x} + 0.4\dot{x} + x + 0.5x^3 = 0.5\cos(0.5t)$$

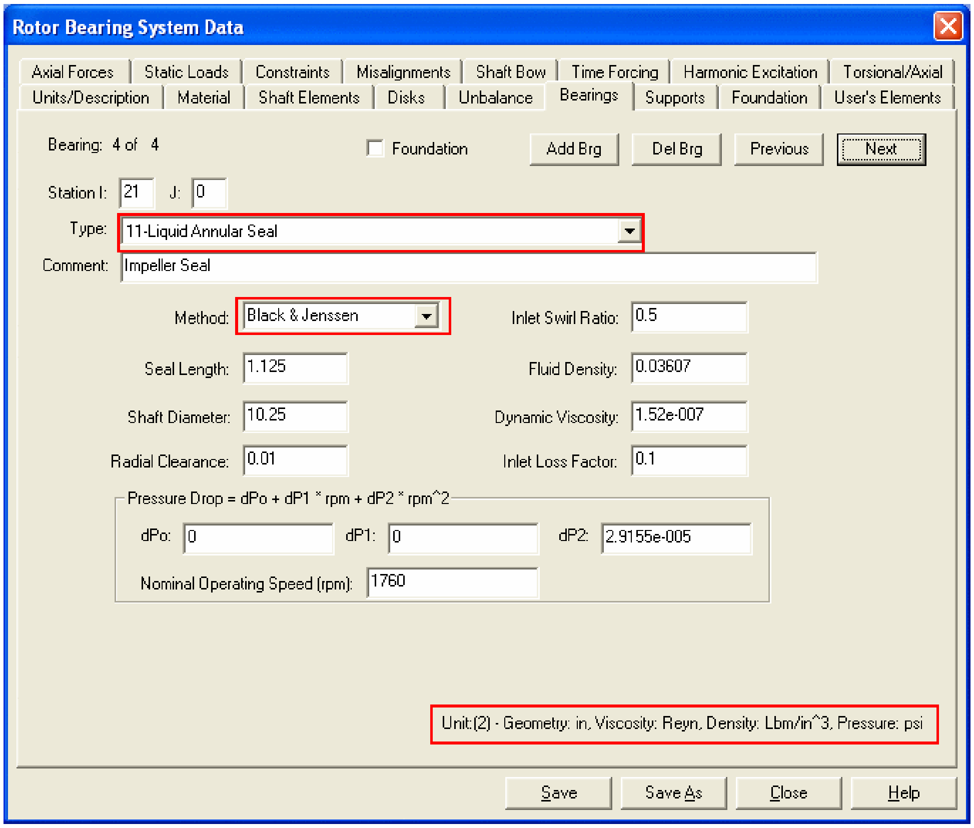
The inputs for the linear damping and nonlinear stiffness are shown below:



See also [Bearings]

Liquid Annular Seal

The liquid annular seals used in the pumps are known to raise the dry critical speeds by a considerable amount. The mathematical model of the liquid annular seal is similar to the bearing model. The dynamic coefficients are calculated from the given seal geometric data and operating condition. Two models are available: one is based on Black & Jenssen and the other is based on Childs 1983. For more information on the liquid seal theory, click Liquid Annular Seal Dynamic Coefficients.

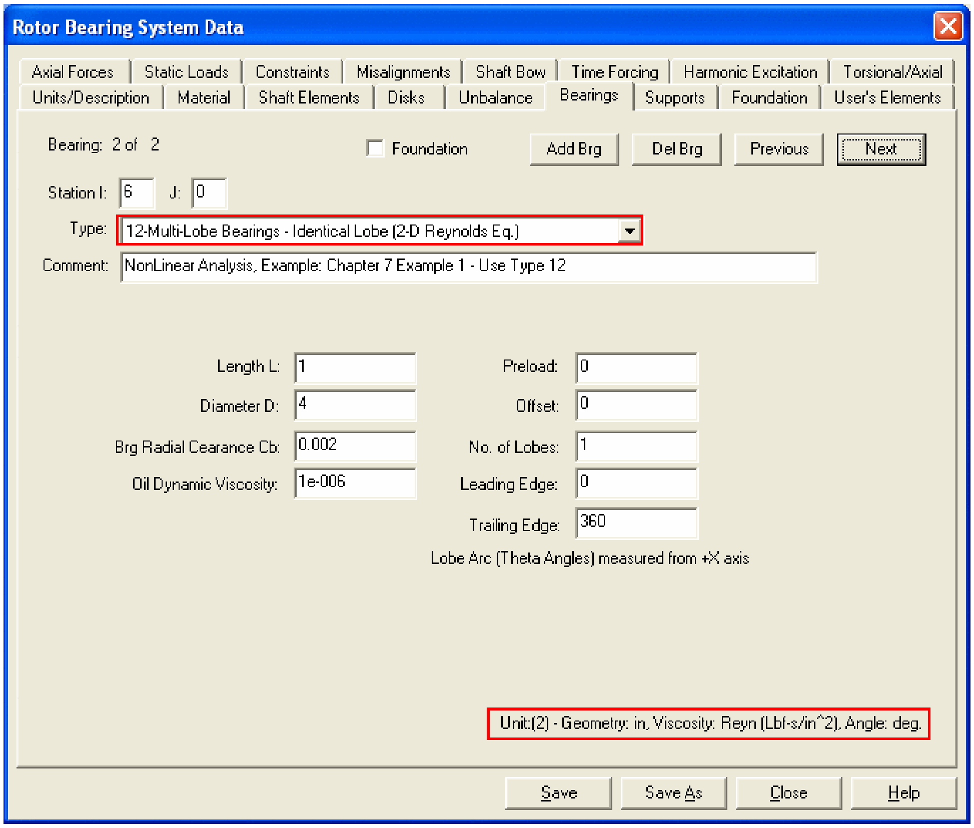


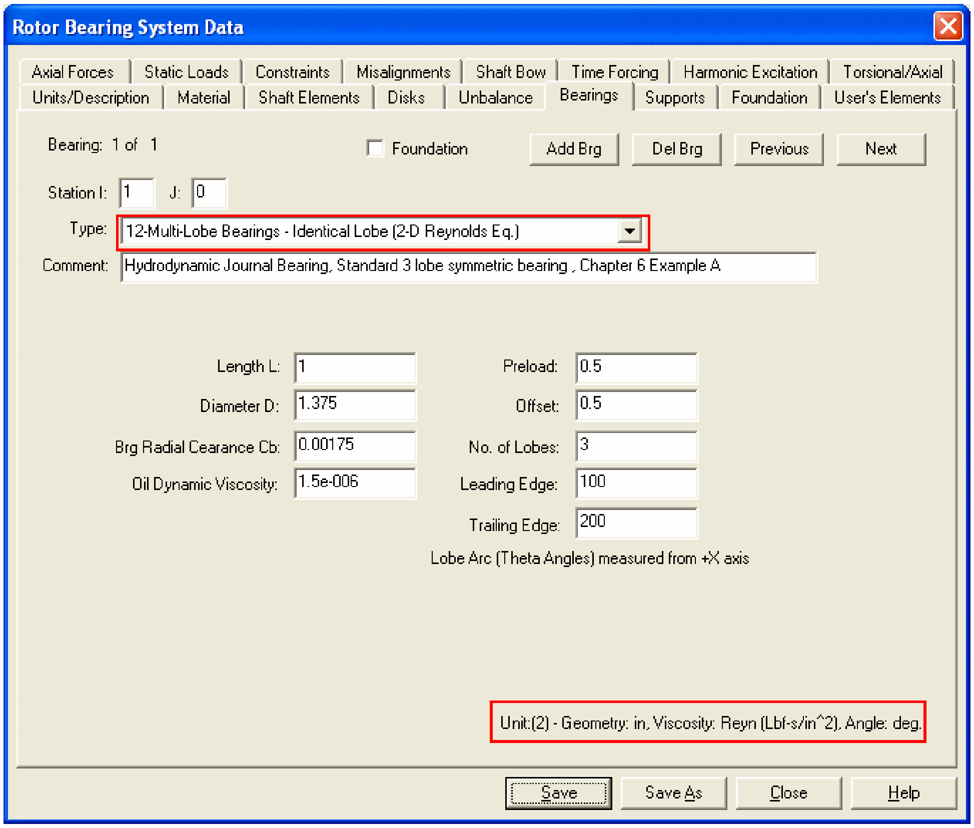
See also Bearings, Liquid Annular Seal Dynamic Coefficients.

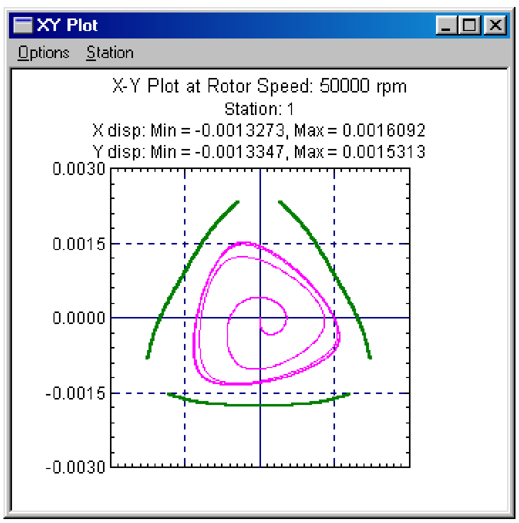
Multi-Lobe Hydrodynamic Bearing

This option allows you to include a multi-lobe bearing which is solved in a complete 2D Reynolds equation coupled with the shaft elastic equations. For more information on the bearing theory, refer to the book titled **Introduction to Dynamics of Rotor-Bearing Systems** by Chen and Gunter (2005). Note that by selecting this bearing type, only transient analysis can be performed. The rotor elastic equations are coupled with the nonlinear Reynolds equations for the bearings.

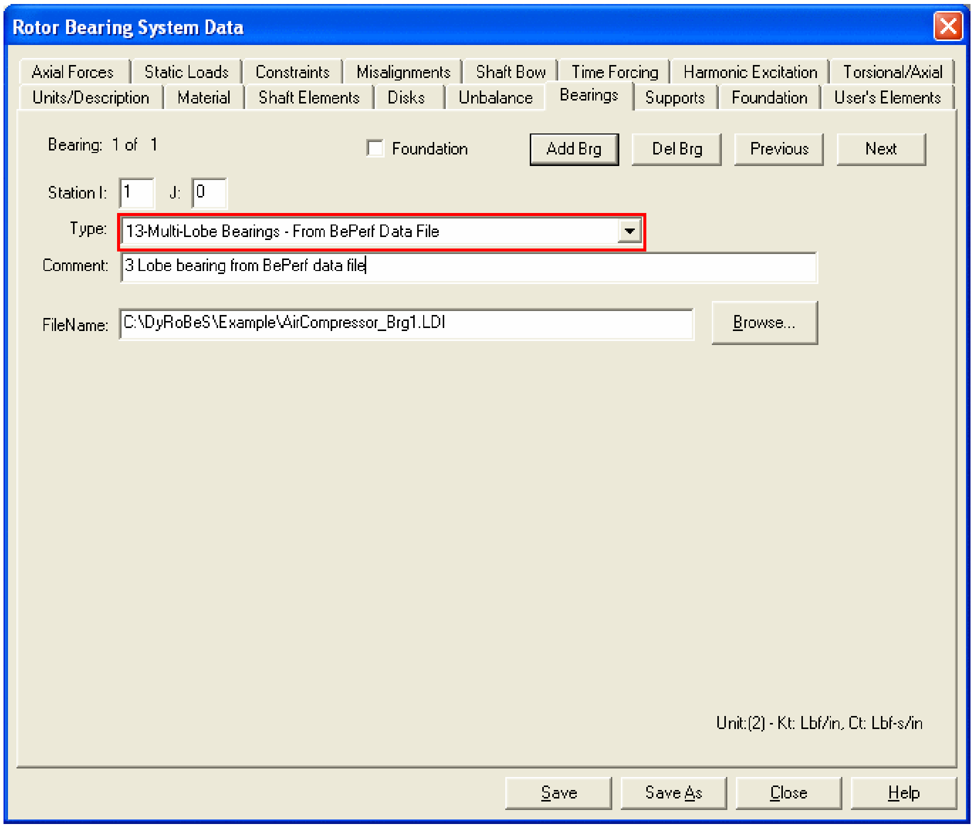
For bearing Type 12, the pads must be identical and the bearing data are entered in this input page as shown below. The angles used to describe the lobe arc are measured from +X axis. The bearing clearance for each pad is continuous and only pressure is the unknown for each finite element node in the Reynolds equation.







For bearing Type 13, it can be any fixed lobe geometric bearing as defined in BePerf. The bearing clearance can be discontinuous in this bearing type and a total of 3 DOFs (pressure and pressure gradients) are defined for each finite element node in the Reynolds equation.



See also [Bearings]() and See also ***[DyRoBeS©_BePerf]()***.

Flexible Supports

Two translational displacements are used to describe the motion of a flexible support, i.e. two degrees-of-freedom at each support station. A support is connected to a rigid ground through the support stiffness and damping. Note that the support stiffness and damping can be zero if the support is not connected to the rigid ground; however, the support mass cannot be zero since it defines the degrees-of-freedom. The flexible supports are considered to be non-rotating components. The rotational displacements of a flexible support are constrained automatically in the modeling process.

Station I: Support station number. Note that a support station has to connect to the rotor station or another support by a bearing.

Comment: Description of this support.

Coefficients: Mass, Damping and Stiffness Coefficients

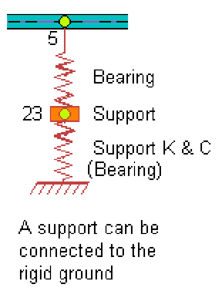
MxxMxyMyxMyy

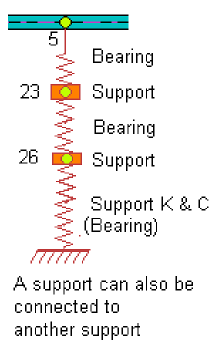
Cxx CxyCyxCyy

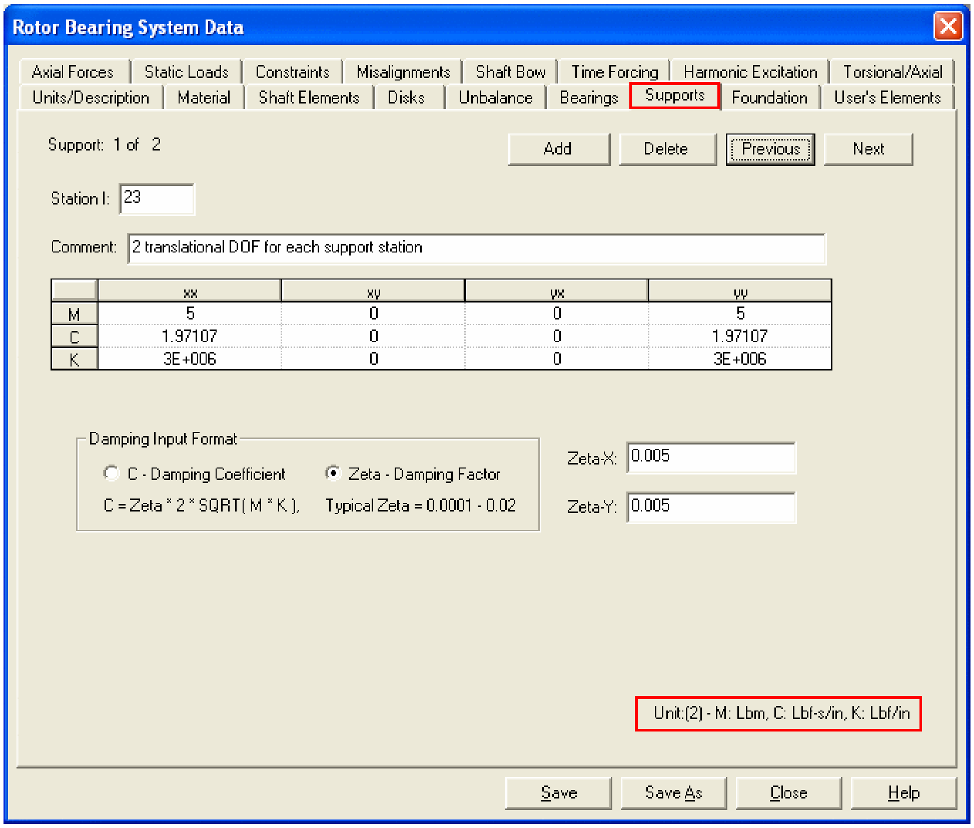
Kxx KxyKyxKyy

Damping Input:

The damping effect can be entered in two ways, that is, either the direct damping coefficients or the modal damping factor. For the structure damping, sometimes it is hard to specify the damping coefficients. Damping factor can be entered as an alternative input. Typical damping factor ranges from 0.0001 to 0.02.







See also Bearings.

Foundation

In contrast to a two (2) degrees-of-freedom flexible support, a foundation can possess NDOFF degrees-of-freedom and NDOFF >= 2. If not specified, foundation is considered to be rigid. The foundation matrices are usually created by using Guyan reduction from general Finite Element Analysis programs, such as NASTRAN or ANSYS, and only the coordinates which are pertinent to the rotor motion are retained. The foundation is connected to the rotor assembly by bearings. The foundation box needs to be checked in the bearing connected to the foundation. The (X,Y,Rx,Ry) are the coordinate entries in the foundation matrices. When foundation is included, one should always review the text output from Lateral Vibration Model Summary, which includes the modal analysis for the foundation alone. This can be used to verify the foundation data. For static deflection and critical speed analysis, foundation is neglected and assumed to be rigid.

```
 ****************************** Foundation Data *****************************
  C:\DyRoBeS\Example\LargeTurbine\Foundation.fdn
  Foundation DOF = 15
 ************************  Foundation Modal Analysis  **********************

   Mode   Damping Factor   Damping Coef.    Damped Freq.(R/S)        Hz

     1     .15534E-01        -96.522          6212.8            988.81
     2     .69470E-02        -96.522          13894.            2211.2
     3     .80217E-02        -128.70          16043.            2553.3
:::::::::::::::::::::::::::::::::::::::::::::::::::::::::::::::::::::::::::::::
:::::::::::::::::::::::::::::::::::::::::::::::::::::::::::::::::::::::::::::::
    15     .21968E-01        -965.22          43926.            6991.1

****************************************************************************
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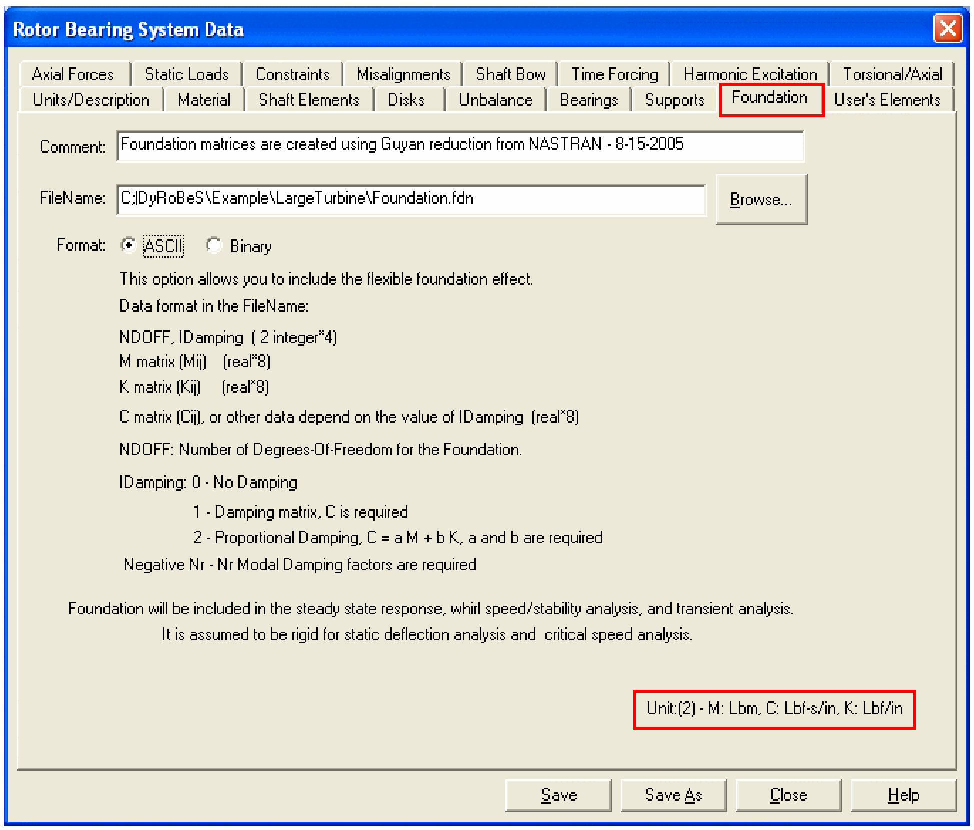
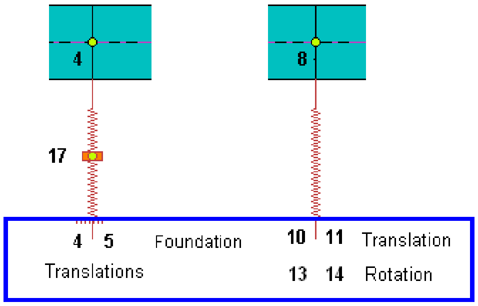
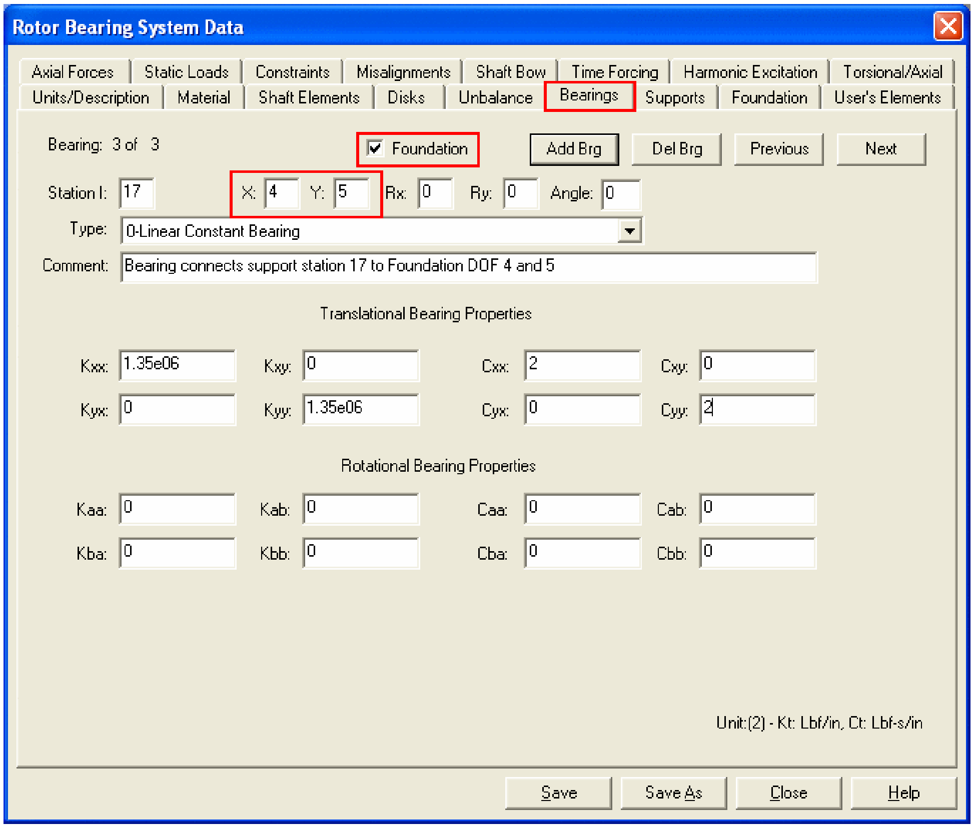
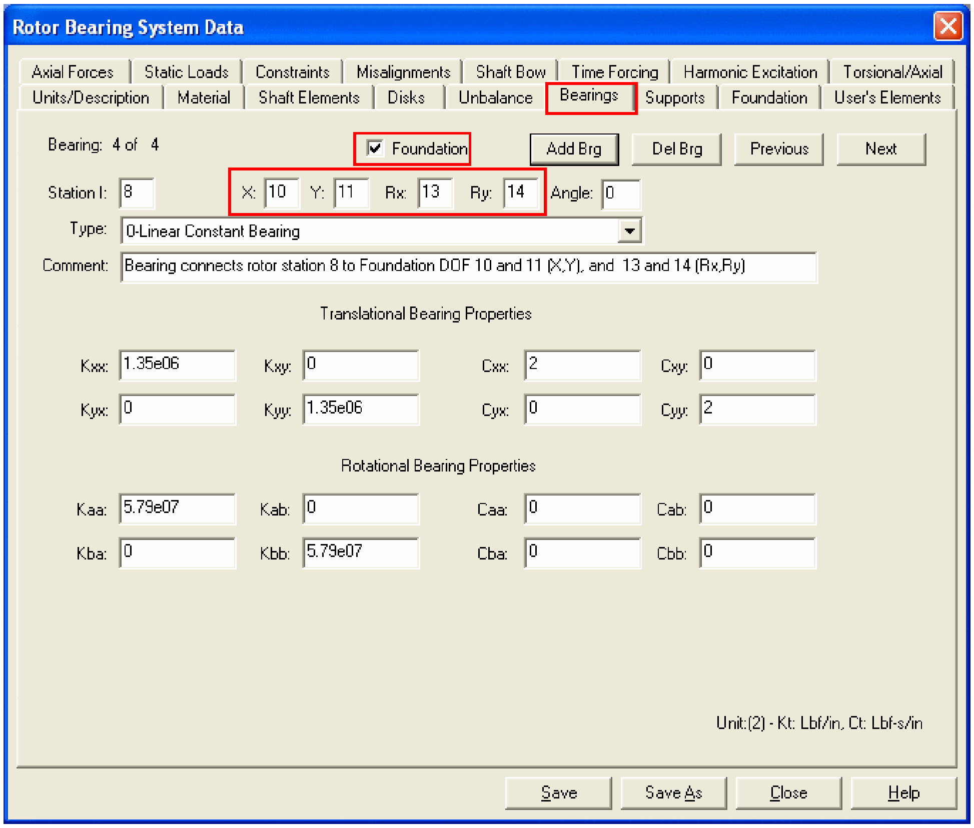
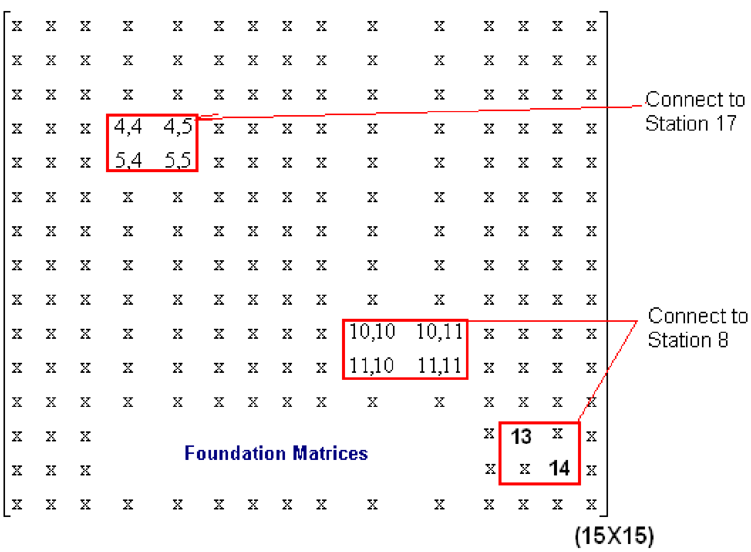


Example: The foundation has 15 degrees-of-freedom and has two connections to the rotor system. The first connection is from a support station 17 to the Foundation DOF 4 and 5. The second connection is from a rotor station 8 to the Foundation DOF 10 and 11 for the X and Y translational displacements, and 13 and 14 for the Rx and Ry rotational displacements.


4
8
17
4 5 Foundation
Translations
10 11 Translation
13 14 Rotation

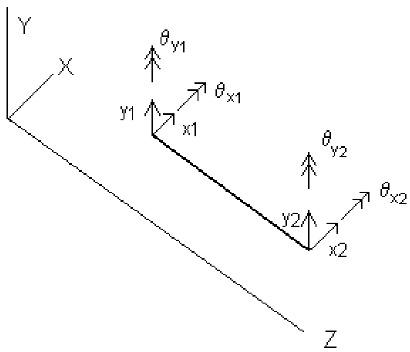

Rotor Bearing System Data
Axial Forces
Static Loads
Constraints
Misalignments
Shaft Bow
Time Forcing
Harmonic Excitation
Torsional/Axial
Units/Description
Material
Shaft Elements
Disks
Unbalance
Bearings
Supports
Foundation
User's Elements
Bearing: 3 of 3
Foundation
Add Brg
Del Brg
Previous
Next
Station I: 17
X: 4 Y: 5 Rx: 0 Ry: 0 Angle: 0
Type: 0-Linear Constant Bearing
Comment: Bearing connects support station 17 to Foundation DOF 4 and 5
Translational Bearing Properties
Kxx: 1.35e06 Kxy: 0 Cxx: 2 Cxy: 0
Kyx: 0 Kyy: 1.35e06 Cyx: 0 Cyy: 2
Rotational Bearing Properties
Kaa: 0 Kab: 0 Caa: 0 Cab: 0
Kba: 0 Kbb: 0 Cba: 0 Cbb: 0
Unit:(2) - Kt: Lbf/in, Ct: Lbf-s/in
Save
Save As
Close
Help


Rotor Bearing System Data
Axial Forces
Static Loads
Constraints
Misalignments
Shaft Bow
Time Forcing
Harmonic Excitation
Torsional/Axial
Units/Description
Material
Shaft Elements
Disks
Unbalance
Bearings
Supports
Foundation
User's Elements
Bearing: 4 of 4
Foundation
Add Brg
Del Brg
Previous
Next
Station I: 8
X: 10
Y: 11
Rx: 13
Ry: 14
Angle: 0
Type: 0-Linear Constant Bearing
Comment: Bearing connects rotor station 8 to Foundation DOF 10 and 11 (X,Y), and 13 and 14 (Rx,Ry)
Translational Bearing Properties
Kxx: 1.35e06
Kxy: 0
Cxx: 2
Cxy: 0
Kyx: 0
Kyy: 1.35e06
Cyx: 0
Cyy: 2
Rotational Bearing Properties
Kaa: 5.79e07
Kab: 0
Caa: 0
Cab: 0
Kba: 0
Kbb: 5.79e07
Cba: 0
Cbb: 0
Unit:(2) - Kt: Lbf/in, Ct: Lbf-s/in
Save
Save As
Close
Help


4,4
4,5
5,4
5,5
Connect to Station 17
10,10
10,11
11,10
11,11
Connect to Station 8
13
14
Foundation Matrices
(15X15)

User's Elements

This option allows you to specify your own subelement matrices. An element is assumed to be isotropic, with the (X-Z) and (Y-Z) planes having identical dynamic properties. The rotor shaft centerline is located along the Z-axis. Since the element is isotropic, the stiffness matrix in the X-Z plane is required only. The coordinates for a typical element are shown in the following figure.



The lumped mass system is used for the Use's Element. The element mass and diametral moment of inertia at both end (1-Left and 2-Right) are entered to establish the mass matrix. The mass matrix in the (X-Z) plane will be:

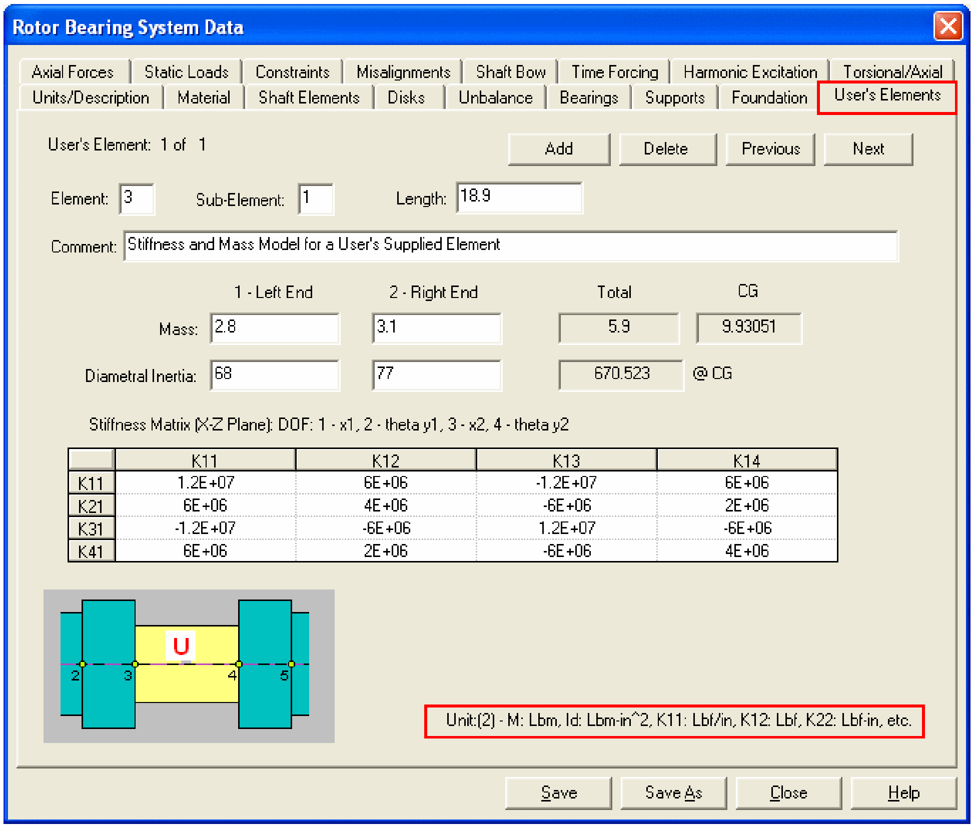
$$\begin{bmatrix} m_L/2 & 0 & 0 & 0 \\ 0 & I_{d,L}/2 & 0 & 0 \\ 0 & 0 & m_R/2 & 0 \\ 0 & 0 & 0 & I_{d,R}/2 \end{bmatrix} \quad \text{for} \quad \begin{Bmatrix} \ddot{x}_1 \\ \ddot{\theta}_{y1} \\ \ddot{x}_2 \\ \ddot{\theta}_{y2} \end{Bmatrix}$$

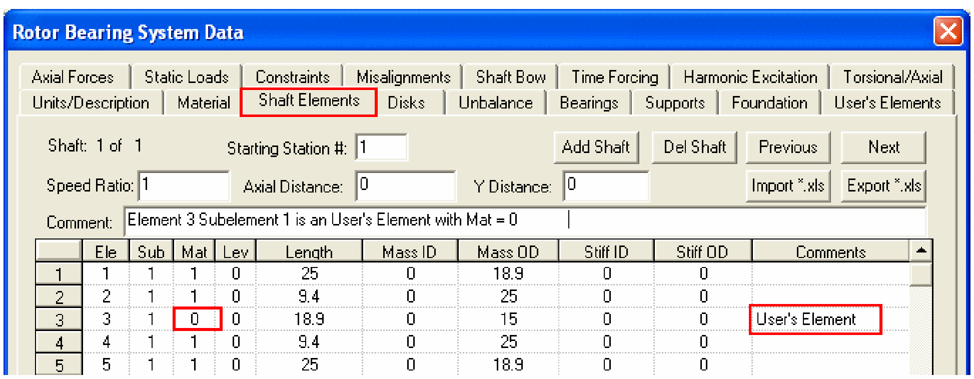
The total element mass, inertia, and CG from the left end are calculated automatically for reference purpose. The element length is needed for the CG and inertia about CG calculation, but, it does not affect the system dynamic results. These mass properties can also be entered in the Rigid/Flexible Disks tab as disks at both ends, and not entered here.

The user supplied (4x4) stiffness matrix in the (X-Z) plane will be:

$$\begin{bmatrix} K_{11} & K_{12} & K_{13} & K_{14} \\ K_{21} & K_{22} & K_{23} & K_{24} \\ K_{31} & K_{32} & K_{33} & K_{34} \\ K_{41} & K_{42} & K_{43} & K_{44} \end{bmatrix} \quad \text{for} \quad \begin{Bmatrix} x_1 \\ \theta_{y1} \\ x_2 \\ \theta_{y2} \end{Bmatrix}$$

The stiffness matrix is symmetric, therefore only the upper half is used in the formulation of the element stiffness matrix. The **Material Number** in the Shaft Elements data page should be set to **0** for User Supplied Elements.





See also Shaft Elements,

Axial Force and Torque

This option allows you to include the axial forces and torques effects in the model. Note that this option is used for the lateral vibration only. For axial force, tension is defined to be positive in magnitude and compression is defined to be negative. For axial torque, Right Hand Rule is used for the sign convention. A positive axial torque vector points in the positive outward normal direction at the boundary element.

1. Stn From: Left starting station number.

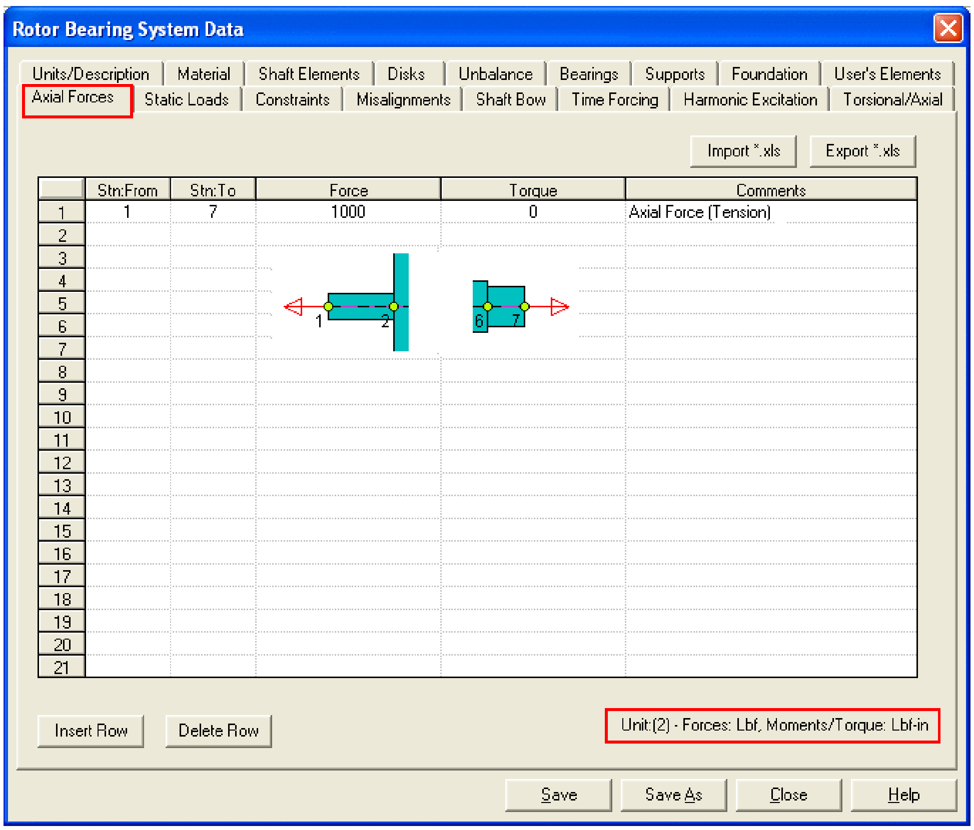
2. Stn To: Right ending station number.

3. Force: Axial force value.

4. Torque: Axial torque value.

5. Comment: Description.

The ***<Import *.xls>*** and ***<Export *.xls>*** buttons allow you to import and export data from and to the MS Excel file.



Static Loads

The static loads are used in the Static Deflection and Bearing Load Analysis, Transient Analysis, and Static Maneuver Load Analysis. The static loads are externally applied loads, such as gear loads. The static loads due to gravity are specified in the run time data folder with gravity constants, not here.

1. Stn: Station number where the static loads are applied.

2. Fx: Force in the X direction.

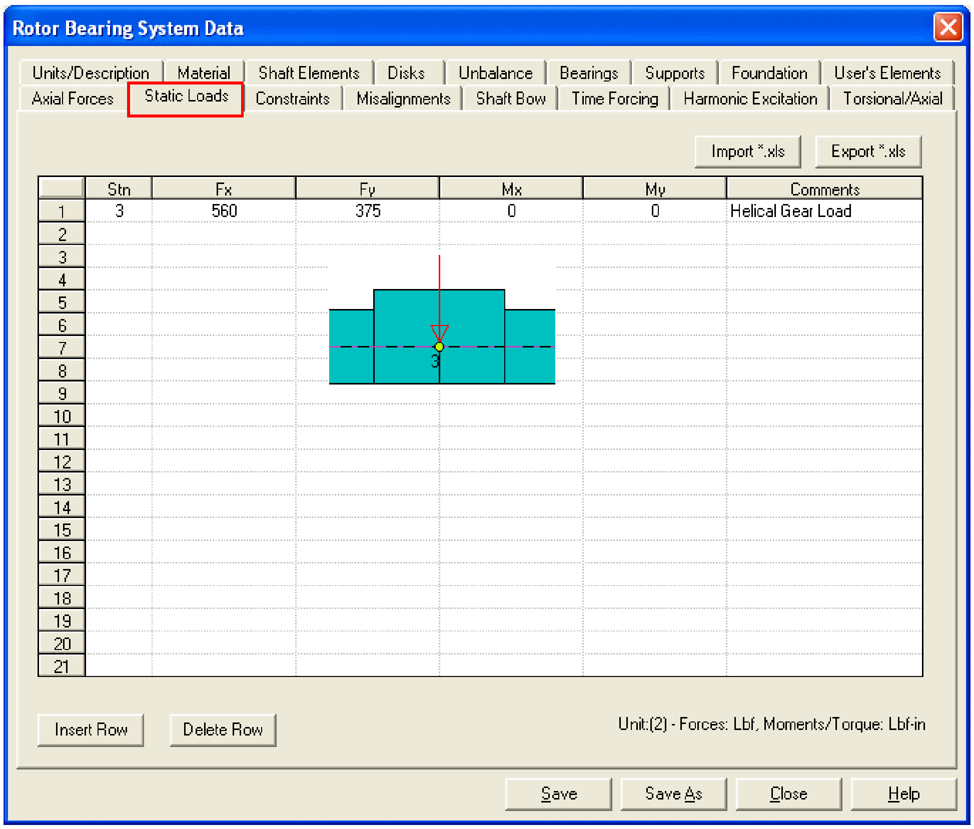
3. Fy: Force in the Y direction.

4. Mx: Moment about the X direction.

5. My: Moment about the Y direction.

6. Comments: Description.

The ***<Import *.xls>*** and ***<Export *.xls>*** buttons allow you to import and export data from and to the MS Excel file.



Constraints

This option allows you to model the geometric and natural boundary conditions. Note that this option is used for the lateral vibration only. For the axial and torsional vibrations, the constraints are entered under the Connectivity of Torsional/Axial Data folder. In the practice of rotordynamics, the geometric constraints are seldom used. They are provided in the program for the theoretical verification with many closed form solutions. The natural boundary conditions (Shear/Moment Release) are used mainly for the simulation of a spline or a flexible **coupling** with moment release.

1. Stn: Station number where the constraint is imposed.

2. x: Translational displacement in the X axis (Fixed or None).

3. y: Translational displacement in the Y axis (Fixed or None).

3. Theta x: Rotational displacement about X axis (Fixed or None).

4. Theta y: Rotational displacement about Y axis (Fixed or None).

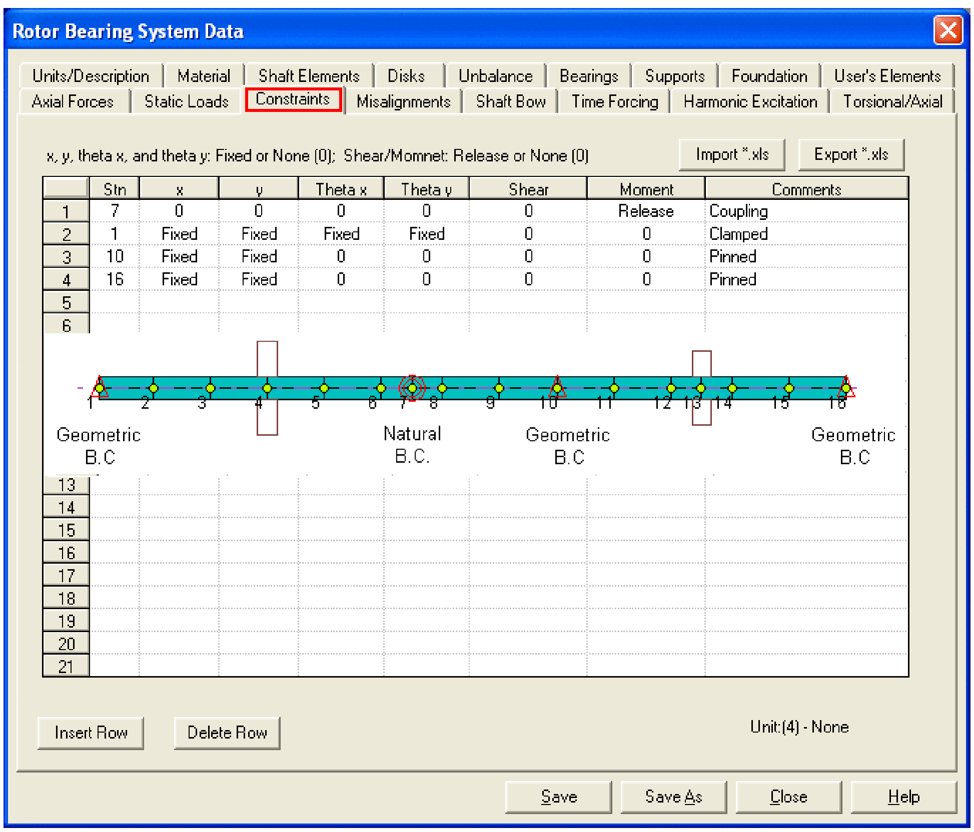
5. Shear: Shear force release (Release or None).

6. Moment: Moment release (Release or None).

7. Comment: Description.

Where **x, y, Theta x,** and **Theta y** are geometric boundary condition. **Shear** and **Moment** are natural boundary condition. If both geometric and natural constraints are applied at the same finite element station, the natural boundary conditions are ignored.

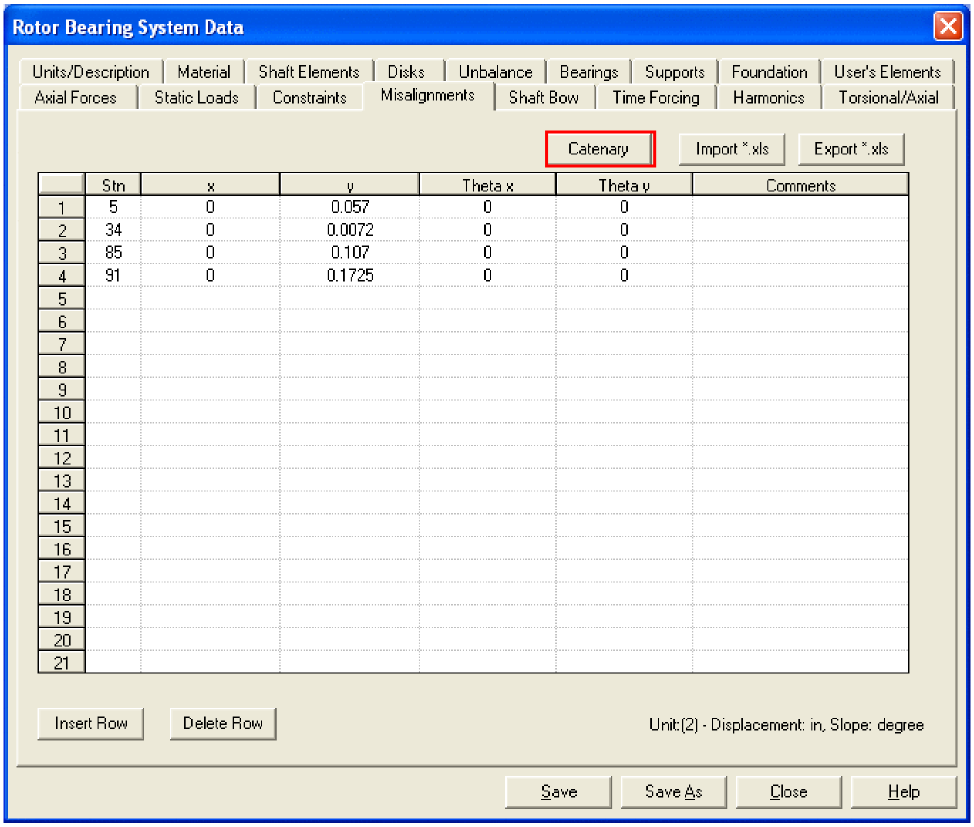
The ***<Import *.xls>*** and ***<Export *.xls>*** buttons allow you to import and export data from and to the MS Excel file.



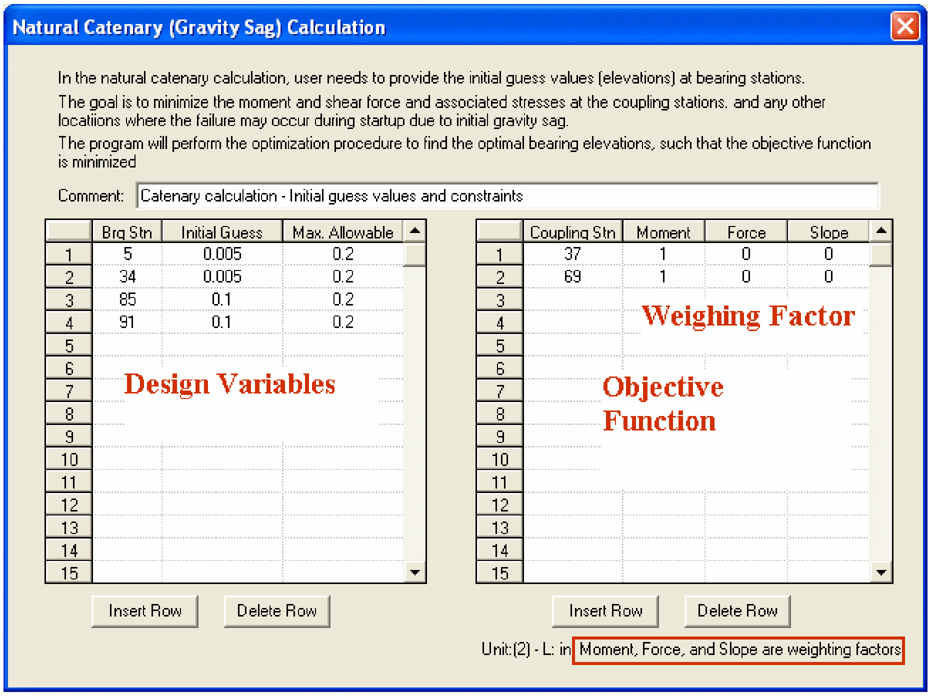
Misalignments

Misalignments on the bearings are commonly performed for a large turbine-generator set to obtain a desirable catenary's curve, which produces minimal bending stresses at the coupling rigid flange faces. Misalignment also creates synchronous excitation.

The ***<Import *.xls>*** and ***<Export *.xls>*** buttons allow you to import and export data from and to the MS Excel file. The ***<Catenary>*** button allows you to perform the automatic Natural Catenary Analysis option (Analysis type == 11) using optimization technique



Catenary button

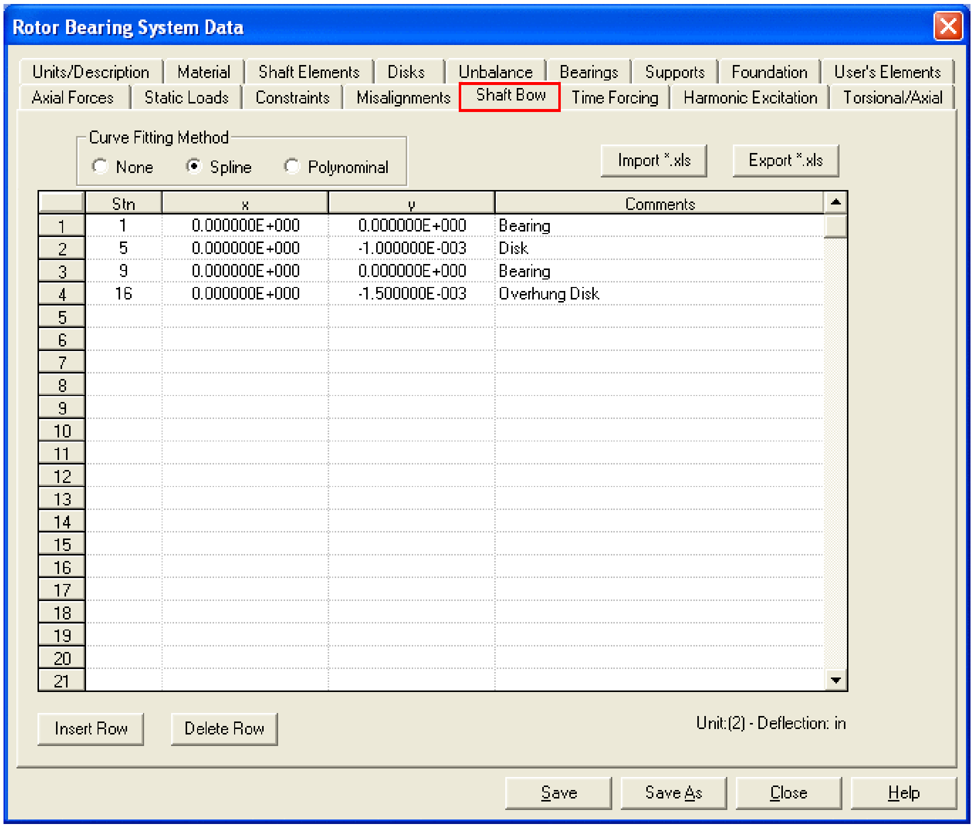


For more catenary (gravity sag) analysis, click Catenary (Gravity Sag) Analysis

Shaft Bow

This option allows you to model the shaft residual bow effect. The shaft bow generates synchronous excitation with speed independent magnitude. Since it is not feasible and not practical to input the residual shaft bow (displacements and slopes) for all the finite element stations, two curve fitting options are provided in the program. By selecting **None**, the unspecified displacements and slopes are set to be zero. By selecting **Spline** or **Polynomial**, the unspecified displacements are interpolated or extrapolated from the given displacements and the slopes are derived from the displacements. The disk skew caused by the shaft bow is added to the total disk skew angle.

The ***<Import *.xls>*** and ***<Export *.xls>*** buttons allow you to import and export data from and to the MS Excel file.



Time Forcing Functions

This option allows you to model almost any type of excitation forces/moments, such as synchronous (unbalance, blade loss, etc.), non-synchronous (gear mesh, etc.), any harmonics, step, constant, impulse, etc. The excitation is applied when time (t) is between t1 and t2. Multiple excitations can be applied at the same station. The combination of various types of excitation allows you to model almost any forms of excitation.

1. Stn: Station number where the excitation is applied.

2. Dir: Coordinate at which the excitation is applied. 1 – force in x direction, 2- force in y direction, 3 – moment about x axis, 4 – moment about y axis. 0 – forces in X and Y directions for type 5 only.

3. Type: Excitation type. Various types are defined below.

4. Start (t1): Starting time at which the excitation is applied.

5. Stop (t2): Stop time at which the excitation is ended.

6. Par 1, 2, 3, 4: Parameters used to define the excitation. They are defined below.

7. Comment: Description.

Various types of excitations are defined below. Note that the excitation is applied when $t_1 \le t < t_2$.

Type = 0 Exponentially Decay Force

$$F = F_m\, e^{(-\lambda(t-t_1))} \cos(\omega_{exc}(t-t_1)+\phi)$$

where, $F_m = \mathrm{Par1},\ \lambda = \mathrm{Par\,2},\ \omega_{exc} = \mathrm{Par\,3\ (rpm)}, \phi = \mathrm{Par4\ (degree)}$ are the input parameters. By properly adjusting these parameters, the excitation can be in many other forms. For example:

if $\lambda = 0$	the force is a purely harmonic force.
If $\lambda = 0,\ \omega_{exc} = 0,\ \phi = 0$	the force is a step constant

Type = 1 Purely Harmonics

$$F = F_c\ \cos(\omega_{exc}(t-t_1)) + F_s\ \sin(\omega_{exc}(t-t_1))$$

where, $F_c = \mathrm{Par1},\ F_s = \mathrm{Par2},\ \omega_{exc} = \mathrm{Par\,3\ (rpm)}$. *Fc* and *Fs* are constant amplitude and ω_{exc} is the excitation frequency.

Type = 2 Step Constant

$$F = F_0$$

where, $F_0 = \mathrm{Par1}$. If t2 = t1, or the time interval is very small, then the excitation becomes impulse force. Caution must be taken in the case of the impulse excitation, that is, t1 must be in the discrete time point.

Type = 3 Linear function

$$F = F_1 \text{ at } \mathrm{t_1} \quad \text{, and} \quad F = F_2 \text{ at } \mathrm{t_2}$$

where, $F_1 = \mathrm{Par1}$, and $F_2 = \mathrm{Par2}$

Type = 4 Polynomial function

$$F = F_0 + F_1(t-t_1) + F_2(t-t_1)^2 + F_3(t-t_1)^3$$

where, $F_0 = \mathrm{Par1},\ F_1 = \mathrm{Par2},\ F_2 = \mathrm{Par3},\ F_3 = \mathrm{Par4}$

Type = 5 Purely Harmonics with speed dependent amplitude

$$F = F_c\ (\mathrm{n}\Omega)^{\mathrm{m}} \cos(\mathrm{n}\Omega(t-t_1)) + F_s\ (\mathrm{n}\Omega)^{\mathrm{m}} \sin(\mathrm{n}\Omega(t-t_1))$$

where, $F_c = \mathrm{Par1},\ F_s = \mathrm{Par2},\ m = \mathrm{Par\,3}, n = \mathrm{Par\,4}$. *Fc* and *Fs* are constant amplitude and Ω is the rotational speed. n is the multiple of the rotational speed, m is the power of the (nΩ). To simulate the blade loss (unbalance), n=1 and m=2, and one needs two entries (one in x direction and one in y direction), such as:

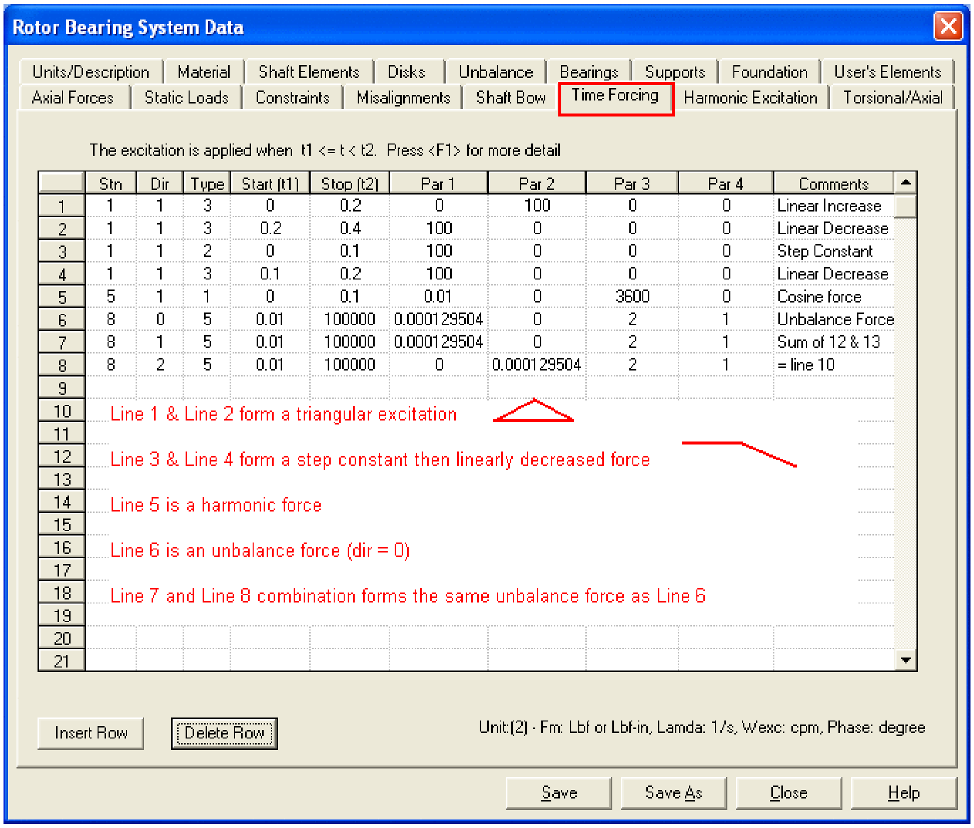
in X-direction: $F_x = |F|\,\Omega^2 \cos(\Omega(t-t_1))$,Par1 =|F|, Par2 = 0, Par3 = 2, Par4=1

in Y-direction: $F_y = |F|\,\Omega^2 \sin(\Omega(t-t_1))$, Par1 =0, Par2 = |F|, Par3 = 2, Par4=1

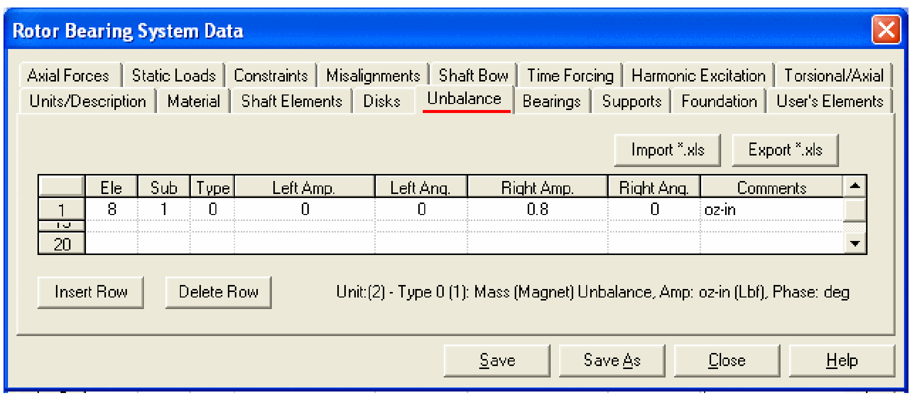
To simulate the blade loss, another option can be used. Set Dir == 0. Then, only one line input is needed. Dir == 0, then, Par1 = |F|, Par2 = Phase (deg), Par3 = 2, Par4 = 1. This single line input will then generate the necessary X- and Y- direction forces as follows:

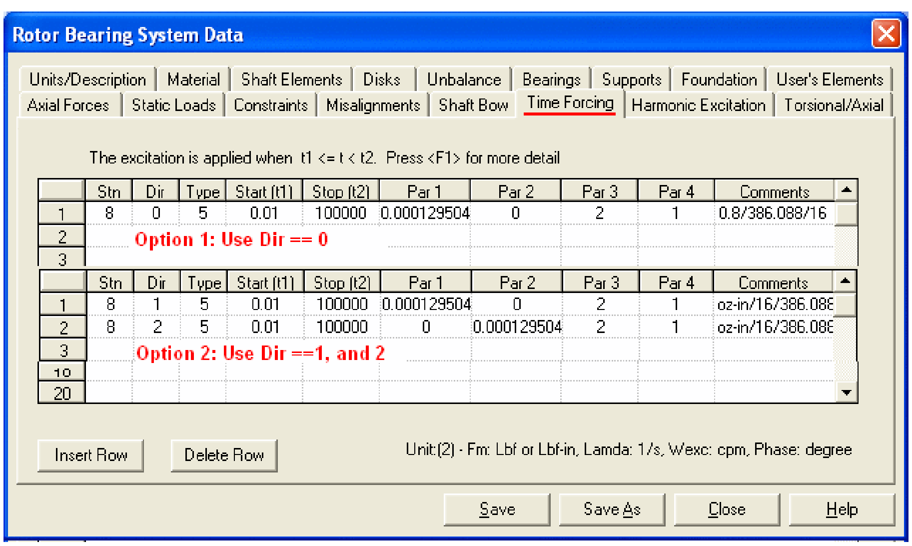
in X-direction: $F_x = |F|\,\Omega^2 \cos(\Omega(t-t_1)+\phi)$

in Y-direction: $F_y = |F|\,\Omega^2 \sin(\Omega(t-t_1)+\phi)$



For a standard unbalance, two options can be used to simulate this unbalance and they are listed below:





Steady State Harmonic Excitations

For steady state harmonic excitation analysis, the excitation is expressed in a general form:

$$Q = |Q| \cdot \cos(\omega t + \alpha)$$

where |Q| is the excitation amplitude and ω is the excitation frequency in rad/sec for the above expression.

For general purpose, the excitation frequency can be any of the following types:

1. Harmonic excitation frequency is a function of rotor speed. This also includes a constant excitation frequency or multiple of the rotor speed. The excitation frequency can be expressed as:

$$\omega_{exc}(\text{cpm}) = \omega_0 + \omega_1 \cdot \text{rpm} + \omega_2 \cdot \text{rpm}^2$$

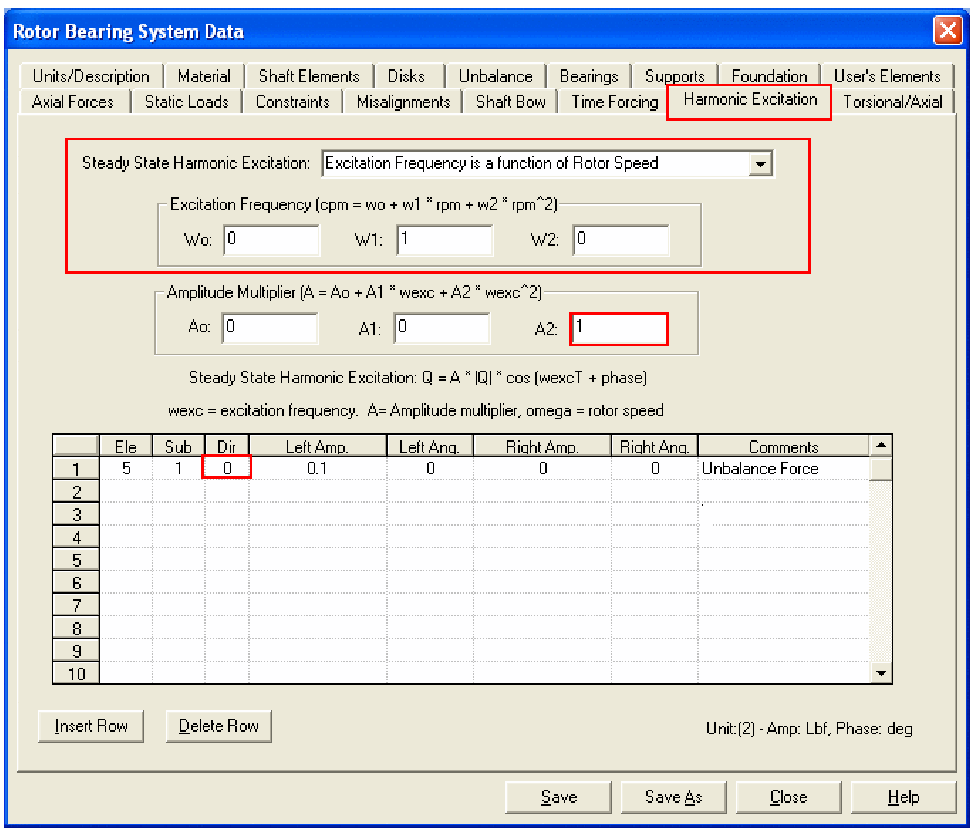
where rpm is the rotor rotational speed.

2. Excitation frequency varies at a constant rotor speed. The excitation frequency is independent of the rotor speed in this option. This excitation is commonly caused by other machines near by the machine under study.

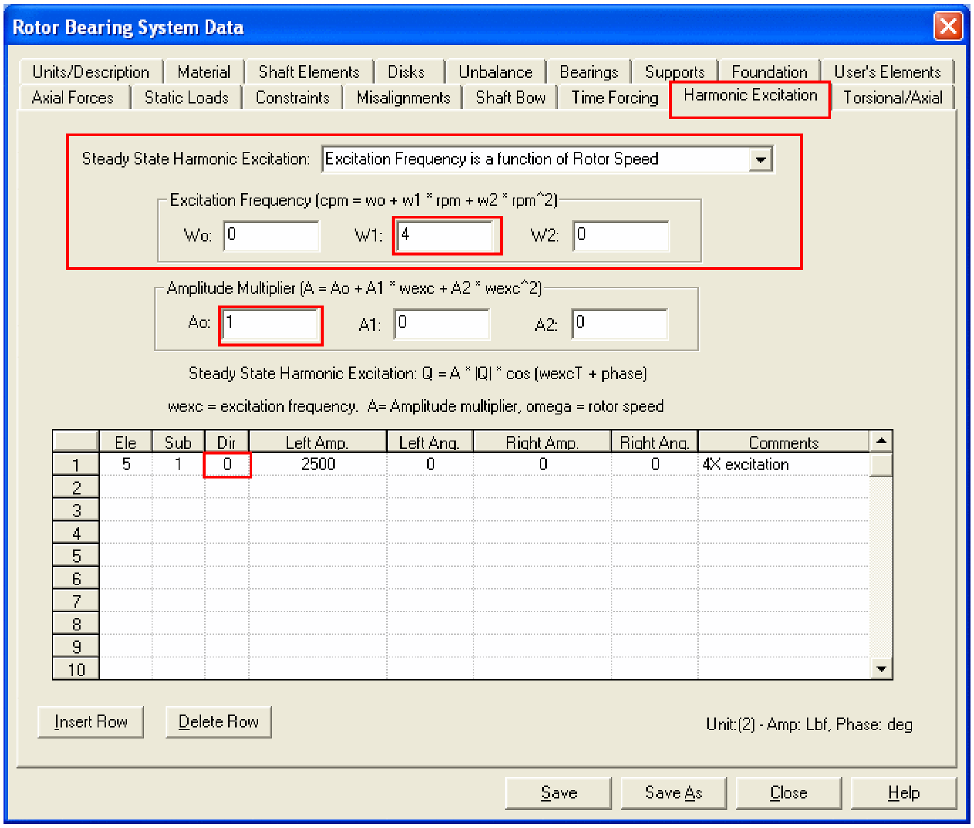
The excitation amplitude can also be expressed in terms of excitation frequency:

$$|Q| = \left(A_0 + A_1 \cdot \omega + A_2 \cdot \omega^2\right) |A|$$

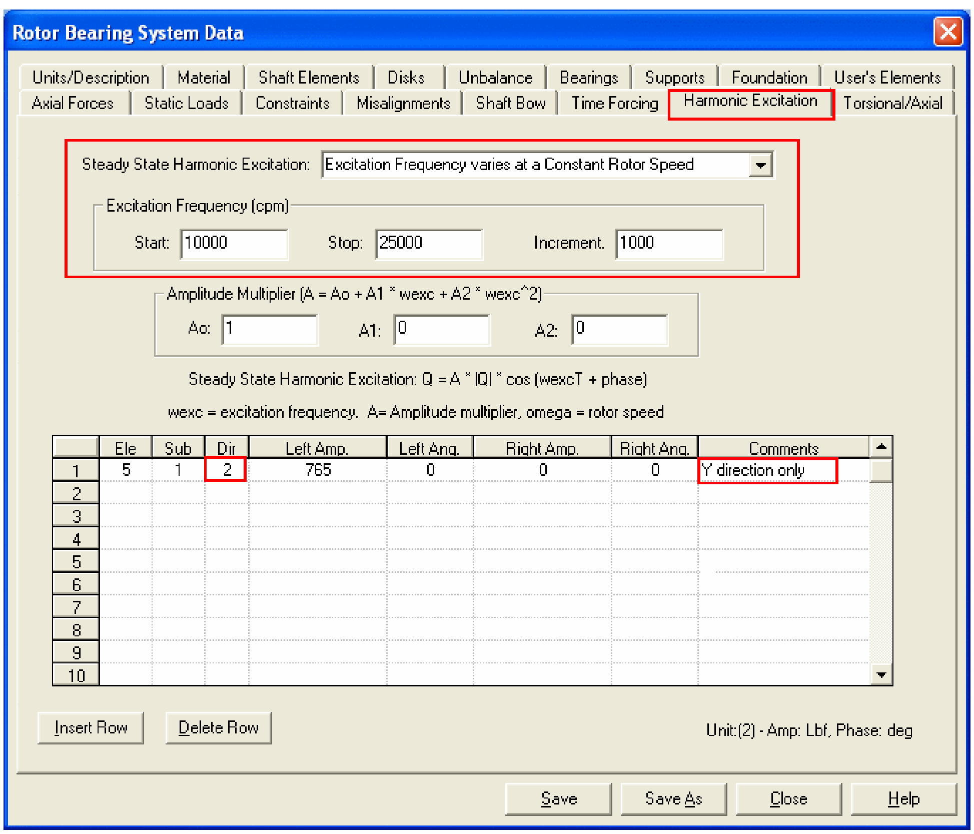
Depending upon the excitation frequency type, the input screen can be different. For a conventional unbalance force:



For a constant harmonic excitation in both X and Y directions with an excitation frequency of 4 X the rotor speed:



Excitation frequency caries at a constant rotor speed is shown below. This is caused by the machine next to the one under study.



Note that Dir parameter can be: 0, 1, 2, 3, and 4. If Dir == 0, the force is a rotating force with

$Q_x = |Q| \cdot \cos(\omega t + \alpha)$ in X direction, and

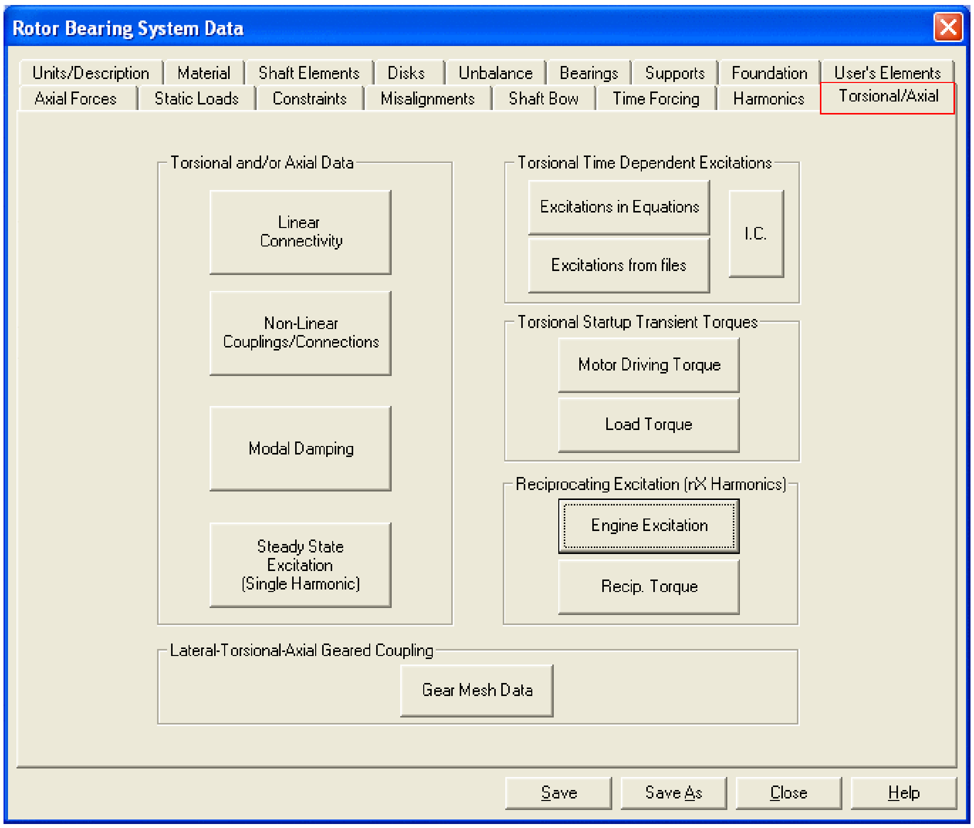
$Q_y = |Q| \cdot \sin(\omega t + \alpha)$ in Y direction

If Dir ==1, 2, 3, and 4, the force (or moment) is acting in x, y, rotation x, and rotation y, respectively. For variable excitation frequency, if w0=w2=0 and w1=1, this will be a synchronous excitation.

Torsional/Axial Data

This folder allows you to input the additional data required for Torsional and/or Axial vibrations. The buttons for Connectivity, Modal Damping, and Steady State Excitation are for torsional or axial vibration. The buttons for Non-Linear Coupling, the time dependent excitations in equations entered in Torsional Excitations in Equation, the time dependent excitations in data files entered in Torsional Excitations in Data Files, Driving Torque, and Load Torque are only for torsional transient analysis. Engine Excitation is the excitation inputs for the gas or diesel engines. Reciprocating Excitation is the steady state torsional excitation from the reciprocating machine. Initial Condition is used to specify the torsional displacement (theta) and velocity (theta dot) initial conditions (rad, rad/sec) for the torsional time transient analysis.

The Connectivity allows you to input the linear stiffness and damping or constraints. The Modal Damping allows you to input the damping factors, if the direct damping is not available. The Non-Linear Connection/Coupling allows you to input the nonlinear couplings and connections. The Steady State Excitation allows you to input the excitations for steady state forced torsional analysis. The time dependent torsional excitations allow you to input the excitation torques in a form of either in equations or from a data file. The Driving Torque and Load Torque are the torques required for the Torsional Transient Startup Analysis, which are speed dependent.



If you want to analyze the coupled Lateral-Torsional-Axiial vibration for a geared system, then you will need to enter Gear Mesh Data..

See also Connectivity, Modal Damping, Non-Linear Coupling, Steady State Excitation, Time Dependent Torsional Excitations, Driving Torque, Load Torque, Engine Excitation, Reciprocating Excitation, Gear Mesh Data.

Torsional/Axial Connectivity Data

This option allows you to input the connectivity and constraints for the torsional and/or axial vibration data. For torsional and axial vibrations, the system can be continuous by using the shaft elements and/or discrete by using the external connectivity, or the combination of continuous and discrete model. The discrete data is entered in this option. The connectivity links station I and station J.

1. T/A: **T**orsional or **A**xial vibration (enter either **T** or **A**).

2. Stn I: Station number I.

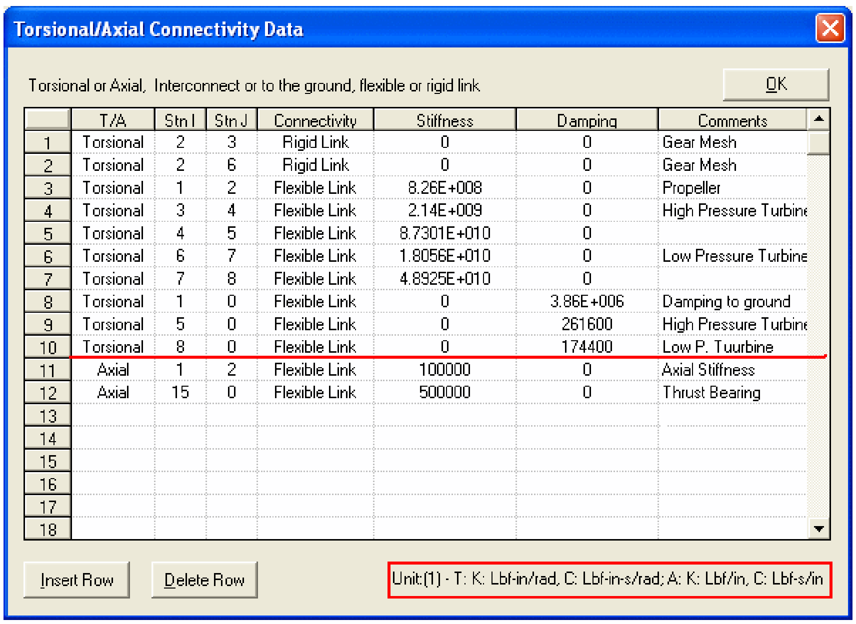
3. Stn J: Station number J. If J = 0, station I is connected to the rigid ground.

4. Connectivity: **R**igid or **F**lexible Link. Rigid link indicates that Station I and Station J are rigidly linked and the displacements at Station I and Station J are identical (for example, rigid gear meshes). Flexible link indicates that the Stations I and J are connected by the external stiffness and damping.

5. Stiffness: Used in Flexible Link to connect Station I and Station J.

6. Damping: Used in Flexible Link to connect Station I and Station J.

7. Comment: Description.



	T/A	Stn I	Stn J	Connectivity	Stiffness	Damping	Comments
1	Torsional	2	3	Rigid Link	0	0	Gear Mesh
2	Torsional	2	6	Rigid Link	0	0	Gear Mesh
3	Torsional	1	2	Flexible Link	8.26E+008	0	Propeller
4	Torsional	3	4	Flexible Link	2.14E+009	0	High Pressure Turbine
5	Torsional	4	5	Flexible Link	8.7301E+010	0	
6	Torsional	6	7	Flexible Link	1.8056E+010	0	Low Pressure Turbine
7	Torsional	7	8	Flexible Link	4.8925E+010	0	
8	Torsional	1	0	Flexible Link	0	3.86E+006	Damping to ground
9	Torsional	5	0	Flexible Link	0	261600	High Pressure Turbine
10	Torsional	8	0	Flexible Link	0	174400	Low P. Tuurbine
11	Axial	1	2	Flexible Link	100000	0	Axial Stiffness
12	Axial	15	0	Flexible Link	500000	0	Thrust Bearing
13							
14							
15							
16							
17							
18							

See other Torsional/Axial Data Torsional/Axial Data.

Torsional NonLinear Connections and Couplings

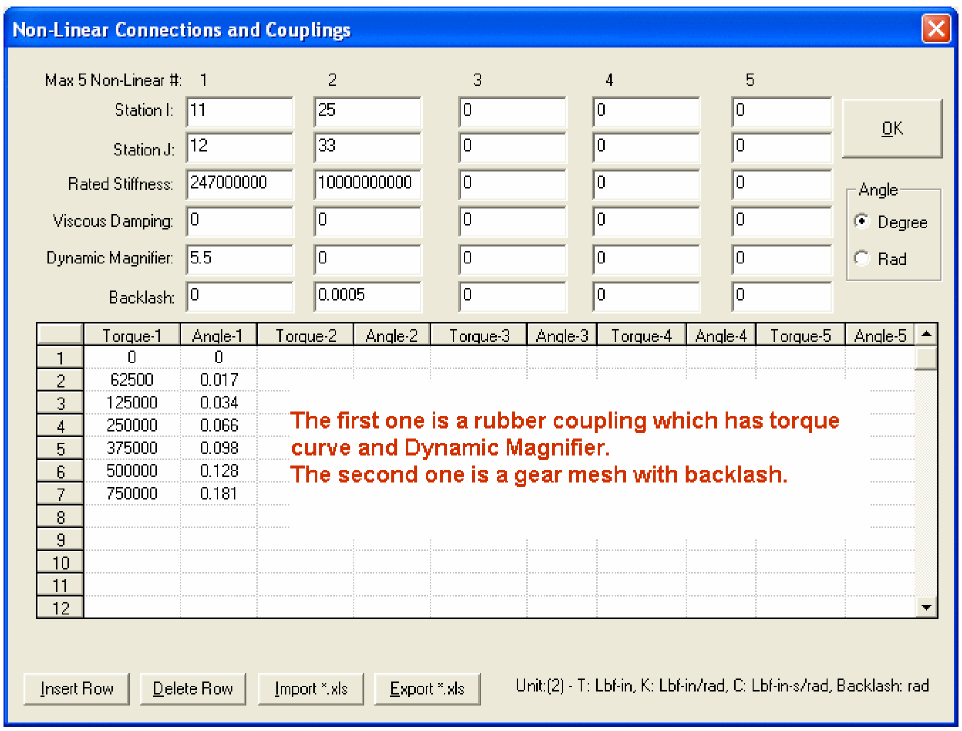
This option allows you to include up to five (5) non-linear couplings (or nonlinear connection and elements) in the torsional model. Each non-linear coupling or connection connects two stations, station I and station J. If station J is zero, that means station I is connected to the ground. Rated stiffness will be used in the linear analysis, such as frequency analysis and steady state response. It will be used in the non-linear analysis if the torque-deflection is not given in the table. Linear viscous damping can be specified in the model. Frequency dependent damping is commonly specified using the dynamic magnifier, which exists in the rubber type coupling. The damping value is obtained as:

The damping varies directly with torsional stiffness (*K*) and inversely with the dynamic magnifier (*Dm*) and excitation frequency (ω). If the torque-deflection is given, the tangent stiffness will be used, otherwise, the rated stiffness will be used in the damping calculation. This frequency dependent damping can be included in the forced response analysis, such as steady state forced response and transient analysis. Backlash is used if the torsional clearance exists, such as in the gear mesh.

Following input shows two nonlinear connections:

1. A rubber type coupling connects station 11 and 12. Both the rated stiffness and non-linear torque-deflection curve are specified. The rated stiffness will be used in the linear analysis and the torque curve will be used in the nonlinear analysis. The damping exists in the form of dynamic magnifier.

2. Second connection is for gear mesh which connects station 25 and 33. A linear stiffness and backlash are specified. Torque curve is not given, therefore, the rated stiffness will be used in both linear and non-linear analyses.



See other Torsional/Axial Data Torsional/Axial Data.

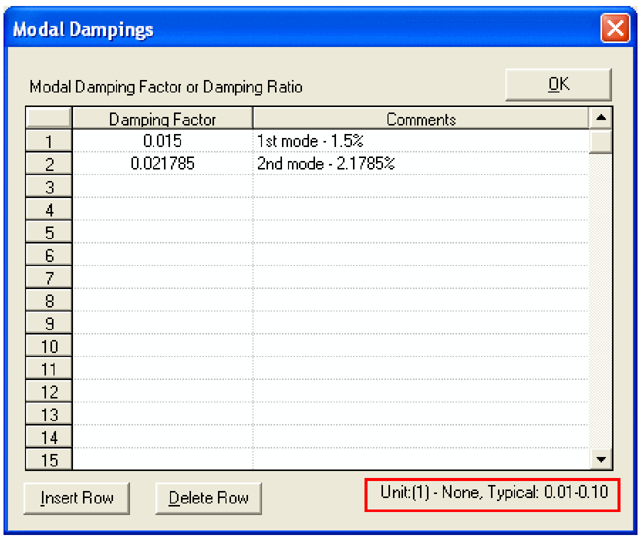
Torsional/Axial Modal Damping

This option allows you to input the modal damping factors, if the direct damping is not available. In general, the torsional damping for a geared train system is not readily available from the element level. The torsional modal damping factors for systems with dry type couplings have been reported in the range of 1-5 percent. For systems with resilient rubber couplings, the modal damping of the coupling modes, where the motions are dominated by the coupling, can go up to 6-10 percent. For practical purpose, only the first several modes are of interest, modal truncation is employed to approximate the physical damping. The modal damping factor is the ratio of the actual modal damping to the critical modal damping for a given mode. If the modal damping factor is equal to and greater than one (1), the associated mode is said to be critically damped. Typical value is between 0.01 (1 percent) to 0.1 (10 percent). The definition of modal damping factor is:

$$\xi_r = \frac{C_r}{2 J_r \omega_r} \qquad r = \text{mode of interest}$$

where

C_r, J_r, ω_r are modal damping, modal mass/inertia, and natural frequency



See other Torsional/Axial Data Torsional/Axial Data.

Torsional/Axial Steady State Excitations

This option allows you to input the torsional and/or axial steady state excitations. For torsional vibration, the excitation is torque. For axial vibration, the excitation is force. The excitation frequency can be a second order polynomial function of rotor speed, although the multiple (or fraction) of the rotor speed is the most common in the rotating machinery. By adjusting the frequency coefficients, the excitation frequency can be a constant, synchronous, or non-synchronous excitation. The torque/force multiplier provides the flexibility of including the frequency into the torque/force amplitude. By adjusting the multiplier coefficients, the excitation amplitude can be a constant, first order, or second order of the frequency (speed).

$$\omega_{exc} = \left(C_0 + C_1\,\Omega + C_2\,\Omega^2\right) \times \ (2\pi/60)$$

$$T = \left(T_c \cos(\omega_{exc} t) + T_s \sin(\omega_{exc} t)\right) \times T_{multiplier}$$

$$T_{multiplier} = M_0 + M_1\,\Omega + M_2\,\Omega^2$$

where Ω is the rotor speed (rpm)

C_0, C_1, C_2: Frequency coefficients.

M_0, M_1, M_2: Torque/force multiplier coefficients.

1. T/A Option: Torsional or Axial vibration.

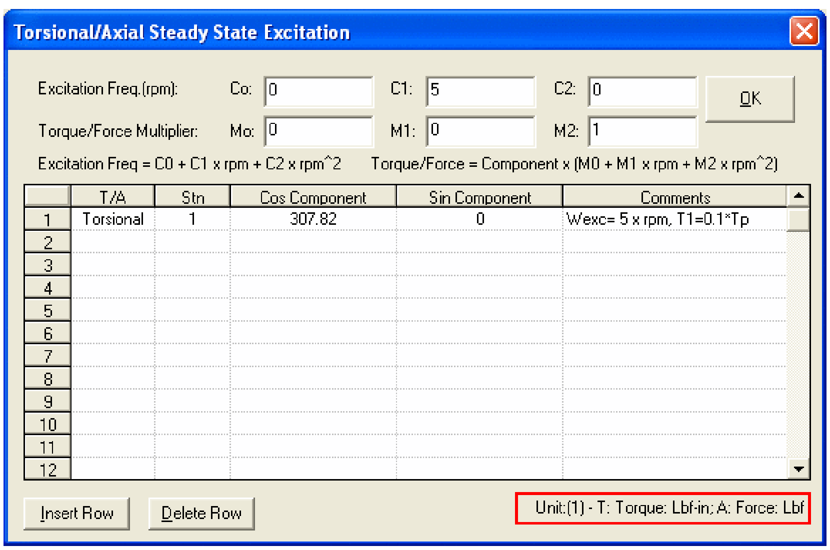
2. Stn I: Station number where the excitation is applied.

3. Cos Component: *Tc*, Cosine component of the excitation amplitude.

4. Sin Component: *Ts*, Sine component of the excitation amplitude.

5. Comment: Description.

For a single excitation, there is no phase difference between the excitations. The sine component of the excitation (*Ts*) can be set to zero and only use the cosine component (*Tc*) for the excitation amplitude.



See other Torsional/Axial Data [Torsional/Axial Data](#).

Torsional Excitations Expressed in Equations

This option allows you to input the short circuit torques and other torsional excitations that can be expressed by computable equations for torsional time transient analysis. Although short circuit torque is the most common torsional excitation given in equations, it is NOT limited by the short circuit torques. Any time dependent excitation torque expressed in the following equation can be modeled as Type zero (0) in the input page.

$$T = T_{rated}\left[T_0\, e^{-a_0 t} + T_1\, e^{-a_1 t}\sin(\omega t + \phi_1) + T_2\, e^{-a_2 t}\sin(2\omega t + \phi_2)\right]$$

where

T_{rated}, T_0, T_1, T_2 are rated torque and torque components.

a_0, a_1, a_2 are time constants.

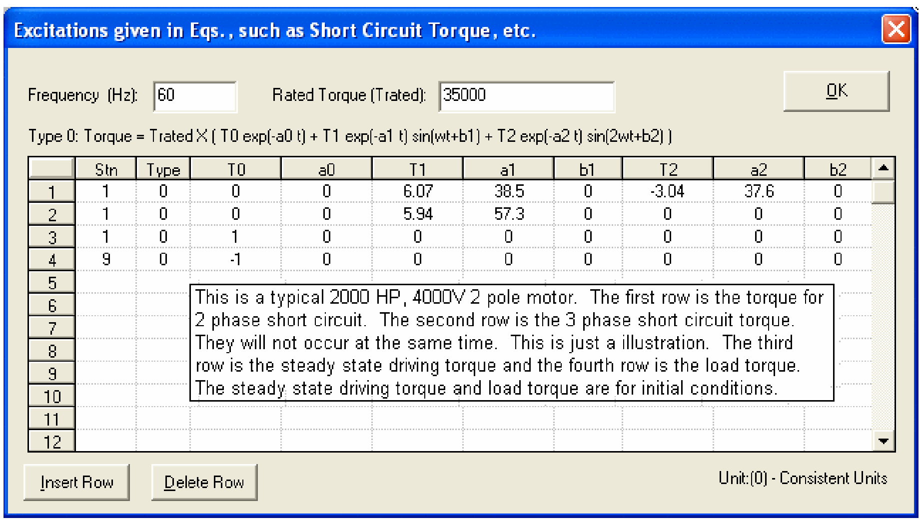
ϕ_1, ϕ_2 are phase angles (degree).

The following input page shows the transient torques for a typical 2000 Hp, 4000V, 2 pole, 60 Hz motor. Please note that the 2-phase short circuit and 3-phase short circuit do not occur at the same time. They are listed simultaneously here only for reference purposes. The steady state driving torque and load torques are also entered in this form. The steady state driving torque and load torques have effects on the initial conditions (acceleration).

Other types of excitations have not been implemented and will be added in the near future.

The torque and horsepower are related by:

$$T_{Lbf-in} = \frac{12 \times 33000 \times hp}{2\pi \times rpm} = \frac{63025 \times hp}{rpm}$$



	Stn	Type	T0	a0	T1	a1	b1	T2	a2	b2
1	1	0	0	0	6.07	38.5	0	-3.04	37.6	0
2	1	0	0	0	5.94	57.3	0	0	0	0
3	1	0	1	0	0	0	0	0	0	0
4	9	0	-1	0	0	0	0	0	0	0
5										
6										
7										
8										
9										
10										
11										
12										

Torsional Excitation data from external data file

This option allows you to input the time dependent torsional excitations from the data files. The data stored in data files are ASCII free format file containing two columns, separated by spaces. One is time in seconds and the other is excitation as shown below:

0 0

0.001 50

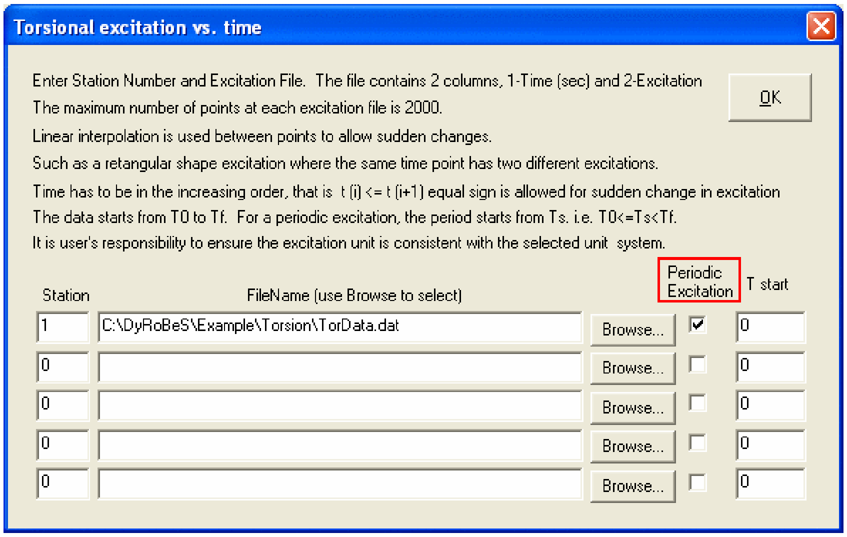
0.002 100

0.005 900

0.010 7000

Be careful to avoid any strange characters or unusual formats in this ASCII data file.

Linear interpolation will be used for sudden changes in excitation, such as step function, triangular shapes, etc… You may specify the starting time to include this excitation in the analysis. If the Periodic Excitation is checked, then the excitation will become periodic and it will repeat when the time is end. The maximum time (time at the last point) is the period. If the Periodic Excitation is NOT checked, the excitation will be a constant when the time exceeds the last point. No extrapolation will be used. Any time larger than the given time in the data file, the last point (torque) will be used when time is greater than the last point.



See other Torsional/Axial Data Torsional/Axial Data.

Torsional Driving Torque

The Torsional Driving Torque and Load Torque are used for torsional startup analysis. It is known that the synchronous motors produce oscillating torques with excitation frequency equal to twice the slip frequency during startup and they can cause serious failure if the system is not properly designed. Induction motors also produce the oscillating torques at start; however, the torque amplitude exponentially decays and the excitation frequencies are equal to the line frequency and twice the line frequency. The data fields are self-explanatory. The synchronous speed (rpm) can be automatically calculated by specifying the number of poles or can be input independently if the driver is not a synchronous motor.

$$N_{syn} = \frac{120 \times LineFrequency\ (\mathrm{Hz})}{No.\ of\ Poles}\ \mathrm{rpm}$$

The **synchronous motor** driving torque at any instant during startup has the form:

$$T = T_{avg} + T_{osc} \sin(\omega_{exc} t)$$

where T_{avg}, T_{osc} are the average and oscillating torques. They are provided by the motor manufacturers as a function of motor speed in percentage with respect to the rated torque. The excitation frequency of the pulsating torque is equal to twice the slip frequency.

$$\omega_{exc} = 2\pi \times 2 \times Line\ Frequency\ \times \left(\frac{Synchronous\ Speed - Motor\ Speed}{Synchronous\ Speed} \right)$$

The torsional resonant (critical) speeds for a synchronous motor startup are given below:

$$N_{cr} = N_{syn} \left(1 - \frac{\omega_i}{4\pi f_L} \right)$$

where

N_{cr} is the critical speed in rpm to be calculated.
N_{syn} is the synchronous speed in rpm.
ω_i is the system natural frequency that is less than 2X line frequency (rad/sec).
f_L is the Line Frequency in Hz (50 or 60 Hz).

The **induction motor** driving torque has the form:

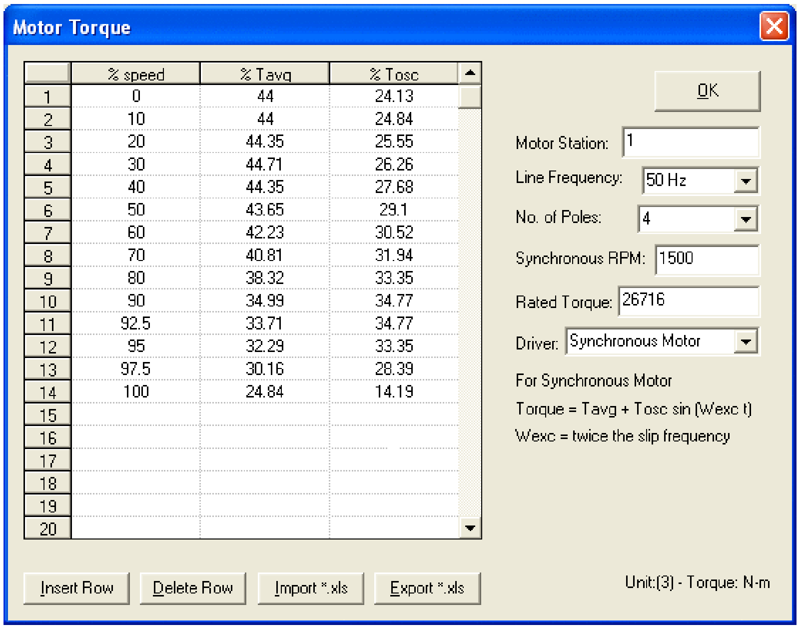
$$T = T_{avg} + T_{osc}$$

where the average torque is given in the same way as the synchronous motor input. The oscillating torque is given in the following expression:

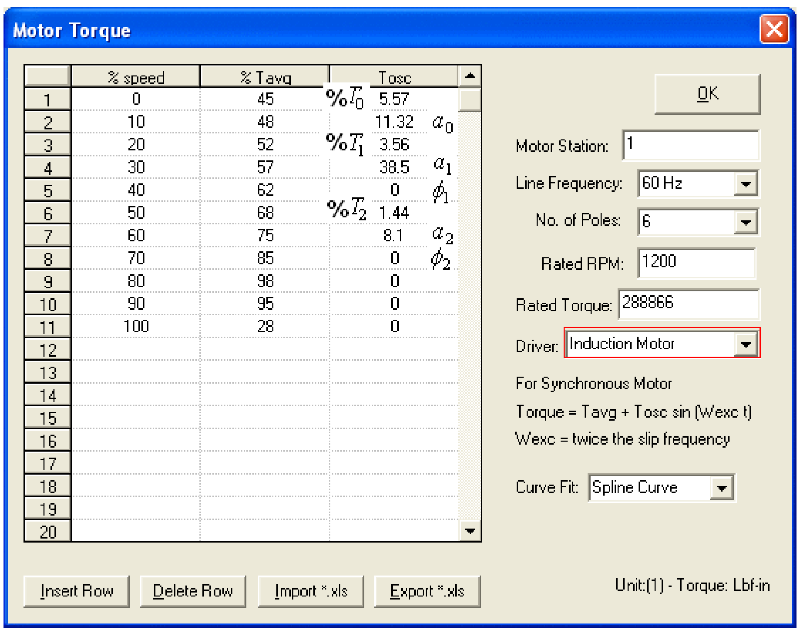
$$T_{osc} = T_{rated} \left[T_0\, e^{-a_0 t} + T_1\, e^{-a_1 t} \sin(\omega t + \phi_1) + T_2\, e^{-a_2 t} \sin(\omega t + \phi_2) + T_3\, e^{-a_3 t} \sin(2\omega t + \phi_3) \right]$$

where the parameters $T_0, a_0, T_1, a_1, \phi_1, T_2, a_2, \phi_2, T_3, a_3, \phi_3$ are entered in the Tosc column in the above sequence. Note that $T_1, a_1, \phi_1, T_2, a_2, \phi_2$ are for the LINE frequency and T_3, a_3, ϕ_3 are for the 2X Line frequency.

For synchronous motor



For induction motor



For GE Power motor

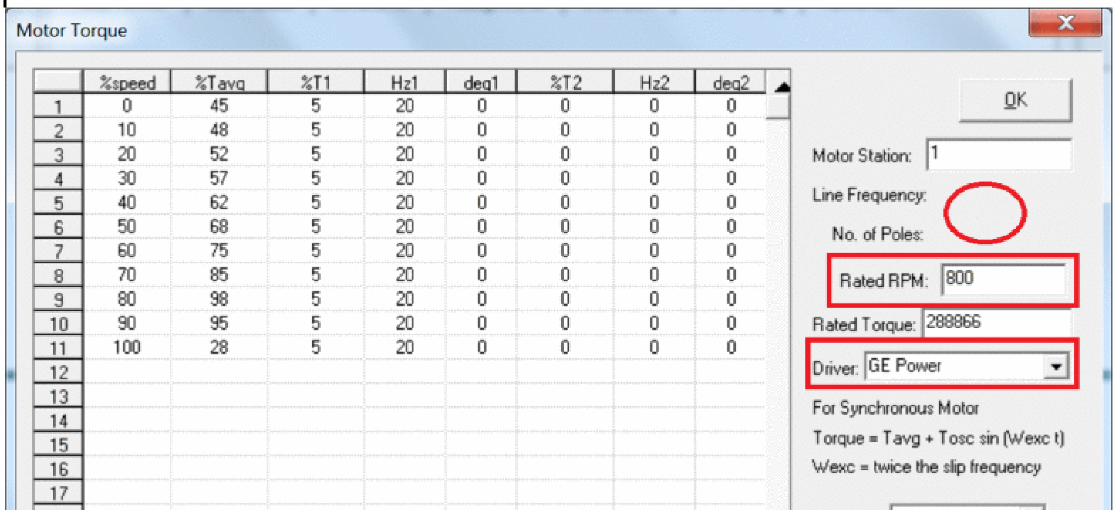
Driver: GE Power

This feature is developed for GE Power Management. The motor rated (synchronous) speed is specified; the line frequency and number of poles are not used. The motor torque during startup, at every speed, is specified by the following equation:

$$T_d = T_{rated}\left[T_{avg} + T_1 \sin\left(\omega_1 t + \phi_1\right) + T_2 \sin\left(\omega_2 t + \phi_2\right)\right]$$

Input: %speed, %Tavg, %T_1, ω_1, ϕ_1, %T_2, ω_2, ϕ_2

%speed – rpm in percentage of the rated speed.
%Tavg, %T_1, %T_2 - Torques in percentage of the rated torque.
ω_1, ω_2 - excitation frequencies in Hz, they can be related, or independent with each other.
ϕ_1, ϕ_2 - phase angle in degree.


Motor Torque

	%speed	%Tavg	%T1	Hz1	deg1	%T2	Hz2	deg2
1	0	45	5	20	0	0	0	0
2	10	48	5	20	0	0	0	0
3	20	52	5	20	0	0	0	0
4	30	57	5	20	0	0	0	0
5	40	62	5	20	0	0	0	0
6	50	68	5	20	0	0	0	0
7	60	75	5	20	0	0	0	0
8	70	85	5	20	0	0	0	0
9	80	98	5	20	0	0	0	0
10	90	95	5	20	0	0	0	0
11	100	28	5	20	0	0	0	0
12								
13								
14								
15								
16								
17								

OK
Motor Station: 1
Line Frequency:
No. of Poles:
Rated RPM: 800
Rated Torque: 288866
Driver: GE Power
For Synchronous Motor
Torque = Tavg + Tosc sin (Wexc t)
Wexc = twice the slip frequency

See other Torsional/Axial Data [Torsional/Axial Data](#).

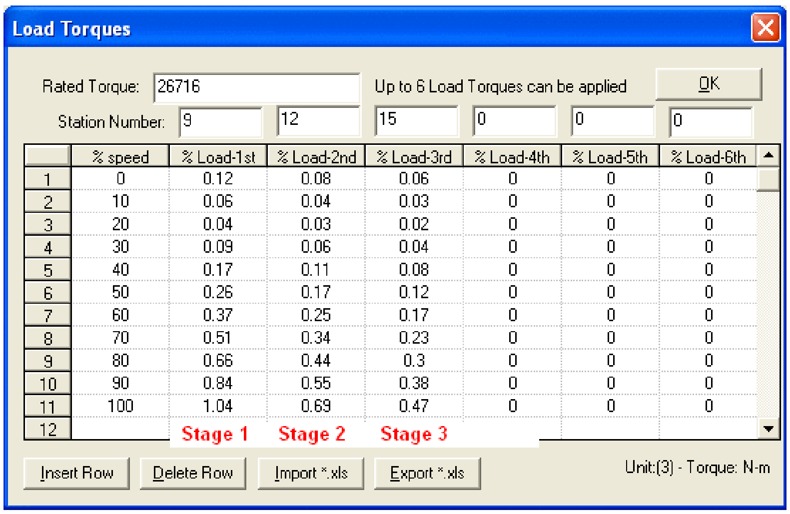
Torsional Load Torque

The load torques include compressor stage power, gear power loss, oil pump, etc. Up to six loads can be input in the analysis. It is common to specify the load torque as a function of speed in percentage. The torques are input in their actual (true) values, not the equivalent torques. The program converts the true torques to the equivalent torques and performs the analysis. The output vibratory torque can be specified either in true or equivalent values.

$$Equivalent\ Torque\ =\ Actual\ Torque \times n$$

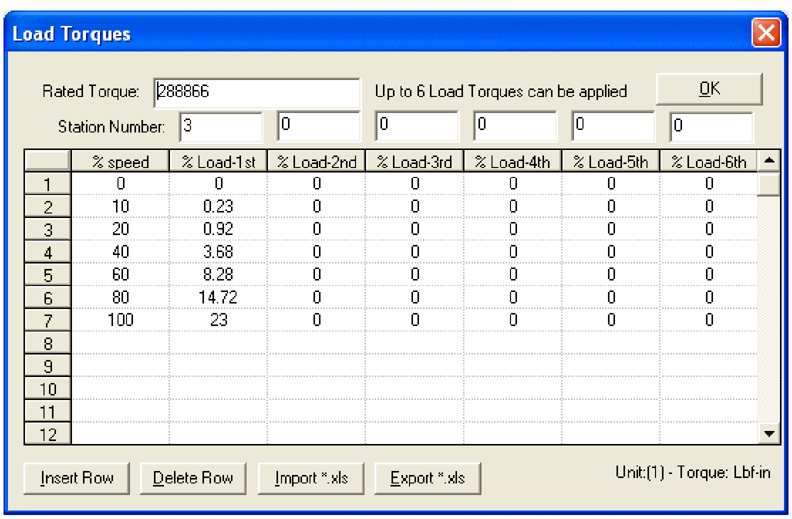
where n is the speed ratio.

Following is an example for 3 stages compressor. All the data are entered in their actual values. Speed ratios must be entered in the Shaft Elements Tab to convert the data into the equivalent values. The speed ratios for the stage 1 to the driving shaft is 8.43, for the stage 2 to the driving shaft is 12.04, and for the stage 3 to the driving shaft is 16.85. The compressor is normally started with inlet valve closed, which is so called **unloaded** condition. At full speed, the total unloaded torque is about 25% of the full load torque.



	% speed	% Load-1st	% Load-2nd	% Load-3rd	% Load-4th	% Load-5th	% Load-6th
1	0	0.12	0.08	0.06	0	0	0
2	10	0.06	0.04	0.03	0	0	0
3	20	0.04	0.03	0.02	0	0	0
4	30	0.09	0.06	0.04	0	0	0
5	40	0.17	0.11	0.08	0	0	0
6	50	0.26	0.17	0.12	0	0	0
7	60	0.37	0.25	0.17	0	0	0
8	70	0.51	0.34	0.23	0	0	0
9	80	0.66	0.44	0.3	0	0	0
10	90	0.84	0.55	0.38	0	0	0
11	100	1.04	0.69	0.47	0	0	0
12		Stage 1	Stage 2	Stage 3			

Following is another example with all the loads lumped at one single station and all data are converted into the equivalent data already. No speed ratio is needed for this equivalent system.

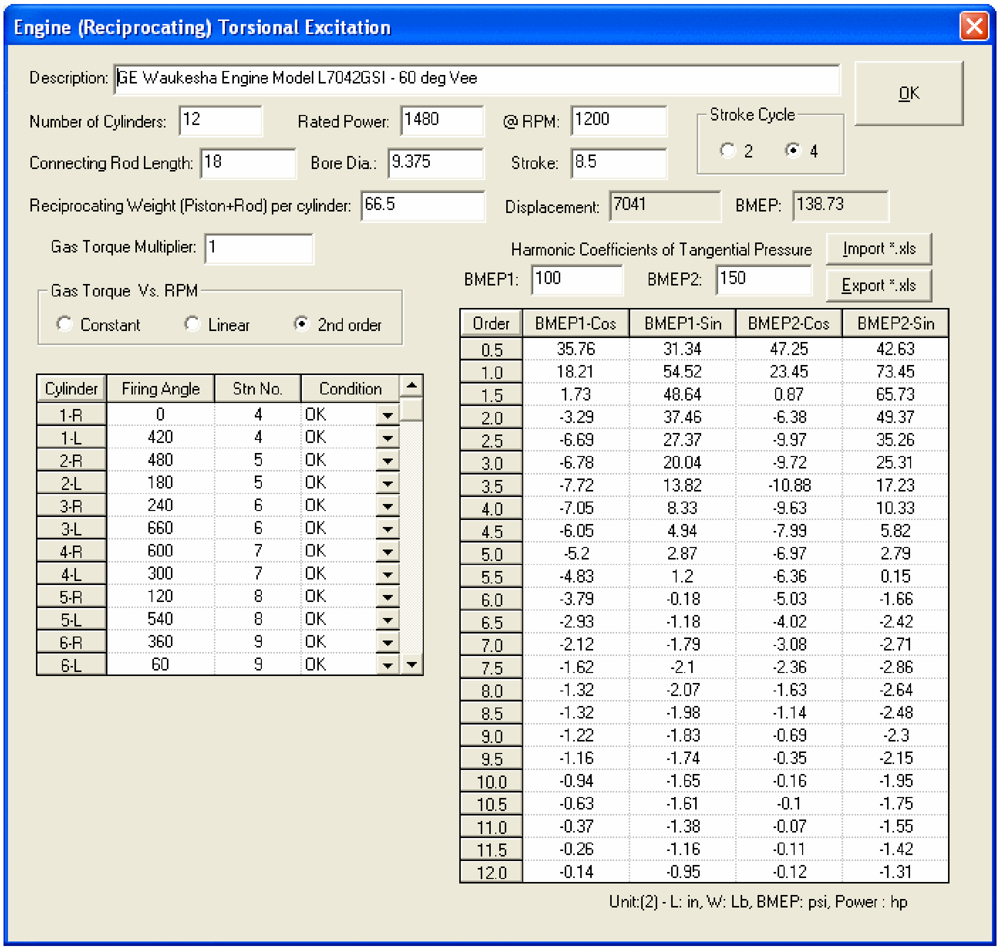


	% speed	% Load-1st	% Load-2nd	% Load-3rd	% Load-4th	% Load-5th	% Load-6th
1	0	0	0	0	0	0	0
2	10	0.23	0	0	0	0	0
3	20	0.92	0	0	0	0	0
4	40	3.68	0	0	0	0	0
5	60	8.28	0	0	0	0	0
6	80	14.72	0	0	0	0	0
7	100	23	0	0	0	0	0
8							
9							
10							
11							
12							

See other Torsional/Axial Data Torsional/Axial Data.

Engine Torsional Excitation

This torsional excitation for a gas or diesel engine is entered in this tab. If a cylinder is misfired, then the gas torque is removed from that cylinder and only the inertia torque is retained. Although the inertia torque in general is very small compared with the gas torque. However, if the cylinder rotor station is negative or zero, which indicates that this particular cylinder is not contributing to any torsional excitation. That is this particular cylinder is removed from the system totally.



Cylinder	Firing Angle	Stn No.	Condition
1-R	0	4	OK
1-L	420	4	OK
2-R	480	5	OK
2-L	180	5	OK
3-R	240	6	OK
3-L	660	6	OK
4-R	600	7	OK
4-L	300	7	OK
5-R	120	8	OK
5-L	540	8	OK
6-R	360	9	OK
6-L	60	9	OK

Order	BMEP1-Cos	BMEP1-Sin	BMEP2-Cos	BMEP2-Sin
0.5	35.76	31.34	47.25	42.63
1.0	18.21	54.52	23.45	73.45
1.5	1.73	48.64	0.87	65.73
2.0	-3.29	37.46	-6.38	49.37
2.5	-6.69	27.37	-9.97	35.26
3.0	-6.78	20.04	-9.72	25.31
3.5	-7.72	13.82	-10.88	17.23
4.0	-7.05	8.33	-9.63	10.33
4.5	-6.05	4.94	-7.99	5.82
5.0	-5.2	2.87	-6.97	2.79
5.5	-4.83	1.2	-6.36	0.15
6.0	-3.79	-0.18	-5.03	-1.66
6.5	-2.93	-1.18	-4.02	-2.42
7.0	-2.12	-1.79	-3.08	-2.71
7.5	-1.62	-2.1	-2.36	-2.86
8.0	-1.32	-2.07	-1.63	-2.64
8.5	-1.32	-1.98	-1.14	-2.48
9.0	-1.22	-1.83	-0.69	-2.3
9.5	-1.16	-1.74	-0.35	-2.15
10.0	-0.94	-1.65	-0.16	-1.95
10.5	-0.63	-1.61	-0.1	-1.75
11.0	-0.37	-1.38	-0.07	-1.55
11.5	-0.26	-1.16	-0.11	-1.42
12.0	-0.14	-0.95	-0.12	-1.31

Engine (Reciprocating) Torsional Excitation

Description: GE Waukesha Engine Model L7042GSI - 60 deg Vee

OK

Number of Cylinders: 12 Rated Power: 1480 @ RPM: 1200

Stroke Cycle: ○ 2 ◉ 4

Connecting Rod Length: 18 Bore Dia.: 9.375 Stroke: 8.5

Reciprocating Weight (Piston+Rod) per cylinder: 66.5 Displacement: 7041 BMEP: 138.73

Gas Torque Multiplier: 1

Gas Torque Vs. RPM: ○ Constant ○ Linear ◉ 2nd order

Cylinder	Firing Angle	Stn No.	Condition
1-R	0	4	OK
1-L	420	4	OK
2-R	480	5	OK
2-L	180	5	OK
3-R	240	6	OK
3-L	660	6	OK
4-R	600	7	OK
4-L	300	7	OK
5-R	120	8	OK
5-L	540	8	OK
6-R	360	9	Misfire
6-L	60	9	OK

Harmonic Coefficients of Tangential Pressure

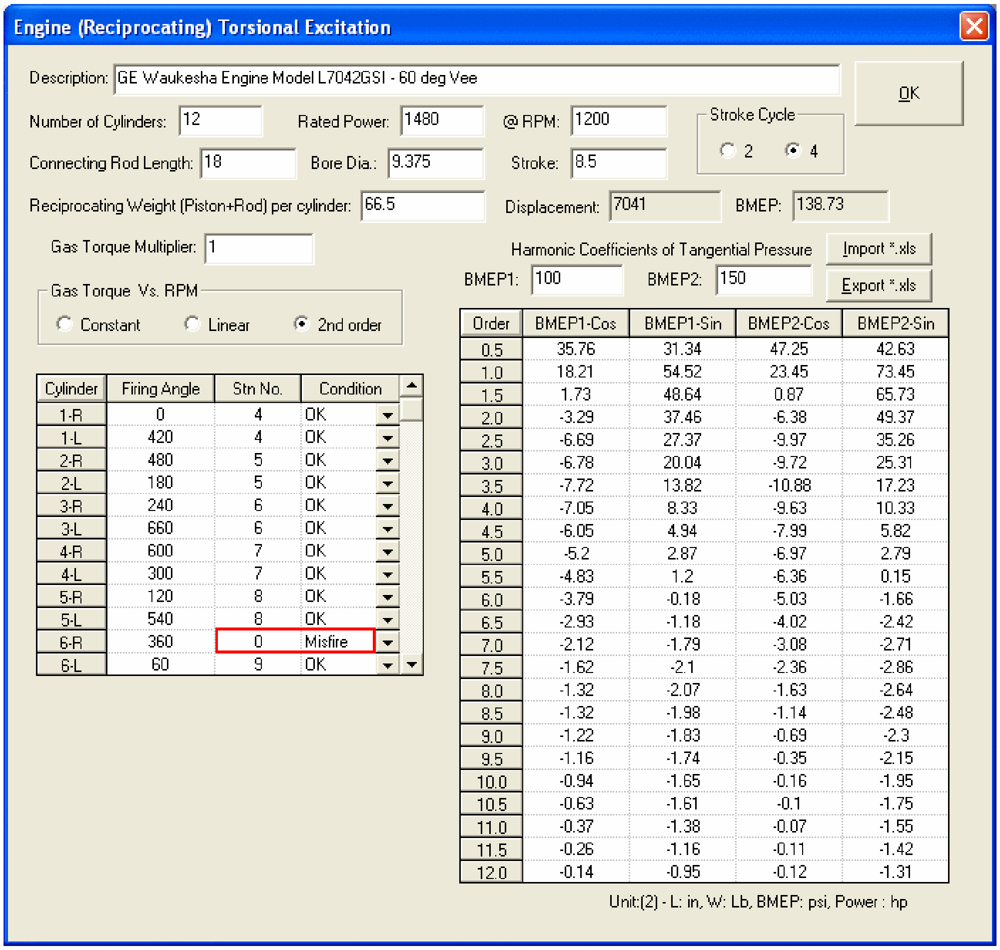
Import *.xls

BMEP1: 100 BMEP2: 150

Export *.xls

Order	BMEP1-Cos	BMEP1-Sin	BMEP2-Cos	BMEP2-Sin
0.5	35.76	31.34	47.25	42.63
1.0	18.21	54.52	23.45	73.45
1.5	1.73	48.64	0.87	65.73
2.0	-3.29	37.46	-6.38	49.37
2.5	-6.69	27.37	-9.97	35.26
3.0	-6.78	20.04	-9.72	25.31
3.5	-7.72	13.82	-10.88	17.23
4.0	-7.05	8.33	-9.63	10.33
4.5	-6.05	4.94	-7.99	5.82
5.0	-5.2	2.87	-6.97	2.79
5.5	-4.83	1.2	-6.36	0.15
6.0	-3.79	-0.18	-5.03	-1.66
6.5	-2.93	-1.18	-4.02	-2.42
7.0	-2.12	-1.79	-3.08	-2.71
7.5	-1.62	-2.1	-2.36	-2.86
8.0	-1.32	-2.07	-1.63	-2.64
8.5	-1.32	-1.98	-1.14	-2.48
9.0	-1.22	-1.83	-0.69	-2.3
9.5	-1.16	-1.74	-0.35	-2.15
10.0	-0.94	-1.65	-0.16	-1.95
10.5	-0.63	-1.61	-0.1	-1.75
11.0	-0.37	-1.38	-0.07	-1.55
11.5	-0.26	-1.16	-0.11	-1.42
12.0	-0.14	-0.95	-0.12	-1.31

Unit:(2) - L: in, W: Lb, BMEP: psi, Power : hp



Engine (Reciprocating) Torsional Excitation

Description: GE Waukesha Engine Model L7042GSI - 60 deg Vee

OK

Number of Cylinders: 12 Rated Power: 1480 @ RPM: 1200

Stroke Cycle: ○ 2 ◉ 4

Connecting Rod Length: 18 Bore Dia.: 9.375 Stroke: 8.5

Reciprocating Weight (Piston+Rod) per cylinder: 66.5 Displacement: 7041 BMEP: 138.73

Gas Torque Multiplier: 1

Harmonic Coefficients of Tangential Pressure

Import *.xls

BMEP1: 100 BMEP2: 150

Export *.xls

Gas Torque Vs. RPM: ○ Constant ○ Linear ◉ 2nd order

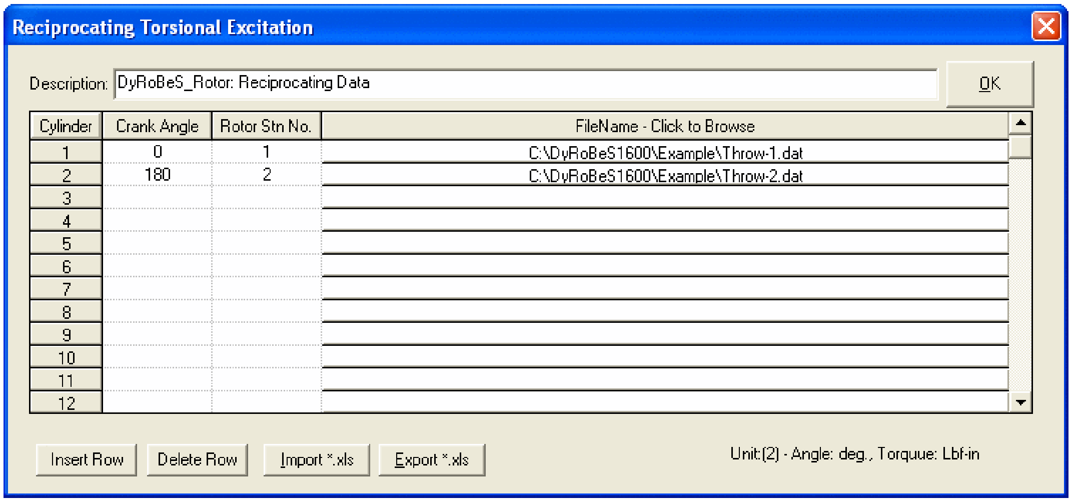
Cylinder	Firing Angle	Stn No.	Condition
1-R	0	4	OK
1-L	420	4	OK
2-R	480	5	OK
2-L	180	5	OK
3-R	240	6	OK
3-L	660	6	OK
4-R	600	7	OK
4-L	300	7	OK
5-R	120	8	OK
5-L	540	8	OK
6-R	360	0	Misfire
6-L	60	9	OK

Order	BMEP1-Cos	BMEP1-Sin	BMEP2-Cos	BMEP2-Sin
0.5	35.76	31.34	47.25	42.63
1.0	18.21	54.52	23.45	73.45
1.5	1.73	48.64	0.87	65.73
2.0	-3.29	37.46	-6.38	49.37
2.5	-6.69	27.37	-9.97	35.26
3.0	-6.78	20.04	-9.72	25.31
3.5	-7.72	13.82	-10.88	17.23
4.0	-7.05	8.33	-9.63	10.33
4.5	-6.05	4.94	-7.99	5.82
5.0	-5.2	2.87	-6.97	2.79
5.5	-4.83	1.2	-6.36	0.15
6.0	-3.79	-0.18	-5.03	-1.66
6.5	-2.93	-1.18	-4.02	-2.42
7.0	-2.12	-1.79	-3.08	-2.71
7.5	-1.62	-2.1	-2.36	-2.86
8.0	-1.32	-2.07	-1.63	-2.64
8.5	-1.32	-1.98	-1.14	-2.48
9.0	-1.22	-1.83	-0.69	-2.3
9.5	-1.16	-1.74	-0.35	-2.15
10.0	-0.94	-1.65	-0.16	-1.95
10.5	-0.63	-1.61	-0.1	-1.75
11.0	-0.37	-1.38	-0.07	-1.55
11.5	-0.26	-1.16	-0.11	-1.42
12.0	-0.14	-0.95	-0.12	-1.31

Unit:(2) - L: in, W: Lb, BMEP: psi, Power : hp

Reciprocating Torsional Excitation

This torsional excitation for a reciprocating machine is entered in this tab.



Note, the data file is in ASCII and free format. The format is:

1. First Line - Description

2. Unit system, 0, 1, 2, 3, or 4. See Unit Systems

3. Second Line - # of speeds (NRPM), # of harmonics (NHAR)

4. Third Line – speed (rpm)

Then followed by the excitation torque

For each harmonics, the torque has the form of Tn = Tcn cos(wnt) + Tsn sin(wnt)

Then repeat the following data for each harmonics

4a. order of harmonics, cos component, sin component

Repeat NHAR times

5. Repeat 4 and 4a for every speed. NRPM times

Example

Ariel Frame JGK/4 Rated rpm 1200, throw 1 data

2 // unit system

3 24 // NRPM, NHAR

1100 // RPM

1 2423 -13467

2 –12867 9207

3 –2973 5670

Other harmonics

24 -1.1 36.3 Total of 24 harmonics

1150

1 2423 -13467

2 –12867 9207

3 –2973 5670

Other harmonics

24 -7.4 31.6

1200

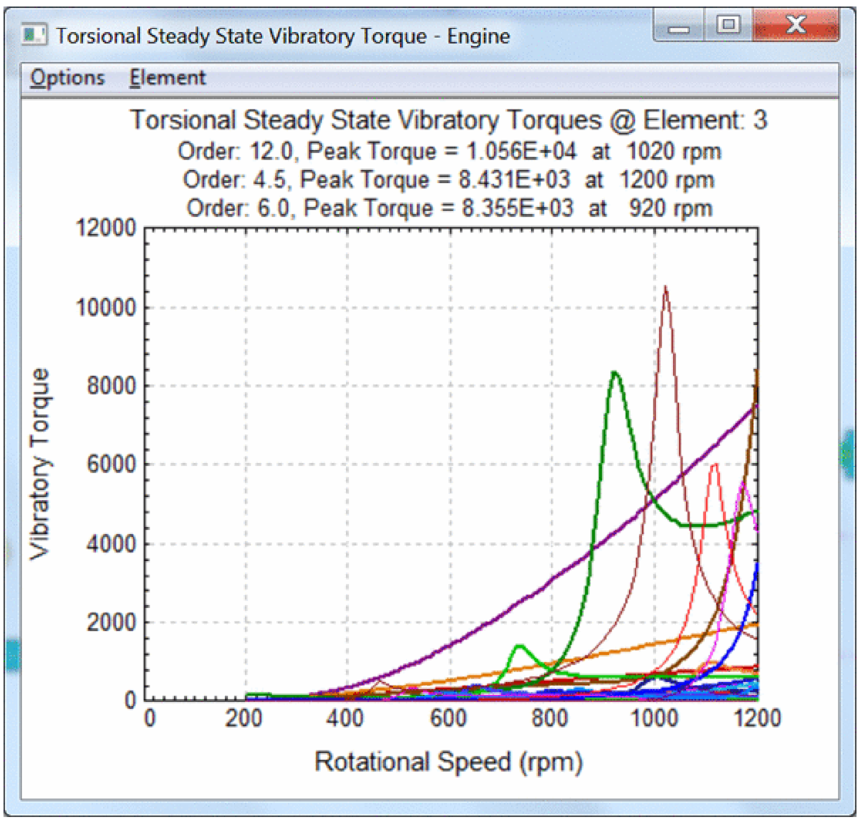
1 2423 -13467

2 –12867 9207

3 –2973 5670

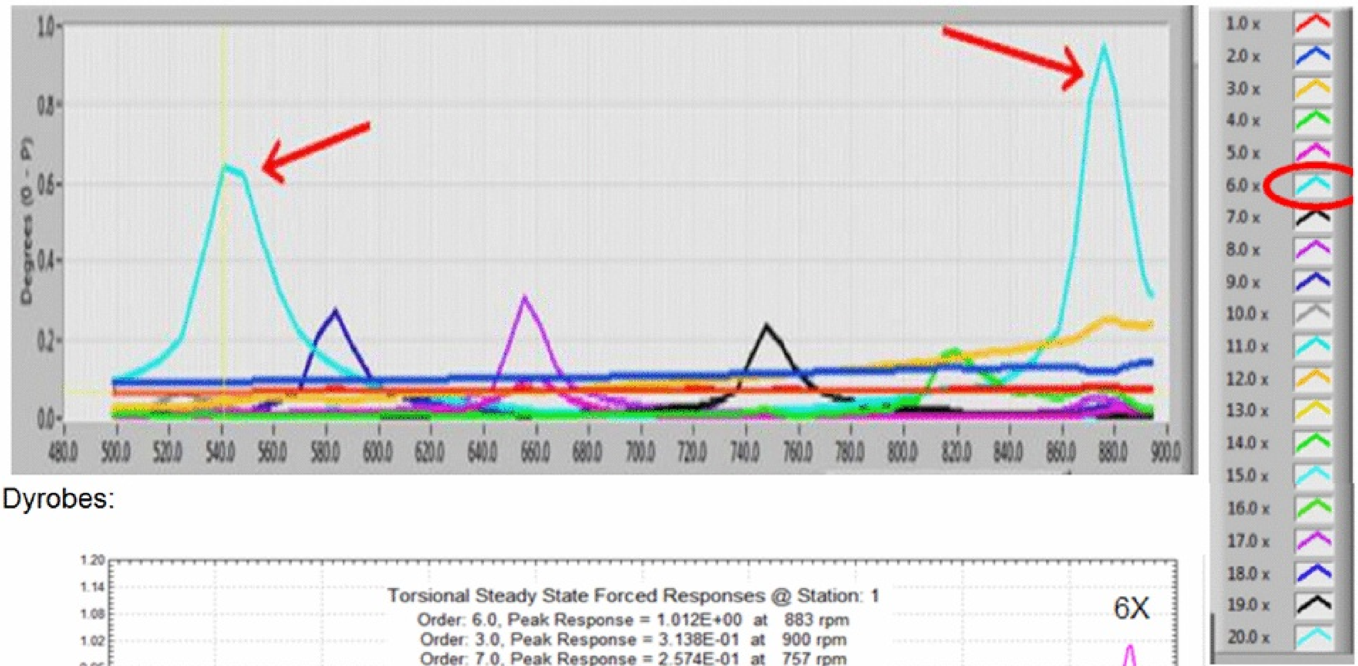
Other harmonics

24 -13.9 25.3

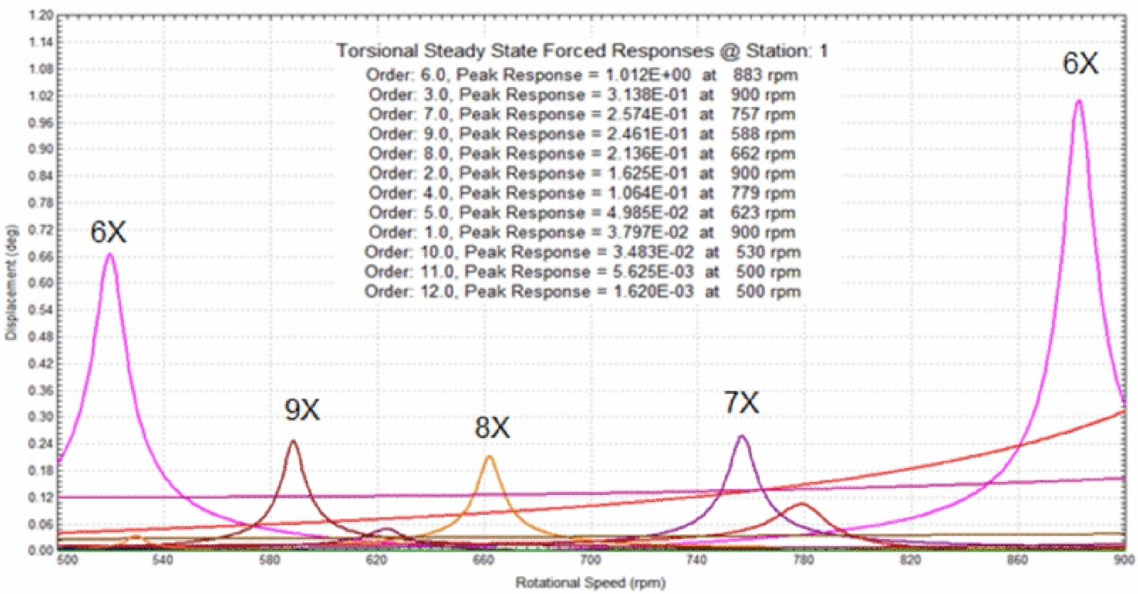


Here are the field and Dyrobes comparison plots.

Field:

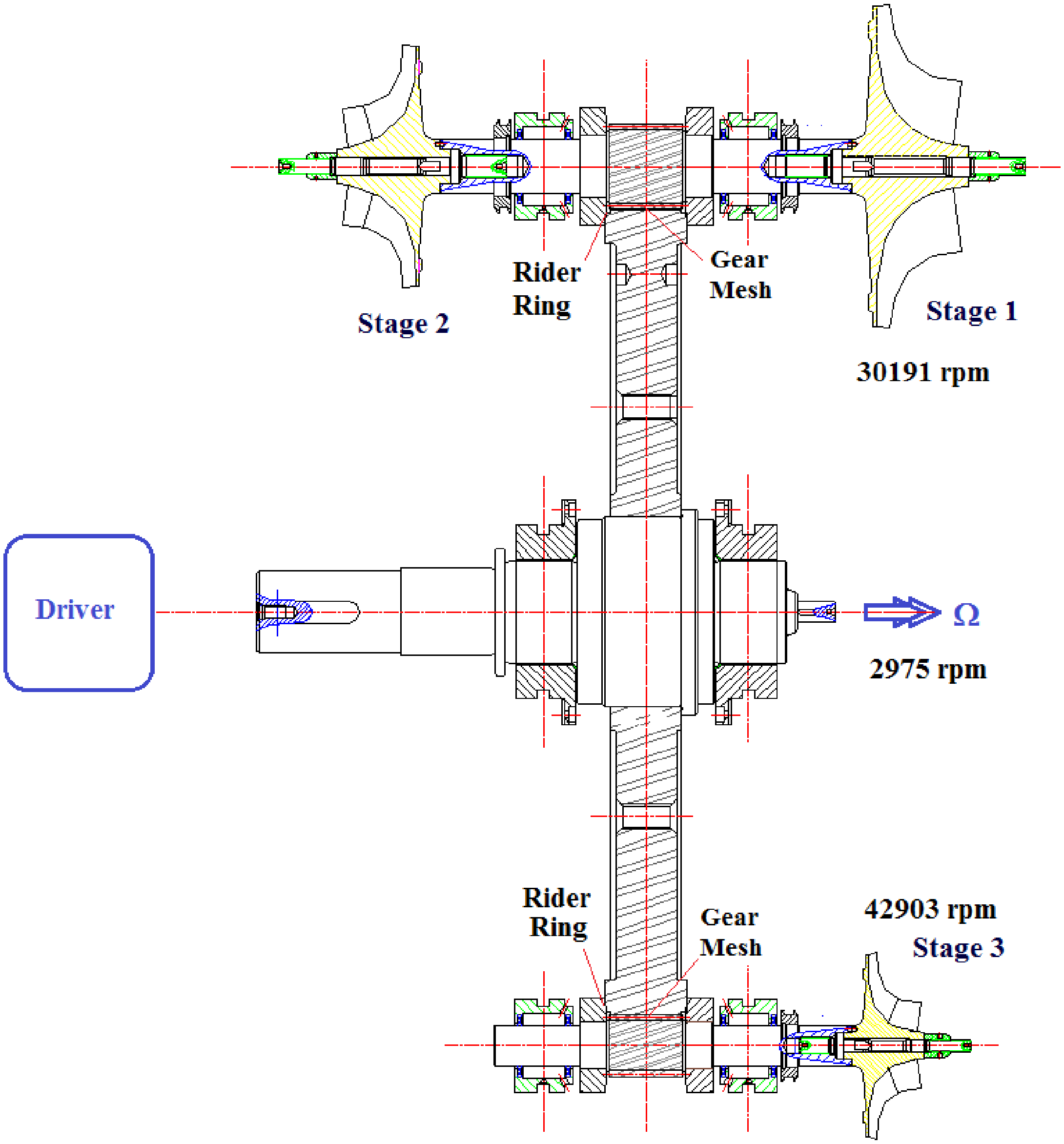


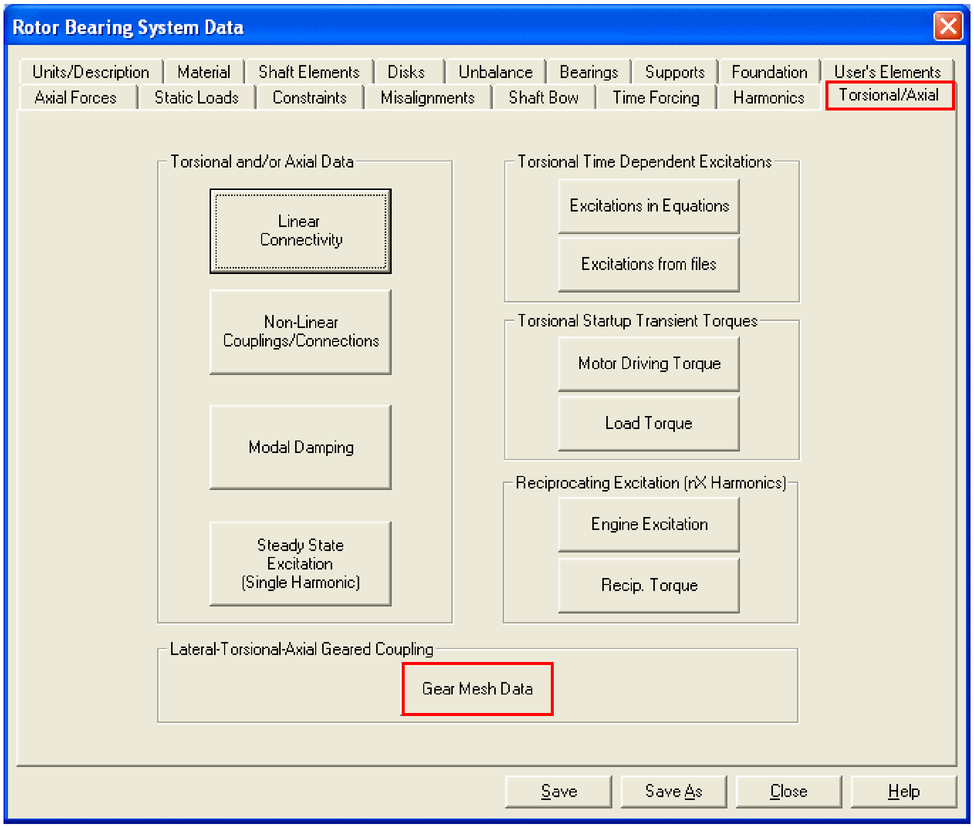
Dyrobes:

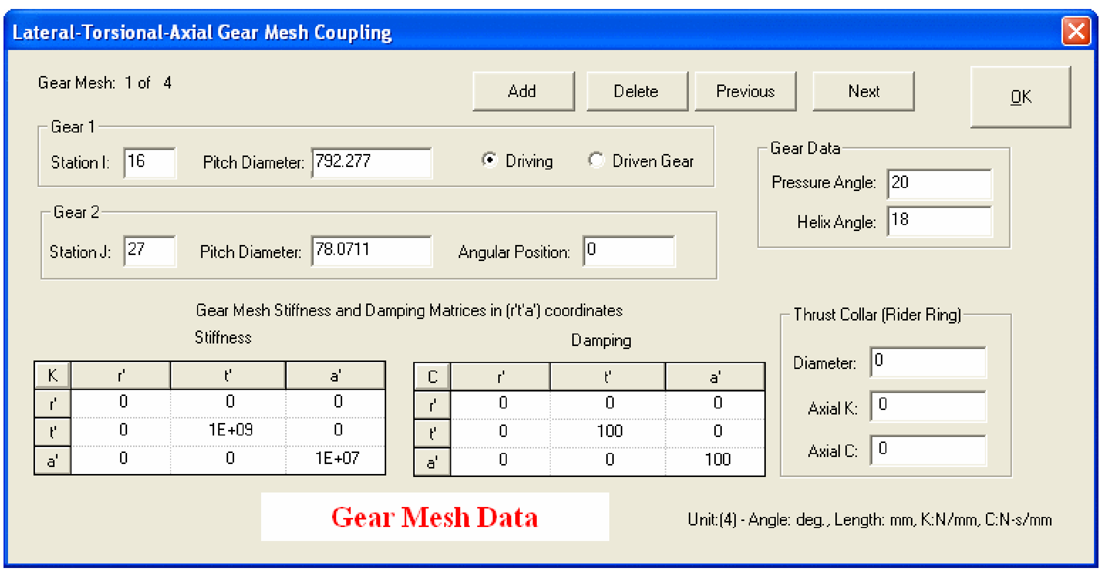


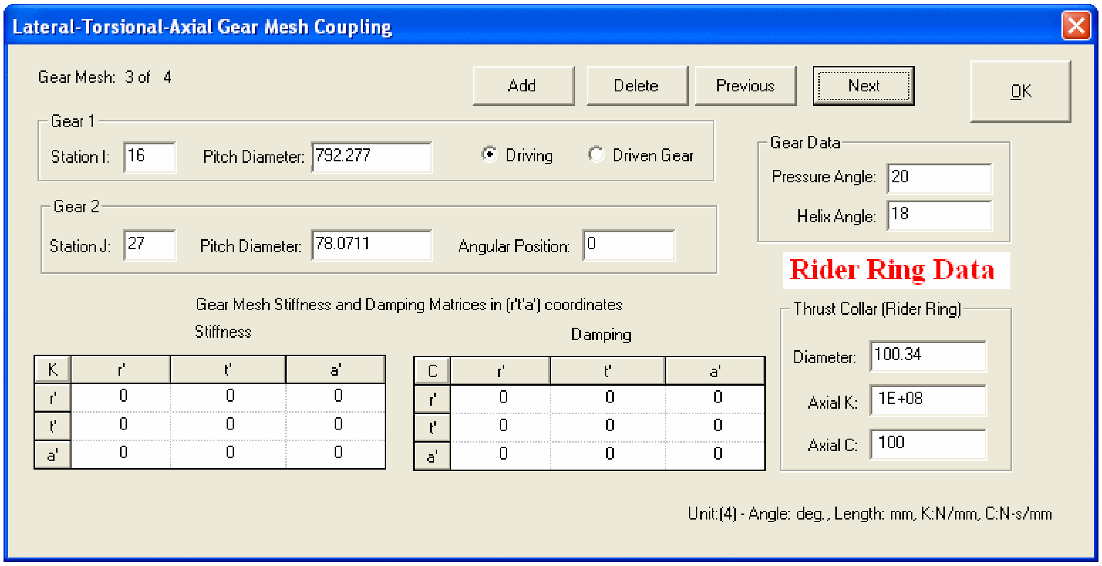
Gear Mesh Data

The gear mesh data is for the coupled Lateral-Torsional-Axial vibration analysis. The motions are coupled through the gear mesh and/or rider ring (thrust collar).




Rotor Bearing System Data
Units/Description
Material
Shaft Elements
Disks
Unbalance
Bearings
Supports
Foundation
User's Elements
Axial Forces
Static Loads
Constraints
Misalignments
Shaft Bow
Time Forcing
Harmonics
Torsional/Axial
Torsional and/or Axial Data
Linear Connectivity
Non-Linear Couplings/Connections
Modal Damping
Steady State Excitation (Single Harmonic)
Torsional Time Dependent Excitations
Excitations in Equations
Excitations from files
Torsional Startup Transient Torques
Motor Driving Torque
Load Torque
Reciprocating Excitation (nX Harmonics)
Engine Excitation
Recip. Torque
Lateral-Torsional-Axial Geared Coupling
Gear Mesh Data
Save
Save As
Close
Help


Lateral-Torsional-Axial Gear Mesh Coupling
Gear Mesh: 1 of 4
Add
Delete
Previous
Next
OK
Gear 1
Station I: 16
Pitch Diameter: 792.277
Driving
Driven Gear
Gear 2
Station J: 27
Pitch Diameter: 78.0711
Angular Position: 0
Gear Data
Pressure Angle: 20
Helix Angle: 18
Gear Mesh Stiffness and Damping Matrices in (r't'a') coordinates
Stiffness
K r' t' a'
r' 0 0 0
t' 0 1E+09 0
a' 0 0 1E+07
Damping
C r' t' a'
r' 0 0 0
t' 0 100 0
a' 0 0 100
Thrust Collar (Rider Ring)
Diameter: 0
Axial K: 0
Axial C: 0
Gear Mesh Data
Unit:(4) - Angle: deg., Length: mm, K:N/mm, C:N-s/mm



For more details, see LTA Analysis

Model Summary

The Model Summary summarizes the system parameters and tabulates the related input data in a very organized ASCII format (text) that allows you to verify your model. This file may be used in a report and reformatted for formal reports. The model summary is useful for details of the rotor weight, inertia, length, and center of gravity location. It is also important to review if the analysis does not run correctly.

See also Analysis, Lateral Vibration Analysis, Torsional Vibration Analysis, Axial Vibration Analysis, PostProcessor, File Extension.

Analysis

Once the rotor-bearing model (.rot file) is built, you now can proceed to the Analysis menu. **DyRoBeS©_Rotor** is capable of performing free and forced vibration analyses for Lateral Vibration, Torsional Vibration, and Axial Vibration.

The lateral vibration of the discretized system is described by two translational (x, y) and two rotational (θ_x, θ_y) coordinates at each finite element station. **Note that Z-axis is the spinning axis.** For torsional vibration, the motion of each finite element station is described by a rotational displacement (θ_z) about the spinning axis. For axial vibration, the motion of each finite element station is described by a translational displacement (z) along the spinning axis.

The analyses for Lateral Vibration Analysis include:

Static Deflection and Bearing/Constraint Reactions

Critical Speed Analysis

Critical Speed Map Analysis

Whirl Speed and Stability Analysis

Steady State Synchronous Response Analysis – Linear System

Steady State Synchronous Response Analysis – NonLinear System

Time Transient Analysis (Time Domain)

Steady State Harmonic Excitation Response Analysis

Steady Maneuver Load Analysis

Time Transient Analysis (Frequency Domain)

Catenary (Gravity Sag) Analysis

The analyses for Torsional Vibration Analysis are:

Damped and Undamped Natural Frequencies and Modes Calculation

Steady State Forced Response Analysis

Transient Analysis (Time Dependent Excitations)

Startup Transient Analysis (Speed Dependent Excitations)

The analyses for Axial Vibration Analysis are:

Damped and Undamped Natural Frequencies and Modes Calculation

Steady State Forced Response Analysis

The analyses for the Coupled Lateral-Torsional-Axial Analysis are:

Whirl Speed and Stability Analysis

Steady State Synchronous Response Analysis

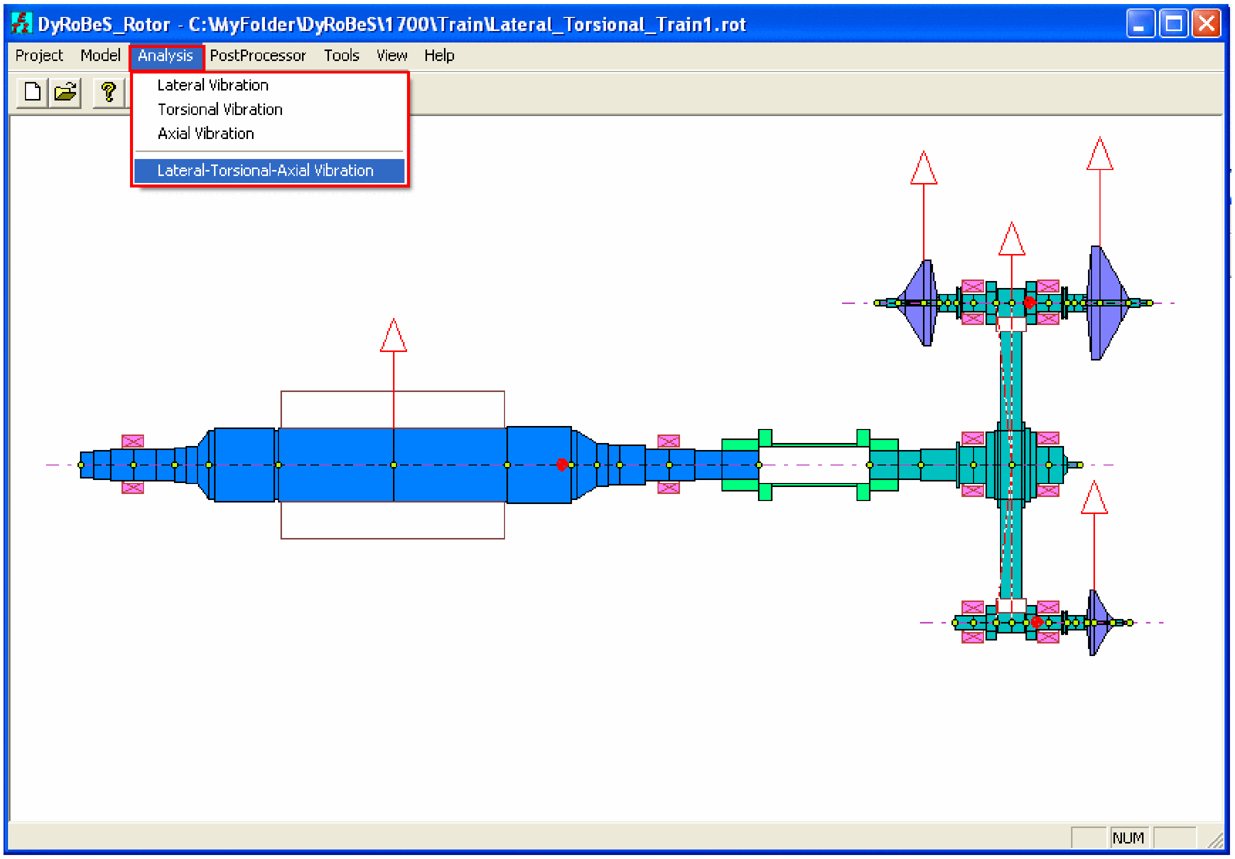
Time Transient Analysis (Time Domain)

Once the analysis is performed, you may go to the corresponding PostProcessor menu to view analysis results.

For more theoretical development, users are encouraged to read rotordynamics books. This help file is only used to guide the software use, not to cover the rotordynamics theory.

Analysis Options and Run Time Data

Under the Analysis menu, there are four options: Lateral Vibration, Torsional Vibration, Axial Vibration, and coupled lateral-torsional-axial vibration, as shown below:



You will need to select the analysis type and enter the run time data to meet your analysis requirement. You may input the run time data only for that selected analysis or you can input all the data for future use. Details on these four different analysis inputs (Run Time Data) are described below.

Lateral Vibration Analysis

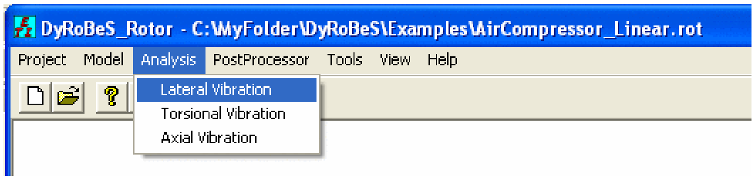
Torsional Vibration Analysis

Axial Vibration Analysis

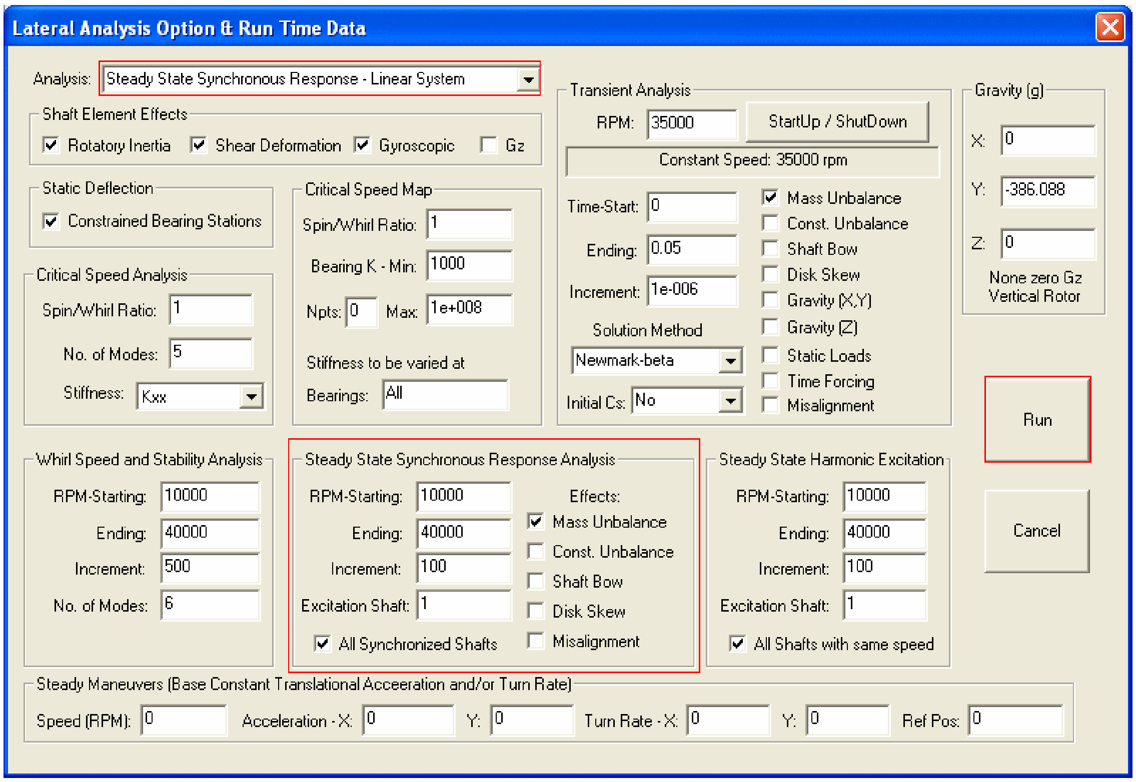
Coupled Lateral-Torsional-Axial Analysis

Lateral Vibration Analysis

To perform the lateral vibration analysis, the rot-bearing data file (.rot) must be built and opened. From the Main Menu, select Analysis and Lateral Vibration as shown below.



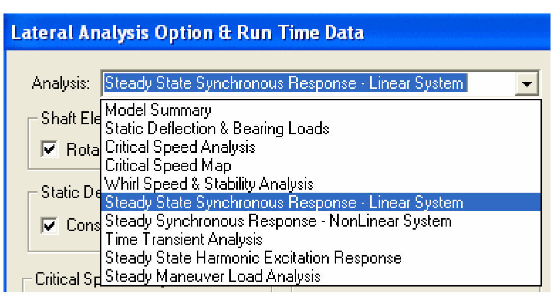
The run time data screen for the Lateral Vibration Analysis is shown below:



The inputs are described:

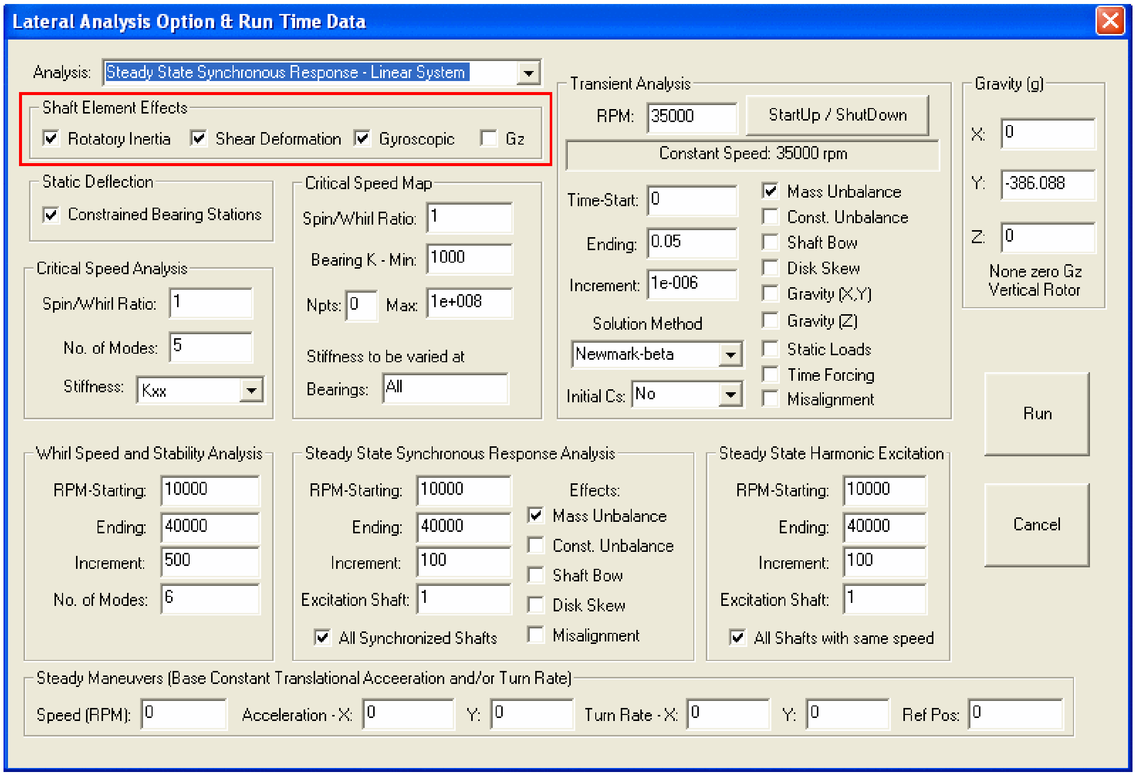
1. Analysis:

Select one analysis at a time as shown below. Once the specific analysis option is selected, the corresponding data are also required.



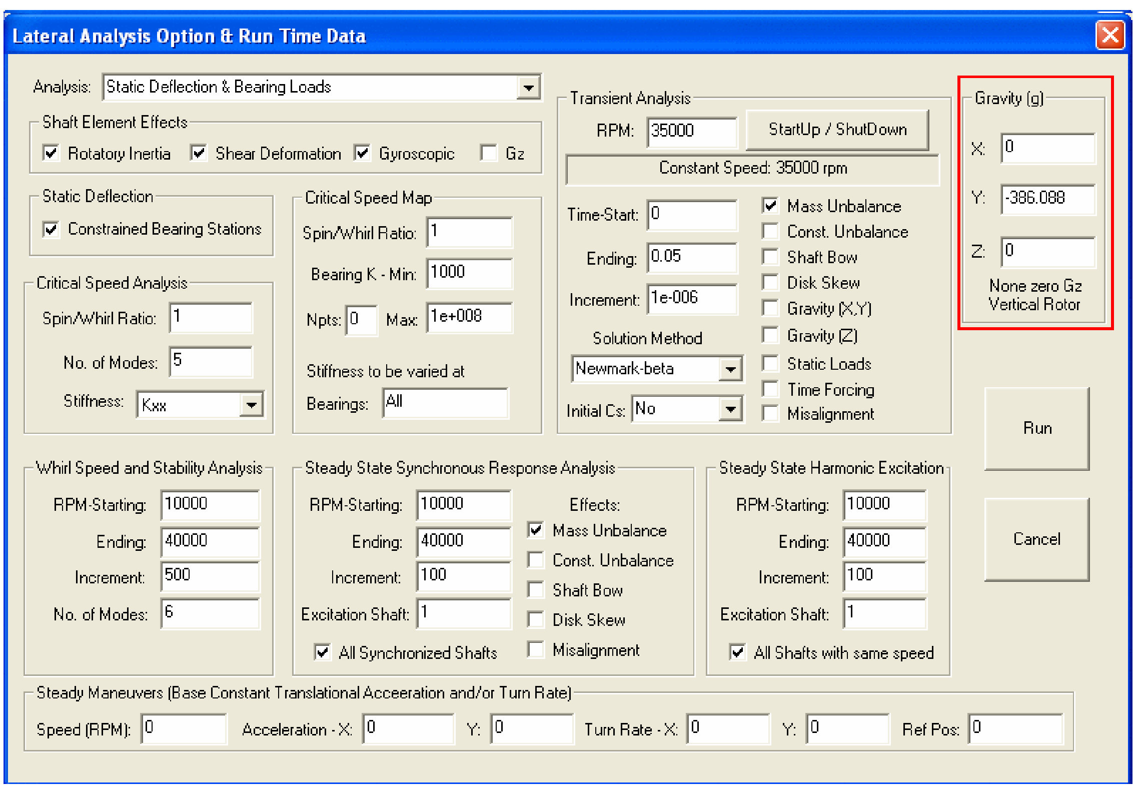
2. Shaft Element Effects

The shaft rotatory inertia, shear deformation, gyroscopic effect, and gravity Z for vertical rotors, can be included or neglected in the calculation by checking the boxes in the run time data. All the effects are included by the default selection.



3. Gravity

Gravity constants in X, Y, and Z directions can be specified. **Note that Z-axis is the spin axis in DyRoBeS.** Since gravity is a vector, the value can be positive or negative. In general, the negative Gy is for the **horizontal rotors**, and non-zero Gz is for the **vertical rotors**. For a horizontal rotor, the gravity X and Y affect the forced rotor response and their effects are entered in the right hand side of the equation of motion as forcing vector. For a vertical rotor, the gravity Z affects the rotor free and forced response. Its effect is entered in the stiffness matrix. Typical value of the gravity constant is 386.088 in/s^2 for Unit Systems 1 and 2, and 9.8066 m/s^2 for Unit System 3, and 9806.6 mm/s^2 for Unit System 4.



See also:

[Static Deflection and Bearing/Constraint Reactions]

Critical Speed Analysis

Critical Speed Map Analysis

Whirl Speed and Stability Analysis

Steady State Synchronous Response Analysis – Linear System

Steady State Synchronous Response Analysis – NonLinear System

Time Transient Analysis

Steady State Harmonic Excitation Response Analysis

Steady Maneuver Load Analysis

Time Transient Analysis (Frequency Domain)

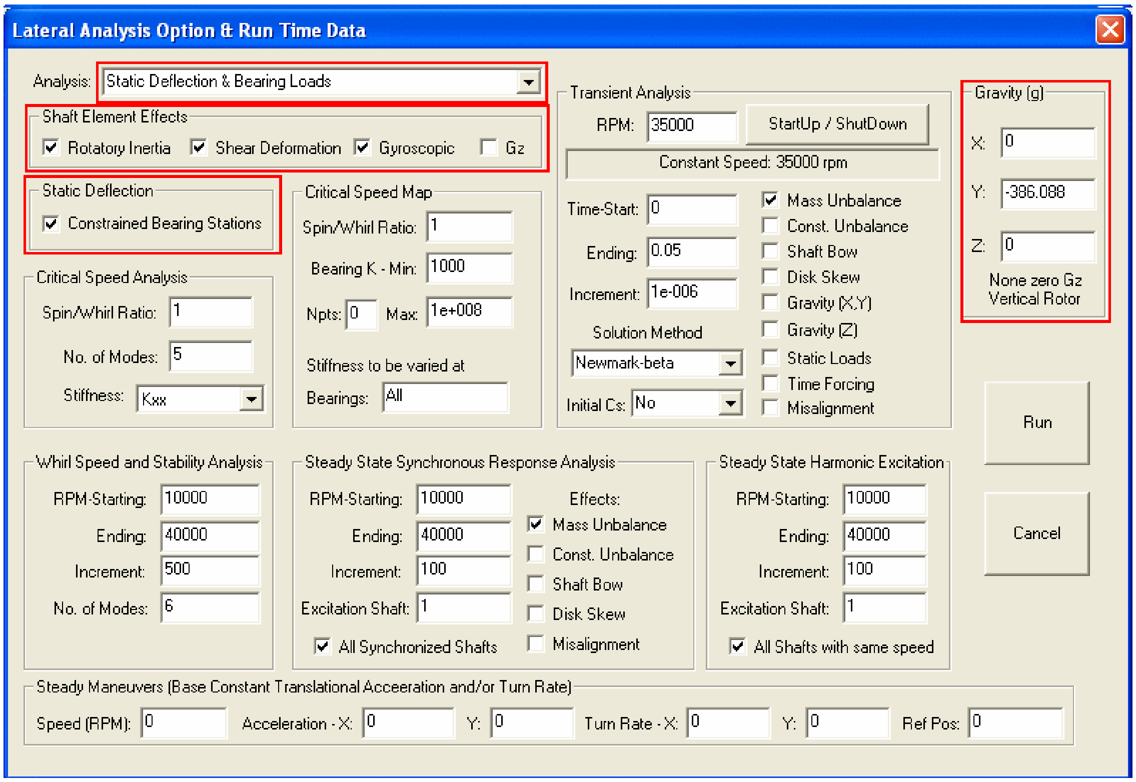
Catenary (Gravity Sag) Analysis

Static Deflection and Bearing/Constraint Reactions

This option calculates the shaft static deflection and bearing/constraint reaction forces for the static determinate and indeterminate problems. The resulting element internal shear forces and bending moments are also derived. The associated stresses are also calculated. The external static loads, gravity, and misalignment are included in this analysis. In the window version of **DyRoBeS©_Rotor**, the finite element stations where linear bearings are located at can be either constrained (zero displacements) or flexible with their bearing stiffnesses being used in the calculation. For multiple speed dependent bearing coefficients, the bearing stiffness at the lowest speed is used. The non-linear bearing and active magnetic bearing stations are still constrained. This window version allows for multiple shaft systems.

Constrained Bearing Stations: If this option is checked, the bearing stations will be constrained (zero displacements) in this static analysis.

The related inputs for the Static Deflection and Bearing Loads Analysis are shown below:



For sample outputs, click Static Deflection PostProcessor.

Critical Speed Analysis

The Critical Speed Analysis calculates the undamped critical speeds, mode shapes and associated kinetic and potential energy distribution, and modal stress. The isotropic system is assumed and the users can select the bearing stiffness direction (Kxx, Kyy, or average) be used in the analysis, in case if the bearing/support is not isotropic. For multiple speed dependent bearing coefficients, the bearing stiffness at the highest speed is used. The flexible support effect is also included in the Critical Speed Analysis.

The undamped critical speeds are determined directly by solving a reduced eigenvalue problem associated with the system equations expressed in a rotating reference frame. These undamped modes are circular relative to the fixed reference frame but are constant relative to the rotating reference frame. Therefore, it is convenient to consider only one of the two planes of motion. This procedure will simplify the calculation and reduce computational time. Due to the simplifying assumptions applied in the undamped critical speed calculation, extreme care must be taken in the preparation and interpretation of these results.

A brief description of the associated run time data for the Critical Speed Analysis is included below:

Spin/Whirl Ratio

This parameter determines the type of critical speeds under calculation. The typical values for the Spin/Whirl Ratio are:

1 = Forward synchronous critical speeds

-1 = Backward synchronous critical speeds

0 = Planar critical speeds for non-rotating systems

2 = Half frequency whirl (Subsynchronous criticals), etc.

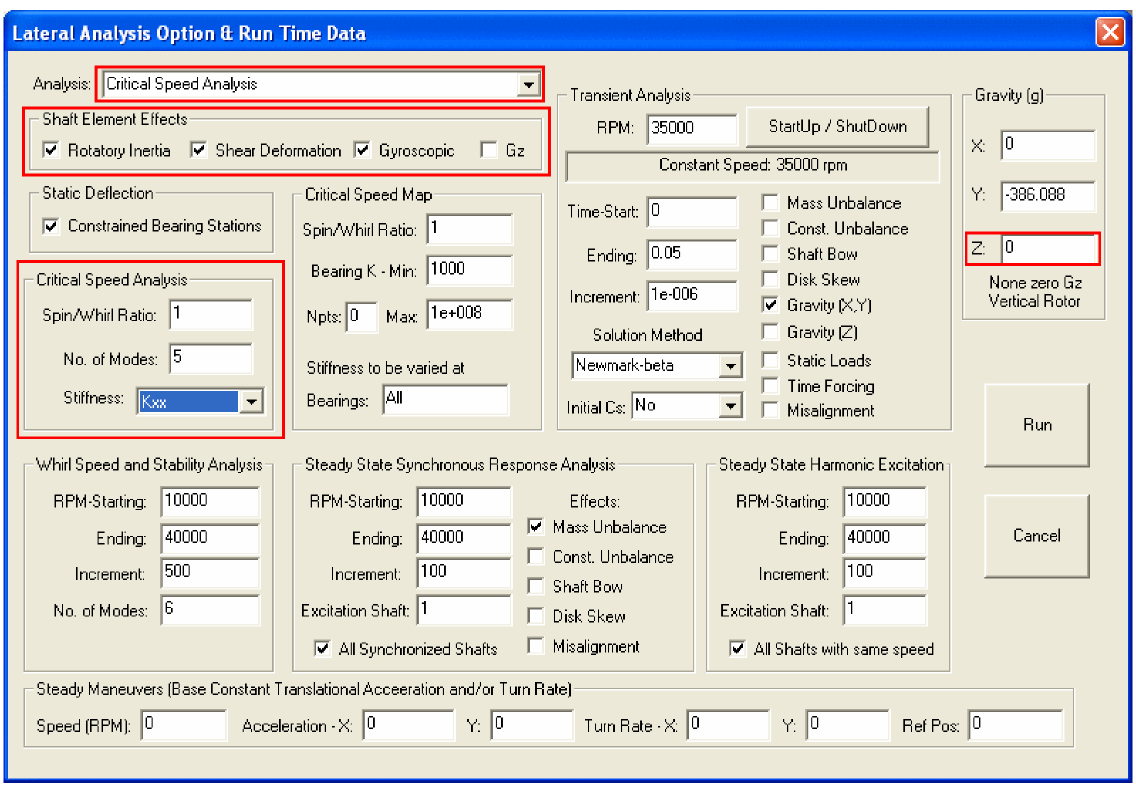
The forward synchronous critical speeds and modes are the most commonly calculated due to unbalance excitation.

Number of Modes

This parameter is used to calculate the selected mode shapes. The value of zero indicates that only the critical speeds are calculated in the analysis and no mode output. In practice, only the lowest 3-5 critical speed modes are of importance. For large systems, more mode shapes may be required in the analysis.

Stiffness

The isotropic system is assumed in the undamped critical speed calculation. However, the bearing stiffness, Kxx and Kyy, in the bearing data may not be the same. This option allows you to select the stiffness to be used in the calculation.



For sample outputs, click Critical Speed Analysis PostProcessor.

Critical Speed Map Analysis

The Critical Speed Map calculates the undamped critical speeds for a given range of bearing stiffnesses. The flexible supports were ignored in the Critical Speed Map Analysis before Ver 16.20. That is, the bearings can not be in series before Ver 16.20. The intention for the Critical Speed Map is to examine the rotor flexibility with the support stiffness. So, the bearing stiffness used here is the total equivalent support stiffness if there are bearings in series. However, one may use this analysis to study the effect of that particular bearing stiffness on the critical speed. So, after Ver 16.20, this restriction has been removed. That is, now, it allows bearings in series. However, if you would like to vary the flexible support stiffness, then create a bearing for the flexible support stiffness in the bearing data, and only enter the flexible support mass in the support data. A check box "**Allow Bearings in Series**" is added for this option. If the box is checked, then it will allow bearings in series (new feature in Ver 16.20), if the box is unchecked, any flexible supports are ignored, just like old versions.

In Version 6.0 and above, it allows you to vary the stiffnesses of several selected bearings and hold the remaining bearing stiffnesses fixed. In Version 5.0 and below; however, all the bearings are varied with the assumption that all the bearings are identical. This enhanced capability allows you to analyze the multiple shaft systems.

The run time data required in this analysis are described below:

Spin/Whirl Ratio

This parameter has been explained in the Critical Speed Analysis.

Bearing K, Min and Max

These two stiffness values (minimum and maximum) define the stiffness range for the analysis.

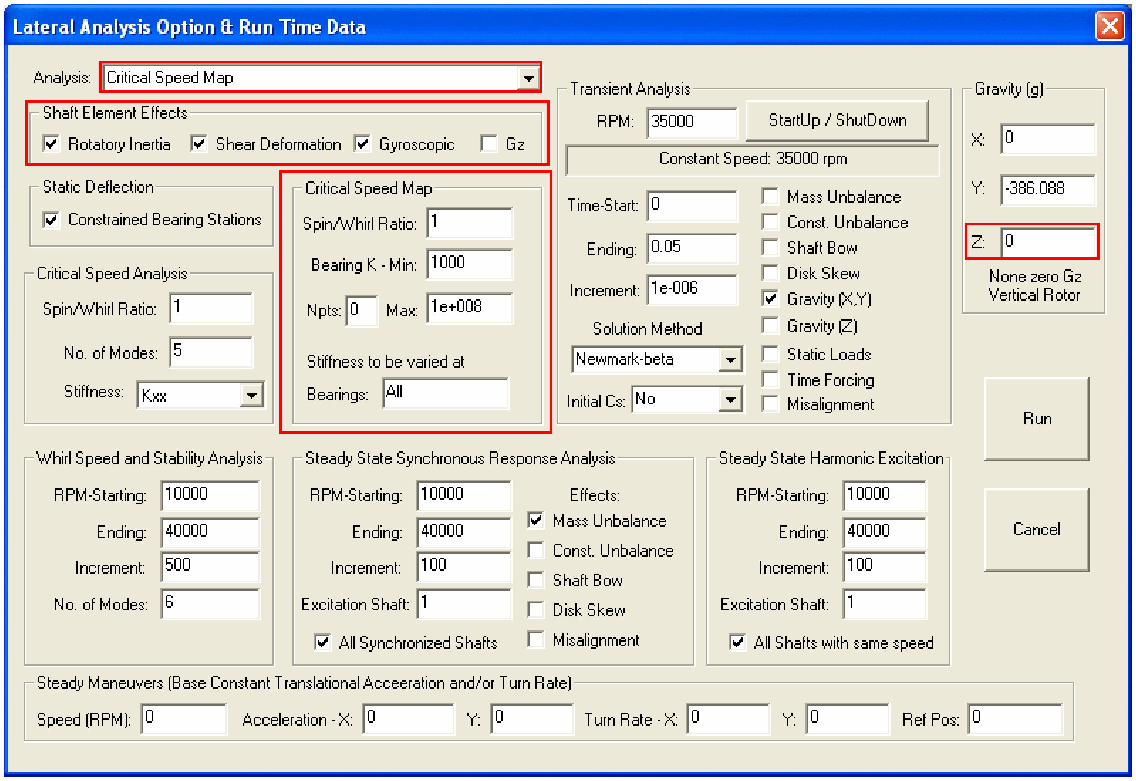
Npts

This number defines the number of stiffness points between the specified stiffness range (***Kmin*** and ***Kmax***) that will be used in the calculation. If Npts = 0, then it will be reset to be:

$$Npts = \log_{10} K_{\max} - \log_{10} K_{\min} + 1$$

Stiffness to be varied at Bearings

If the input is zero (**0**) or **All**, then all the bearings will be assumed to be identical and their stiffnesses will be varied according to the ***Kmin***, ***Kmax*** and **Npts** inputs. **Or, you can select up to 5 bearings** to be varied by using the ***Kmin***, ***Kmax*** and **Npts** inputs and hold the remaining bearing stiffnesses constant by using the regular bearing stiffnesses (Kxx) data provided in the bearing input tab. The bearings to be varied are input as bearing numbers (not bearing stations) and separated by commas. For example, the string **1,3,4** indicates that the bearing numbers 1, 3, and 4 are to be varied. This allows you to skip the bearings, which are not REAL but modeled as bearings, such as pseudo bearings caused by aerodynamic cross-coupling, etc..



For sample outputs, click Critical Speed Map.

Whirl Speed and Stability Analysis

The Whirl Speed and Stability Analysis calculates the damped natural frequencies (whirl speeds), damping coefficients, logarithmic decrements, damping factors, and precessional mode shapes. The QR algorithm is utilized in the calculation of eigenvalues and eigenvectors. This algorithm has been well proven to be reliable and numerically stable. The imaginary parts of the eigenvalues are the system damped natural frequencies and they can be used to determine the damped critical speeds. The real parts of the eigenvalues are the system damping coefficients which can be used to determine the system stability. A positive damping coefficient indicates system instability. Very often, the logarithmic decrements or damping factors are used to determine the system stability. A negative logarithmic decrement or damping factor indicates system instability. When the value of a logarithmic decrement exceeds 1, that particular mode is considered to be well damped. Due to the non-symmetric properties of the bearing coefficients and the gyroscopic effect, the Whirl Speed Map and Stability Map are very complex in nature, caution must be taken when preparing these maps.

The run time data required in this analysis are shaft rotational speeds (**starting**, **ending**, and **incremental speeds**) and **number of modes**. For multiple shaft systems, the shaft speed is referred to the shaft number 1 speed. For single speed calculation, enter the speed as the starting speed and set the ending speed and incremental speed to zero. The **number of modes** is used to specify the number of precessional modes that mode shapes (eigenvectors) are calculated for. The value of zero indicates that only the eigenvalues are calculated and no mode shapes (eigenvectors) are calculated. Generally, only the lowest 4 to 6 precessional modes are of importance in practice. More modes are required for large systems.

A complex eigenvalue is given by:

$$\lambda_i = \sigma_i + j\,\omega_{di}$$

where the subscript *I* is the mode number. If the damped natural frequency is a non-zero value, this mode is a precessional mode with an oscillating frequency equals to the damped natural frequency. If the damped natural frequency equals to zero, this mode is a real mode (pure rigid body mode) or non-oscillating mode. The logarithmic decrement and damping factor of a precessional mode are defined to be:

Logarithmic Decrement:

$$\delta = \frac{-2\pi\,\sigma}{\omega_d} = \frac{2\pi\,\xi}{\sqrt{1-\xi^2}}$$

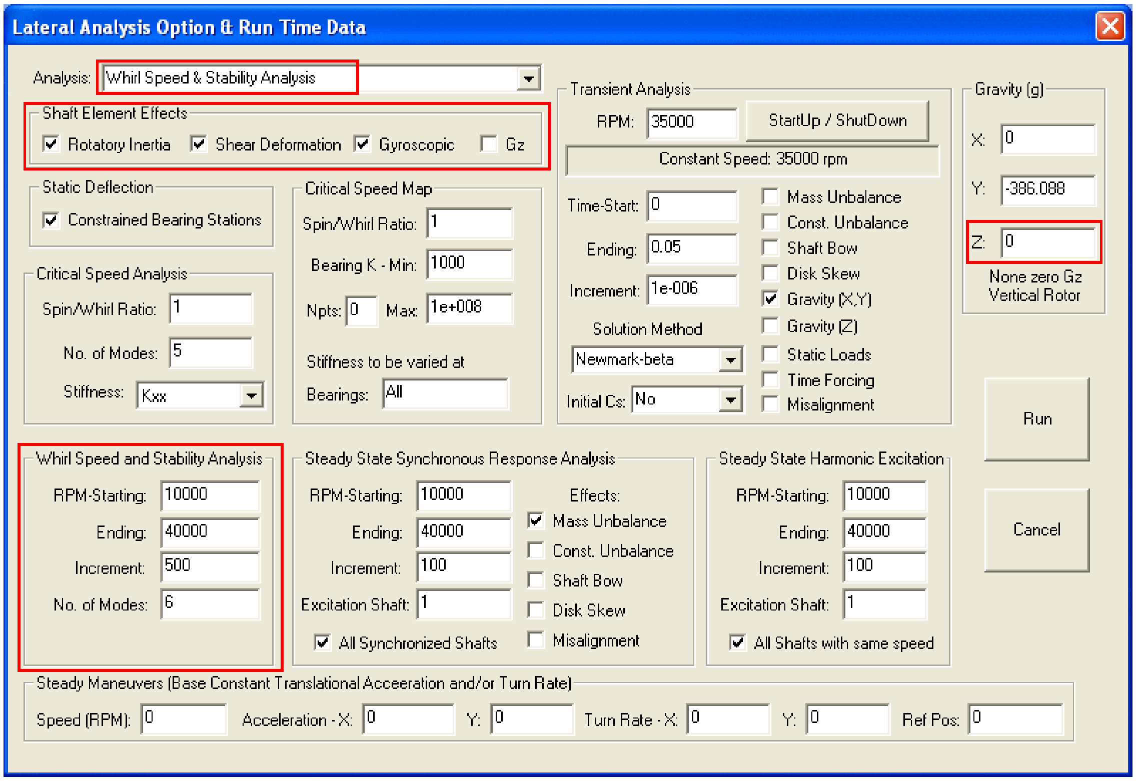
Damping Factor:

$$\xi = \frac{\delta}{\sqrt{(2\pi)^2 + \delta^2}}$$

where

ξ	damping factor, or damping ratio
$\delta\ >0$	Stable or damped system
$\delta\ =0$	threshold of instability
$\delta\ <0$	unstable system

The logarithmic decrement is commonly used in the lateral vibration and damping factor is commonly referred in the torsional analysis.



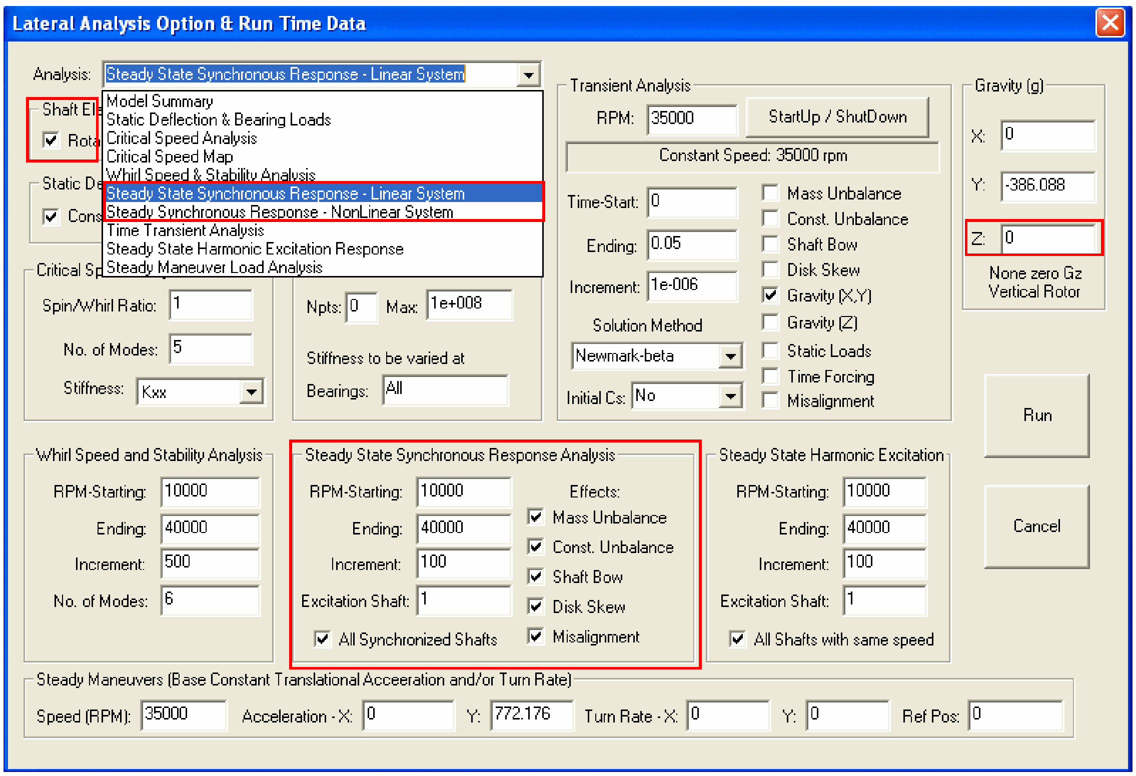
For sample outputs, click Whirl Speed and Stability Maps.

Steady State Synchronous Response Analysis

The Steady State Synchronous Response Analysis calculates the steady state synchronous response due to unbalance force (mass or/and magnet), shaft bow, disk skew, and misalignment for systems with **Linear Bearings** or with **Non-Linear Isotropic Bearings** (Bearing Type 4 - Squeeze Film Dampers, and Bearing Type 6 – general Non-Linear Isotropic Bearing). For linear systems, the response orbits are elliptical in general. However, for non-linear isotropic systems, centered circular orbits are assumed in the analysis. This analysis is commonly used to determine the rotor speeds that produce the peak response (separation margin) and the corresponding levels of vibration (amplification factor).

The shaft rotational speeds (**starting**, **ending**, and **incremental speeds**) are required in the run time data. For multiple shaft systems, the shaft speed is referred to the shaft number 1 speed. For single speed calculation, enter the speed as the starting speed and set the ending speed and incremental speed to zero. The **Excitation Shaft** indicates the shaft number to be considered with excitation for multi-shaft systems. The effects due to unbalance (mass or magnet), shaft bow, disk skew, and misalignment, can be included (box checked) or neglected (box unchecked). Note that if shaft bow and fitting method are given, then the disk skew due to shaft bow effect is added to the total disk skew. The check box for the **All Synchronized Shafts** is used if the excitations from all other shafts, which have the same rotational speed with the **Excitation Shaft** are also included in the analysis.

The output can be presented in many forms, such as, Bode plot, polar plot, elliptical orbit axes plot, bearing/support transmitted force, response at multiple stations, element shear forces and moments, shaft response and displacement orbit at specified speed. The relative displacement, velocity and acceleration are also available.



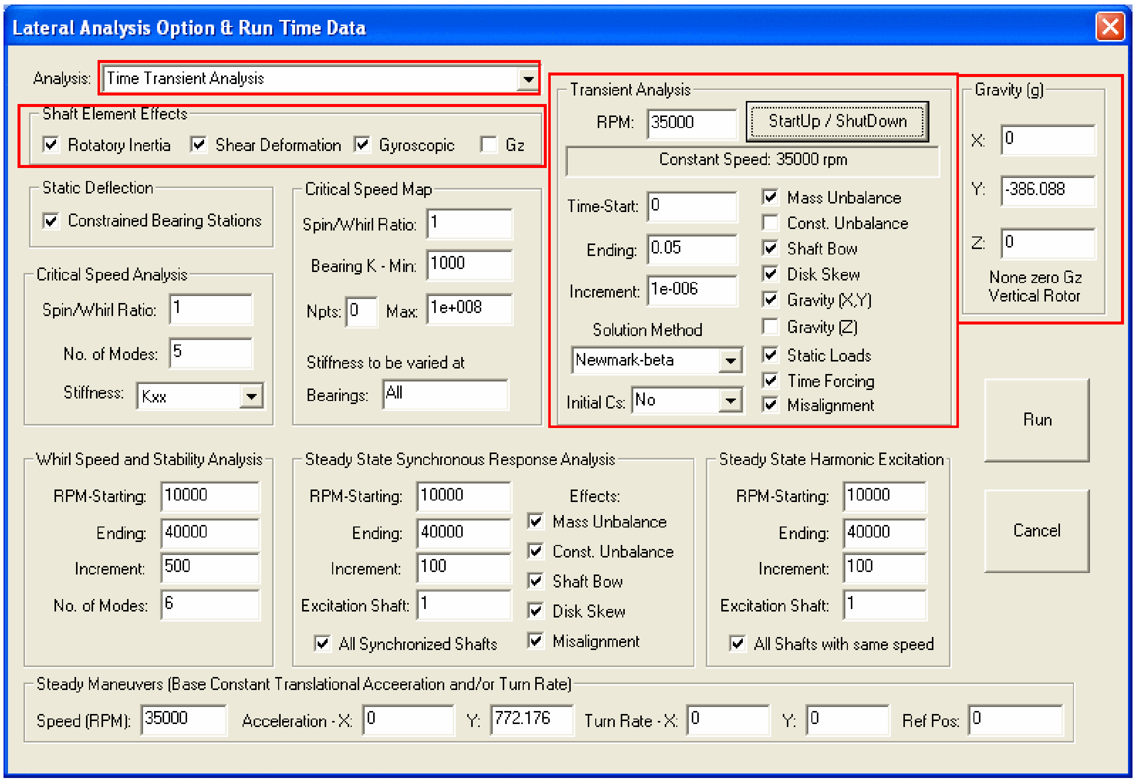
For sample outputs, click Steady State Response Plots.

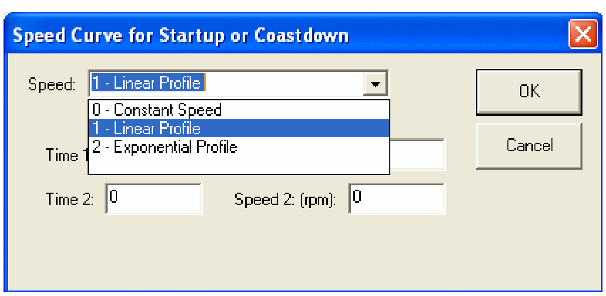
Time Transient Analysis (Time Domain)

The Time Transient Analysis calculates the transient response for a given constant rotor speed or variable speeds and a specified time interval. The system can be linear or non-linear. The effects of unbalance force (mass and magnet), shaft bow, disk skew, gravity, external static loads, time forcing functions, and misalignment can be turn on or off in the run time data by checking the box. A time interval must be specified for the numerical integration and they are: **Start**, **ending**, and **incremental time**. If you are not sure what time step size (time increment) should be used in the integration, then enter zero in the time increment data field. A default time step size will be provided by the program and then used in the analysis. If you enter the time step size is larger than the recommended value, a message box will prompt you and ask you if you want to change the value. One must exhibit some care with time transient analysis so that the computational times are not excessive. However, for the highly non-linear case, a small time interval is necessary for the solution convergence.

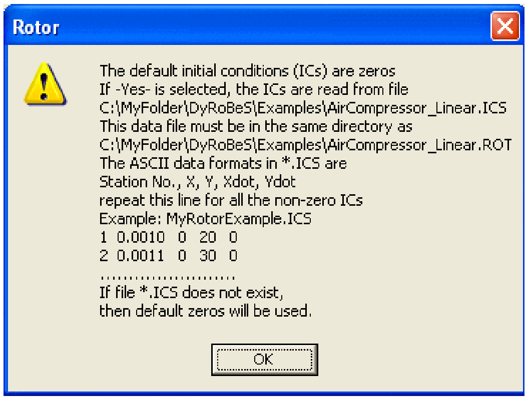
Five **solution methods** are provided: Gear's Method, Runge-Kutta, Newmark-beta, Wilson-theta, and Newmark-Modified methods. In general, Wilson-theta provides good convergence. The **initial conditions** can be specified for the numerical integration, or leave as zeros, or use the previous last point as the initial condition for this run.

For a constant speed, enter **RPM** as the rotor speed. For startup and shutdown analysis, click the **Startup/Shutdown** button for more input. Two speed profiles are available: Linear and Exponential. For more information on the speed curves, click the Speed Curvers here.

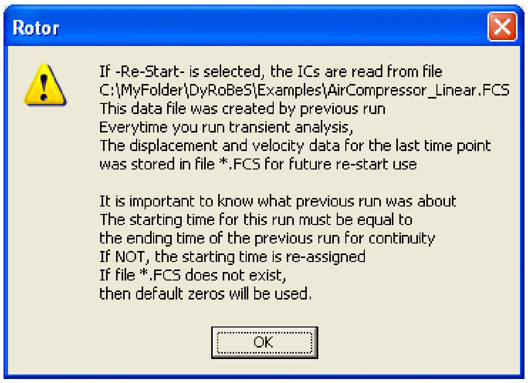




Initial Conditions-YES, a data file containing the initial conditions (displacement and velocity) must be provided.



Initial Conditions-Re-Start



For sample outputs, click Transient Response Plots.

Steady State Harmonic Response Analysis

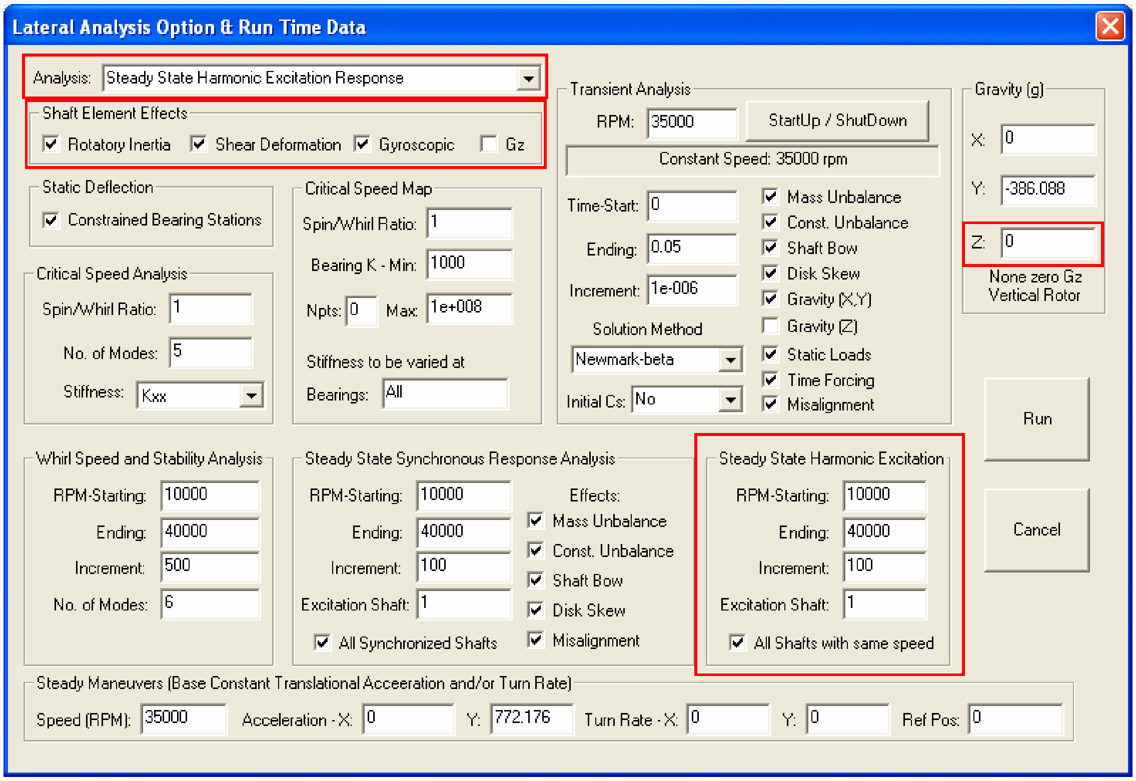
For steady state harmonic excitation analysis, the excitation is expressed in a general form:

$$Q = |Q| \cdot \cos(\omega t + \alpha)$$

where $|Q|$ is the excitation amplitude and the omega is the excitation frequency. If the excitation frequency coincides with the rotor speed, it is called synchronous excitation. However, the harmonic excitation frequency does not have to be the same as the rotor speed. It can be a multiple or fraction of the rotor speed, or completely independent from the rotor speed. The response is also a harmonic motion with the form:

$$q = |q| \cdot \cos(\omega t + \phi)$$

For the harmonic excitation input, click Harmonic Excitations.



For sample outputs, click Steady Harmonic Response Plots.

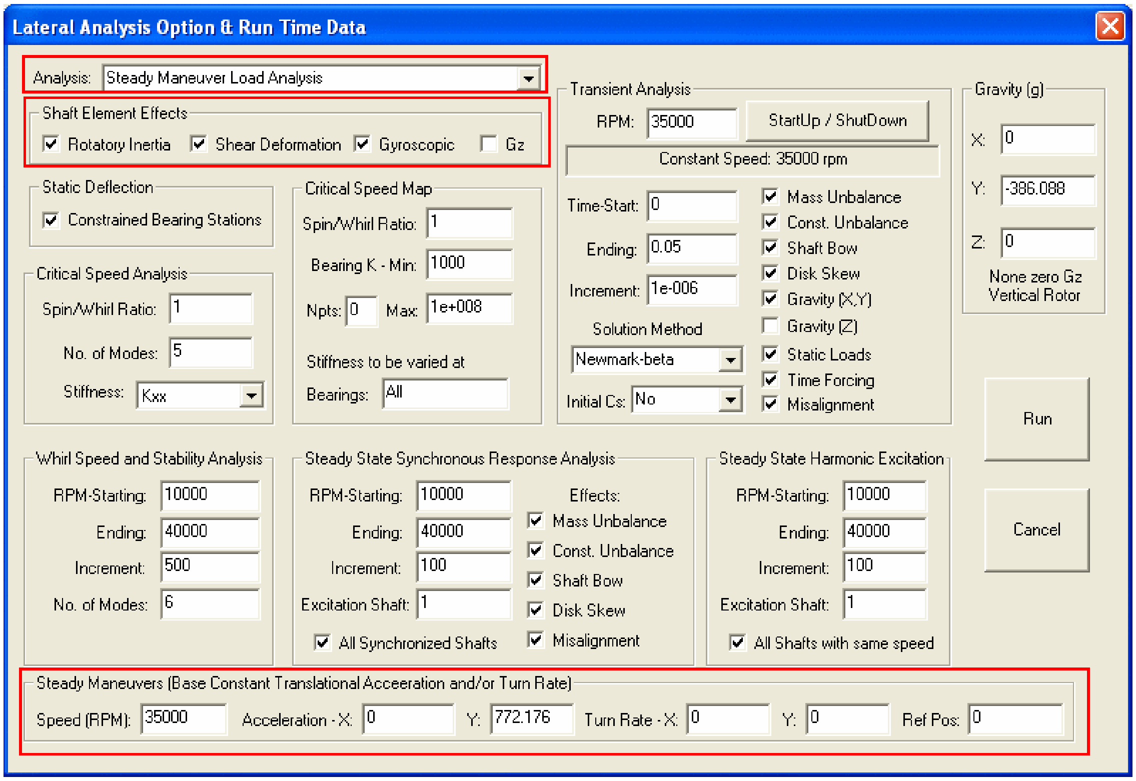
Steady Maneuver Load Analysis

This option calculates the steady maneuver response and bearing loads due to constant base translational acceleration and turn rate. The rotating assembly is mounted in a rigid base through bearings. The base motion is described by constant accelerations, where the magnitude is commonly specified by a multiple of gravity constant, or a constant turn rate. Following example is a 2G acceleration applied for the static maneuver analysis.

Units:

Acceleration: L/T^2 (for unit systems 1 and 2: inches/sec^2, for unit system 3: m/sec^2, for unit system 4: mm/sec^2).

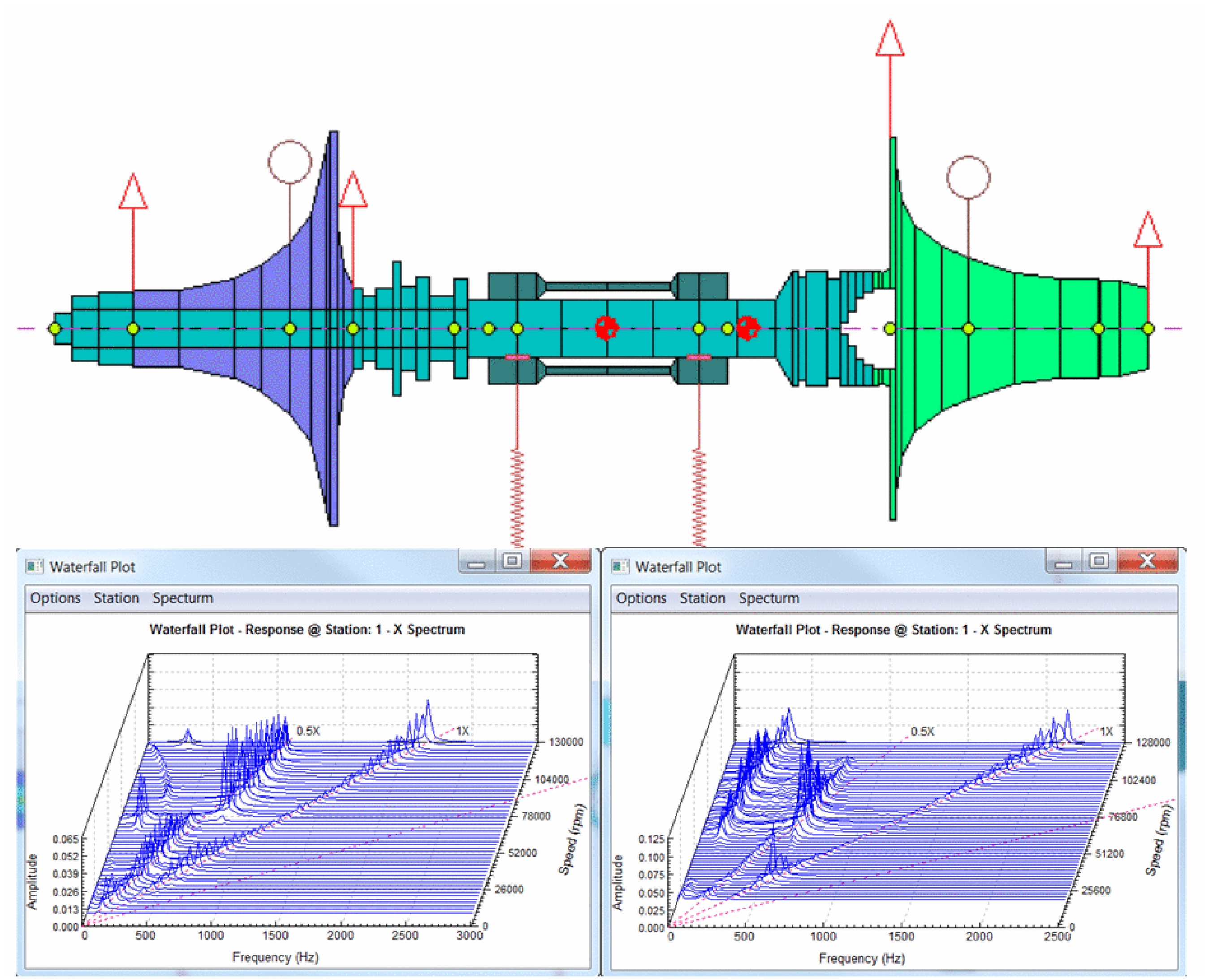
Turn rate: rad/sec.

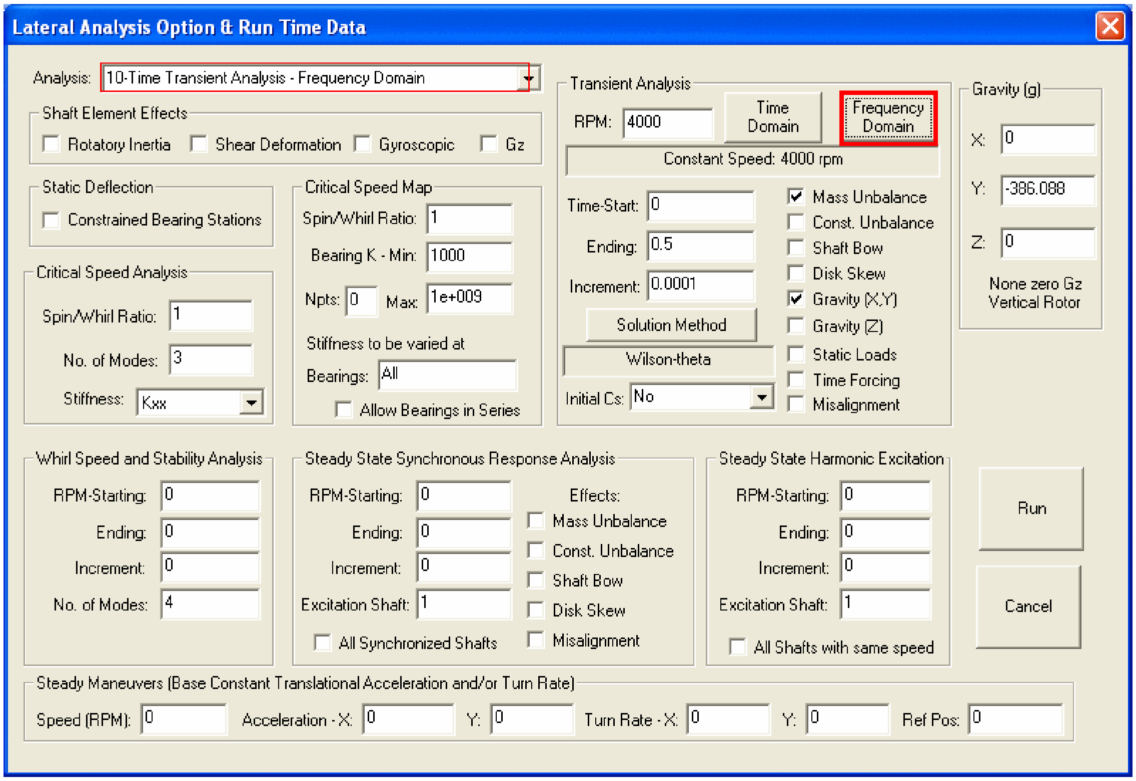


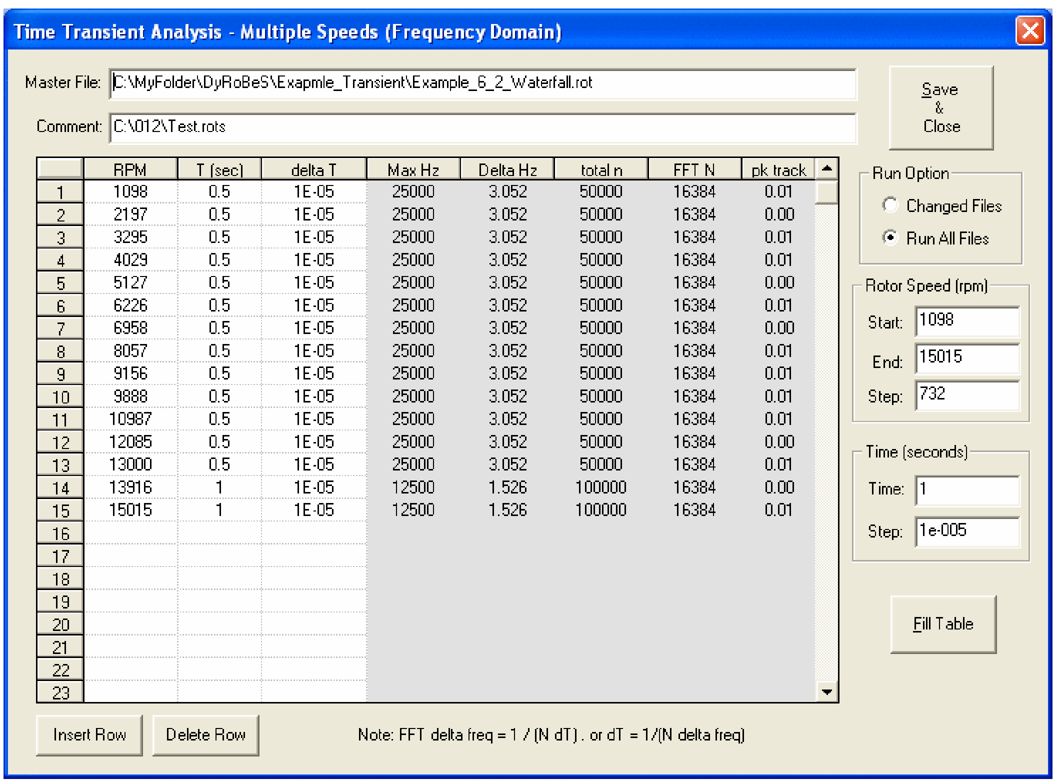
For sample output, click Steady Maneuver Load Response Plot.

Time Transient Analysis – Frequency Domain

This option allows for the transient analysis in multiple speeds and the results can be displayed in waterfall or cascade plot, response spectral intensity plot, order tracking plot, and shaft center line plot. This is a very time consuming task, so it is not a common analysis. However, it is widely used in small turbocharger rotor dynamic analysis since most of the small turbocharger rotor systems are nonlinear in nature.







For more information in Transient Analysis, click Time Transient Analysis (Time Domain).

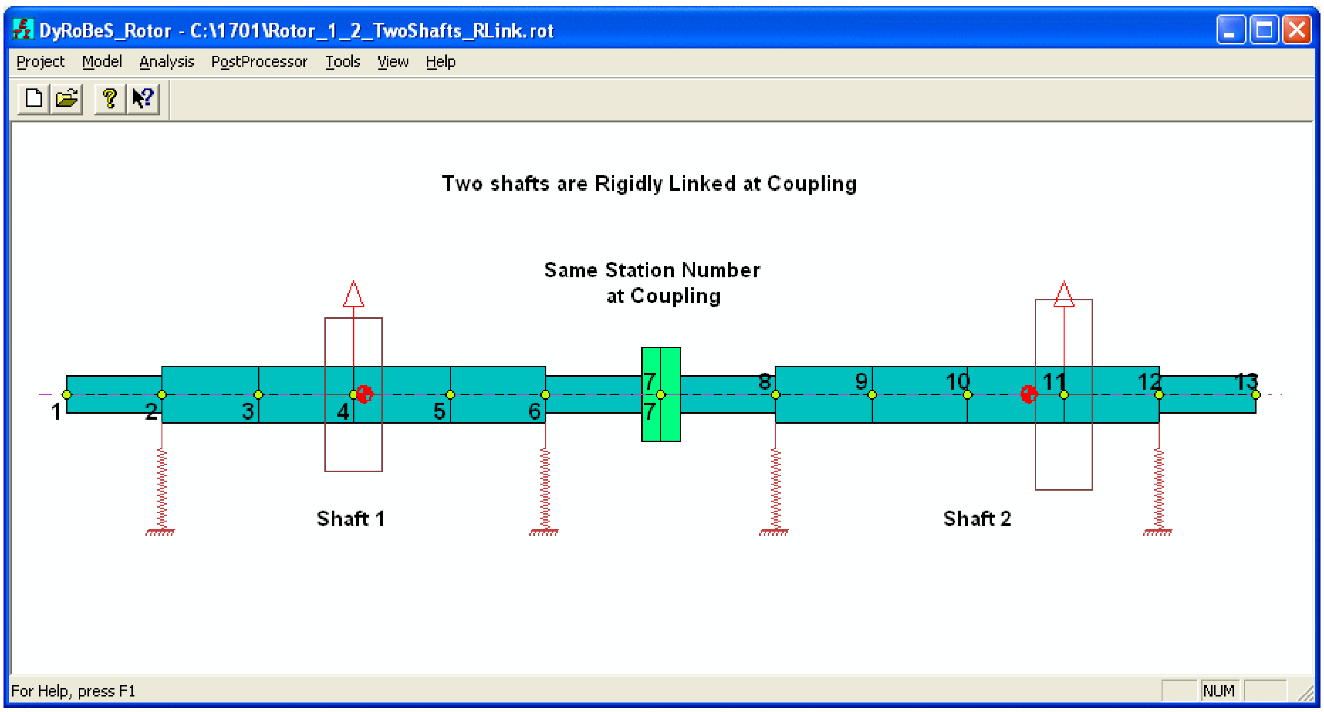
Natural Catenary (Gravity Sag) Analysis

This option performs optimization procedure to find the optimal bearing elevations such that the bending moment and/or shear force due to rotor weight (gravity sag) at coupling station are minimized. For a large turbine-generator system, the moment and force at coupling station, or near coupling location can experience large moment (stress) during startup. Therefore, at the construction of a turbine-generator system, the level of each bearing-center is adjusted (elevated) to minimize the bending moment and shear force at each coupling or locations where the potential failure may occur.

Ideally, the weight of each rotor should be supported entirely by its own bearings. The ideal bearing reactions are therefore equal to the bearing reactions present when the rotors are uncoupled. When the weight of each rotor is supported by its own bearings, the resulting shape of the full-shaft-line caused by the elevations of the bearings is often called Catenary Curve. Please note that the term of catenary used here has little to do with the catenary defined in mathematics or civil engineering due to the complexity of the rotor configurations.

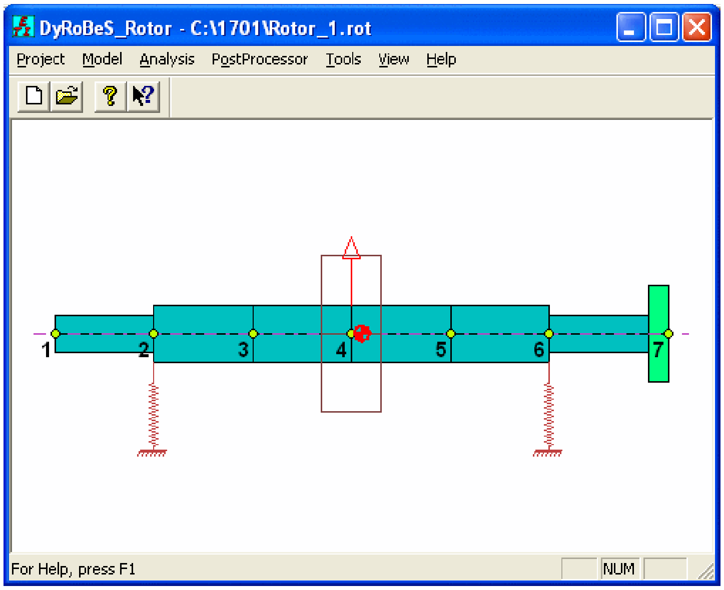
Caution must be taken when performing this task. Misalignment of the bearings will eliminate the high stress at or near couplings during the initial start up. However, this rotor bow also creates a synchronous excitation in addition to the mass unbalance excitation.

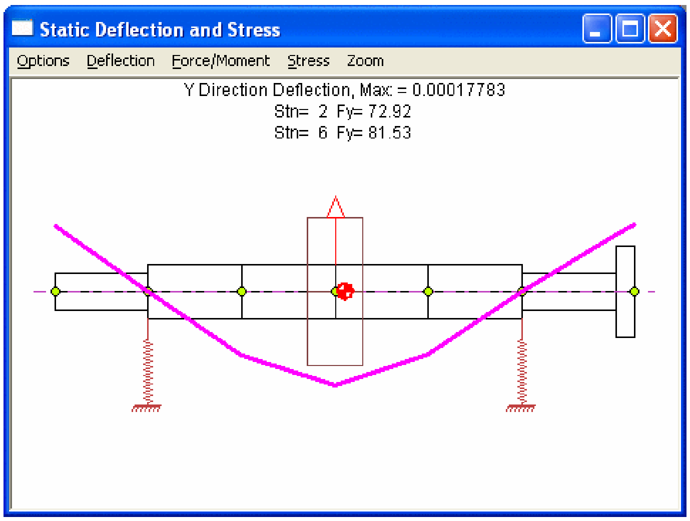
For each shaft supported by two bearings, the catenary curve can be obtained by the following procedure which will result in the rotor static weight of each rotor being supported extirely by its own bearings. An example is used here for illustration purposes:



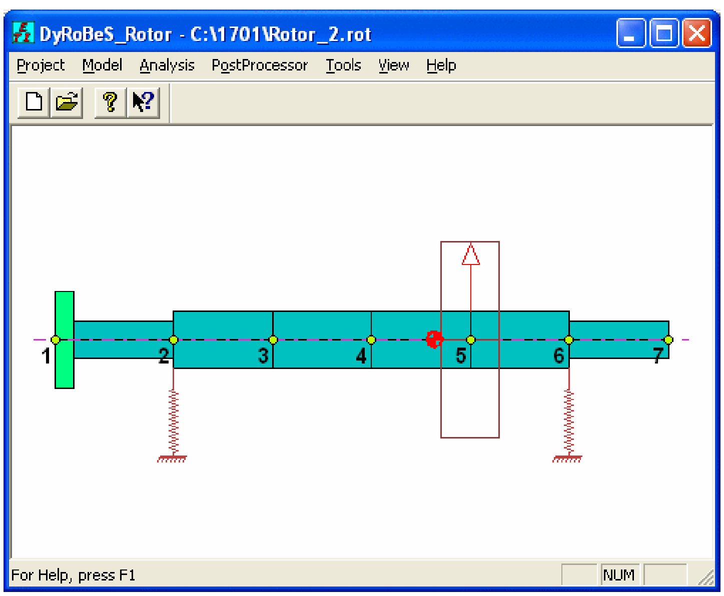
Step 1. Calculates the bearing reactions of the uncoupled rotor due to rotor weight only using Analysis Option 1 – Static Deflection and Bearing Loads.

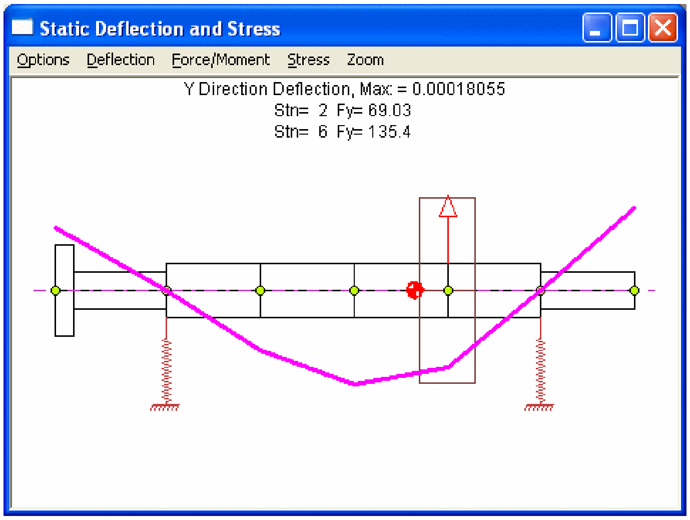
Rotor 1


DyRoBeS_Rotor - C:\1701\Rotor_1.rot
Project Model Analysis PostProcessor Tools View Help
1
2
3
4
5
6
7
For Help, press F1
NUM


Static Deflection and Stress
Options Deflection Force/Moment Stress Zoom
Y Direction Deflection, Max = 0.00017783
Stn= 2 Fy= 72.92
Stn= 6 Fy= 81.53

Rotor 2

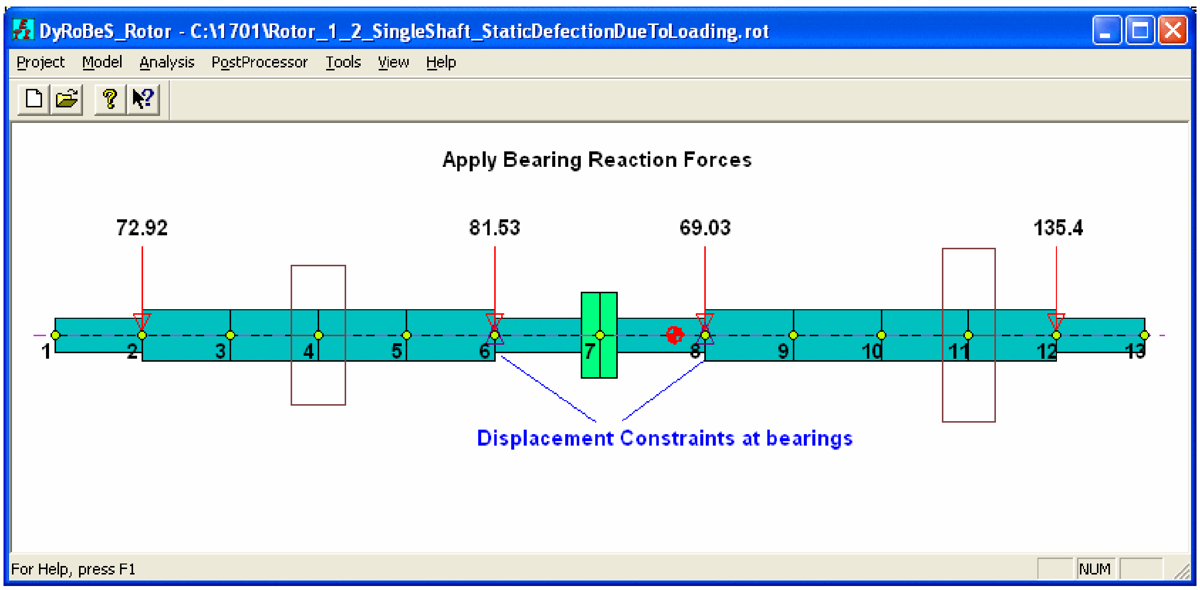

DyRoBeS_Rotor - C:\1701\Rotor_2.rot
Project Model Analysis PostProcessor Tools View Help
1
2
3
4
5
6
7
For Help, press F1
NUM

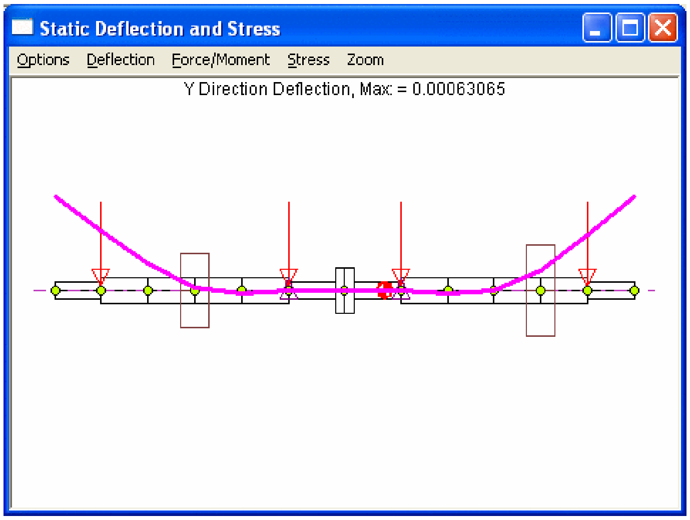


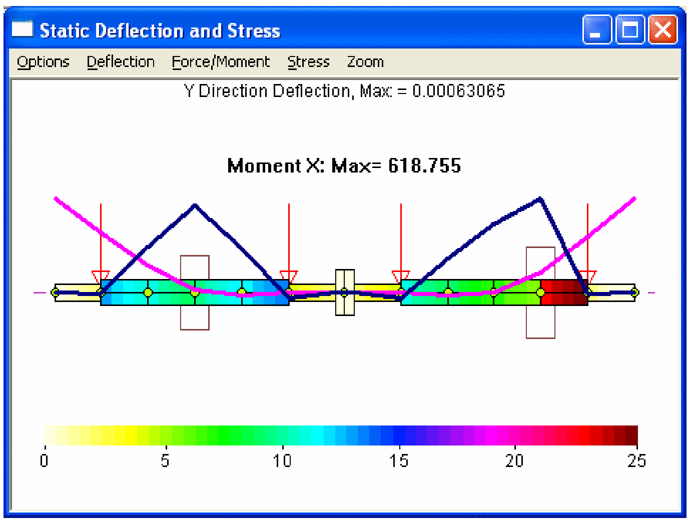
Note that the summation of 2 bearings reaction forces is the rotor static weight.

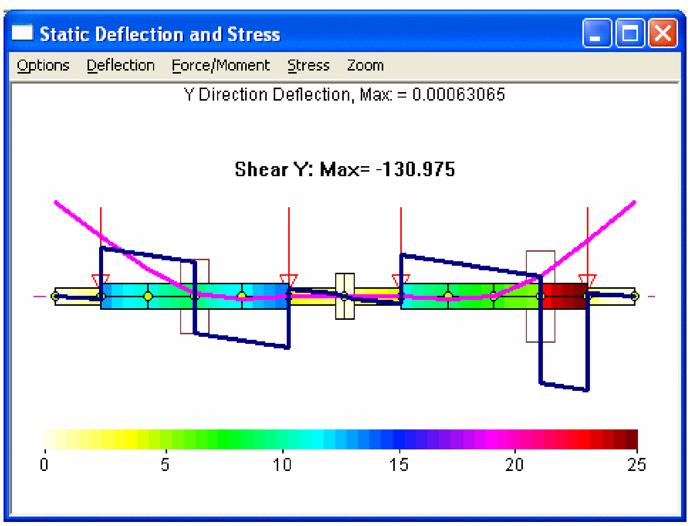
Step 2. Apply this bearing reaction forces on the combined rotor. Note that in this step, since the rotor is unconstrained, the solution is possible only after rigid body motion is eliminated. Since the rotor has translational and rotational motions for each plane, we will need at least two constraints. Most common is to constrain the displacements at two bearings near coupling, in this case, stations 6 and 8. Or, one may consider to restrict the displacement and slope at coupling. Both results from the Analysis Option 1 - Static Deflection and Bearing Loads are shown below:

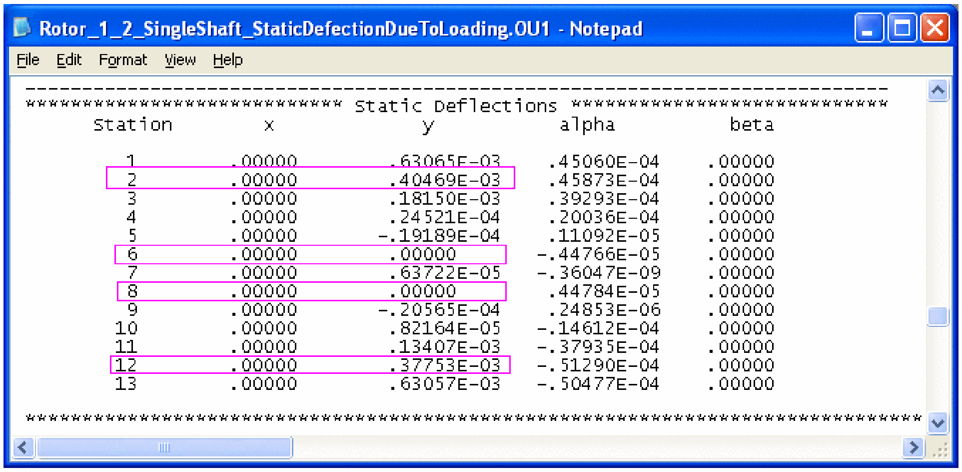
Case 1 – Fix the displacements at bearings – stations 6 & 8.




Static Deflection and Stress
Options Deflection Force/Moment Stress Zoom
Y Direction Deflection, Max = 0.00063065


Static Deflection and Stress
Options Deflection Force/Moment Stress Zoom
Y Direction Deflection, Max = 0.00063065
Moment X: Max= 618.755
0
5
10
15
20
25


Static Deflection and Stress
Options Deflection Force/Moment Stress Zoom
Y Direction Deflection, Max = 0.00063065
Shear Y: Max= -130.975
0
5
10
15
20
25



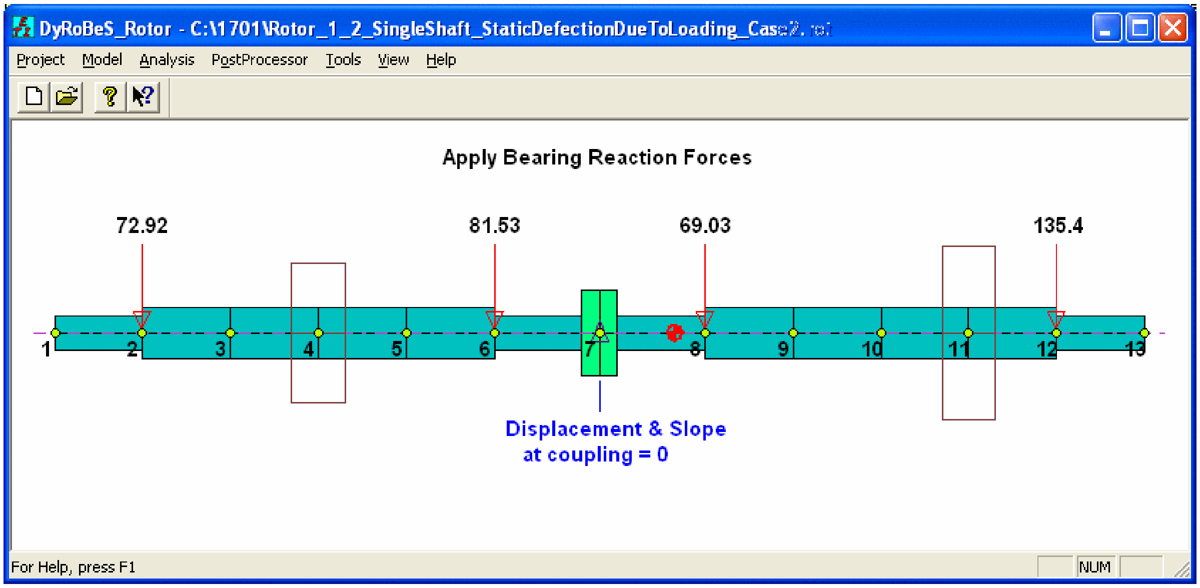
Rotor_1_2_SingleShaft_StaticDefectionDueToLoading.OU1 - Notepad

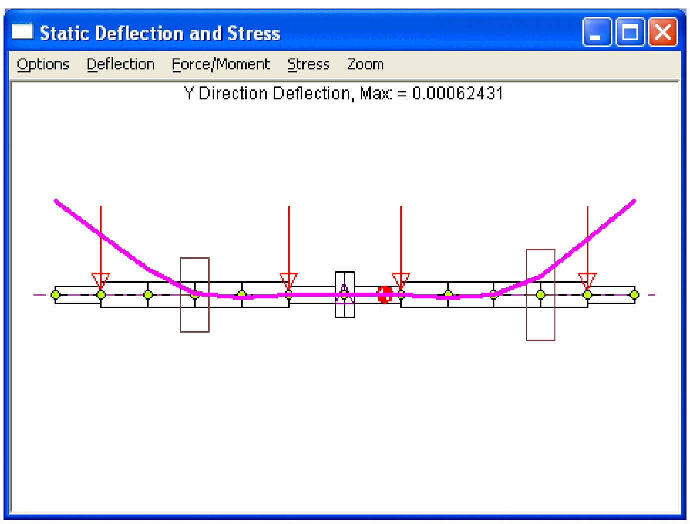
File Edit Format View Help

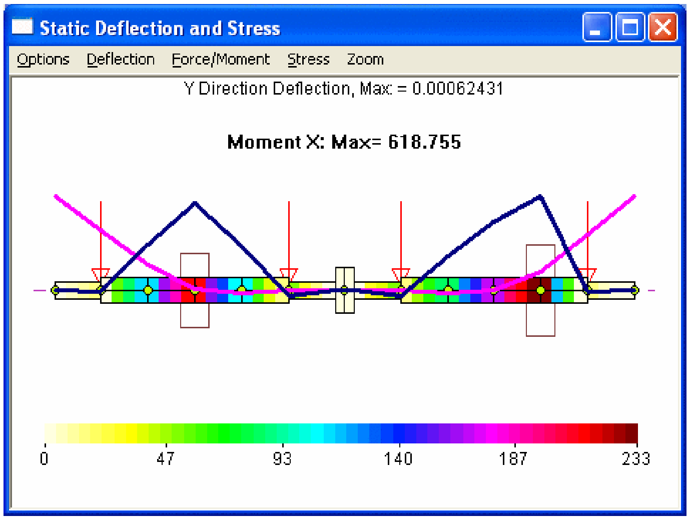
***************************** Static Deflections *****************************

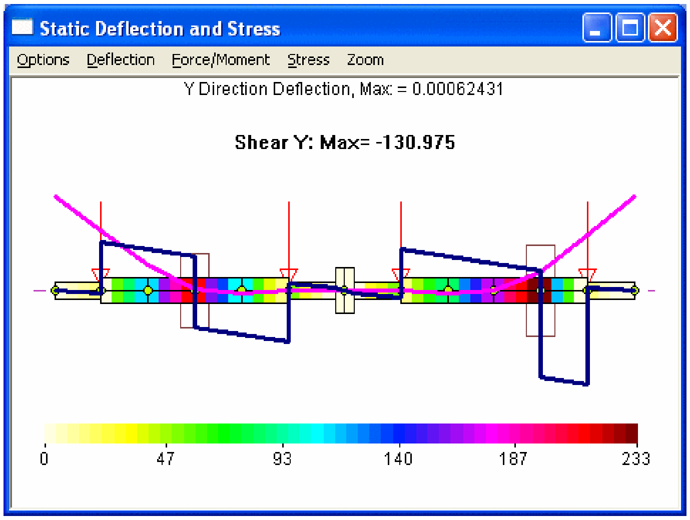
Station	x	y	alpha	beta
1	.00000	.63065E-03	.45060E-04	.00000
2	.00000	.40469E-03	.45873E-04	.00000
3	.00000	.18150E-03	.39293E-04	.00000
4	.00000	.24521E-04	.20036E-04	.00000
5	.00000	-.19189E-04	.11092E-05	.00000
6	.00000	.00000	-.44766E-05	.00000
7	.00000	.63722E-05	-.36047E-09	.00000
8	.00000	.00000	.44784E-05	.00000
9	.00000	-.20565E-04	.24853E-06	.00000
10	.00000	.82164E-05	-.14612E-04	.00000
11	.00000	.13407E-03	-.37935E-04	.00000
12	.00000	.37753E-03	-.51290E-04	.00000
13	.00000	.63057E-03	-.50477E-04	.00000

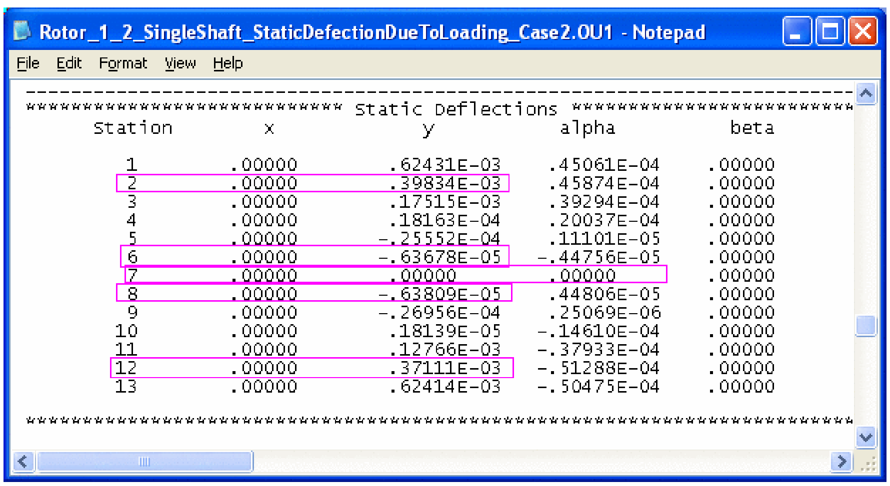
Case 2 – Fix the displacement and slope at coupling – station 7










Rotor_1_2_SingleShaft_StaticDefectionDueToLoading_Case2.OU1 - Notepad

File Edit Format View Help

```
-------------------------------------------------------------------------------
***************************** Static Deflections **************************
     Station        x              y           alpha            beta

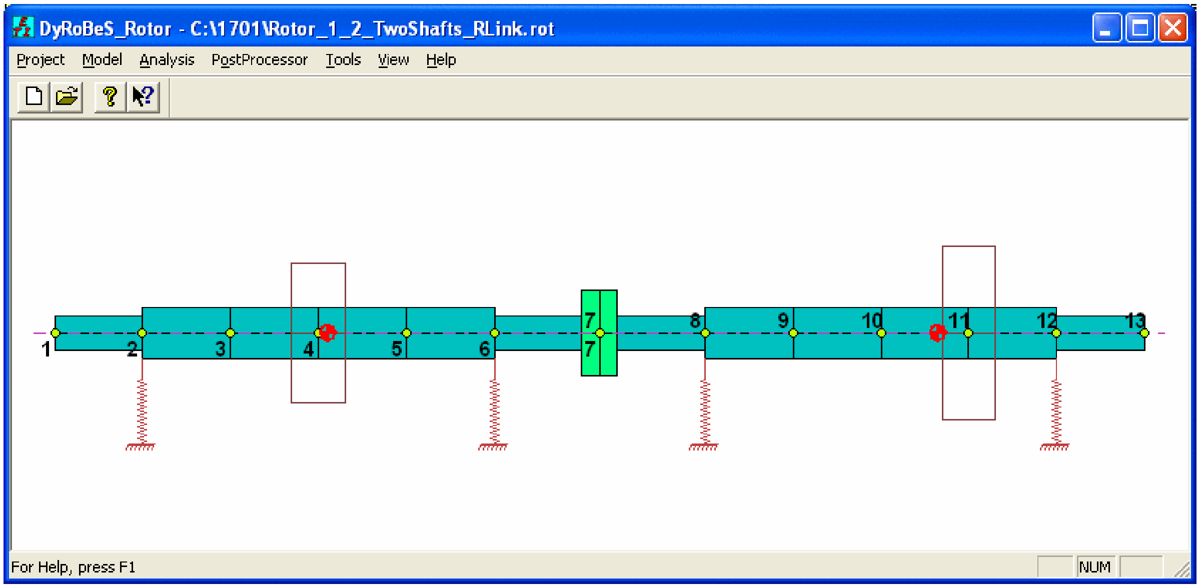
        1         .00000        .62431E-03    .45061E-04      .00000
        2         .00000        .39834E-03    .45874E-04      .00000
        3         .00000        .17515E-03    .39294E-04      .00000
        4         .00000        .18163E-04    .20037E-04      .00000
        5         .00000       -.25552E-04    .11101E-05      .00000
        6         .00000       -.63678E-05   -.44756E-05      .00000
        7         .00000        .00000        .00000          .00000
        8         .00000       -.63809E-05    .44806E-05      .00000
        9         .00000       -.26956E-04    .25069E-06      .00000
       10         .00000        .18139E-05   -.14610E-04      .00000
       11         .00000        .12766E-03   -.37933E-04      .00000
       12         .00000        .37111E-03   -.51288E-04      .00000
       13         .00000        .62414E-03   -.50475E-04      .00000

*******************************************************************************
```

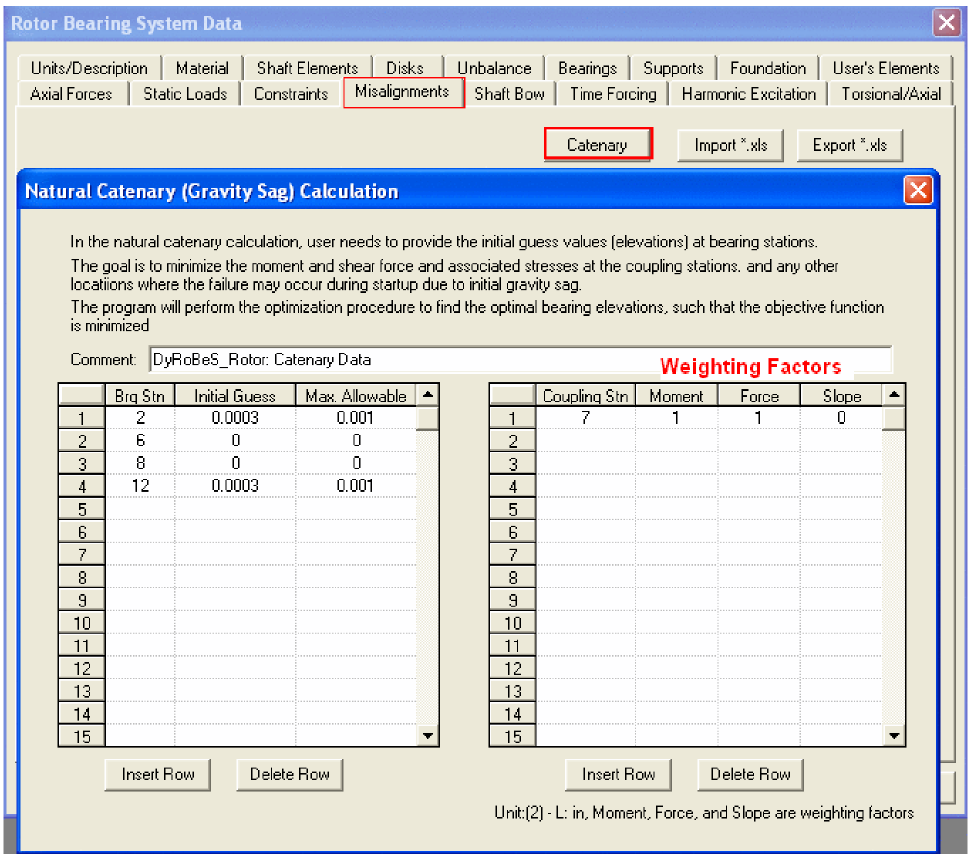
Note that both cases yield nearly identical results in the deflection curve, moment, force, and stress. However, case 1 needs to raise 2 bearings at station 2 and 12 only. But case 2 needs to change the bearing elevations for all 4 bearings, which may not be so desirable, although changes in bearings at stations 6 and 8 are extremely small.

The catenary (gravity sag) analysis provided in DyRoBeS utilizes the optimization techniques to find the optimal elevations for the selected bearings with specified upper and lower bounds such that the moments, and/or forces, and/or slopes at selected stations (may be couplings, bearings, or the weakest link locations) are minimized. One can also use the weighting factors to enhance their objective in moment, or force, or slope.

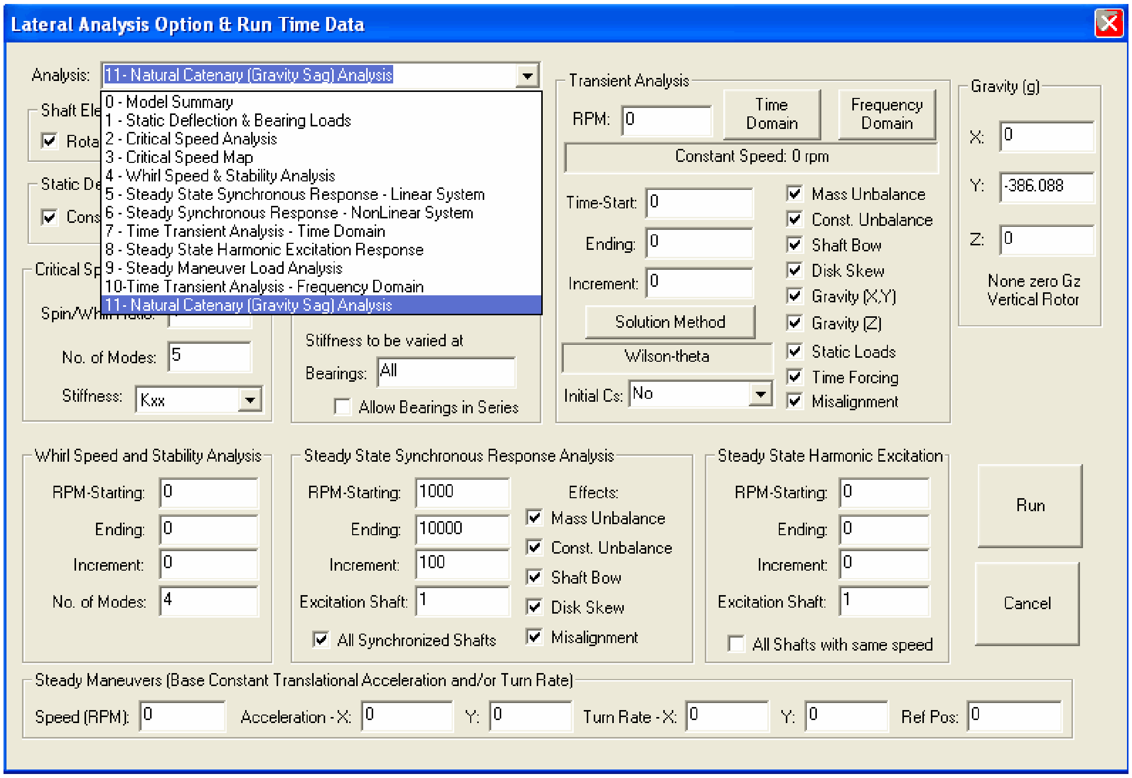
In this same example, let us model the rotor system in 2 shafts, as shown below. Note that if the system is modeled as a single shaft, same result can be obtained.



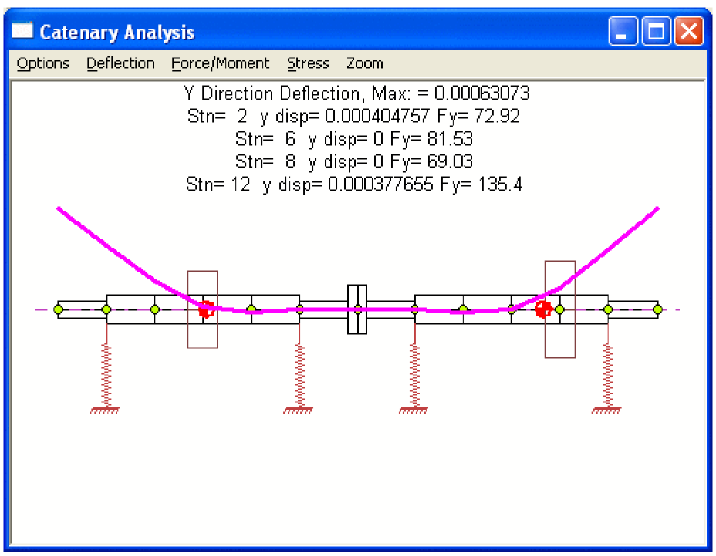
The initial guess values and objective are defined under Misalignment Tab – Catenary button as shown below. It shows that at bearing stations 2 & 12, the initial guess values are 0.0003 inches and maximum allowable values are 0.001 inches. At bearing stations 6 and 8, the initial values are zero and the maximum allowable values are also zero. It indicates that these two bearings are not the design variables and their elevations will not be changed. Coupling station 7 is specified such that the moment and force are minimized. Note that any station or stations can be used to minimize their moments and/or force, and/or slope.



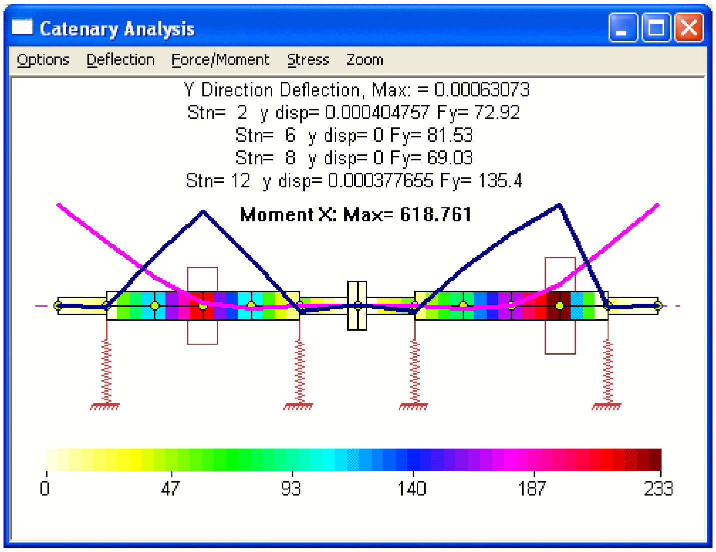
The analysis option is 11 – Natural Catenary (Gravity Sag) Analysis. Once this analysis option is selected, the program will perform the optimization procedure. The results can be viewed just like the results from the Analysis Option 1 – Static Deflection. The final converged results are shown below. It shows that the program converged to the bearing elevations extremely close to the results from Case 1 studied above using manual procedure, which is not surprising. Also, by using these bearing elevations, the bearing reaction forces are identical to the bearing forces from the Step 1 of the manual procedure.

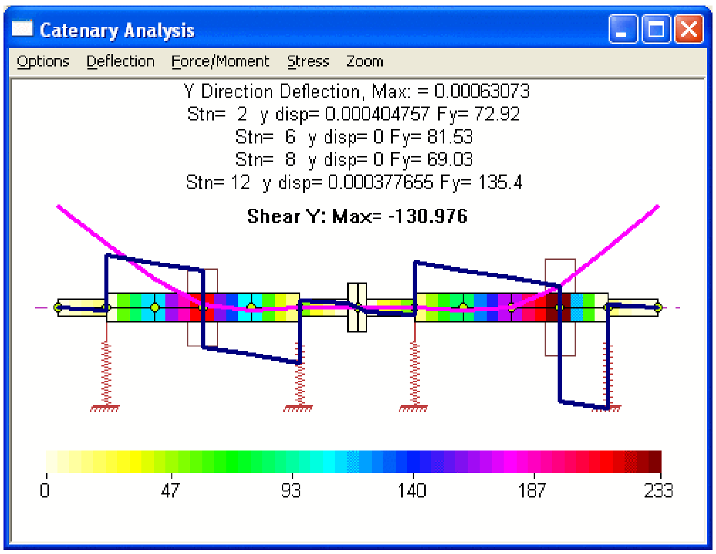


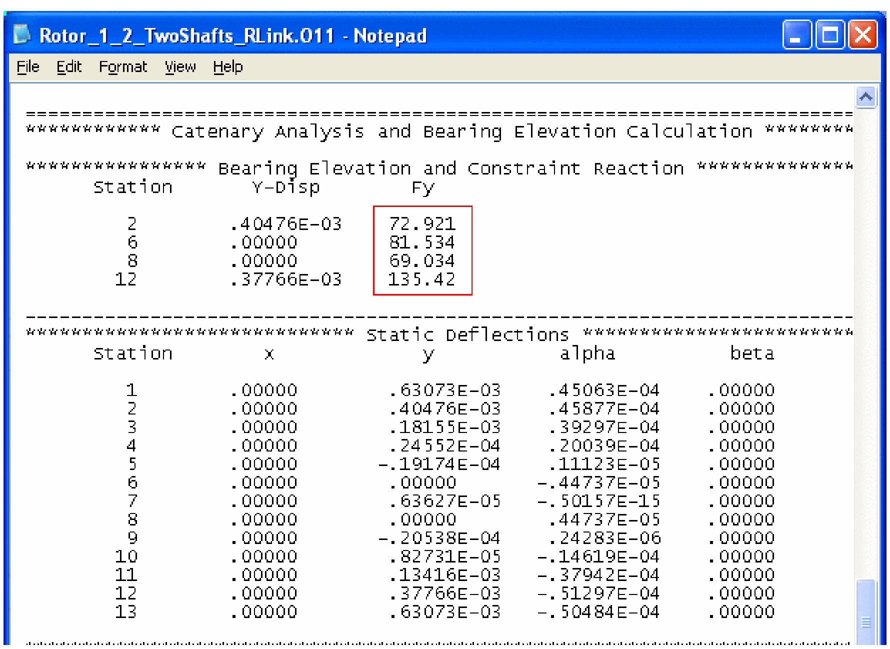
Lateral Analysis Option & Run Time Data
Analysis: 11- Natural Catenary (Gravity Sag) Analysis
0 - Model Summary
1 - Static Deflection & Bearing Loads
2 - Critical Speed Analysis
3 - Critical Speed Map
4 - Whirl Speed & Stability Analysis
5 - Steady State Synchronous Response - Linear System
6 - Steady Synchronous Response - NonLinear System
7 - Time Transient Analysis - Time Domain
8 - Steady State Harmonic Excitation Response
9 - Steady Maneuver Load Analysis
10-Time Transient Analysis - Frequency Domain
11- Natural Catenary (Gravity Sag) Analysis
No. of Modes: 5
Stiffness: Kxx
Stiffness to be varied at
Bearings: All
Allow Bearings in Series
Transient Analysis
RPM: 0
Time Domain
Frequency Domain
Constant Speed: 0 rpm
Time-Start: 0
Ending: 0
Increment: 0
Solution Method
Wilson-theta
Initial Cs: No
Mass Unbalance
Const. Unbalance
Shaft Bow
Disk Skew
Gravity (X,Y)
Gravity (Z)
Static Loads
Time Forcing
Misalignment
Gravity (g)
X: 0
Y: -386.088
Z: 0
None zero Gz Vertical Rotor
Whirl Speed and Stability Analysis
RPM-Starting: 0
Ending: 0
Increment: 0
No. of Modes: 4
Steady State Synchronous Response Analysis
RPM-Starting: 1000
Ending: 10000
Increment: 100
Excitation Shaft: 1
All Synchronized Shafts
Effects:
Mass Unbalance
Const. Unbalance
Shaft Bow
Disk Skew
Misalignment
Steady State Harmonic Excitation
RPM-Starting: 0
Ending: 0
Increment: 0
Excitation Shaft: 1
All Shafts with same speed
Run
Cancel
Steady Maneuvers (Base Constant Translational Acceleration and/or Turn Rate)
Speed (RPM): 0
Acceleration - X: 0
Y: 0
Turn Rate - X: 0
Y: 0
Ref Pos: 0



Catenary Analysis
Options Deflection Force/Moment Stress Zoom
Y Direction Deflection, Max: = 0.00063073
Stn= 2 y disp= 0.000404757 Fy= 72.92
Stn= 6 y disp= 0 Fy= 81.53
Stn= 8 y disp= 0 Fy= 69.03
Stn= 12 y disp= 0.000377655 Fy= 135.4






Rotor_1_2_TwoShafts_RLink.011 - Notepad

File Edit Format View Help

************* Catenary Analysis and Bearing Elevation Calculation ********

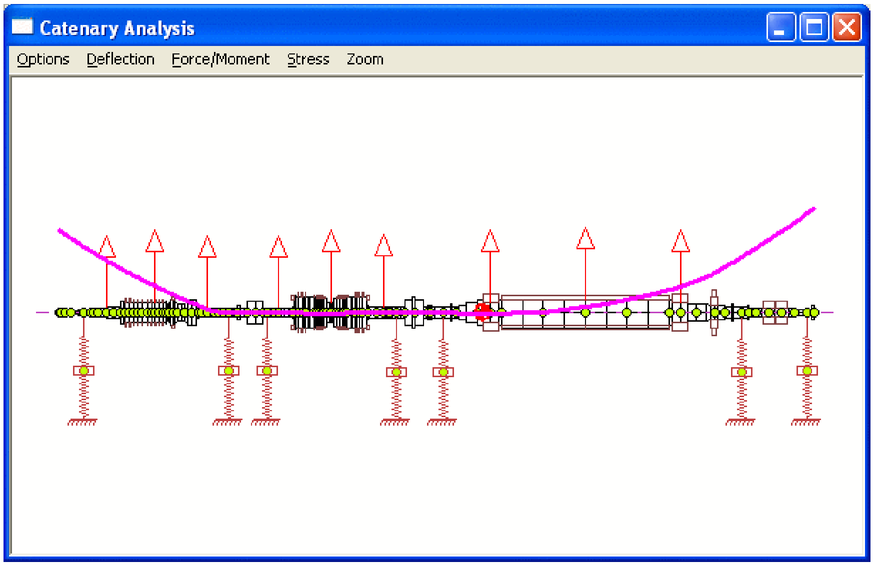
**************** Bearing Elevation and Constraint Reaction **************

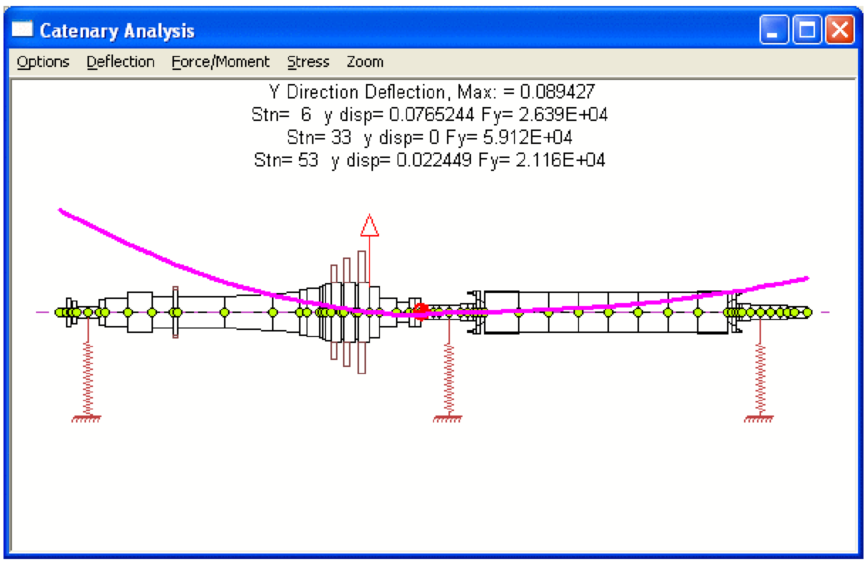
Station	Y-Disp	Fy
2	.40476E-03	72.921
6	.00000	81.534
8	.00000	69.034
12	.37766E-03	135.42

***************************** Static Deflections ************************

Station	x	y	alpha	beta
1	.00000	.63073E-03	.45063E-04	.00000
2	.00000	.40476E-03	.45877E-04	.00000
3	.00000	.18155E-03	.39297E-04	.00000
4	.00000	.24552E-04	.20039E-04	.00000
5	.00000	-.19174E-04	.11123E-05	.00000
6	.00000	.00000	-.44737E-05	.00000
7	.00000	.63627E-05	-.50157E-15	.00000
8	.00000	.00000	.44737E-05	.00000
9	.00000	-.20538E-04	.24283E-06	.00000
10	.00000	.82731E-05	-.14619E-04	.00000
11	.00000	.13416E-03	-.37942E-04	.00000
12	.00000	.37766E-03	-.51297E-04	.00000
13	.00000	.63073E-03	-.50484E-04	.00000

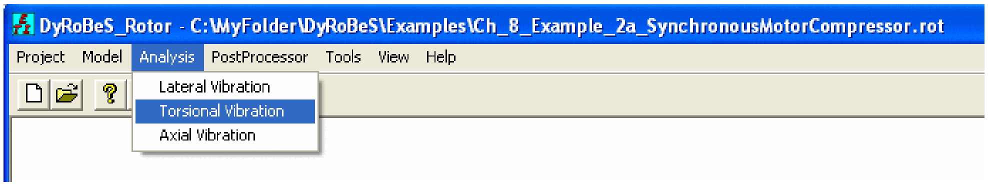
The major benefit of this optimization procedure is not for systems with 2 bearings at each rotor as illustrated in this example, but it is for more complicated systems with many bearings or less bearings for each rotor as shown below.


Catenary Analysis
Options Deflection Force/Moment Stress Zoom

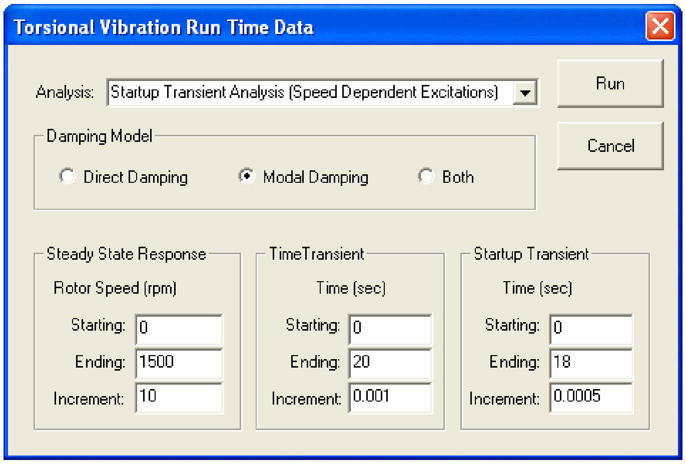

Catenary Analysis
Options Deflection Force/Moment Stress Zoom
Y Direction Deflection, Max: = 0.089427
Stn= 6 y disp= 0.0765244 Fy= 2.639E+04
Stn= 33 y disp= 0 Fy= 5.912E+04
Stn= 53 y disp= 0.022449 Fy= 2.116E+04

Torsional Vibration Analysis

To perform the torsional vibration analysis, the rot-bearing data file (.rot) must be built and opened. From the Main Menu, select Analysis and Torsional Vibration as shown below.



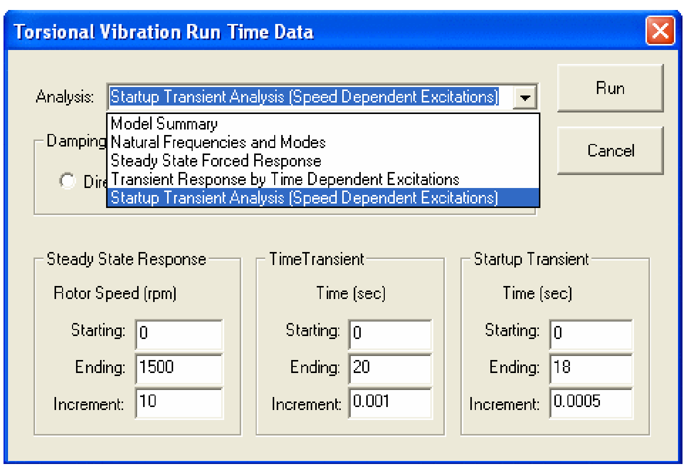
The run time data screen for the Torsional Vibration Analysis is shown below:



The inputs are described:

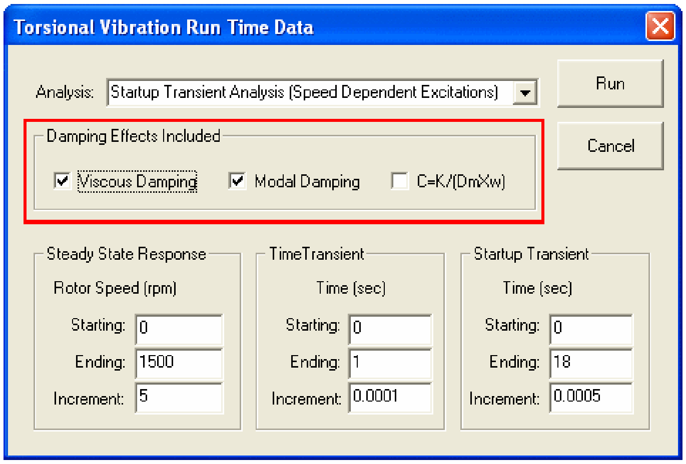
1. Analysis:

Select one analysis at a time as shown below. Once the specific analysis option is selected, the corresponding data are also required.



2. Damping Model:

You can select the damping model to be used in the analysis. Three types of damping model can be included in the analysis: Viscous damping, Modal damping, and Frequency dependent damping. Viscous damping used widely in the lateral vibration analysis is not commonly available in the torsional analysis. Modal damping is widely used in torsional analysis. Frequency dependent damping is commonly used in the rubber type coupling and can only be used in the forced response. Details on the modal damping model are explained in the Modal Damping



For a geared system (multi-branch system), one shaft (normally the shaft number 1) is selected to be the reference shaft and all other shaft properties are converted into the equivalent properties referenced to this reference shaft. The conversion between the actual (physical) system and equivalent (mathematical) system for a geared system is listed below:

Parameter	Actual (physical)	Equivalent (mathematical)
Displacement	q	q/n
Inertia	J	Jn^2
Stiffness	K	Kn^2
Damping	C	Cn^2
Torque	T	Tn

Where n is the speed ratio of the branch rotor to the reference rotor.

See also:

Damped and Undamped Natural Frequencies and Modes Calculation

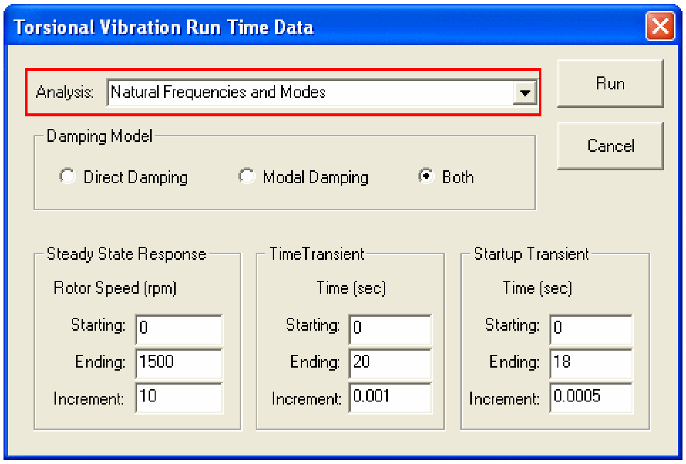
Steady State Forced Response Analysis

Transient Analysis (Time Dependent Excitations)

Startup Transient Analysis (Speed Dependent Excitations)

Torsional Natural Frequencies and Modes

This option calculates the torsional damped and undamped natural frequencies, mode shapes, and associated modal stress. It also provides the frequency interference diagram.



For sample outputs, click Torsional Natural Frequency and Mode Shape Plot.

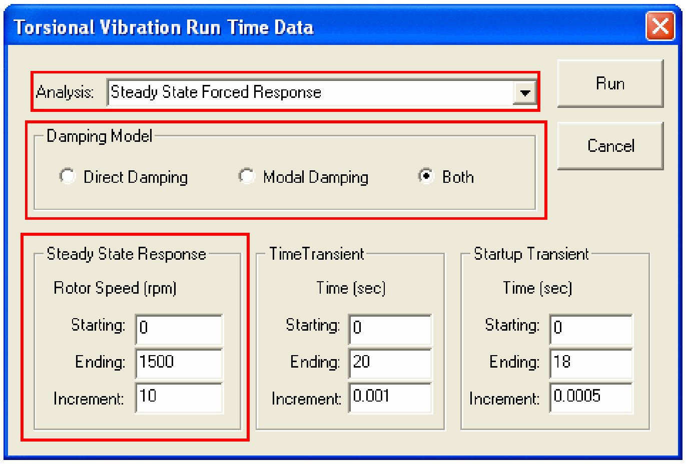
Torsional Steady State Forced Response

This option calculates the torsional steady state forced response for a given range of rotor speed due to Steady State Excitation. The Steady state Forced Response Analysis will be performed only when the steady state excitation exists. The excitation amplitude and frequency can be functions of rotor speed. Details are explained in the Modeling-Data Editor.

The element vibratory torque is calculated by:

$$T = K\Delta\theta + C\Delta\dot{\theta}$$

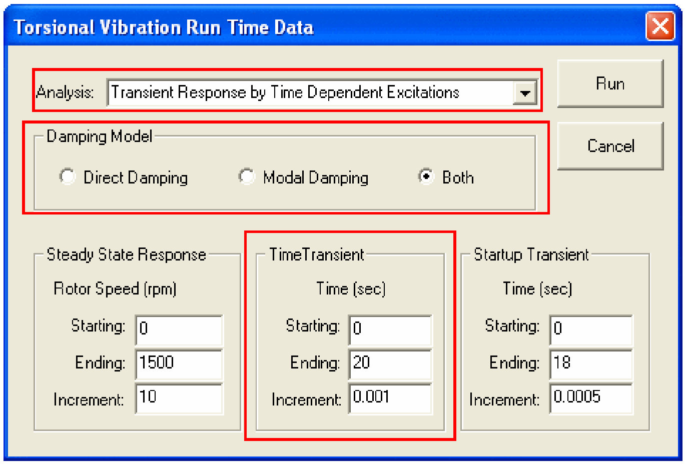
If modal damping is specified, then the vibratory torque caused by the damping, i.e., the second term in the above equation, will not be included.



For sample outputs, click Torsional Steady State Response.

Torsional Transient Analysis with Time Dependent Excitations including Short Circuit Analysis

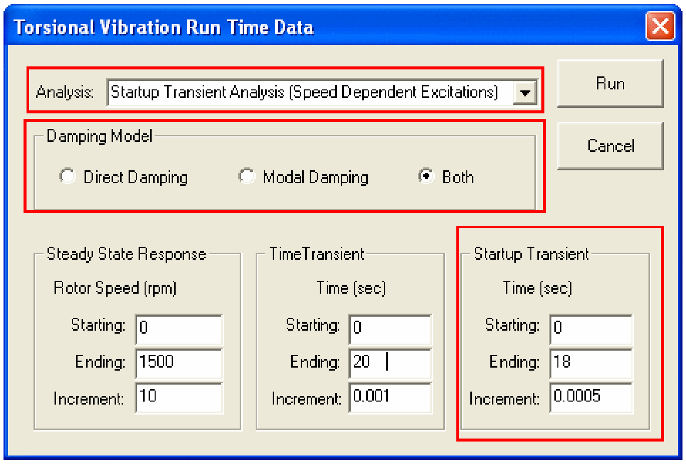
This option performs the torsional transient analysis with time dependent excitations including short circuit torques for a given range of time interval. The time dependent excitation can be entered in the equation forms or from data files. The time dependent excitations in equations are entered in Torsional Excitations in Equation. The time dependent excitations in data files are entered in Torsional Excitations in Data Files. Again, for the transient analysis, the modal damping can be specified if the direct damping is not readily available. Details on the damping model are explained in the Modal Damping.



For sample outputs, click Torsional Transient Response

Torsional Transient Startup Analysis with Speed Dependent Excitations

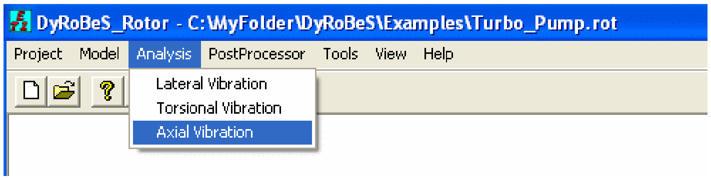
This option performs the torsional transient startup analysis for a given range of time interval. The driving and load torques are speed dependent and entered in Torsional Driving Torque and Load Torque. For startup transient analysis, the excitations are given in terms of rotor speed. For the transient analysis, the modal damping can be specified if the direct damping is not readily available. Details on the damping model are explained in the Modal Damping.



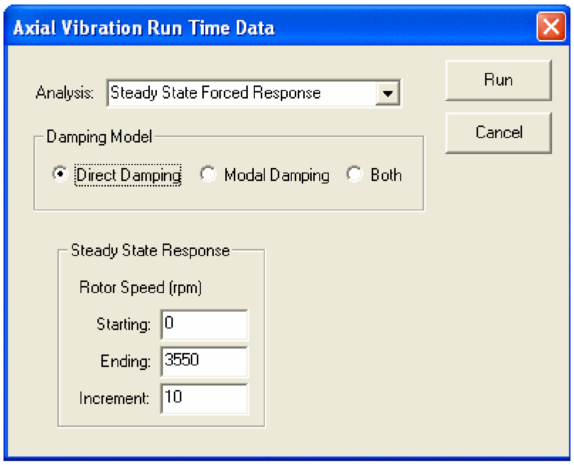
For sample outputs, click Torsional Transient Response

Axial Vibration Analysis

To perform the axial vibration analysis, the rot-bearing data file (.rot) must be built and opened. From the Main Menu, select Analysis and Axial Vibration as shown below.



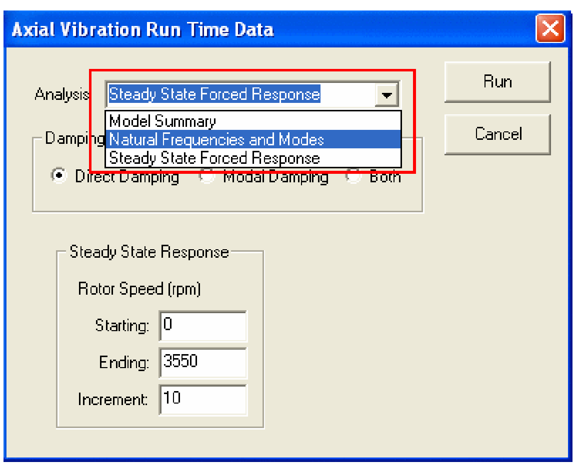
The run time data screen for the Axial Vibration Analysis is shown below:



The inputs are described:

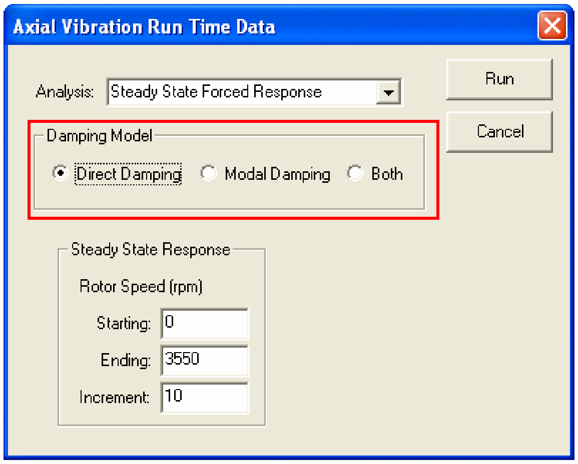
1. Analysis:

Select one analysis at a time as shown below. Once the specific analysis option is selected, the corresponding data are also required.



2. Damping Model:

You can select the damping model to be used in the analysis. You can use either the direct damping, or modal damping, or the combination of both. For forced response analyses including the transient analysis, the modal damping can be specified if the direct damping is not readily available. Details on the damping model are explained in the Modal Damping



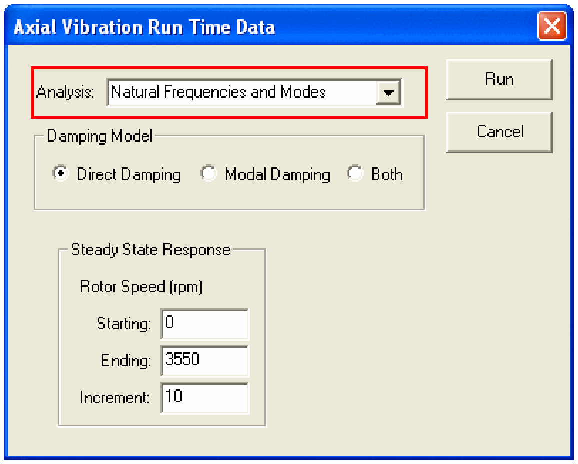
See also:

[Damped and Undamped Natural Frequencies and Modes Calculation]

[Steady State Forced Response Analysis]

Axial Natural Frequencies and Modes

This option calculates the axial damped and undamped natural frequencies and mode shapes.

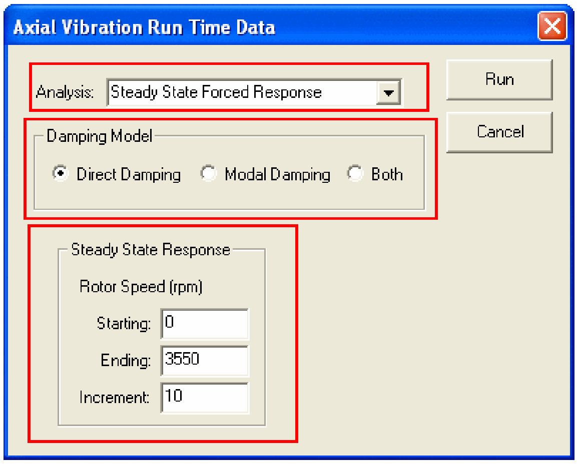


For sample outputs, click Torsional/Axial Natural Frequency and Mode Shape Plot.

Axial Steady State Forced Response

This option calculates the axial steady state forced response for a given range of rotor speed due to Steady State Excitation. It will be calculated only when the excitation exists. The excitation amplitude and frequency can be functions of rotor speed.

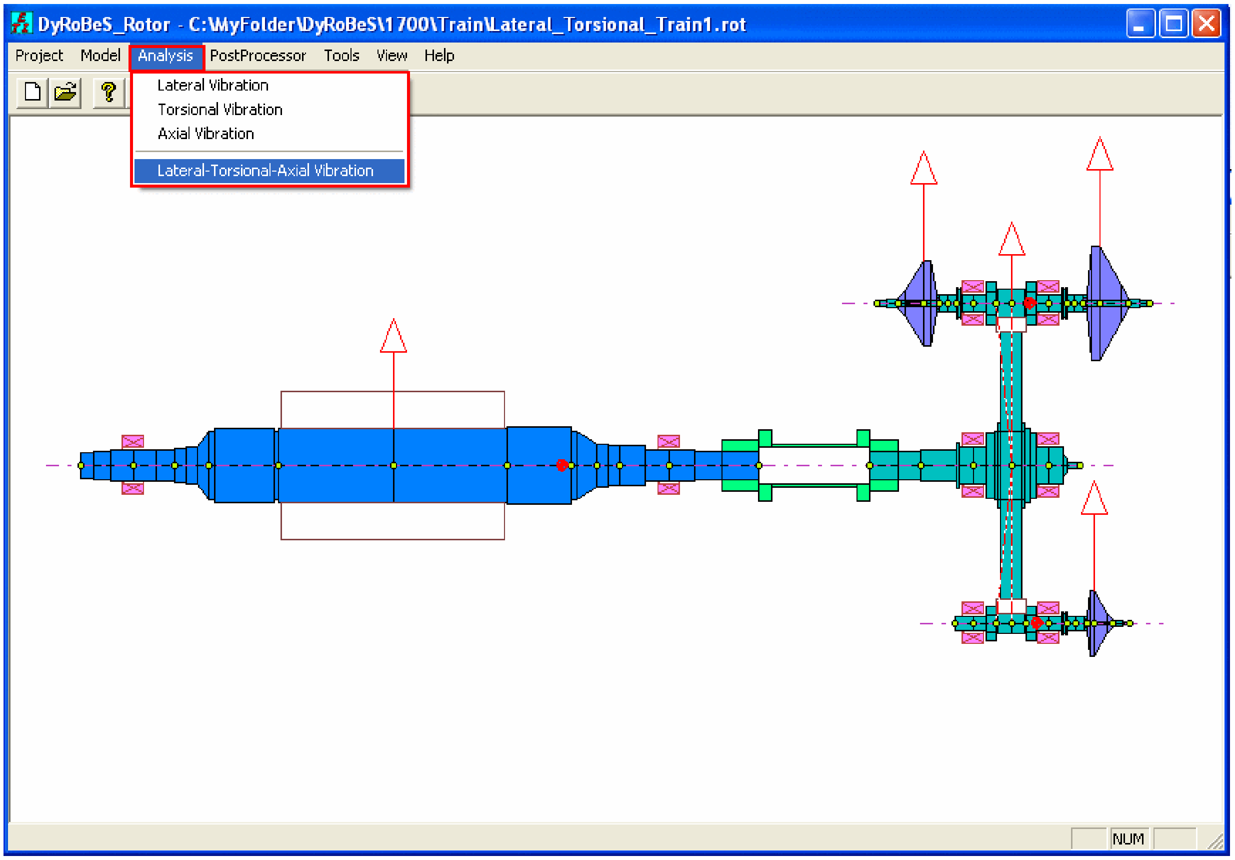
For forced response analysis, the modal damping can be specified if the direct damping is not readily available. Details on the damping model are explained in the Modal Damping. If modal damping is specified, then the vibratory force caused by the damping, i.e., the second term in the above equation, will not be included.

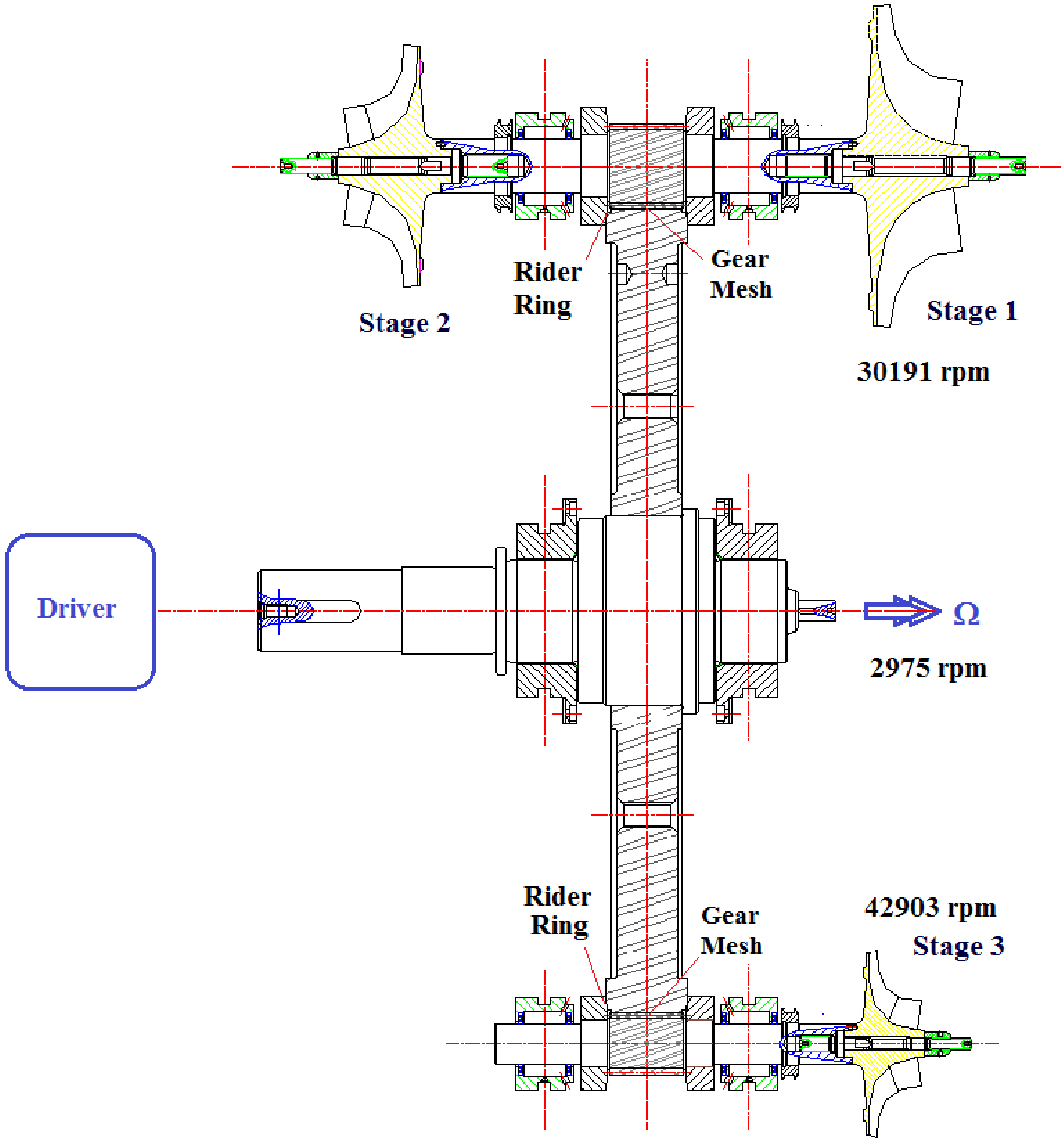


For sample outputs, click Torsional Steady State Response.

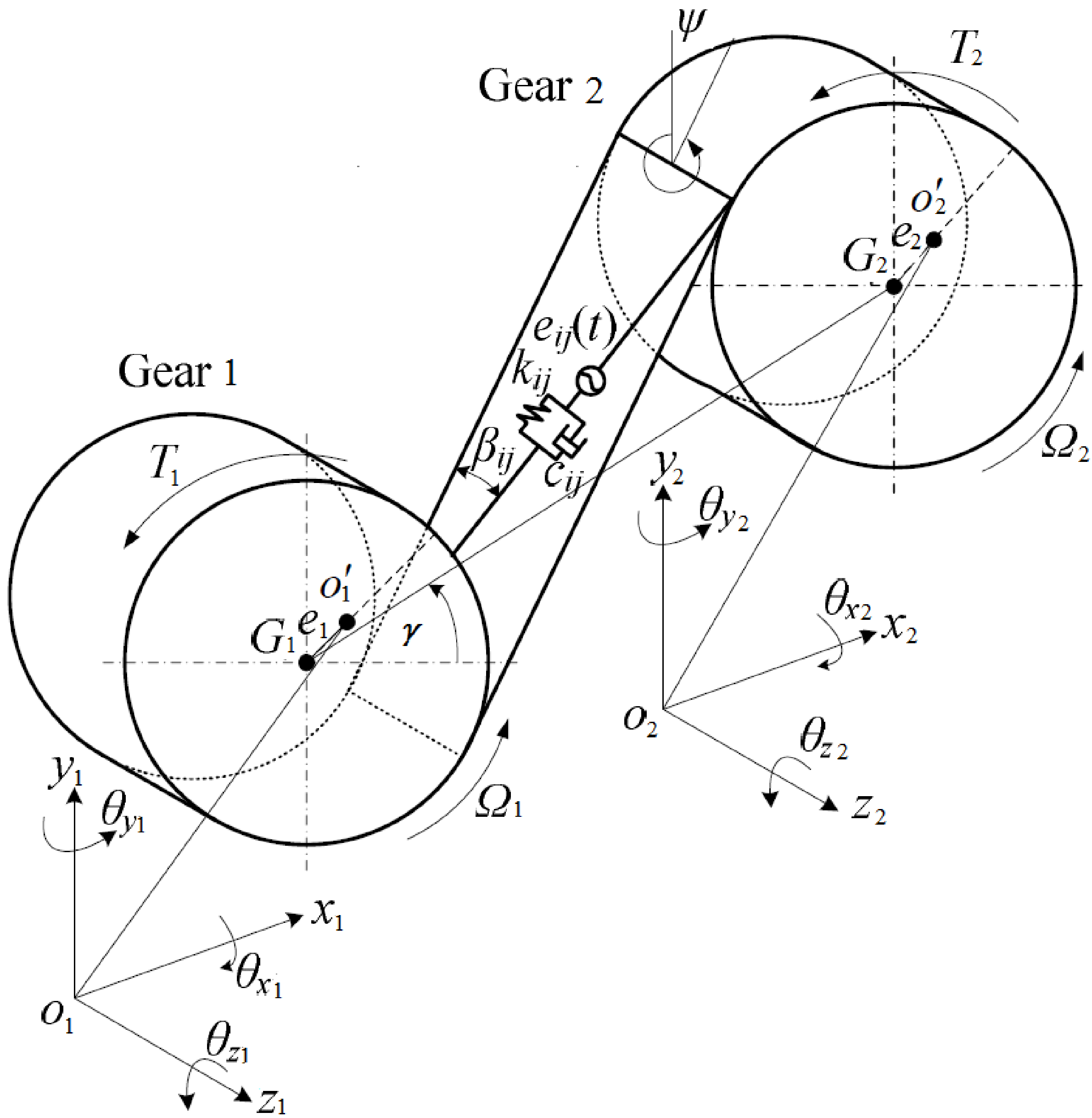
Coupled Lateral-Torsional-Axial Vibration Analysis

For a geared system, the lateral, torsional, and axial motions are coupled though the gear mesh and/or rider ring (thrust collar).

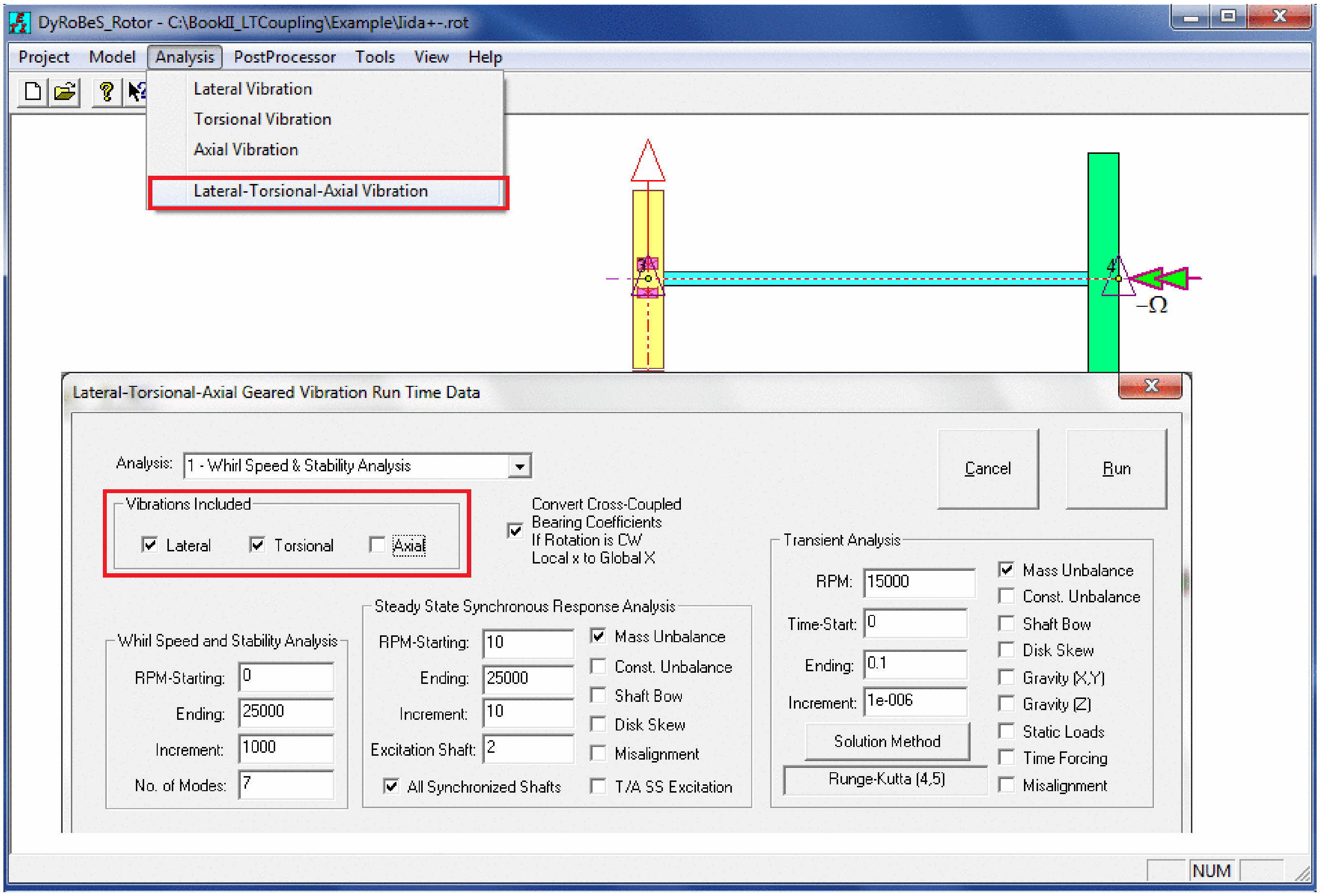




To be consistent with the coordinate system, the generalized displacements at each station contain 3 translational (x,y,z) and 3 rotational (θx, θy, θz) displacements., as shown below:



Note that in this coupled lateral-torsional-axial vibration analysis, one has the option to include all 3 vibrations, or just one vibration, or any combinations.



When analyzing the geared rotor systems and the motion at each finite element station is defined in above figure, then the following cautions must be taken:

Treat the rotor rotational speed ? as a vector, positive if it rotates in the positive Z direction (CCW for a right hand ruled Cartesian coordinate system) and negative if it rotates in the negative Z direction (CW). This will take care of the gyroscopic matrix and related forcing function involved rotational speed.

Change the cross-coupled bearing stiffness and damping coefficients *if necessary* when the rotor speed is negative, since almost all the bearing coefficients are obtained by assuming the rotor speed is positive in the positive Z direction. This is particularly important when performing the stability analysis since cross-coupled stiffness coefficients, including the aerodynamic cross-coupling term, significantly affect the system stability.

When viewing the results for lateral motion, the direction of whirling needs to take into account the rotor speed. That is, the forward or backward whirl is defined by the product of and , not just as used for a single rotor system.

For more details on this coupled lateral-torsional-axial vibration, please reference the document.

See also:

Whirl Speed and Stability Analysis

Steady State Synchronous Response Analysis

Time Transient Analysis

PostProcessor

The assessment of the analysis results constitutes an important aspect of the entire simulation process. The PostProcessor allows you to view the results in the ASCII (text) format and/or the graphics format. ***DyRoBeS©_Rotor*** provides a large number of post processing tools for graphically displaying the results. You can open the Child Windows (PostProcessor graphics) as many as you like to help you to interpret and understand the analysis results. When you open a post processing Child Window, some default initial settings are used to display the results. To modify these settings, select the **Settings** under the **Options** to make necessary changes. Whenever possible, **Scale Factor** is introduced in the **Settings**. This scale factor allows you to change their Y axis data scale, such as changing units from inches to mils or to mm, from mm to inches, etc. A new feature **Export Data** has been also implemented in Version 6.0, this option allows you to export the graphic data into an ASCII text file so you can plot this data using other graphic software. You are strongly recommended to try the graphic **Settings** under the **Options** menu to see their effects and to resize the Child Window to redraw (update) the picture. **Animation** can be an excellent tool to understand the rotor behavior. Whenever possible, animation is provided in the postprocessor.

For better understanding the analysis results, please refer to the book by Chen and Gunter (2005).

The following is a list of the post processing graphic features:

Torsional Vibration Analysis

Torsional Frequency Interference Diagram

Torsional Mode Shapes

Torsional Modal Stress

Torsional Steady State Response

Torsional Steady State Element Vibratory Torque

Torsional Steady State Element Vibratory Stress

Torsional Steady State Excitations

Torsional Transient Response (Time Dependent Excitations)

Torsional Transient Response - Displacement

Torsional Transient Response - Velocity

Torsional Transient Element Deflection

Torsional Transient Element Deflection

Torsional Transient Element Vibratory Torque

Torsional Transient Element Vibratory Stress

Torsional Transient Applied Torques

Torsional Transient Response FFT Analysis

Torsional Transient Response (Speed Dependent Excitations)

Torsional Startup Speed vs. Time

Torsional Startup Transient Element Deflection

Torsional Startup Transient Element Vibratory Torque

Torsional Startup Transient Element Vibratory Stress

Torsional Startup Driving and Load Torques

Axial Vibration Analysis

Axial Frequency Interference Diagram

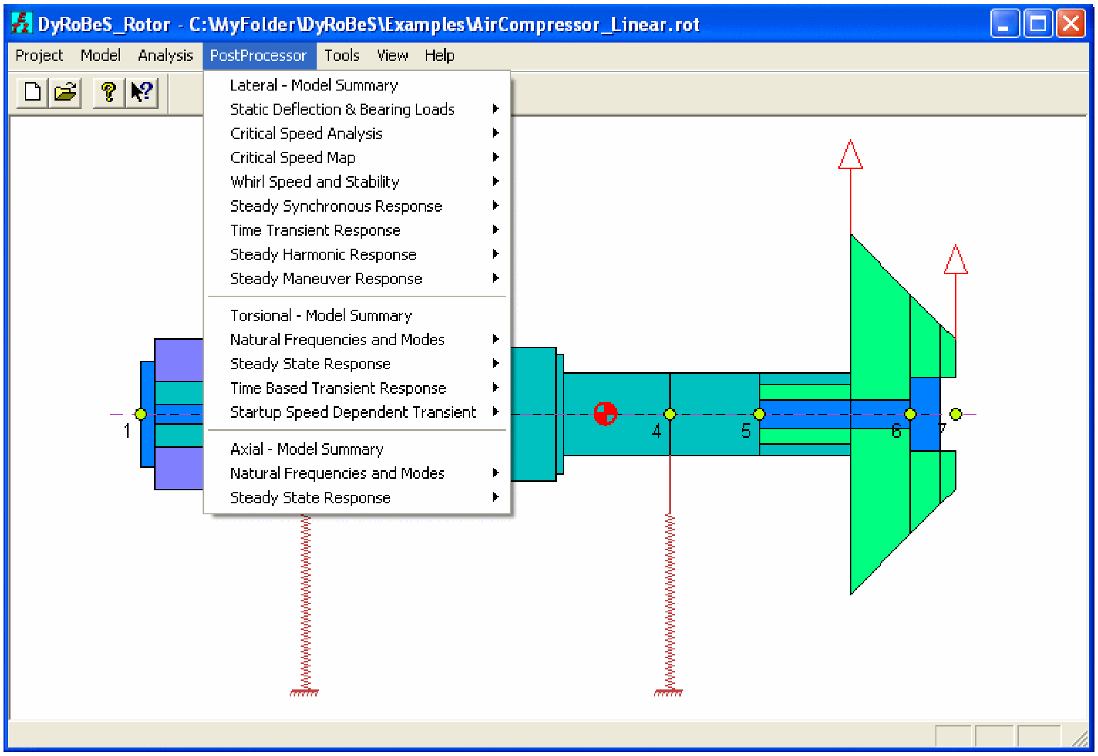
Axial Mode Shapes

Axial Steady State Response

Axial Steady State Element Vibratory Force

Axial Steady State Excitations

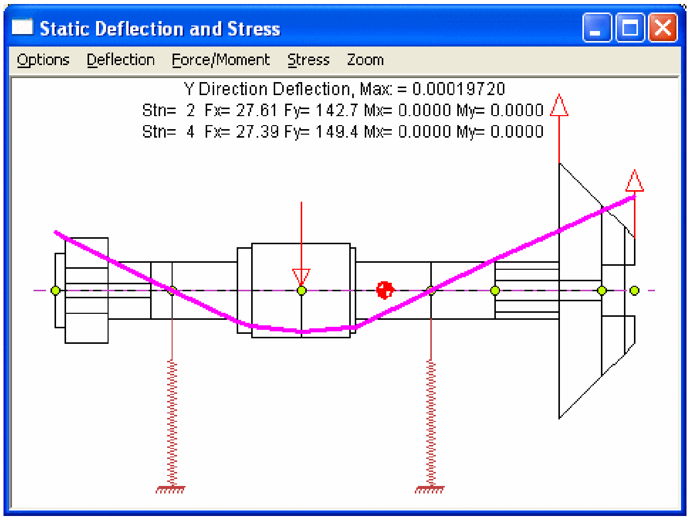
The following figure shows the various options for post processing.

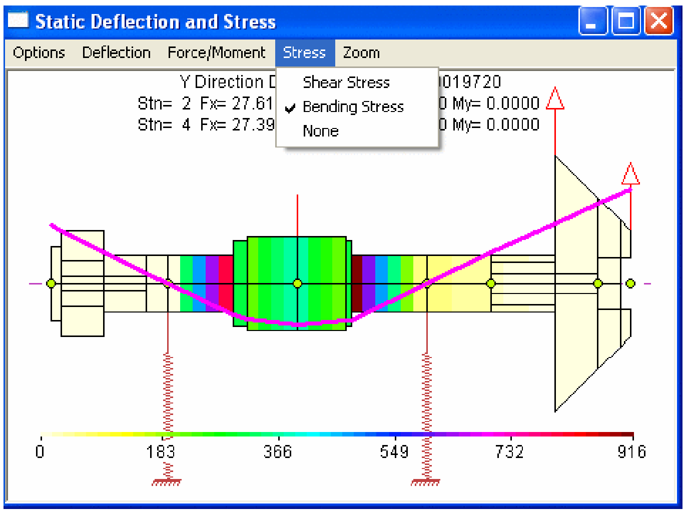


The line Control Data allows you to change the line thickness and color. The plotted curves on a laser printer or ink jet printer will not be as thick as the lines shown on the computer screen. Therefore, if a hardcopy from printers is needed, the line thickness should be increased. The lines may appear too thick on the computer screen, but it will result in a more satisfactory graphical printout. See **Graphic Preferences Settings** under Project.

Static Deflection Curve

This plot displays the shaft static deflection curve. The default deflection is in the Y direction. You can select deflection in the X direction by selecting X direction in the **Deflection** menu. These settings are self-explanatory. The **Scale** is the plotting scale factor (a real positive number) that is used to scale the deflection for better graphic presentation.



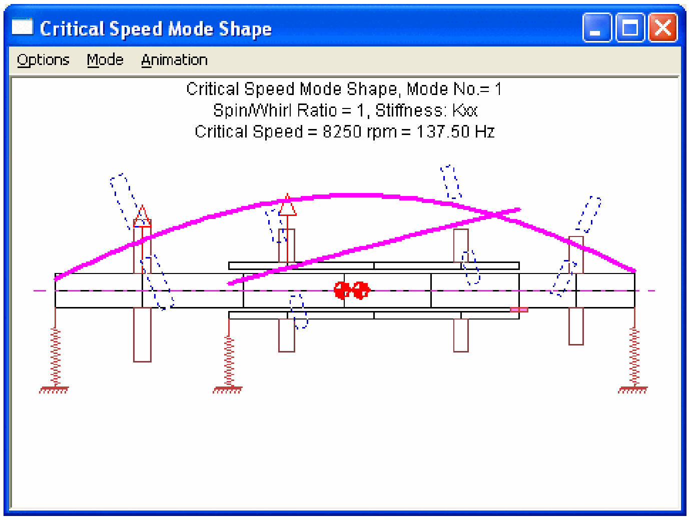


See also [Lateral Vibration Analysis], [Static Deflection and Bearing/Constraint Reactions Analysis].

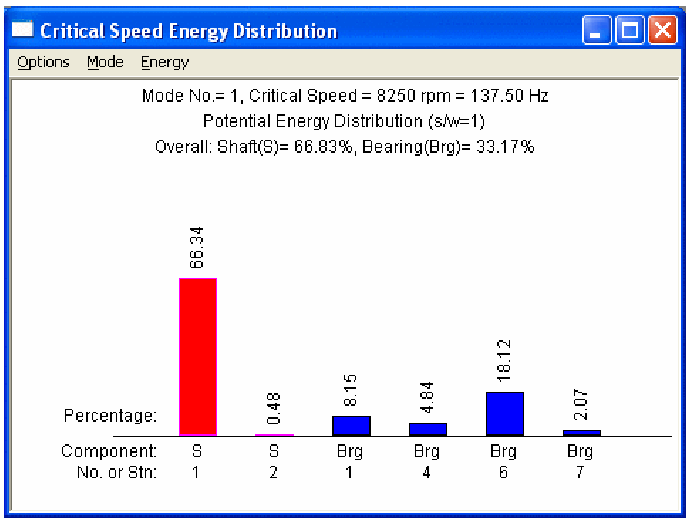
Critical Speed Mode Shape and Energy Distribution

This plot displays a 2-D undamped critical speed mode shape. The graphic **Settings** under **Options** menu are self-explanatory, try them yourself to see their effects. The **Mode** menu allows you to switch the mode number quickly without entering the Setting dialog. **Animation** option allows you to animate the motion.

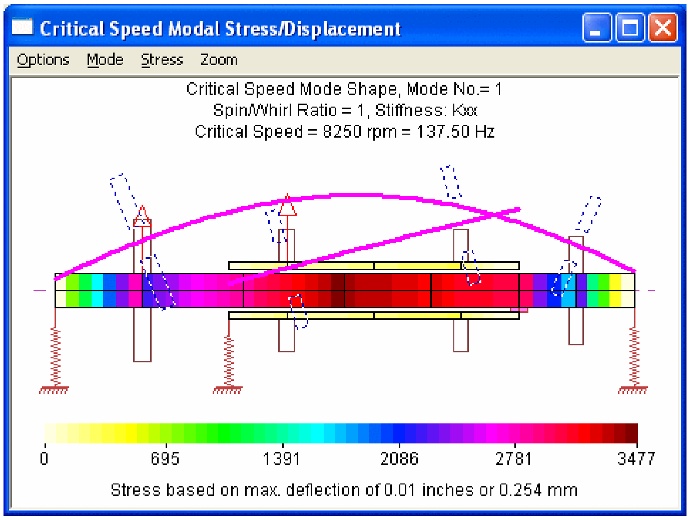
Mode Shape



Energy Distribution



Modal Stress



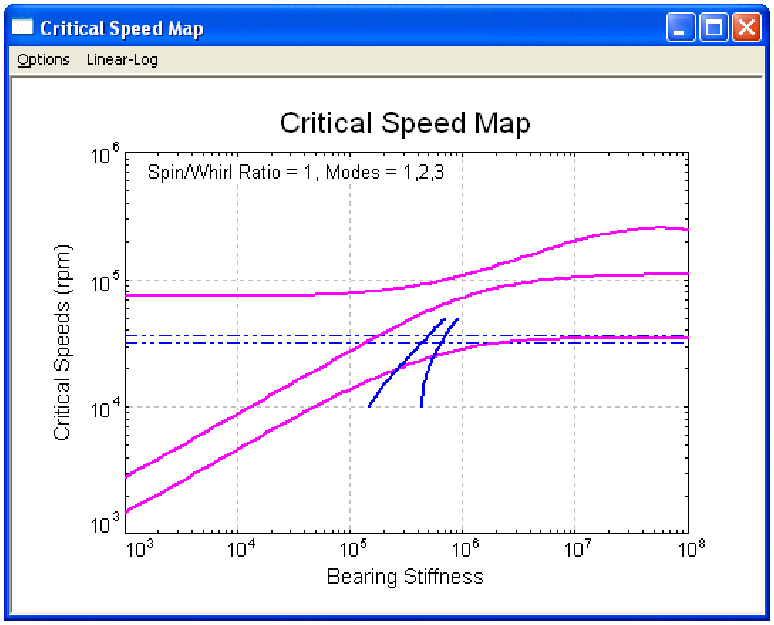
The above figure shows the forward synchronous critical speed mode shape of a two spool dual rotor at 8250 rpm. The bearing stiffnesses Kxx was selected in the run time data. The mode shape is relative displacement and normalized eigenvector which does not have dimension. The kinetic energy is divided into translational and rotational energies. The translational kinetic energy is generated by the mass, and the rotational kinetic energy is generated by the combination of the rotating inertia energy and the gyroscopic energy. Note that the rotational energy is negative for this forward synchronous mode. The negative effect is caused by the gyroscopic energy which causes an increase in the forward synchronous critical speed. For the forward precessional modes, the gyroscopic effect contributes negative kinetic energy and tends to raise the corresponding forward whirl frequency. For the backward precessional modes, the gyroscopic effect contributes positive kinetic energy and tends to lower the corresponding backward whirl frequency. The gyroscopic effect can be significant in the study of high speed overhung rotor system where a large wheel is mounted outside the bearing span.

There is no potential or strain energy in the disks. This is because the disks are assumed as rigid. There is no kinetic energy in the bearings. This is because the bearings are assumed as massless. The first shaft has a strain energy of bending of 66.34% and the second shaft has a strain energy of 0.48%. This indicates that the bending occurs in the first shaft and the second shaft is essentially rigid. This result can also be observed from the stress output.

See also Lateral Vibration Analysis, Critical Speed Analysis.

Critical Speed Map

The Critical Speed Map is normally displayed in a log-log graph of the undamped critical speeds vs. bearing stiffnesses. You can change the graph title, labels, number of modes, and many others by changing the default settings in the **Setting** dialog under **Options** menu. You can overlap your bearing stiffnesses under the Setting dialog to overlap the stiffness curve with the critical speed curve.



The above figure represents the critical speed map of a typical rotor in which the bearing stiffness varies from 1000 to 1.0E08 Lb/in. For stiffness range from 1.0E03 to 1.0E05 Lb/in, the first two modes are essentially straight lines on the log-log scale. These two frequencies in this stiffness range represent rigid body cylindrical and conical modes. The third mode is unaffected by this range of stiffness and represents the first free-free mode.

Note that above a stiffness of 1.0E07 Lb/in the first and second critical speeds do not increase with a further increase in stiffness. The bearings have become node points with zero displacement (rigid bearing). These are called rigid bearing critical speeds. Operation of the rotor in this stiffness range is dangerous since the bearings will provide no damping.

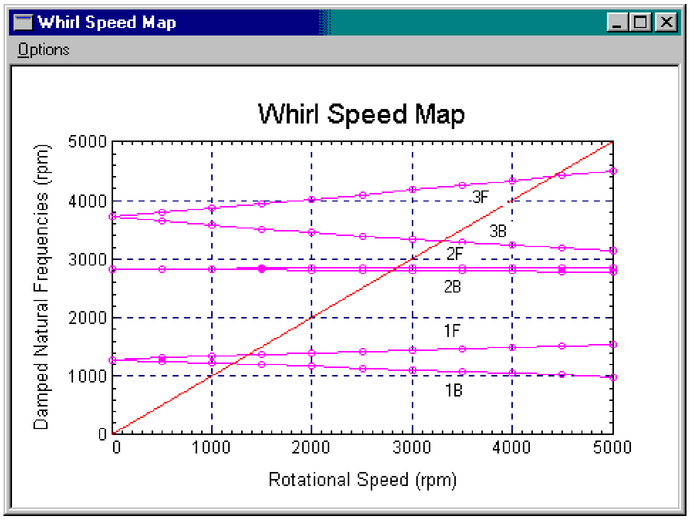
See also Lateral Vibration Analysis, Critical Speed Map Analysis.

Whirl Speed and Stability Results

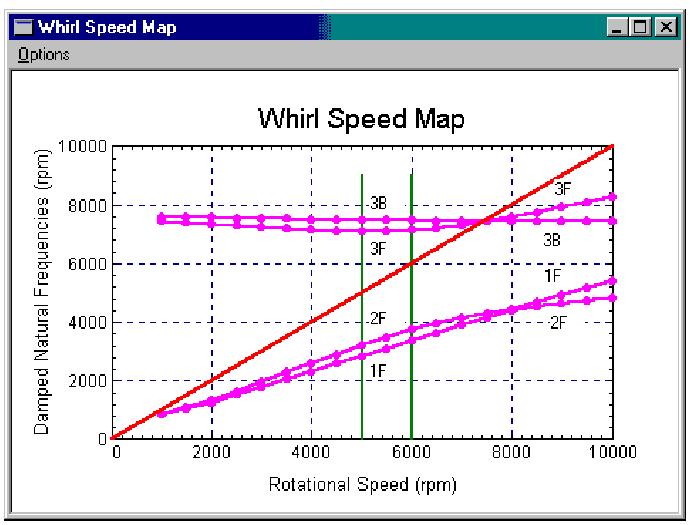
Whirl Speed Map is a linear plot of damped natural frequencies vs. shaft rotational speeds. The damped natural frequency is usually referred to as whirl speed. The **damped critical speeds** and any excitation resonance speeds are determined by noting the coincidence of the shaft speed with the system natural frequencies for a given excitation line in the Whirl Speed Map. A value of 1 in the excitation slope is associated with the synchronous excitation. Up to 5 excitation lines can be displayed in the plot by giving the different excitation slopes. The multiple excitation slopes are separated by commas. For example, the **Excitation Slopes** for the following display is set to be 1.0, 0.5. You can change the graph title, labels, number of modes, and many others by changing the default settings in the **Setting** dialog under **Options** menu.

Due to the non-symmetric properties of the bearing coefficients and the gyroscopic effect, the Whirl Speed Map can be very complicated, caution must be taken when preparing this map.

The following whirl speed map is generated for a typical rotor system with isotropic and constant bearing stiffness. At zero shaft speed, the forward and backward frequencies are identical (repeated eigenvalues). As the speed increases, each vibration mode is split into two modes known as forward and backward precessional modes due to the gyroscopic effect. For the forward precessional modes, the gyroscopic effect contributes negative kinetic energy and tends to raise the corresponding forward whirl frequency. For the backward precessional modes, the gyroscopic effect contributes positive kinetic energy and tends to lower the corresponding backward whirl frequency. Thus, it is the forward modes getting the gyroscopic stiffening effect and the backward modes getting the gyroscopic softening effect. The intersection between the synchronous excitation line and the damped natural frequencies are referred as damped critical speeds.



The following whirl speed map is generated for a rotor system supported by fluid film bearings. At the rotor speed of 5000 rpm, the first mode is a forward conical mode and the second mode is a forward translational mode. The first two backward modes are overdamped (real modes) in this example and are not shown in this map. The third mode is a forward bending mode and the fourth mode is a backward bending mode. However, at the rotor speed of 10000 rpm, the first mode becomes a forward translational mode and the second mode becomes conical mode. The third mode now is a backward bending mode and the fourth mode is a forward bending mode. The direction of precession (forward or backward) and the type of modes (rigid body, bending) should be determined from the mode shapes not from the whirl speed map.

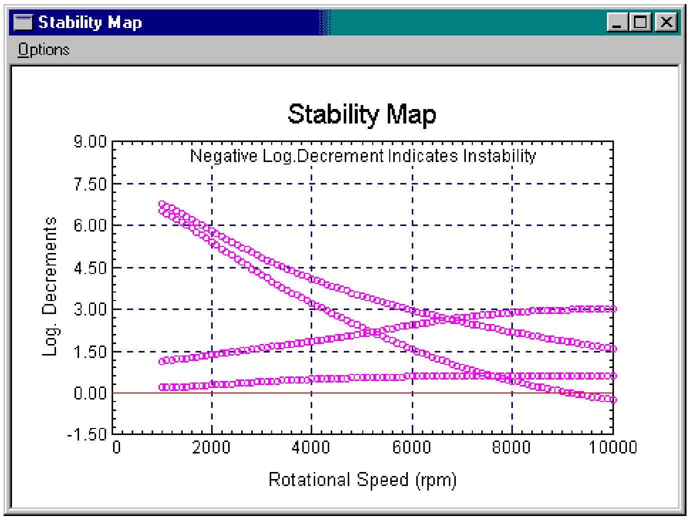


Stability Map

Stability Map is a linear plot of logarithmic decrements (or damping factors) vs. shaft rotational speeds. Negative logarithmic decrement indicates system instability. When the value of logarithmic decrement exceeds 1, that particular mode is well damped. As shown in the figure, the logarithmic decrement of the first mode approaches to zero as the rotor speed approaches to 9100 rpm, and then becomes negative as the rotor speed increases. This speed is known as the **instability threshold**. Self-excited instability occurs at the speeds above the instability threshold.

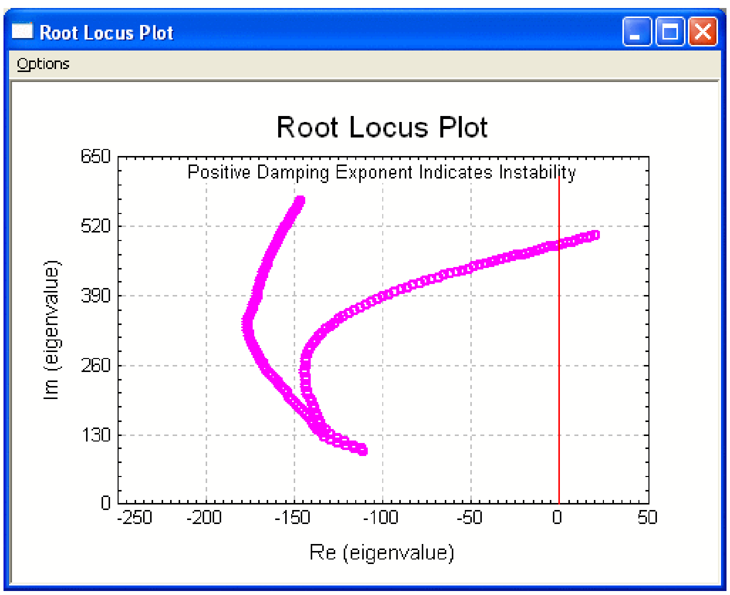
You can change the graph title, labels, number of modes, and many others by changing the default settings in the **Setting** dialog under **Options** menu.

Due to the non-symmetric properties of the bearing coefficients and the gyroscopic effect, the Stability Map can also be very complicated, caution must be taken when preparing this map.



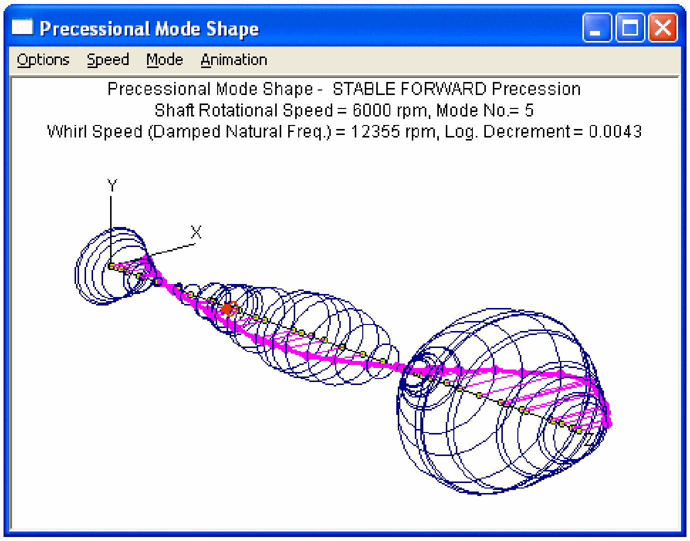
Root Locus Plot

Root locus plot is also commonly used in the system stability evaluation, particularly by control engineers. It is a plot with imaginary part of eigenvalue vs. the real part of eigenvalue for a range of rotor speed as shown below. In DyRoBeS, the imaginary part can also be the cycle per minute or Hz and the real part can be the logarithmic decrement or damping factor.



3D Precessional Mode Shapes

This plot displays a three-dimensional precessional mode shape. The 3-D mode shapes can be displayed in different views by adjusting the **Display Projection** option in the Settings. You can quickly switch the speed and mode number by using the **Speed** and **Mode** options. You can change many other features by changing the default settings in the **Setting** dialog under **Options** menu. **Animation** option allows you to animate the motion and it is easy to see the forward, backward, or mixed precession.

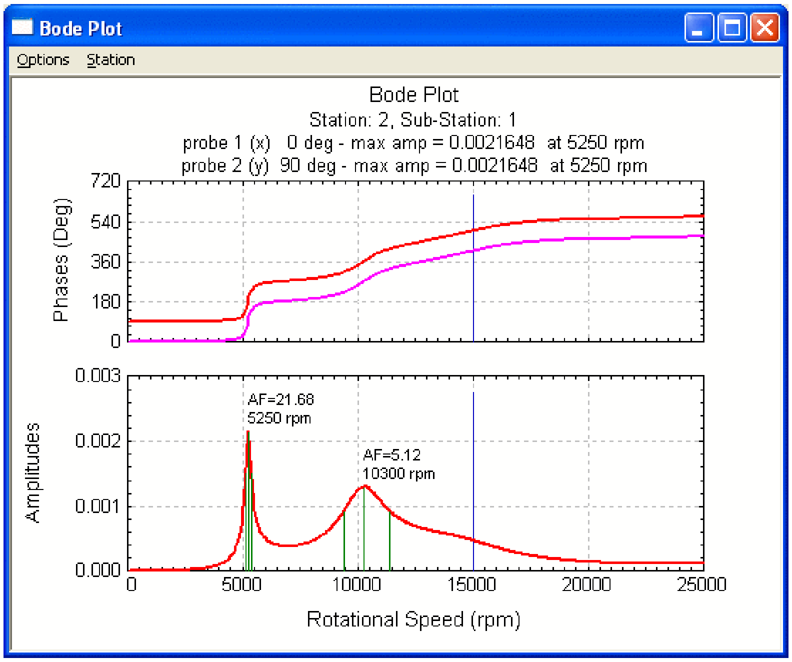


See also Lateral Vibration Analysis, Whirl Speed and Stability Analysis.

Steady State Synchronous Response Results

Bode Plot

A Bode plot is a linear plot of amplitude and phase vs. shaft rotational speeds. You can change the graph title, labels, scale, number of divisions, probes angles and many others by changing the default settings in the **Setting** dialog under **Options** menu. Relative displacement can be specified in the Setting.

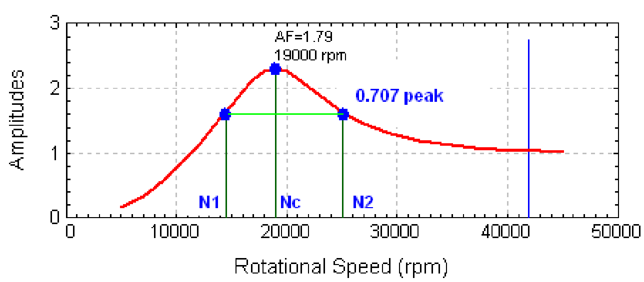


The Bode plot displays the response amplitude and phase at a specified finite element station due to the synchronous excitation (unbalance, shaft bow, disk skew). In the above figure, the amplitude and phase are shown for both the x and y directions. The amplitudes of motion are slightly different due to the effects of asymmetric bearing properties. Thus, the observed critical speeds and vibration amplitude may be different as observed from the x or y probes.

Amplification Factor

The program can allow you to label the Amplification Factors. The Amplification Factor (AF) is defined based on API Standards, such as API 611 for steam turbines or API 672 for geared compressors. The Amplification factor (AF) is defined as:

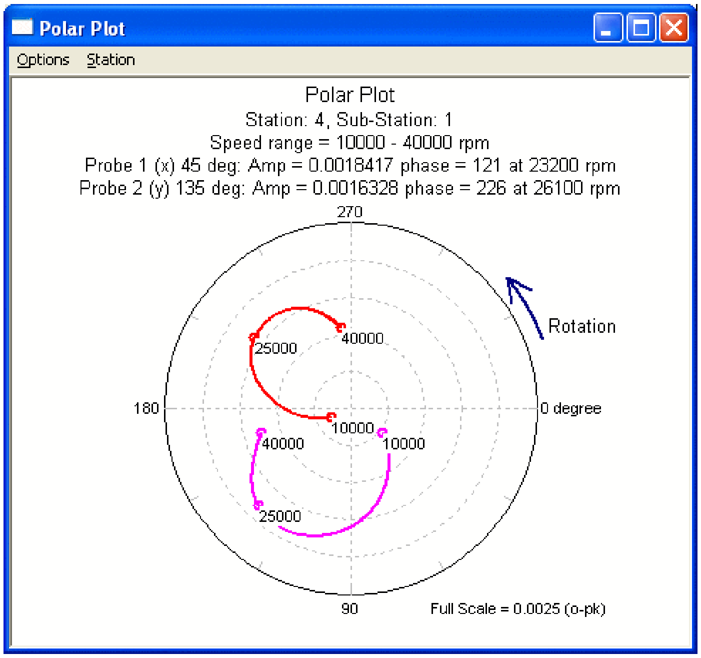
$$AF = \frac{N_c}{N_2 - N_1}$$



As stated in API Standards, When the rotor amplification factor, as measured at the vibration probe, is greater than or equal to 2.5, that frequency is called critical, and the corresponding shaft rotational frequency is called a ***critical speed***. A critically damped system is one in which the amplification factor is less than 2.5.

Polar Plot

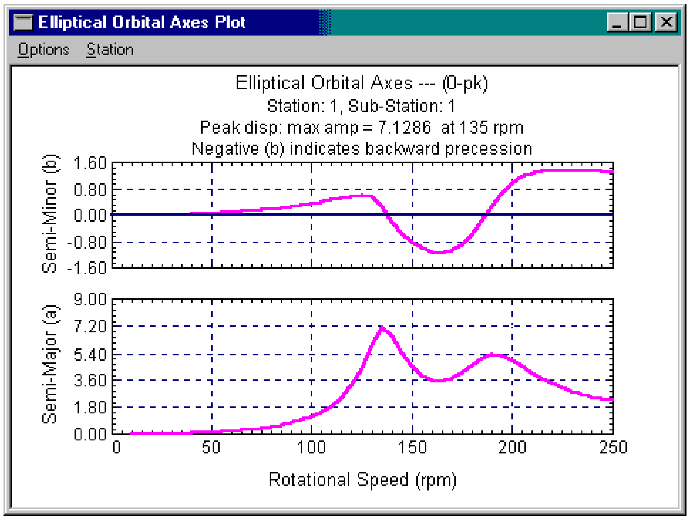
Polar Plot shows amplitude and phase in the polar coordinates over a speed range for a given finite element station. The shaft rotational direction is counterclockwise. You can change the graph title, labels, scale, number of divisions, probes angles and many others by changing the default settings in the **Setting** dialog under **Options** menu.



Elliptical Orbit Axes Plot

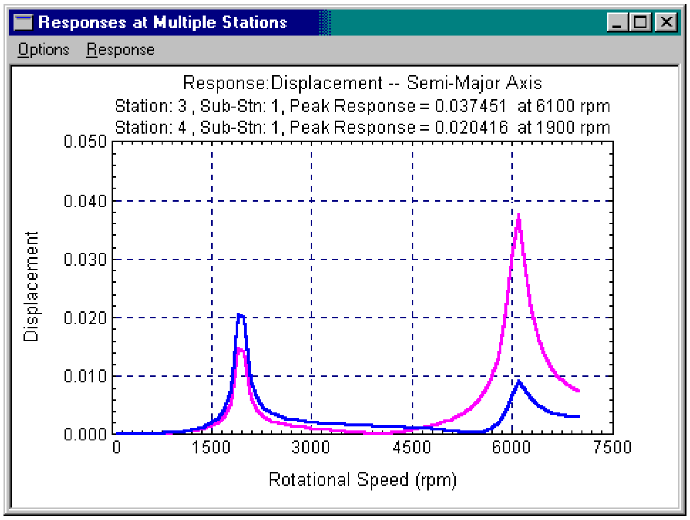
The Elliptical Orbit Axes Plot is a linear plot of semi-major axis amplitudes and semi-minor axis amplitudes vs. shaft rotational speeds. Positive semi-minor axis indicates the **Forward Precession** orbit. Negative semi-minor axis indicates the **Backward Precession** orbit.

The following figure is for a rotor system with asymmetric bearings. It shows that there are two distinct critical speeds split by bearing asymmetry in this example. Since the amplitudes of semi-major and semi-minor axes are different, the orbits are elliptical. When the rotor speed is either below the first critical speed or above the second critical speed, the rotor is in a state of forward synchronous whirl. There are backward precessions when the rotor is rotating with a speed between two critical speeds. Again, the presence of dampings and bearing cross-coupling stiffness, the backward precession may not exist.



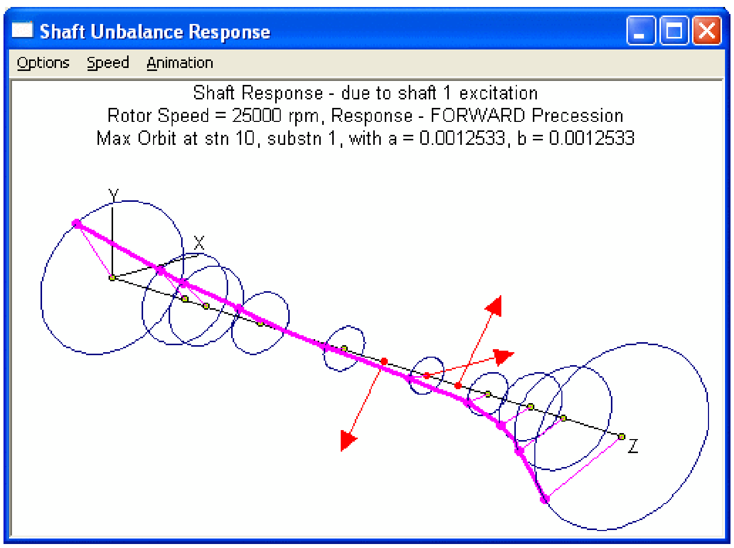
Responses at Multiple Stations

This option allows you to display the steady state synchronous response at multiple stations (up to 5 stations) simultaneously in order to compare the response along the rotor. You have the choice of plotting the **displacement**, **velocity**, or **acceleration** for the semi-major axis, the x motion, or the y motion. The response at multiple stations along the shaft is important from the standpoint of the positioning of monitoring probes or non-contact probes along the rotor to detect critical speeds. For example, a probe at the center of a rotor may observe the rotor first critical speed, but not the second; whereas a probe placed near the rotor end at a bearing may not see much amplitude at the first critical speed, but would detect the monitor at the second critical speed. You can change the graph title, labels, scale, number of divisions and many others by changing the default settings in the **Setting** dialog under **Options** menu.



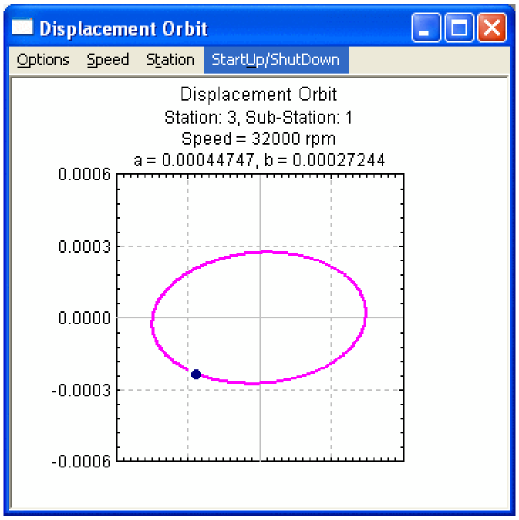
Shaft Response 3D Animation

The option displays the steady state shaft response at a specified speed and/or startup/shutdown animation. This plot would be equivalent to taking an instantaneous strobe flash of the rotor and freezing it in time. At each station, the dots will move around the orbit with synchronous motion and will maintain their relative phase of motion with respect to each other. This orbital motion follows an elliptical path at each station. For symmetric bearing coefficients (isotropic system), the motion at each station reduces to circular. It is also apparent from the observation of the timing marks at each station that the response shape frozen in time is in the same plane. Note that a vibration probe placed at the station 3 would show little vibration regardless of the magnitude of unbalance. **Animation** option allows you to animate the motion at a single speed or during startup/shutdown.



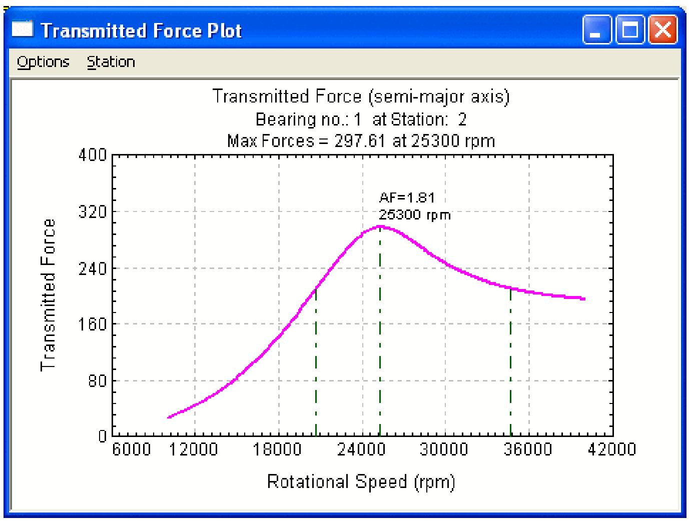
Displacement Orbit at Specified Speed and Station

The Displacement Orbit Plot displays the response displacement at a specified speed and station. (a) is semi-major axis, (b) is semi-minor axis. You can change the graph title, labels, scale, number of divisions and many others by changing the default settings in the **Setting** dialog under **Options** menu. **Animation** option allows you to animate the motion during startup/shutdown.



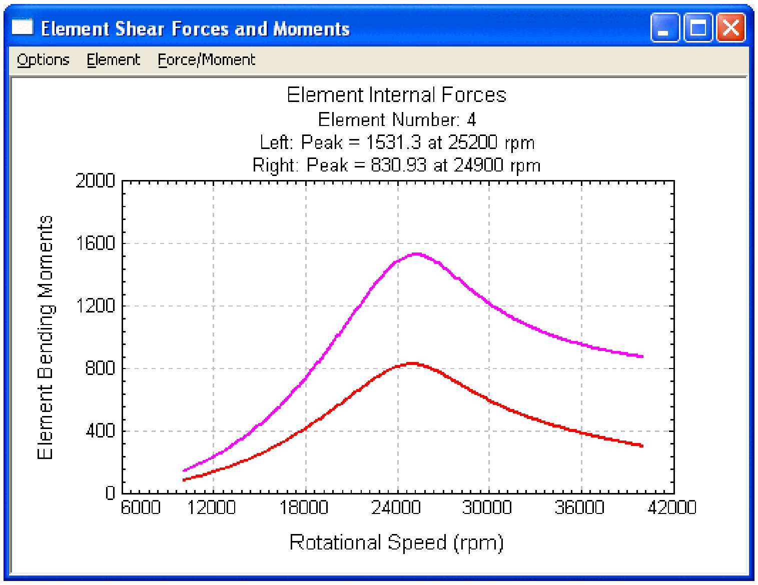
Bearing/Support Transmitted Forces

This plot displays the support transmitted forces vs. shaft rotational speeds. The transmitted bearing force can be an important design parameter. It should be noted that operation away from the critical speed can still result in large bearing forces transmitted.



Element Shear Forces and Moments

The option displays the element internal shear forces and moments due to steady state excitations. Shear forces are the default setting. You can display the moments by selecting **Moments** in the **Force/Moment** menu. You can also change the graph title, labels, scale, number of divisions and many others by changing the default settings in the **Setting** dialog under **Options** menu.

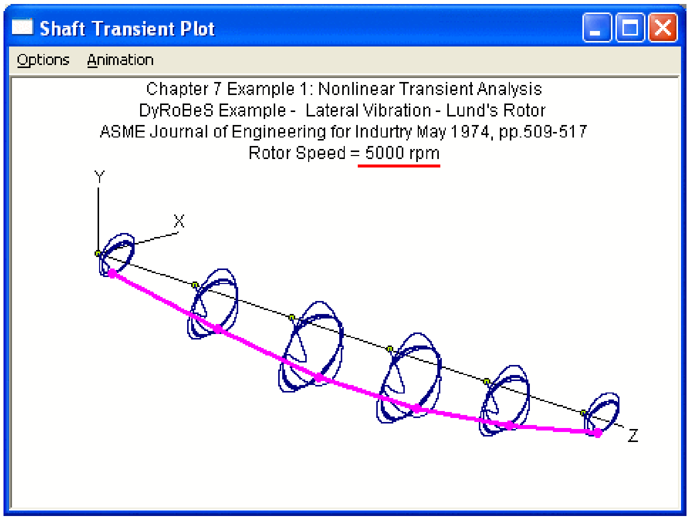


See also Lateral Vibration Analysis, Steady State Synchronous Response Analysis – Linear System, Steady State Synchronous Response Analysis – NonLinear System .

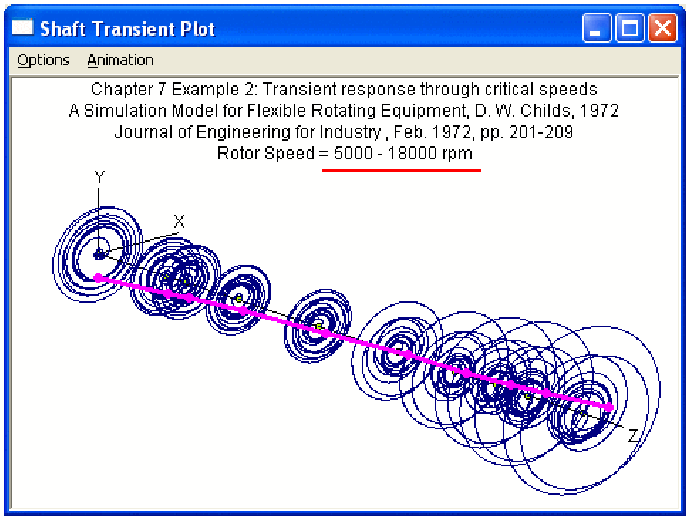
Time Transient Shaft Response

For the time transient analysis, it can be performed at a constant speed or at various speed during startup or shutdown. Again, **Animation** feature plays an important role in visualization of the rotor motion.

The shaft response at a constant speed is shown below:



The shaft response during startup is shown below:

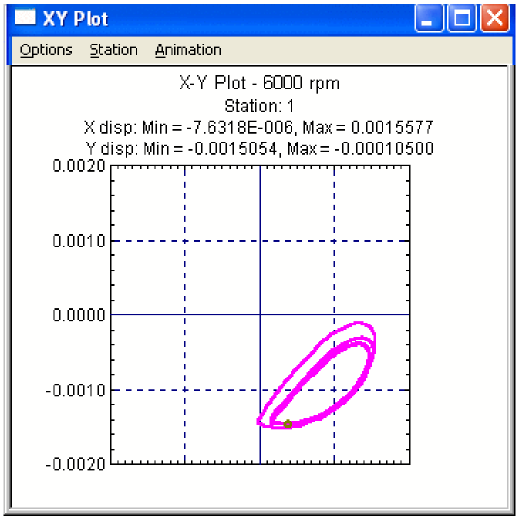


Time Transient X-Y Plot

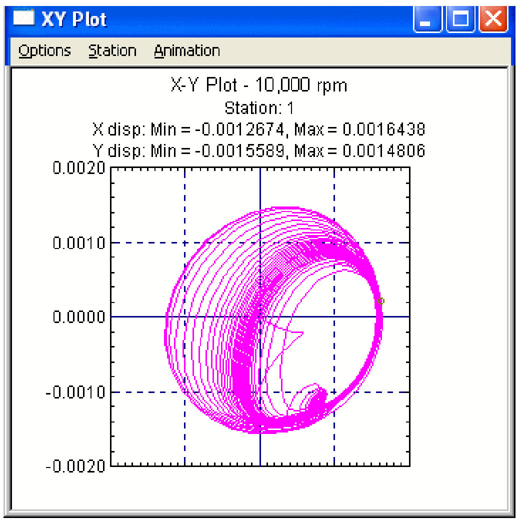
This plot displays the time transient orbit displacement. This option is useful in the investigation of the initial response of a rotor to a sudden excitation and also for the investigation of nonlinear effects such as caused by nonlinear bearings.

The following figure is a X-Y displacement plot for a fluid film bearing with unbalance and gravity effects. At 6000 rpm, the rotor is stable with a nearly elliptical orbit. At 10,000 rpm, the rotor just passes the instability threshold and the orbit is growing with time. The motion will eventually approach the **limit cycle motion**.

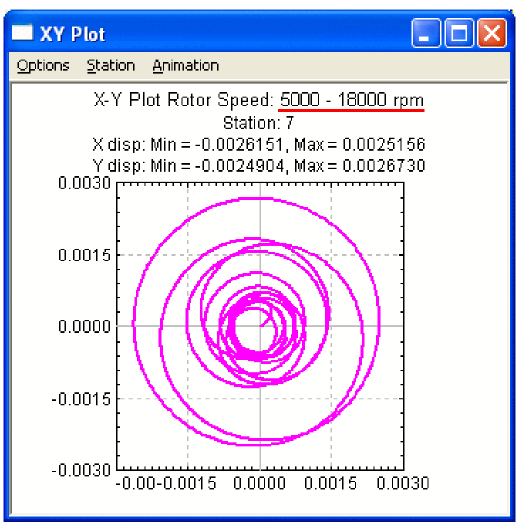
At 6000 rpm



At 10000 rpm



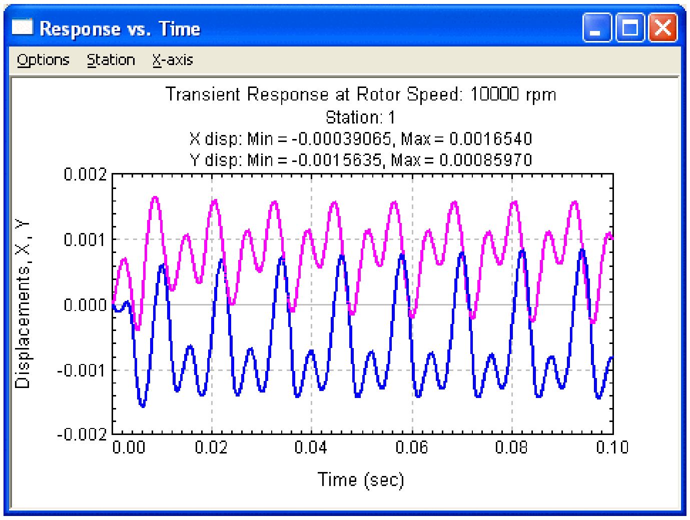
Following picture shows the rotor motion from 5000 rpm to 18000 rpm.



Transient Response vs. Time (Speed)

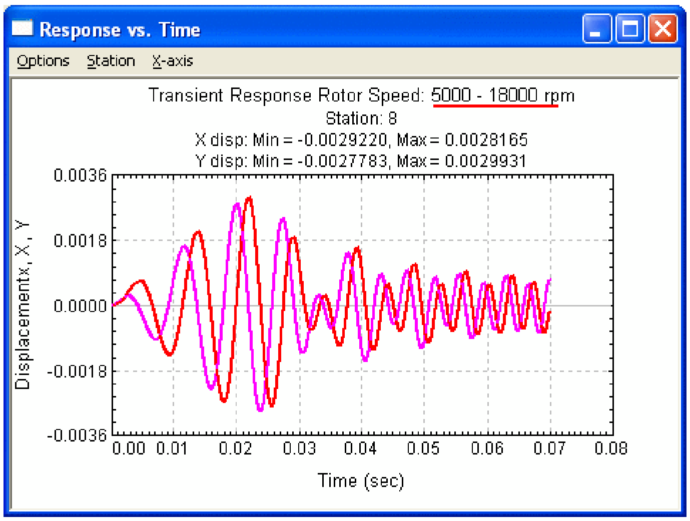
This plot displays the transient response vs. time. In the setting, you can display X displacement, Y displacement, and/or radial displacement. You can change the graph title, labels, scale, number of divisions and many others by changing the default settings in the **Setting** dialog under **Options** menu. For startup/shutdown, the X-axis can be either **time** or **speed**. However, at constant speed transient analysis, the X-axis can only be the time.

Following is a sample plot at 10,000 rpm where the sub-synchronous vibration component is higher than the synchronous vibration component.

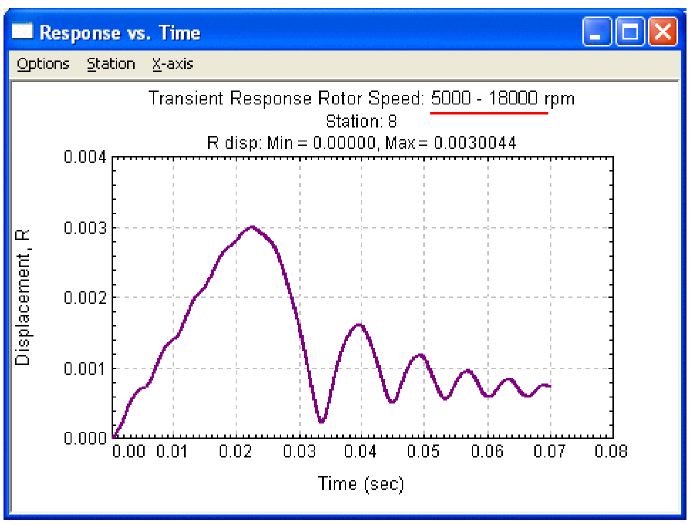


Followings are results for startup/shutdown transient analysis.

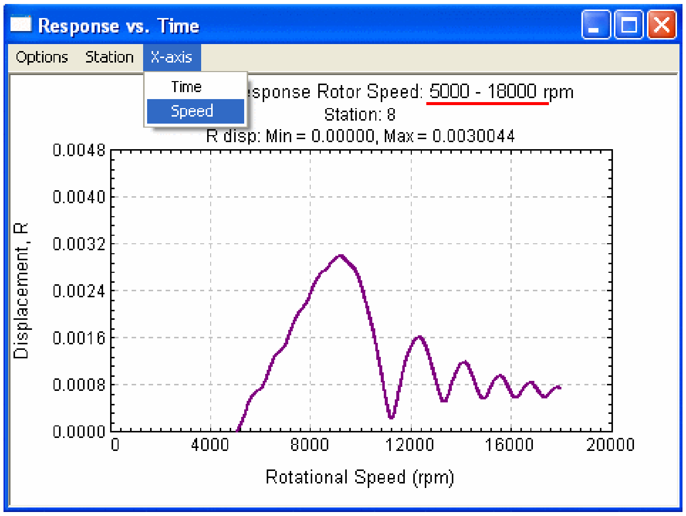
X, Y displacements vs. time



R displacement vs. time



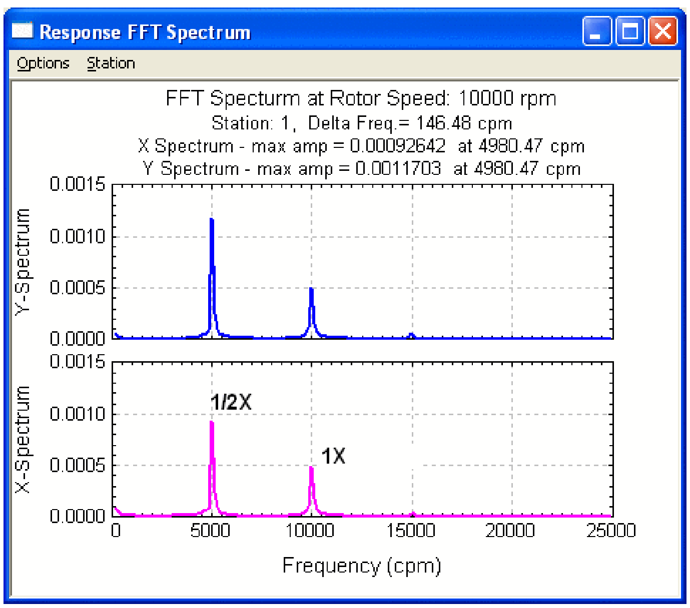
R displacement vs. speed



FFT Spectrum

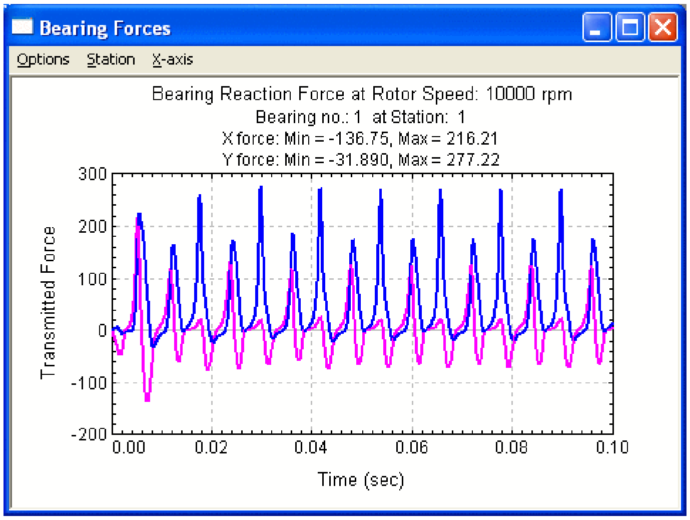
This plot displays the transient response in spectrum format. You can change the graph title, labels, scale, number of divisions and many others by changing the default settings in the **Setting** dialog under **Options** menu. In the default setting, the zero harmonic component (DC level) is not shown in the plot. To include the zero harmonic component, simply check the box in front of the DC Level. This option is available only at constant speed transient analysis.

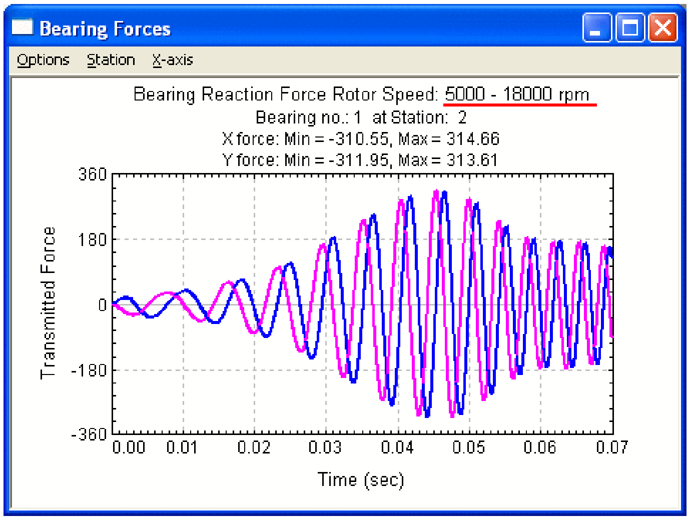
The following figure shows a rotor with a subsynchronous vibration component. The rotor is operated at 10,000 rpm, the subsynchronous vibration (4980 rpm) is larger than the synchronous vibration. This indicates that the rotor is operated at the unstable condition.

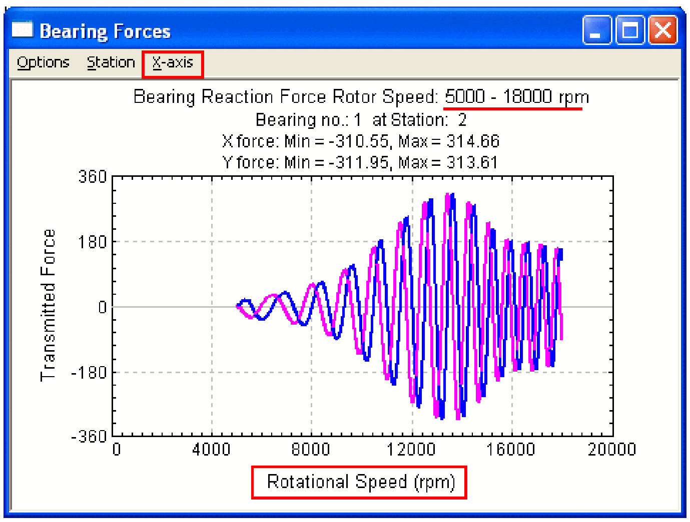


Transient Bearing Reaction Forces

This plot displays the transient bearing reaction forces in X direction, Y direction and/or radial amplitude vs. time (or speed if startup/shutdown analysis is performed). You can change the graph title, labels, scale, number of divisions and many others by changing the default settings in the **Setting** dialog under **Options** menu.

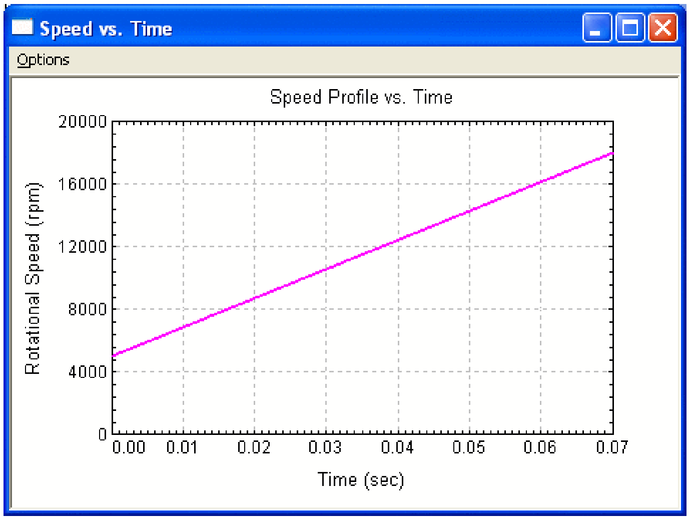






Transient Speed vs. Time

This plot displays the speed vs. time for startup/shutdown. This is used for verification purpose to double-check the speed input.

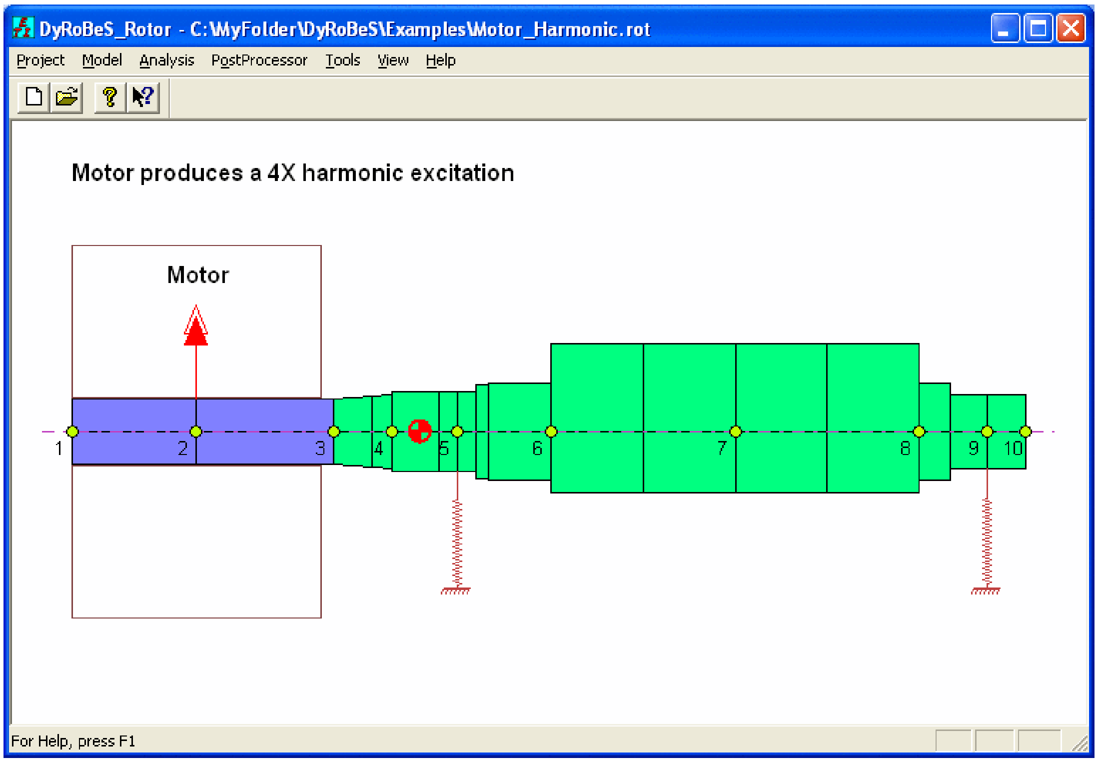


See also Lateral Vibration Analysis, Time Transient Analysis.

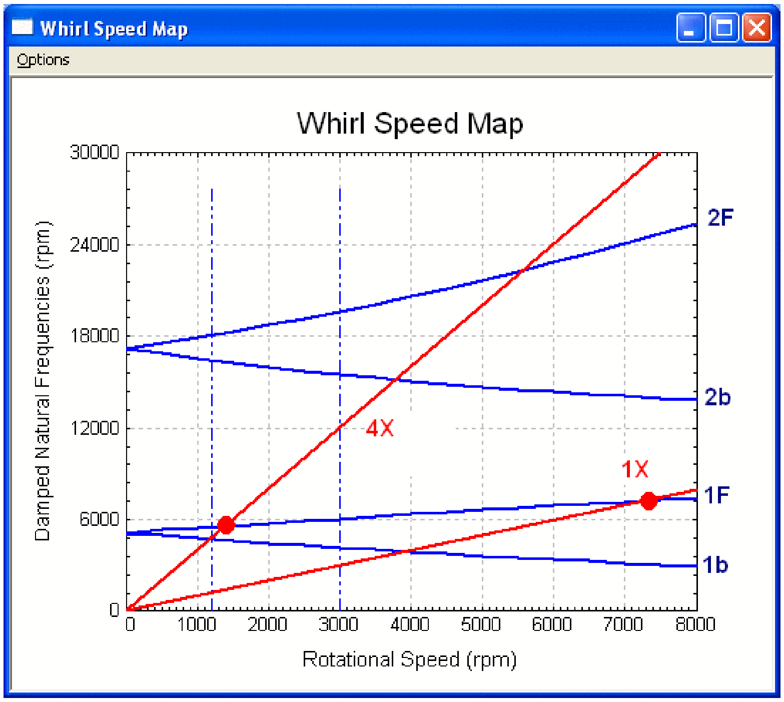
Steady Harmonic Response

This option is for the steady state harmonic response. A compressor is directly coupled with a motor as shown below. The operating speed range is between 1200 rpm to 3000 rpm, depending on the system requirement. The rotor was designed such that the first synchronous critical speed is about 7200 rpm, which is much higher than the operating speed range. No conventional critical speed (due to mass unbalance) was anticipated during the development stage, ball bearings were used. However, the motor was directly mounted into the rotor and due to the wiring design and mounting eccentricity between rotor and stator, the motor produces a 4X excitation. That means the excitation frequency is 4X of the rotor speed. From the whirl speed map, it shows that the first forward synchronous critical speed is around 7200 rpm, however, the first forward 4X critical speed is around 1385 rpm. That means when the rotor is around 1385 rpm, the 4X excitation from the motor excites the first forward natural frequency. These can also be found using the critical speed analysis with spin/whirl ratio of 1 and ¼. The response shows the peak response occurs around 1385 rpm, which falls into the operating speed range. The problem was resolved by re-design the motor with different wiring method. The X-axis can be plotted either in rotor speed or excitation frequency.

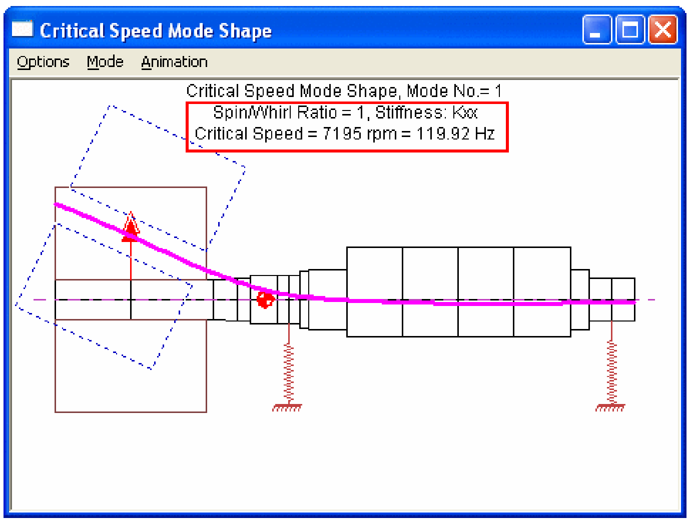
Rotor system



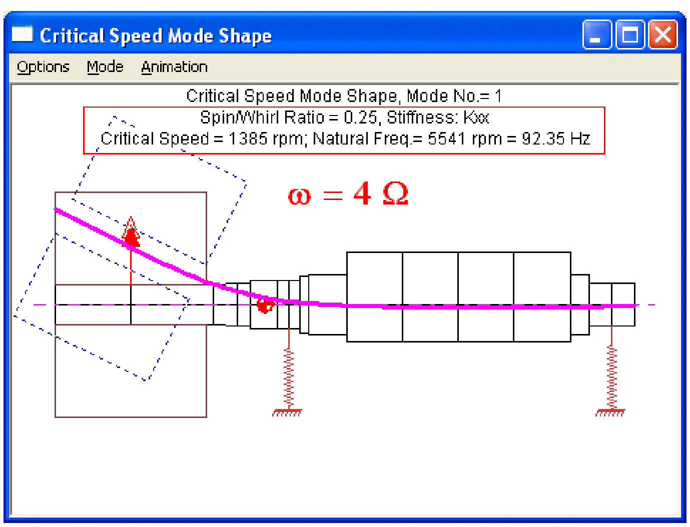
Whirl Speed Map



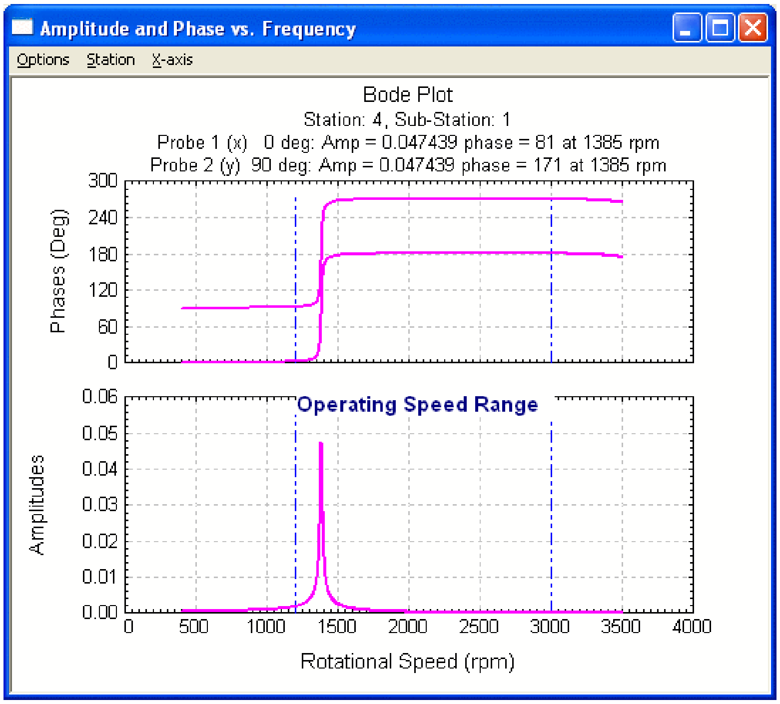
First undamped forward synchronous critical speed, spin/whirl ratio = 1



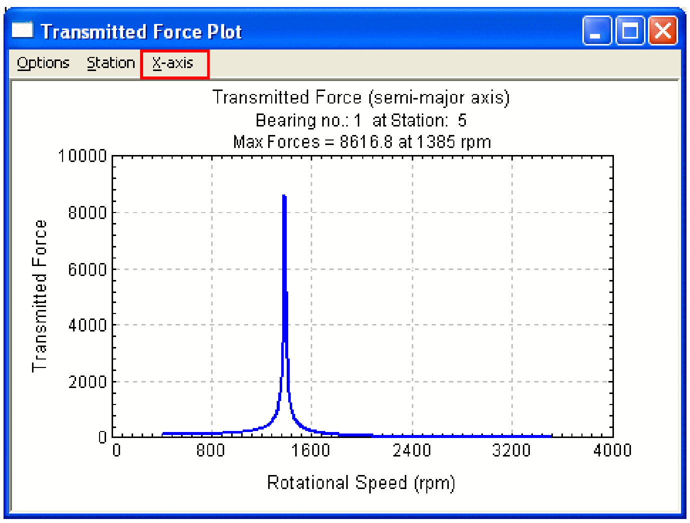
First undamped forward 4Xsynchronous critical speed, spin/whirl ratio = 1/4



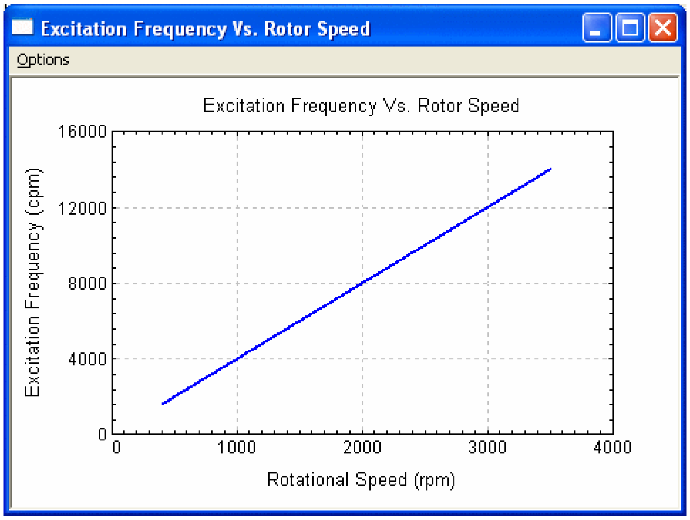
Response amplitude



Bearing force



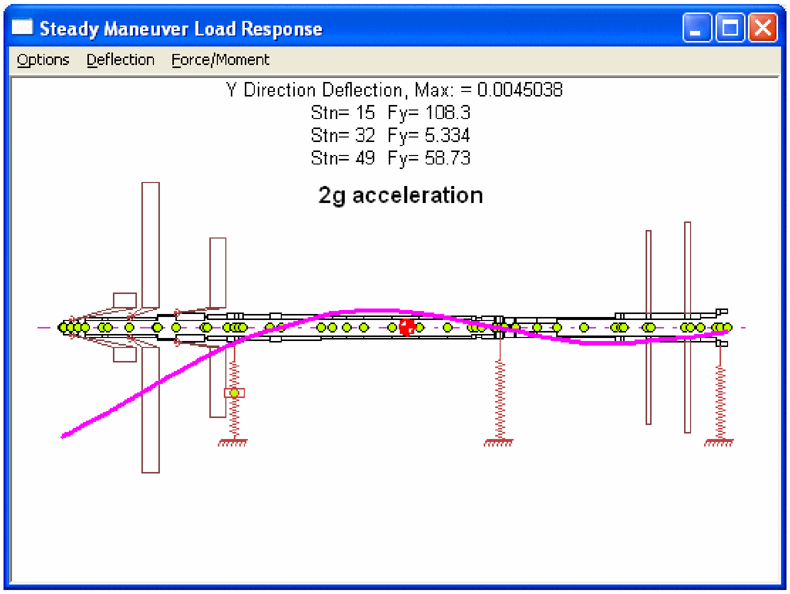
Excitation frequency vs. rotor speed



See also Lateral Vibration Analysis, Steady State Harmonic Excitation Response Analysis.

Steady Maneuver Load Response

This option displays the rotor deflection and bearing loads due to constant acceleration and turn rate.

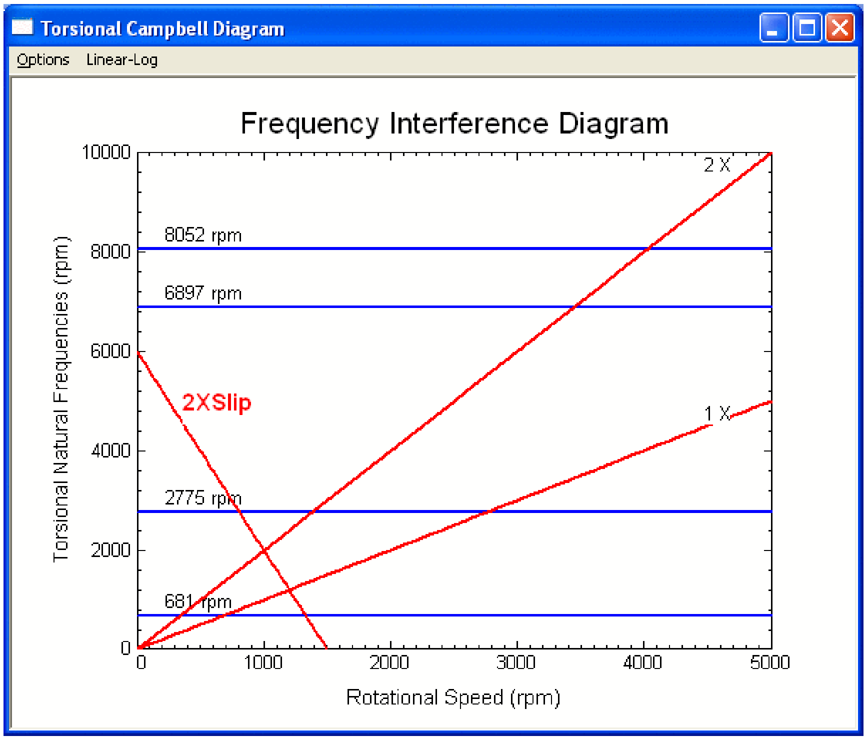


See also Lateral Vibration Analysis, Steady Maneuver Load Analysis.

Torsional/Axial Frequency Analysis Results

Interference Diagram (Campbell Diagram)

This plot displays the Frequency Interference Diagram. It is commonly called Campbell diagram. Up to 5 excitation lines can be displayed and each excitation slope is separated by a comma in the Excitation Slopes data entry. For example, the Excitation Slopes for the following display is set to be 1, 2. Since this is a synchronous motor driven train, twice the slip frequency is also shown in the figure. You can change the graph title, labels, scale, number of divisions and many others by changing the default settings in the **Setting** dialog under **Options** menu.

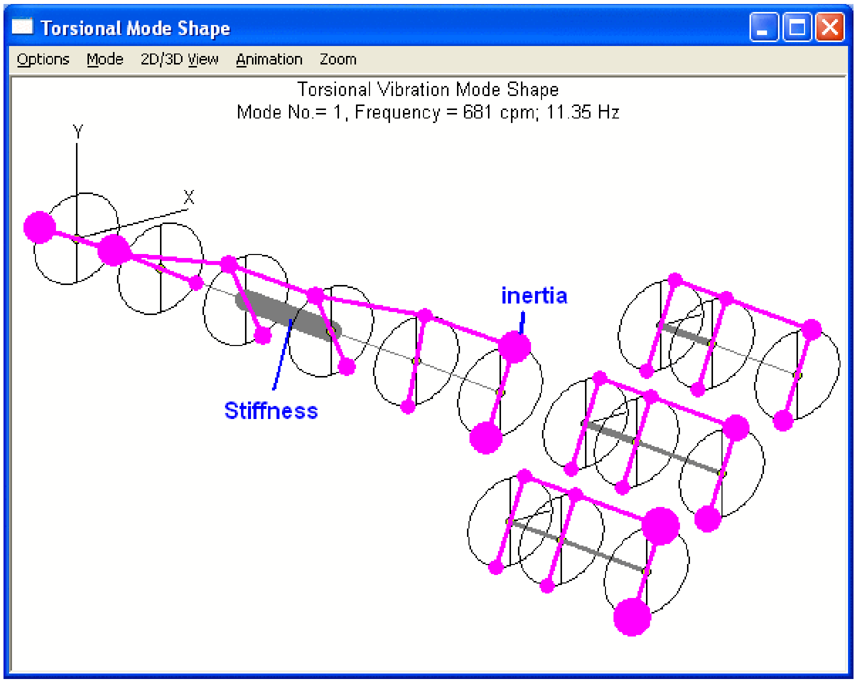


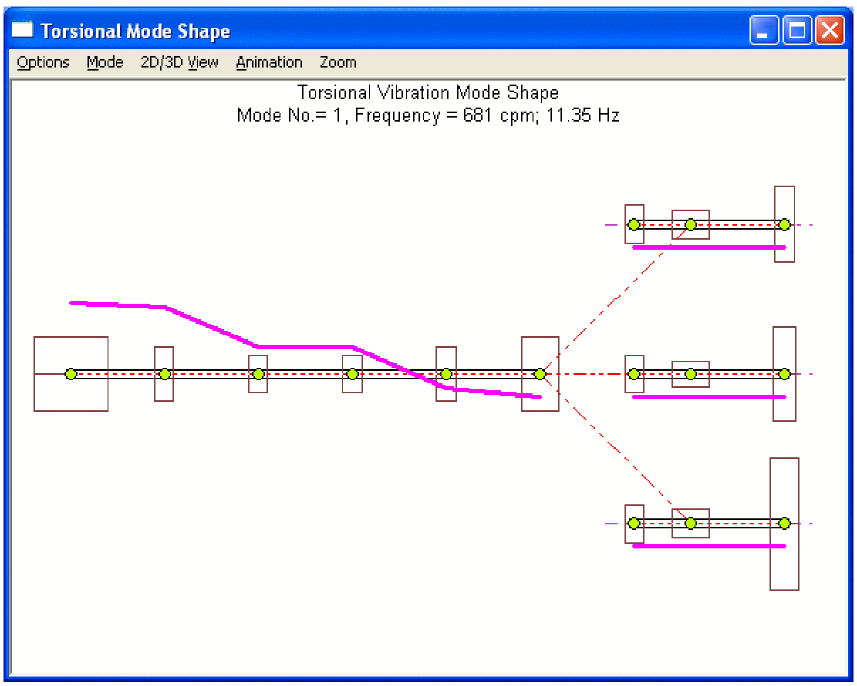
Note: for torsional analysis, the outputs can be referenced to the actual or equivalent system. For details on this conversion, click Torsional Vibration Analysis.

Torsional/Axial Mode Shapes

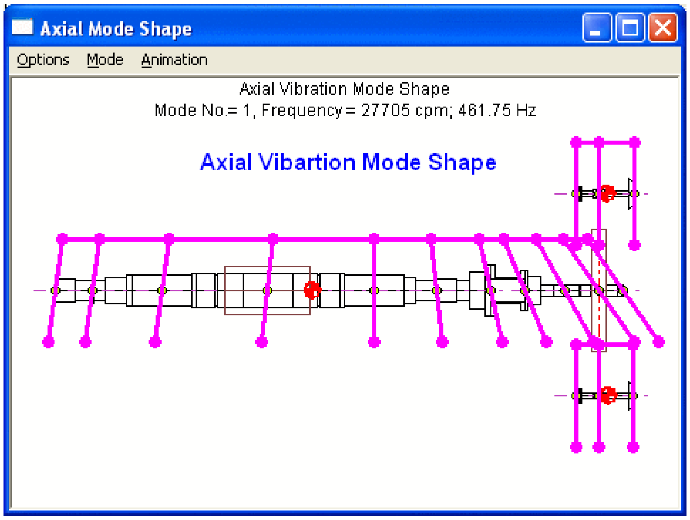
This option displays the torsional or axial vibration mode shape. You can change the graph title, labels, scale, and many others by changing the default settings in the **Setting** dialog under **Options** menu. **Animation** option allows you to animate the motion.

The mode shape can be displayed in a 2D or 3D format. For a torsional 3D mode shape plot, the relative size of the inertia and stiffness are also shown in the plot



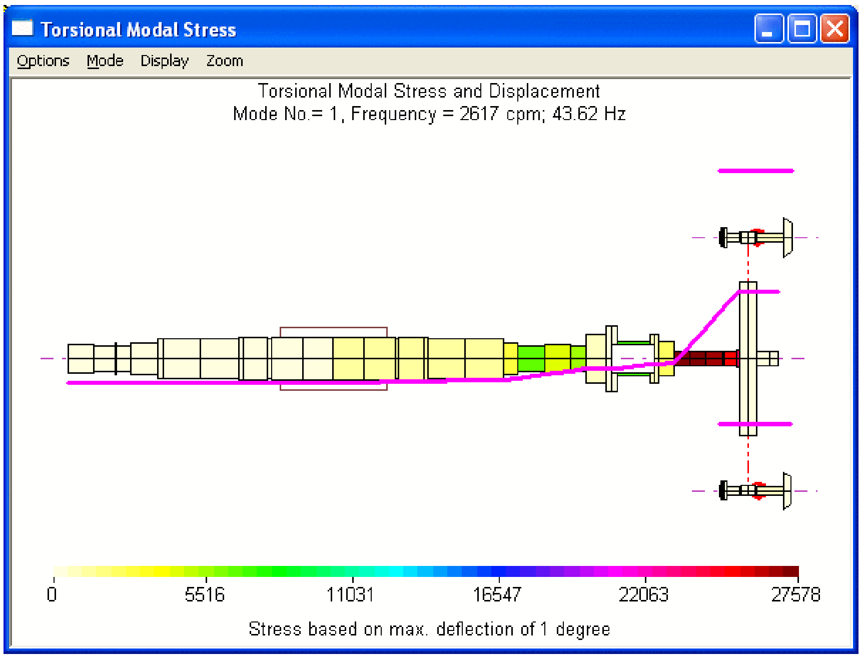


Axial vibration mode shape



Torsional Modal Stress

Modal stress based on 1 degree of maximum deflection is shown below:



See also Torsional Vibration Analysis, Torsional Damped and Undamped Natural Frequencies and Modes Calculation, Axial Damped and Undamped Natural Frequencies and Modes Calculation.

Torsional/Axial Steady State Response

This option displays the torsional or axial steady state forced response and associated vibratory torque (force) and stress. For the torsional analysis, the results can be displayed in the equivalent coordinate or physical coordinate. For torsional vibration, the mass, stiffness, damping, and torque are converted into the equivalent mass, stiffness, damping, and torque referred to shaft 1 for analysis. However, the results can be displayed in either coordinate system. For details on this conversion, click Torsional Vibration Analysis. You can change the graph title, labels, scale, number of divisions and many others by changing the default settings in the **Setting** dialog under **Options** menu.

Figure shows the **Displacements** at stations 9 and 10

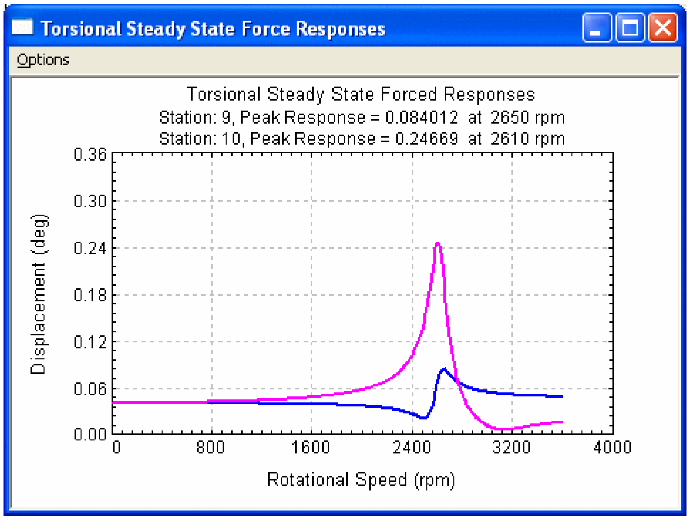
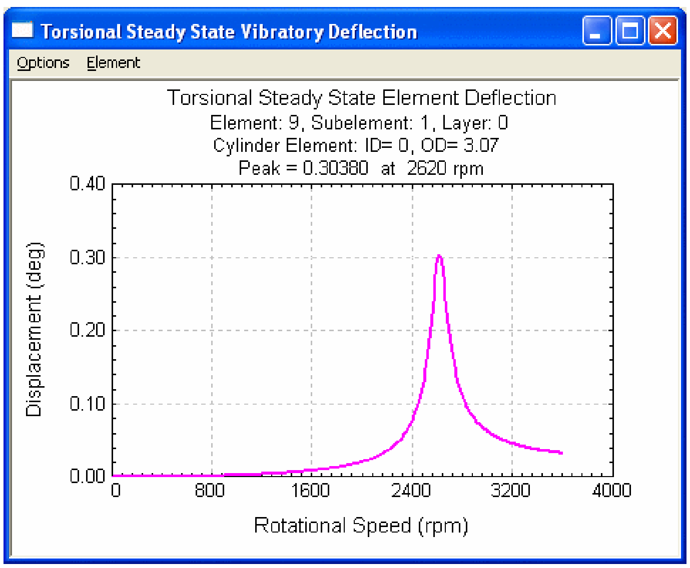


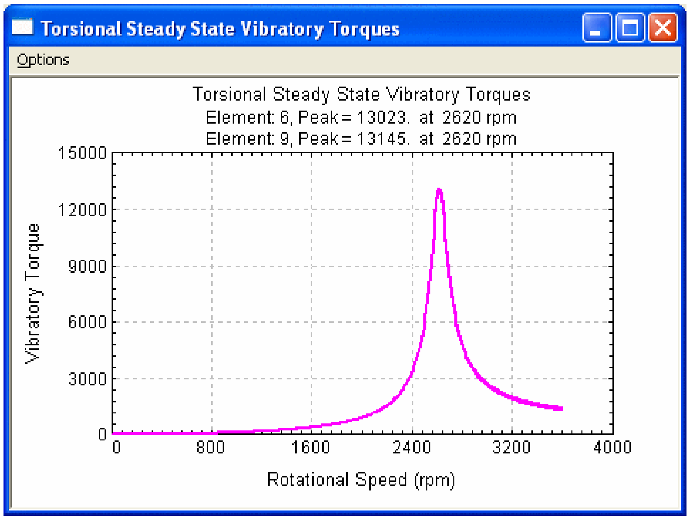
Figure shows the **Deflection** at Element 9 = abs(displacement 9 – displacement 10)



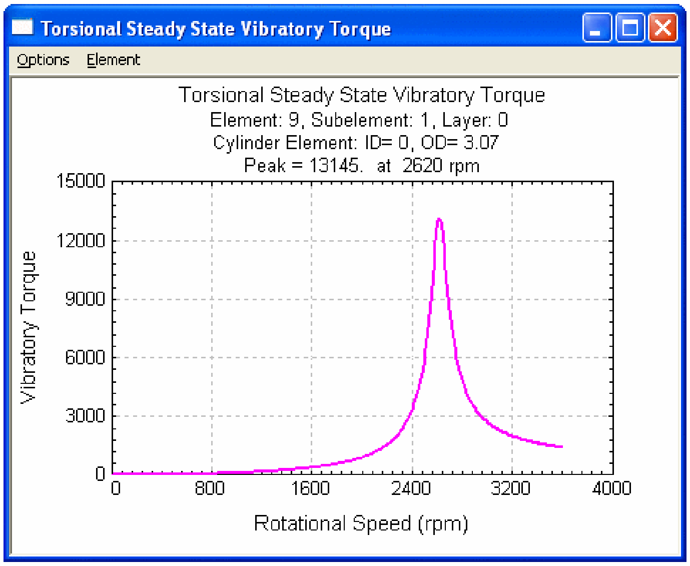
Torsional/Axial Vibratory Torque/Forces

This option displays the element vibratory torque (torsional vibration) or vibratory force (axial vibration). The results for multiple elements can be displayed in the same plot. Caution must be taken for multi-shaft systems since the element number does not exist for the last station of each shaft. For multi-shaft systems, the station numbers are consecutive and the element numbers are not continuous. See Shaft Elements session for better description. You can change the graph title, labels, scale, number of divisions and many others by changing the default settings in the **Setting** dialog under **Options** menu.

Multiple Vibratory Torques at Elements 6 and 9 (Element):



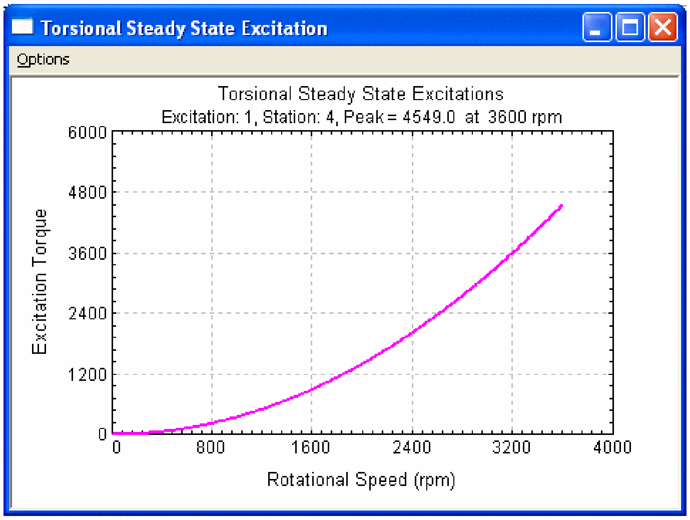
Vibratory Torque at Element 9, Subelement 1, Layer 0 (this option goes to the subelement level):



Torsional/Axial Steady State Excitations

This plot displays the steady state excitations used in the torsional or axial analysis. This is used to double-check the excitation input. Fore more information on the excitation, click Torsional Steady State Excitation. You can change the graph title, labels, scale, number of divisions and many others by changing the default settings in the **Setting** dialog under **Options** menu.

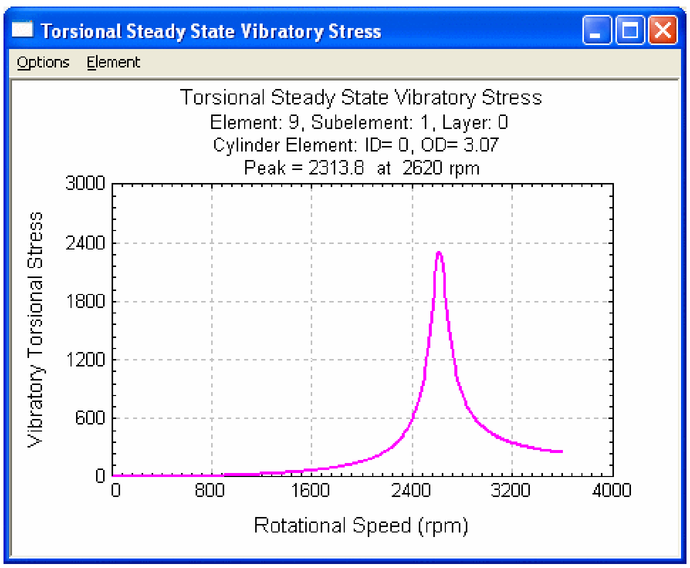
Excitation amplitude vs. speed



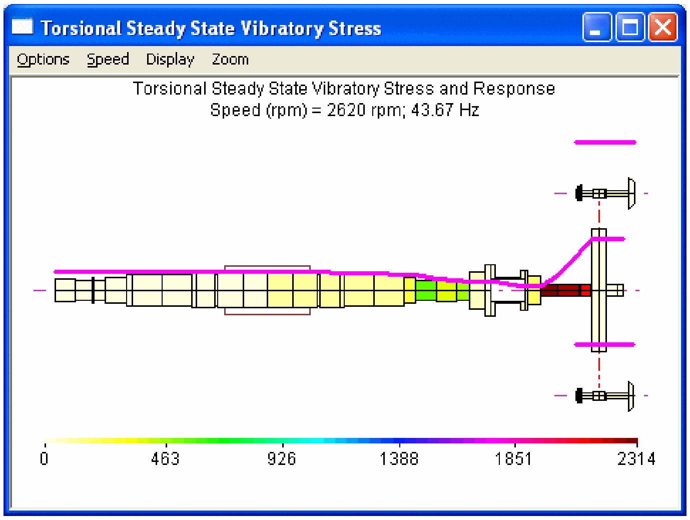
Torsional Vibratory Stress

This option displays the element vibratory stress vs. speed or the element stress for a given speed. Caution must be taken for multi-shaft systems since the element number does not exist for the last station of each shaft. For multi-shaft systems, the station numbers are consecutive and the element numbers are not continuous. See [Shaft Elements] session for better description. You can change the graph title, labels, scale, number of divisions and many others by changing the default settings in the **Setting** dialog under **Options** menu.

Element 9 stress vs. speed



Element stresses at 2620 rpm



See also [Torsional Vibration Analysis](), [Steady State Forced Response Analysis]().

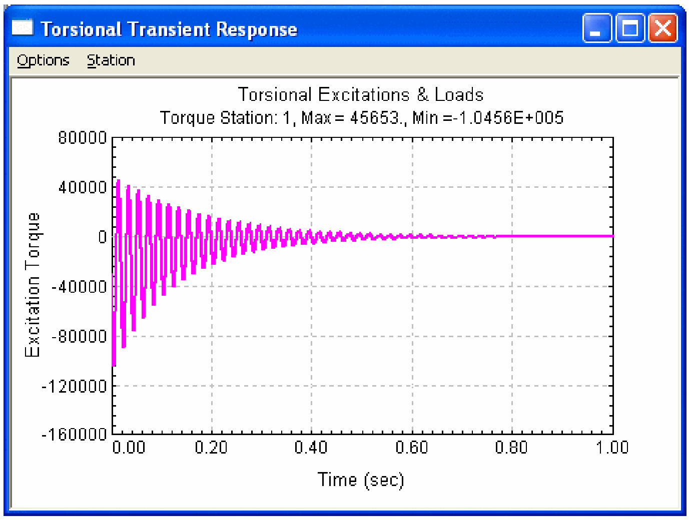
Torsional Transient Response – Time Dependent Excitations

For torsional transient analysis, DyRoBeS allows two types of excitations. One is the time dependent excitation, such as short circuit torque, and the other is the speed dependent excitation, such as synchronous motor startup. For time dependent excitation, the excitation input can be either in the equation format or from a data file. For more information on the time dependent inputs, click Torsional Excitations in Equations and Torsional Excitations in Data Files. For speed dependent excitation, the driving and load torques are entered in Torsional Driving Torque and Load Torque.

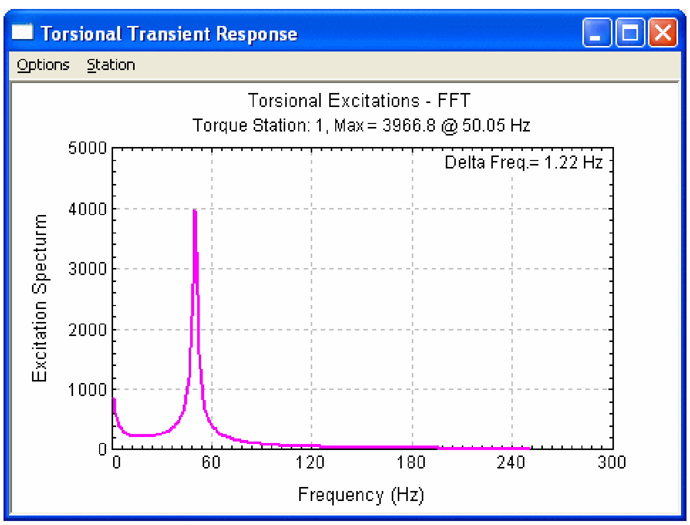
Consider a 4200 kW, 50 Hz, 4 Poles motor with a 3-phase fault, the short circuit excitation torque is shown below:

Torsional Short Circuit Applied Torque

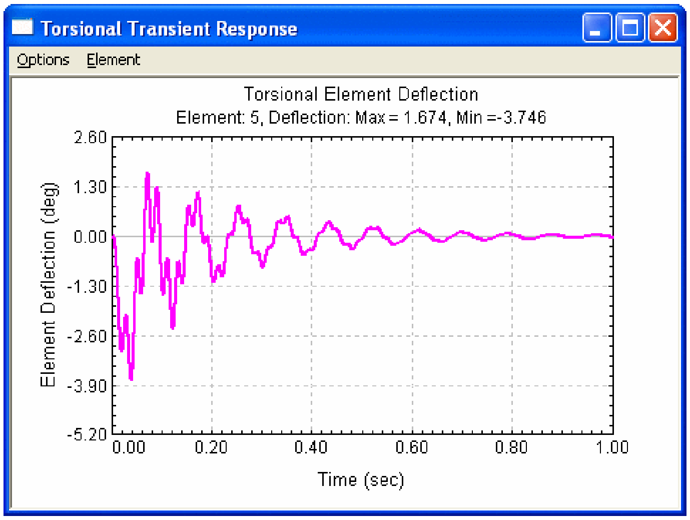
This option displays the torsional short circuit torque vs. time. Note that in a typical torsional short circuit excitation, there is a large initial transient that dies out exponentially.



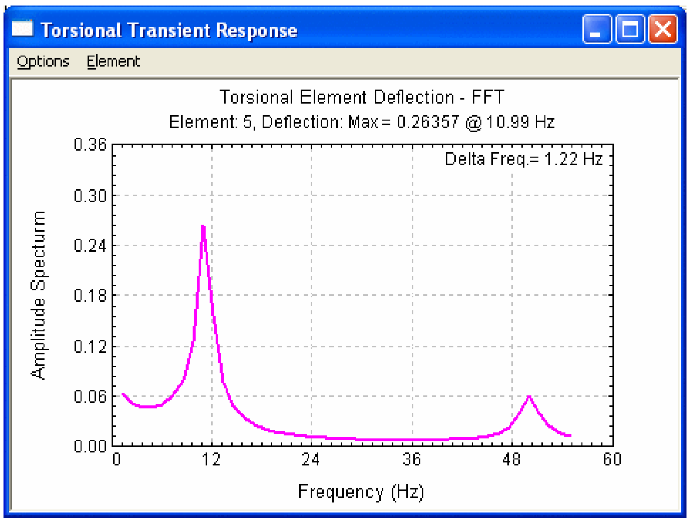
FFT of the excitation, it shows that the excitation is mainly the 50 Hz line frequency.



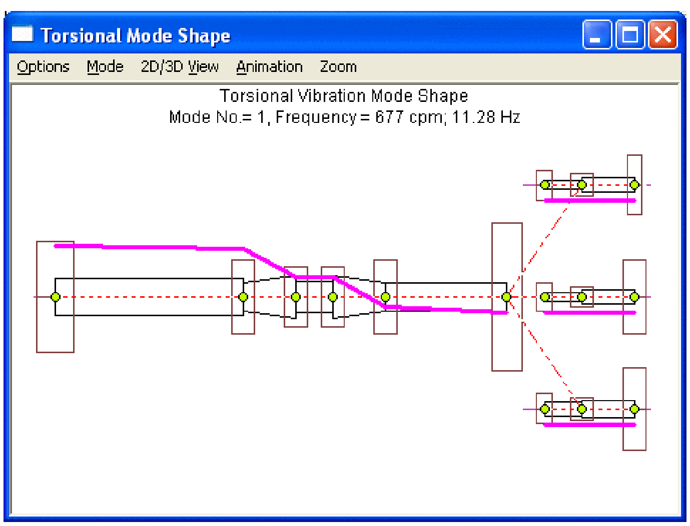
Torsional Short Circuit Transient Element Deflection

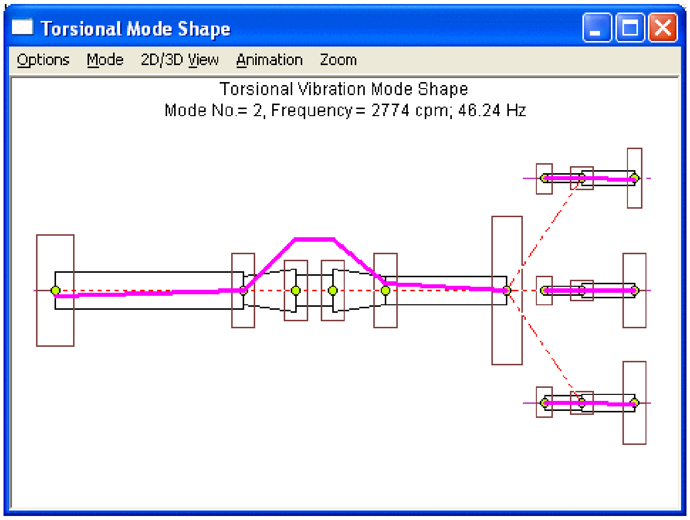


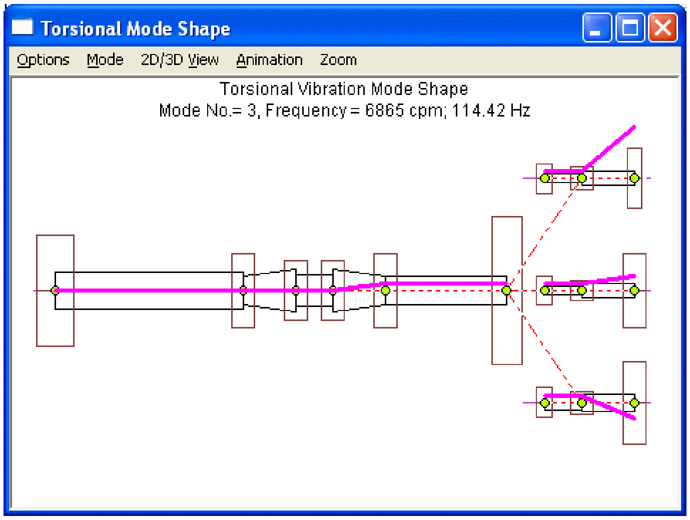
FFT of the response, it shows that the response has two major frequency components. One is 11 Hz, which is the system natural frequency and is excited by the initial conditions. The other one is 50 Hz which is the excitation frequency caused by the short circuit.



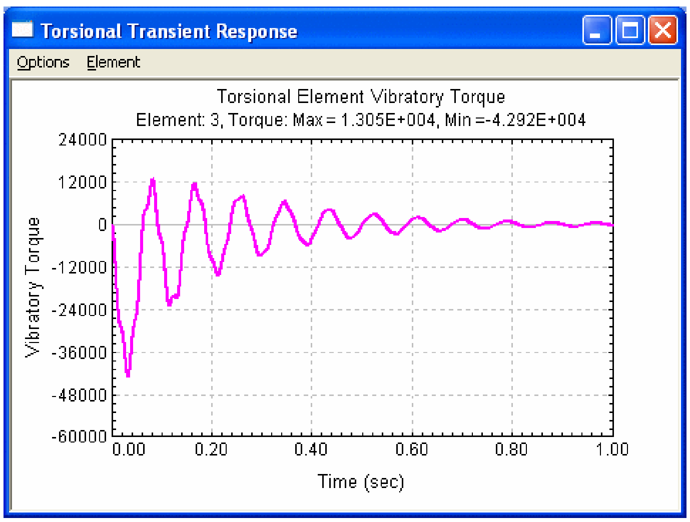
The mode shapes for the first three natural modes are shown below:



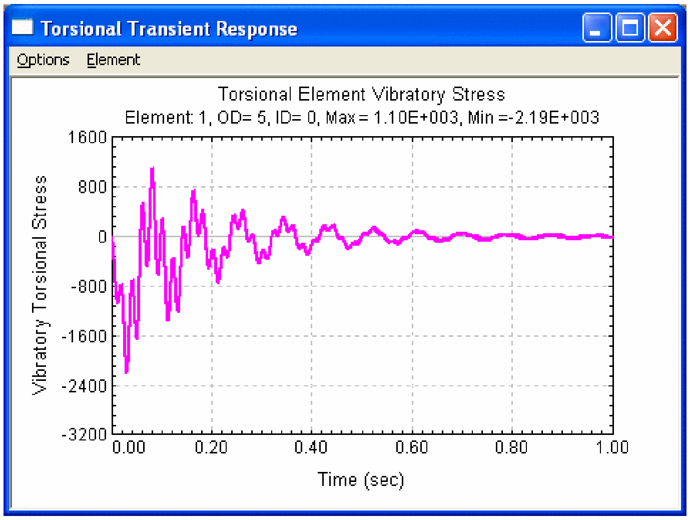




Torsional Short Circuit Transient Element Vibratory Torque



Torsional Short Circuit Transient Element Vibratory Stress



See also Torsional Vibration Analysis, Transient Analysis (Time Dependent Excitations).

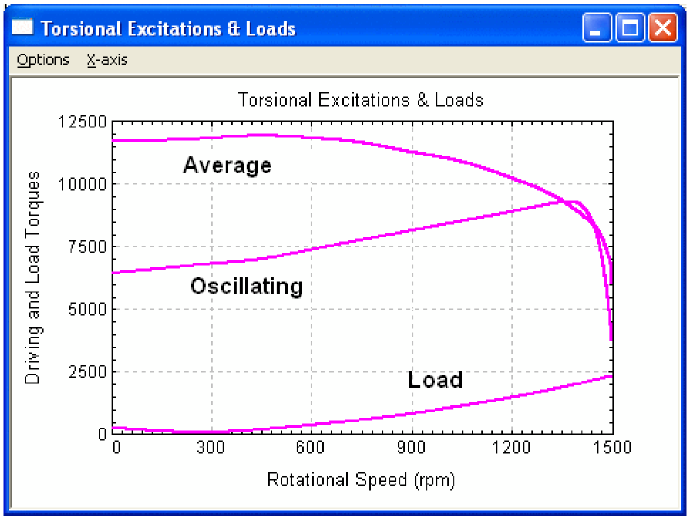
Torsional Transient Response – Speed Dependent Excitations

For torsional transient analysis, DyRoBeS allows two types of excitations. One is the time dependent excitation, such as short circuit torque, and the other is the speed dependent excitation, such as synchronous motor startup. For time dependent excitation, the excitation input can be either in the equation format or from a data file. For more information on the time dependent inputs, click Torsional Excitations in Equations and Torsional Excitations in Data Files. For speed dependent excitation, the driving and load torques are entered in Torsional Driving Torque and Load Torque.

Results for a compressor driven by a 4200 kW, 50 Hz, 4 Poles synchronous motor during startup transient are presented below. Note that the results can be displayed vs. time or speed.

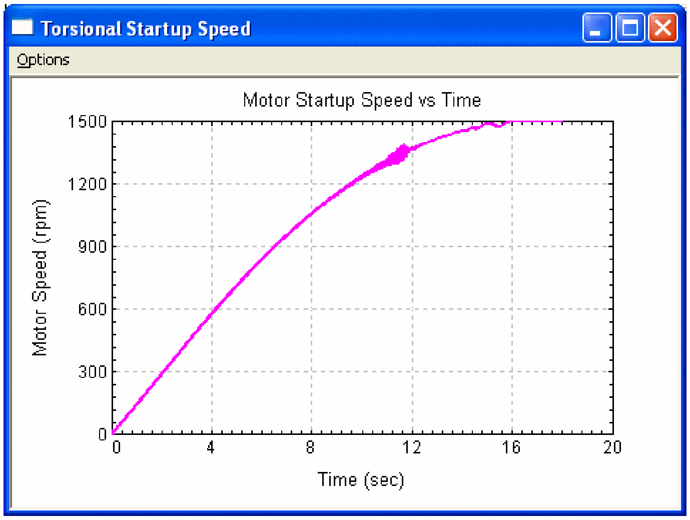
Torsional Startup Driving and Load Torques

This option displays the torsional startup driving and load torques vs. time or speed. This option is used to check the input torques. The following figure shows a plot of the synchronous motor starting average torque and oscillating torque and the compressor load torque The time required to accelerate the system up to the synchronous speed is determined by the numerical integration of the torsional equation of motion.



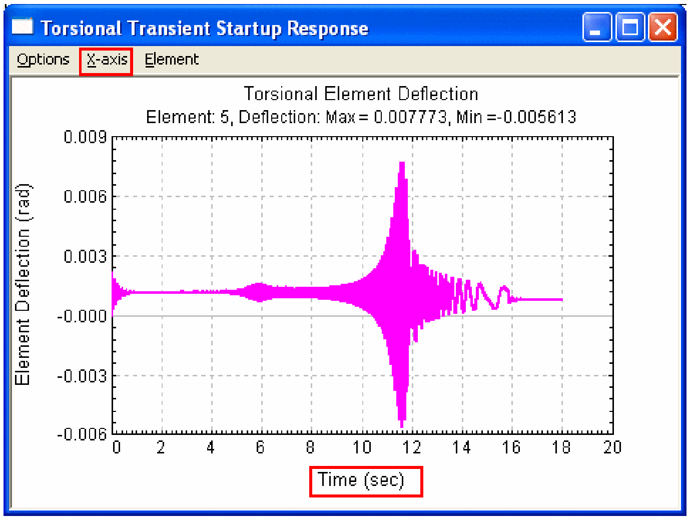
Torsional Startup Speed vs. Time

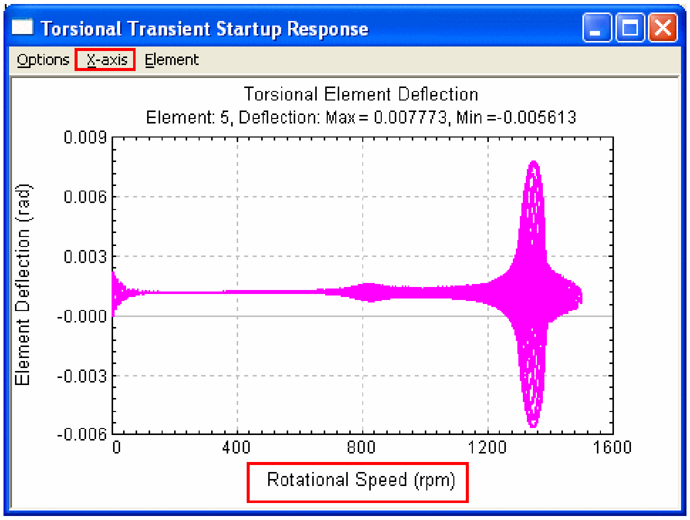
This option displays the torsional transient startup driver speed vs. time. You can change the graph title, labels, number of divisions and many others by changing the default settings in the **Setting** dialog under **Options** menu.



Torsional Startup Transient Element Deflection

This option displays the torsional startup transient element deflection vs. **time** or **speed**. The following figure shows that the first resonance occurs at about 5 seconds (800 rpm) after the startup and the second resonance occurs at about 11 seconds (1330 rpm) after the startup.





The torsional resonant (critical) speeds for a synchronous motor startup are given below:

$$N_{cr} = N_{syn}\left(1 - \frac{\omega_i}{4\pi f_L}\right)$$

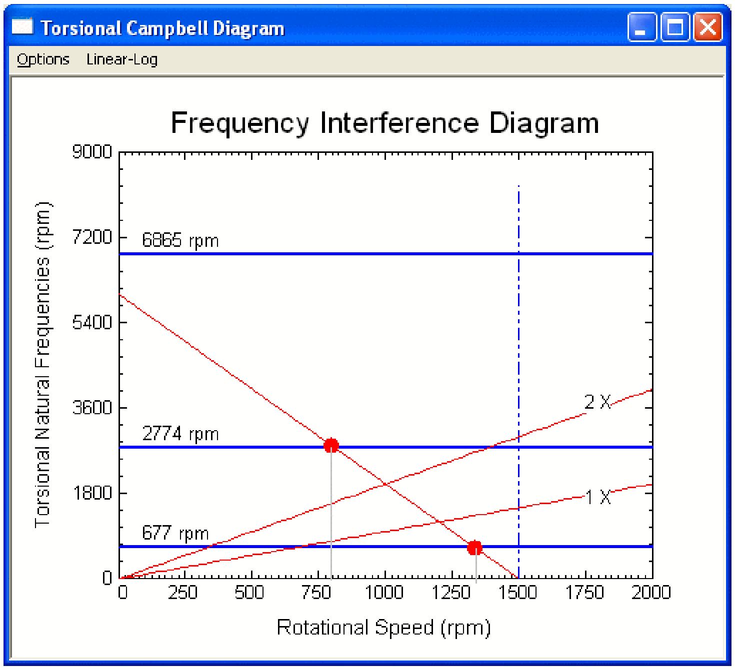
where

N_{cr} is the critical speed in rpm to be calculated.
N_{syn} is the synchronous speed in rpm.
ω_i is the system natural frequency that is less than 2X line frequency (rad/sec).
f_L is the Line Frequency in Hz (50 or 60 Hz).

There are two frequencies below 2X line frequency. They are:

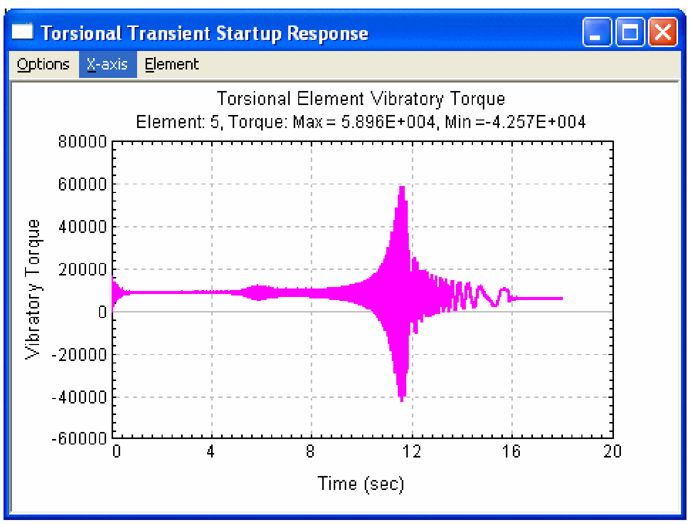
1. 70.89 rad/sec = 11.28 Hz = 677 rpm
2. 290.52 rad/sec = 46.24 Hz = 2774 rpm

Therefore, the two resonant speeds are: 806 rpm and 1331 rpm.

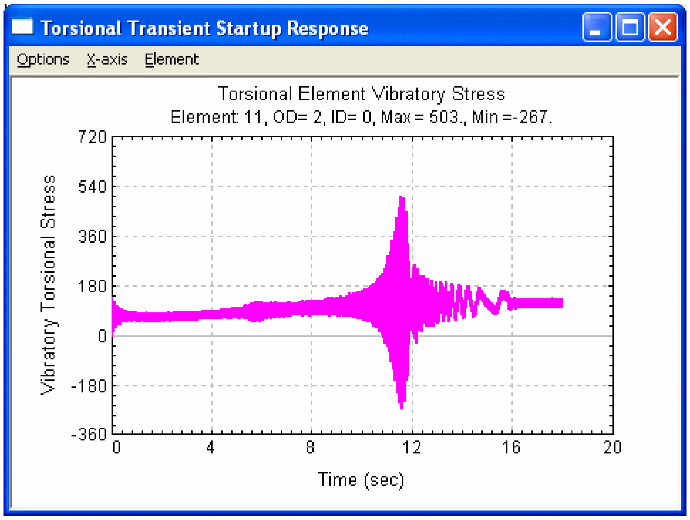


Torsional Startup Transient Element Vibratory Torque

This option displays the torsional startup transient element vibratory torque vs. time or speed.



Torsional Startup Transient Element Vibratory Stress



See also Torsional Vibration Analysis, Startup Transient Analysis (Speed Dependent Excitations).

Tools

A number of engineering tools are included in the program for your convenience. To activate each tool, select the desirable tool from the main Tools menu.

The Alford and Wachel formulations are provided for the aerodynamic cross-coupling calculation. It calculates the cross-coupled stiffness coefficient and it can be a major source of destabilizing force in the turbomachinery.

A simple way to estimate the bearing deflection and stiffness for the rolling element bearings is also provided. The approximate stiffness can be used for the preliminary design study where high accuracy is not required.

The flow rate and dynamic coefficients of liquid annular seal and gas laby seal are also included. Two models are implemented in the liquid seal calculation: Black and Childs models. This effect is also known as Lomakin effect. The gas laby seal tool is provided by the courtesy of Dr. R. G. Kirk of Virginia Tech.

Another design tool is for the squeeze film damper design. It calculates the stiffness and damping for a give damper geometry and operation conditions. It is used for the preliminary design of the squeeze film damper. The complete rotor analysis is required for understanding the effect of the squeeze film damper in the system.

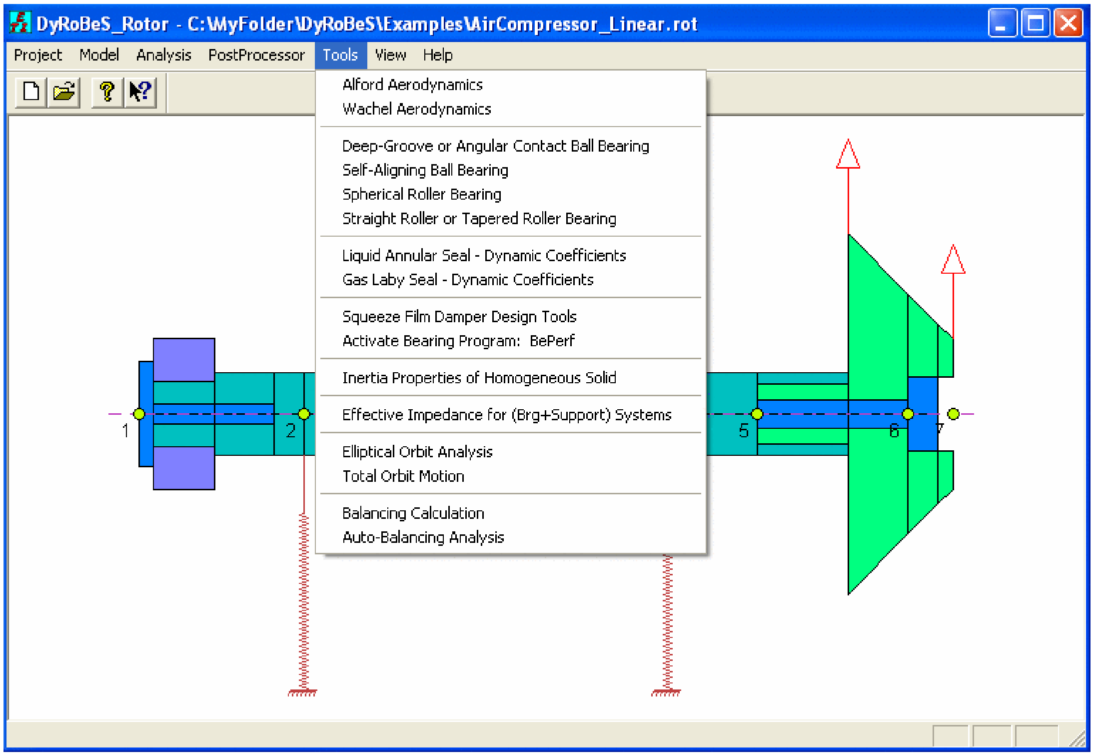
The Mass/Inertia properties calculation for a homogeneous cylinder or cone is also provided for easy reference.

A tool is provided to reduce the 4 DOF journal-bearing-support system to an equivalent 2 DOF journal-bearing system with the reduced equivalent bearing-support dynamic coefficients. Normally, the shaft rotational speed is selected to be the reduction frequency. Thus, the reduced bearing coefficients are called the **synchronously** reduced coefficients. ***DyRoBeS-Rotor*** is capable of handling the flexible supports, therefore, this reduction is not recommended. This reduction procedure is commonly used to demonstrate the effects of the flexible support. However, it is not advisable to perform this reduction in the analysis of complete rotor-bearing systems due to the single harmonic motion assumptions.

Rotor elliptical orbit analysis tool is provided for easy visualization of the rotor precessional motion and it provides some basic understandings of the rotor forward and backward motion. Total orbit motion can also be used to visualize the total rotor motion with various harmonic motions.

Balancing calculation is a tool that is based on the least square method (Influence coefficients method). It performs the balancing calculation to identify the balancing corrections without any knowledge in the rotor system. Auto-Balancing Analysis is another tool for balancing calculation. It assumes the rotor model is known and synchronous excitations are identified. However, the correction planes may not be in the right places. This auto-balancing program can help you to find the optimal corrections to minimize the rotor response due to synchronous excitations.

You may also activate the fluid film bearings program **BePerf** from the rotor program under tools menu.


DyRoBeS_Rotor - C:\MyFolder\DyRoBeS\Examples\AirCompressor_Linear.rot
Project Model Analysis PostProcessor Tools View Help
Alford Aerodynamics
Wachel Aerodynamics
Deep-Groove or Angular Contact Ball Bearing
Self-Aligning Ball Bearing
Spherical Roller Bearing
Straight Roller or Tapered Roller Bearing
Liquid Annular Seal - Dynamic Coefficients
Gas Laby Seal - Dynamic Coefficients
Squeeze Film Damper Design Tools
Activate Bearing Program: BePerf
Inertia Properties of Homogeneous Solid
Effective Impedance for (Brg+Support) Systems
Elliptical Orbit Analysis
Total Orbit Motion
Balancing Calculation
Auto-Balancing Analysis
1
2
5
6
7

Aerodynamic Cross Coupling

The aerodynamic excitation caused by the impeller clearance variation is a destabilizing force:

$$\begin{Bmatrix} F_x \\ F_y \end{Bmatrix} = -\begin{bmatrix} 0 & K_{xy} \\ K_{yx} & 0 \end{bmatrix} \begin{Bmatrix} x \\ y \end{Bmatrix}$$

Alford has proposed the cross-coupled stiffness:

$$Q = K_{xy} = -K_{yx} = \frac{\beta\, T}{D\, H}$$

where

K_{xy}	cross coupled stiffness (Lbf/in)
T	stage torque (lbf-in)
D	blade pitch (mean) diameter (in)
H	blade height (in)
β	efficiency factor (design parameter)

The Alford equation was proposed for axial flow turbines and compressors, it has been also used for centrifugal compressors (Kirk and Donald). The value for the efficiency factor (β) for different types of machines has been suggested by Researchers at Texas A&M University and listed below for reference:

β	= 0.5	for shrouded axially bladed disks
β	= 1.5	for un-shrouded axially bladed disks
β	= 2 - 3	for un-shrouded radial flow impellers
β	= 5 - 10	for extreme cases, overhung impellers

Wachel has proposed an empirical formula for estimating the aerodynamic cross-coupled stiffness based on the several instability problems:

$$Q = K_{xy} = -K_{yx} = \frac{6300\, hp\, Mw}{D\, H\, Rpm} \cdot \frac{\rho_d}{\rho_s}$$

where

K_{xy}	cross coupled stiffness (Lbf/in)
hp	stage horsepower
Mw	molecular weight Air = 28.966, N2 = 28.0134, CO2 = 44.01, O2 = 31.9988, Natural Gas = 19.00
D	impeller diameter (in)
H	restrictive dimension in flow path (in)
Rpm	stage speed
ρ_d	density of fluid at discharge
ρ_s	density of fluid at suction

See also Pseudo bearing.

References for Aerodynamic Cross Coupling

Alford, J., 1965, Protecting Turbomachinery from Self-Excited Rotor Whirl, Journal of Engineering for Power, pp.333-334.

Kirk, R. G. and Donald, G.H., Design Criteria for Improved Stability of Centrifugal Compressors, ASME Rotor Dynamical Instability, AMD-Vol. 55, 1983.

Wachel, J. C., 1983, Compressor Case Histories, presented at Rotating Machinery and Controls (ROMAC) Short Course, University of Virginia, June 8-10.

Rolling Element Bearings

The rolling element bearing stiffness estimation provided in this program is based on Gargiulo's paper. This approximation provides a convenient way to estimate the bearing stiffness since it requires minimum data input.

Deep-Groove or Angular-Contact Radial Ball Bearings

$$\delta = 46.2E-06 \sqrt[3]{\frac{F^2}{DZ^2 \cos^5 \alpha}}$$

$$K = 0.0325E06 \sqrt[3]{DFZ^2 \cos^5 \alpha}$$

Self-Aligning Ball Bearings

$$\delta = 74.0E-06 \sqrt[3]{\frac{F^2}{DZ^2 \cos^5 \alpha}}$$

$$K = 0.0203E06 \sqrt[3]{DFZ^2 \cos^5 \alpha}$$

Spherical Roller

$$\delta = 14.5E-06 \sqrt[4]{\frac{F^3}{L^2 Z^3 \cos^7 \alpha}}$$

$$K = 0.0921E06 \sqrt[4]{FL^2 Z^3 \cos^7 \alpha}$$

Straight Roller or Tapered Roller

$$\delta = 3.71E-06 \frac{F^{0.9}}{L^{0.8} Z^{0.9} \cos^{1.9} \alpha}$$

$$K = 0.300E06 \; F^{0.1} Z^{0.9} L^{0.8} \cos^{1.9} \alpha$$

where
δ Radial Deflection
K Radial Stiffness (Lbf/in)
F External Radial Force (Lbf)
D Ball Diameter (in)
Z Number of Rolling Elements
L Roller Effective Length (in)
α Contact Angle (rad)

See also Generalized non-linear isotropic bearing.

Reference on roller element bearing stiffness estimation

Gargiulo, E.P.,Jr., A Simple Way to Estimate Bearing Stiffness, Machine Design, 1980, pp.107-110.

Liquid Annular Seals Calculation

The liquid annular seals used in the pumps are known to raise the **dry** critical speeds by a considerable amount. Due to the pressure drop across the seal, the pressure difference develops a strong radial restoring force opposing to the shaft displacement. This effect is known as **Lomakin effect**. Black and Jenssen have extended Lomakin's theory in the development of the rotordynamics coefficients by using the bulk flow analysis. Later Childs formulated a more complete bulk flow model using Hirs' lubrication equation which includes the influence of fluid inertia terms and inlet swirl. For small motion, the forces generated in an annular seal are:

$$\begin{Bmatrix} -F_x \\ -F_y \end{Bmatrix} = \begin{bmatrix} K_d & k_c \\ -k_c & K_d \end{bmatrix} \begin{Bmatrix} x \\ y \end{Bmatrix} + \begin{bmatrix} C_d & c_c \\ -c_c & C_d \end{bmatrix} \begin{Bmatrix} \dot{x} \\ \dot{y} \end{Bmatrix} + \begin{bmatrix} m_d & 0 \\ 0 & m_d \end{bmatrix} \begin{Bmatrix} \ddot{x} \\ \ddot{y} \end{Bmatrix}$$

The restoring forces are proportional to the displacement, velocity and acceleration. In this aspect, the seals resemble fluid-film bearings. However, the governing equations for an annular seal are quite different from the normal turbulent Reynolds equations used in the bearing analysis. In DyRoBeS, there are two ways to include the seal dynamic coefficients. One explicit approach is using the TOOLS in the main menu. One can enter the seal data, total pressure drop, speed, and calculation method (Black or Childs), the dynamic coefficients and leakage rate are calculated and shown in the screen. Since the pressure drop depends upon the rotor speed, it is convenient to include the seal calculation inside the rotordynamics code. Therefore, another implicit approach is to enter the seal data under BEARING tab in the rotor bearing data editor (bearing type 11), then the seal dynamic coefficients are calculated inside the rotordynamics code for various analyses.

A fundamental relationship between the total axial pressure drop and the mean axial flow velocity is:

$$\Delta P = \frac{1}{2}(1 + \xi + 2\sigma)\rho V^2$$

where ξ is the entrance loss coefficient (typical value = 0.1)

ρ is the fluid density

$\sigma = \lambda \dfrac{L}{C}$

L is the seal length

C is the seal radial clearance

λ is the frication loss factor which is a function of Reynolds number

V is the mean axial flow velocity

and $\frac{1}{2}(1+\xi)\rho V^2$ defines the inlet pressure drop.

Since σ is a nonlinear function of the mean axial velocity, the mean axial velocity must be determined iteratively.

The circumferential velocity of the fluid entering the seal is commonly expressed as a fraction of shaft surface speed:

$$u_c = \alpha R\Omega$$

where alpha is the inlet swirl ratio. In Black's model, the swirl ratio is 0.5. The value can be lower if a swirl break is used. In Childs' model, the inlet swirl ratio can be varied. It can be seen that inlet swirl has a significant influence on the cross-coupled stiffness and that therefore stability can be improved by reducing the inlet swirl.

Black Model

The seal dynamic coefficients based on Black's model are well documented in references (1,3,4,5) and listed here for reference:

$$K_d = \mu_3 \left(\mu_0 - 0.25\mu_2\Omega^2 T^2\right)$$

$$k_c = \mu_3 \left(0.5\mu_1 \, \Omega T\right)$$

$$C_d = \mu_3 \, \mu_1 \, T$$

$$c_c = \mu_3 \, \mu_2 \, \Omega T^2$$

$$m_d = \mu_3 \, \mu_2 \, T^2$$

with

$$\mu_0 = \frac{(1+\xi)\sigma^2}{(1+\xi+2\sigma)^2}$$

$$\mu_1 = \frac{(1+\xi)^2\sigma + (1+\xi)(2.33+2\xi)\sigma^2 + 3.33(1+\xi)\sigma^3 + 1.33\sigma^4}{(1+\xi+2\sigma)^3}$$

$$\mu_2 = \frac{0.33(1+\xi)^2(2\xi-1)\sigma + (1+\xi)(1+2\xi)\sigma^2 + 2(1+\xi)\sigma^3 + 1.33\sigma^4}{(1+\xi+2\sigma)^4}$$

$$\mu_3 = \frac{\pi R \Delta P}{\lambda}$$

$$\lambda = 0.079 \; R_a^{-0.25}\left[1+\left(\frac{7R_c}{8R_a}\right)^2\right]^{0.375}$$ Friction Loss Factor

$$R_a = \frac{2\rho VC}{\mu}$$ Axial Reynolds number

$$R_c = \frac{\rho \mathrm{R}\Omega C}{\mu}$$ Circumferential Reynolds number

μ is the fluid viscosity

$$T = \frac{L}{V}$$

The above parameters are based on short seal solution. The corrected parameters for finite length seal are:

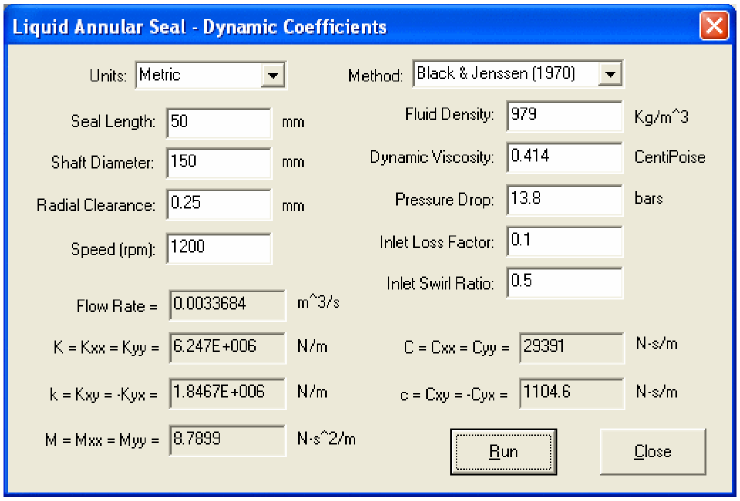
$$\mu_0 = \frac{\mu_0}{1+0.28(L/R)^2}$$

$$\mu_1 = \frac{\mu_1}{1+0.23(L/R)^2}$$

$$\mu_2 = \frac{\mu_2}{1+0.06(L/R)^2}$$

Since the friction loss factor is a function of Reynolds number that is dependent upon the velocity, therefore, an iterative process is needed to solve the mean axial velocity.

Example taken from reference (3) is used to demonstrate this calculation.



The small differences between DyRoBeS and reference (3) are due to:

1. DyRoBeS uses finite length corrected parameters. However, short seal theory was used in the reference. It is known that short seal solutions tend to overestimate the dynamic coefficients of finite length seals.

2. In reference paper (2), although iteration was used to determine the average axial velocity, however, the initial σ was used to calculate the dynamic coefficients. In DyRoBeS, the final converged σ is used in the following calculation.

Childs Model

Childs formulated the seal dynamic coefficients based on Hirs' lubrication equation. The fluid inertia terms are included in the momentum equations and the inlet swirl is also included. The short seal theory is used. The coefficients are summarized in the following:

$$K_d = \left(\frac{\pi R\Delta P}{\lambda}\right)\frac{2\sigma^2}{(1+\xi+2\sigma)}\left\{1.25E - \frac{(\Omega T)^2}{4\sigma}\left\{\frac{1}{2}\left(\frac{1}{6}+E\right)+\frac{2v_0}{a}\left[\left(E+\frac{1}{a^2}\right)(1-e^{-a})-\left(\frac{1}{2}+\frac{1}{a}\right)e^{-a}\right]\right\}\right\}$$

$$k_c = \left(\frac{\pi R\Delta P}{\lambda}\right)\frac{\sigma^2\Omega T}{(1+\xi+2\sigma)}\left\{\frac{E}{\sigma}+\frac{B}{2}\left(\frac{1}{6}+E\right)+\frac{2v_0}{a}\left\{EB+\left(\frac{1}{\sigma}-\frac{B}{a}\right)\left[(1-e^{-a}\left(E+\frac{1}{2}+\frac{1}{a}\right)-1\right]\right\}\right\}$$

$$C_d = \left(\frac{\pi R\Delta P}{\lambda}\right)\frac{2\sigma^2 T}{(1+\xi+2\sigma)}\left[\frac{E}{\sigma}+\frac{B}{2}\left(\frac{1}{6}+E\right)\right]$$

$$c_c = \left(\frac{\pi R\Delta P}{\lambda}\right)\frac{2\sigma\,\Omega T^2}{(1+\xi+2\sigma)}\left\{\frac{1}{2}\left(\frac{1}{6}+E\right)+\frac{v_0}{a}\left[(1-e^{-a}\left(E+\frac{1}{2}+\frac{1}{a^2}\right)-\left(\frac{1}{2}+\frac{e^{-a}}{a}\right)\right]\right\}$$

$$m_d = \left(\frac{\pi R\Delta P}{\lambda}\right)\frac{\sigma\left(\frac{1}{6}+E\right)}{(1+\xi+2\sigma)}\cdot T^2$$

where

$\lambda = 0.066\, R_a^{-0.25}\left(1+\frac{1}{4b^2}\right)^{0.375}$ Friction Loss Factor

$R_a = \frac{\rho VC}{\mu}$ Axial Reynolds number

(note: it is a half of the Reynolds number defined by Black)

$$R_c = \frac{\rho R\Omega C}{\mu}$$

$$b = \frac{R_a}{R_c} = \frac{V}{R\Omega}$$

$$a = \sigma(1+0.75\beta)$$

$$\beta = \frac{1}{1+4b^2}$$

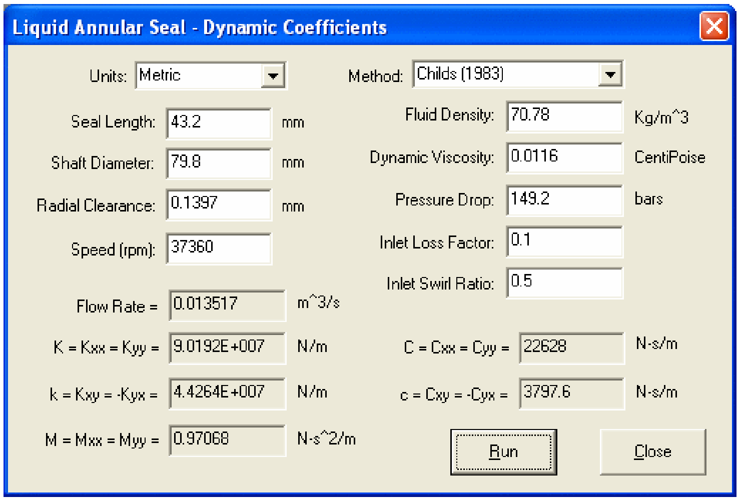
$$B = 1+4b^2\beta(0.75)$$

$$E = \frac{(1+\xi)}{2(1+\xi+B\sigma)}$$

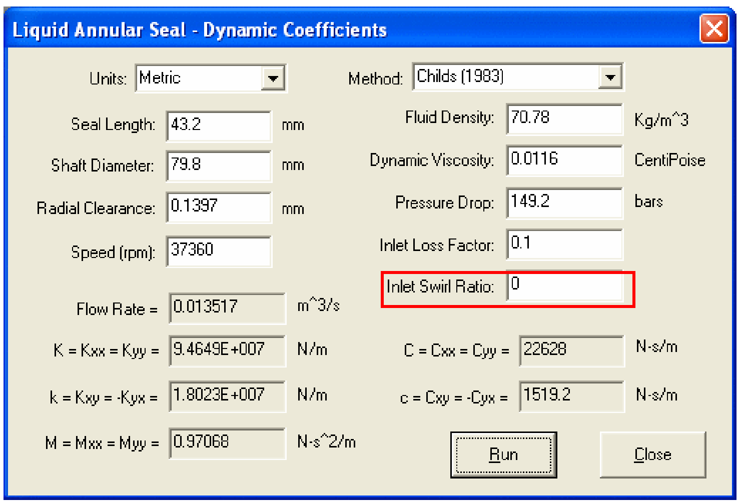
$$T = \frac{L}{V}$$

An interstage seal of the High Pressure Hydrogen Turbopump (HPFTP) of the Space Shuttle Main Engine (SSME) used in reference (2) is presented below.

For an inlet swirl ratio of 0.5, that is, the inlet circumferential velocity is a half of the surface speed, the results are listed below.



For $u_o = 0$, the results are listed below:

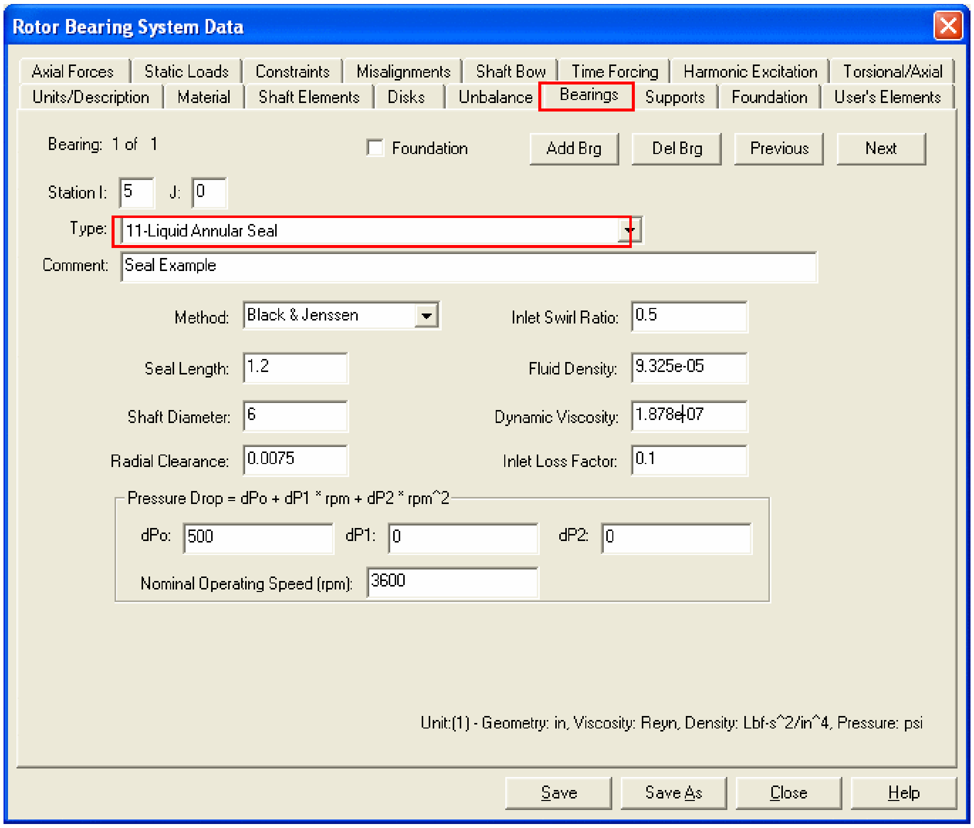


Note that in reference (2), $v_0 = 0$ indicates $u_c = 0.5\, R\Omega$ and $v_0 = -0.5$ indicates $u_c = 0$.

The DyRoBeS results are in good agreement with the results of the reference (2).

Data input under Bearing Tab

The seal dynamic coefficients can be calculated internally when performing the rotordynamic analyses. Due to the similarity with bearing data, the seal data are entered under the Bearing tab. A brief description of the data input is listed below:



There are two methods provided in DyRoBeS: Black and Jenssen, and Childs (1983). Typical value for the Inlet Swirl Ratio is 0.5; that is, the circumferential velocity of the fluid entering the seal is a half of the shaft surface speed. To improve the rotor stability by decreasing the seal cross-coupling stiffness, a swirl break can be used to lower the swirl ratio. For the Black model, 0.5 is always used. Typical value for the inlet loss factor is 0.1. The inlet pressure drop as the fluid entering the seal is defined by $\frac{1}{2}(1+\xi)\rho V^2$. Since the total pressure drop (or the pump discharge pressure) is a function of rotor speed, therefore, a second order polynomial is used to calculate the seal total pressure drop for a given speed. The nominal operating speed in this data input page is only used to calculate the seal coefficients for the Critical Speed Analysis since no speed is given in the analysis input. For the Static Deflection and Bearing Loads calculation, seal data are not used. For the Whirl Speed/Stability Analysis, Unbalance Response Analysis, and Transient Analysis, the seal coefficients are calculated for the speeds given in the each Analysis input.

Note:

Two theoretical methods are provided in DyRoBeS to be used in the rotordynamics calculation. In most publications the theoretical models are compared with the test experiments and good corrections were obtained, although certain differences remain. Further differences are evident in the industrial application; therefore, the precaution must be taken while using these coefficients. A tolerance on these calculated coefficients should be applied when the accurate critical speed and stability margins are critical.

References:

1. Black, H. F. and Jenssen, D. N., 1970, Dynamic Hybrid Bearing Characteristics of Annular Controlled Leakage Seals, Proc Instn Mech Engrs, Vol. 184, pp. 92-100.

2.Childs, D. W., 1983, Dynamic Analysis of Turbulent Annular Seals Based On Hirs' Lubrication.Equaion, ASME Journal of Lubrication Technology, Vol. 105, pp.429-436.

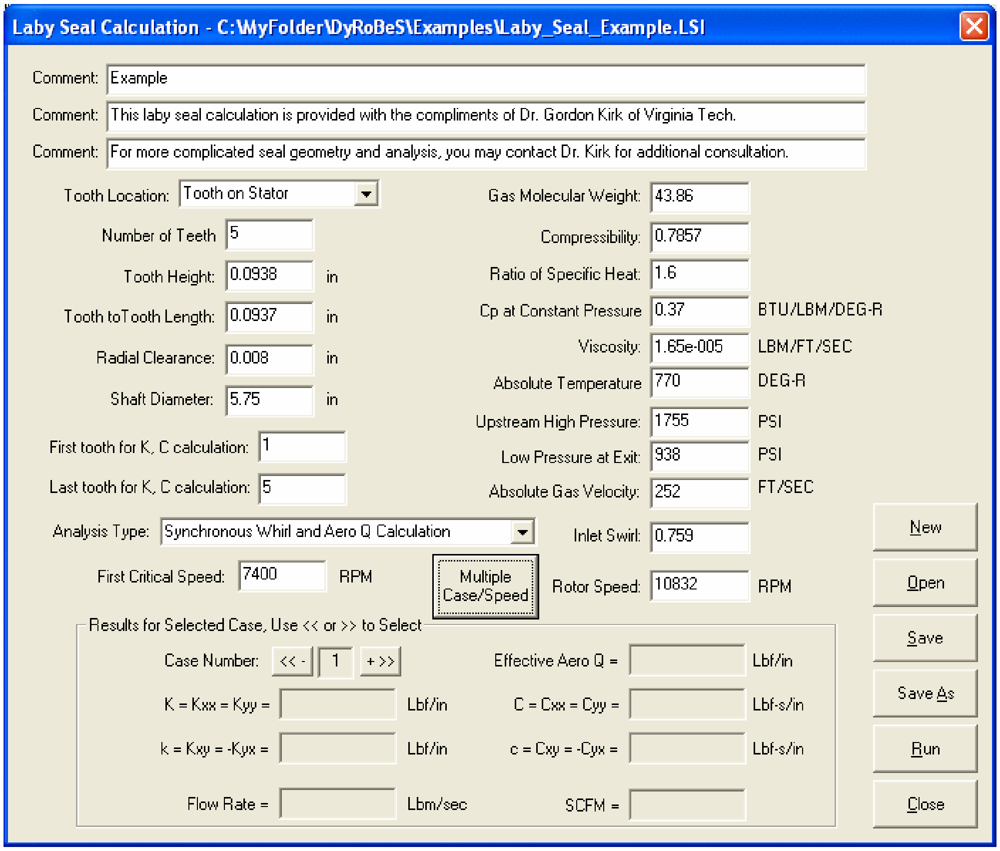
3. Barrett, L. E., 1984, Turbulent Flow Annular Pump Seals: A Literature Review, Shock and Vibration Digest, pp. 3-13.

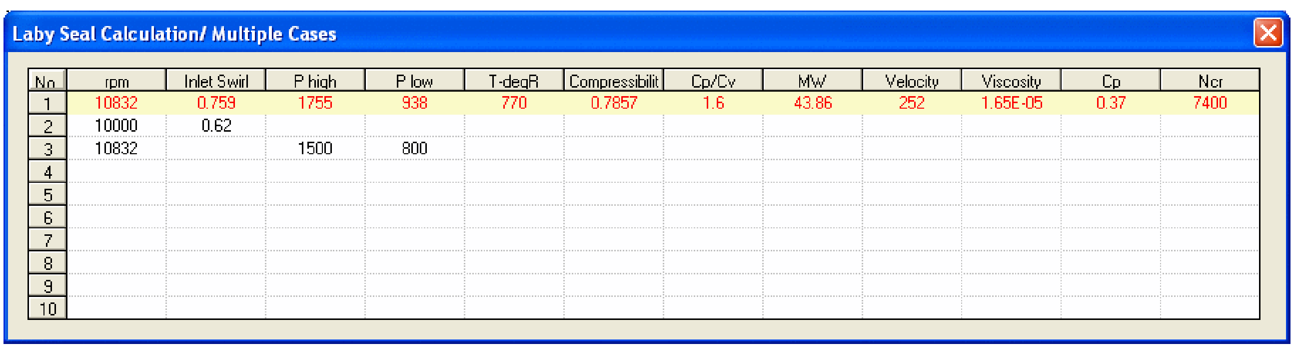
4. Diewald, W. and Nordmann, R., 1989, Dynamic Analysis of Centrifugal Pump Rotors with Fluid-Mechanical Interactions, ASME Journal of Vibration, Acoustics, Stress, and Reliability in Design, Vol. 111, pp.370-378.

5. Corbo, M. A. and Malanoski, S. B., 2003, Pump Rotordynamics Made Simple, Proceeding of The 15^{th} International Pump Users Symposium, pp. 167-204.

Gas Laby Seals

This analysis is provide by Dr. R. G. Kirk of Virginia Tech. It calculates the flow rate and dynamic coefficients of a gas laby seal. It also allows you to run multiple cases/speeds to compare the results.





No	rpm	Inlet Swirl	P high	P low	T-degR	Compressibilit	Cp/Cv	MW	Velocity	Viscosity	Cp	Ncr
1	10832	0.759	1755	938	770	0.7857	1.6	43.86	252	1.65E-05	0.37	7400
2	10000	0.62										
3	10832		1500	800								
4												
5												
6												
7												
8												
9												
10												

Squeeze Film Design Tool

The damper stiffness and damping may be used to size the dimensions for a suitable damper design. After the damper is sized, then a full nonlinear analysis must be performed to determine its suitability.

For the steady state response, the following table summarizes the equivalent stiffness and damping for the cases of circular synchronous motion about the origin and pure radial motion with no precession for the conditions of cavitation (pi film) and no cavitation (2pi film).

Bearing	Film	Motion	Stiffness	Damping
Short Bearing	π film	Circular Synchronous Precession	$\frac{2\mu R L^3 \varepsilon \omega}{C^3 (1-\varepsilon^2)^2}$	$\frac{\mu R L^3 \pi}{2C^3 (1-\varepsilon^2)^{3/2}}$
	2π film		0	$\frac{\mu R L^3 \pi}{C^3 (1-\varepsilon^2)^{3/2}}$
	π film	Pure Radial Squeeze Motion	0	$\frac{\mu R L^3 \pi (2\varepsilon^2+1)}{2C^3 (1-\varepsilon^2)^{5/2}}$
	2π film		0	$\frac{\mu R L^3 \pi (2\varepsilon^2+1)}{C^3 (1-\varepsilon^2)^{5/2}}$
Long Bearing	π film	Circular Synchronous Precession	$\frac{24\mu R^3 L \varepsilon \omega}{C^3 (2+\varepsilon^2)(1-\varepsilon^2)}$	$\frac{12\mu R^3 L \pi}{C^3 (2+\varepsilon^2)(1-\varepsilon^2)^{1/2}}$
	2π film		0	$\frac{24\mu R^3 L \pi}{C^3 (2+\varepsilon^2)(1-\varepsilon^2)^{1/2}}$

Where R = damper radius
L = damper axial length
C = radial clearance
ω = whirl speed
μ = oil viscosity
ε = eccentricity ratio

Note that for the circular synchronous motion, the equivalent stiffness term is a highly nonlinear function of eccentricity and may lead to a nonlinear jump phenomenon under high rotor unbalance. Caution must be taken while designing the damper, since it can significantly either improve or degrade the dynamic characteristics of the rotor system.

See also Squeeze film damper.

DyRoBeS©_BePerf

See Full DyRoBeS©_BePerf Manual at dyrobes.com

DyRoBeS©_BePerf computer program has been developed to analyze the **Be**aring steady state and dynamic **Perf**ormance of fixed lobe, flexural pad, and tilting pad hydrodynamic journal bearings based on Finite Element Analysis (FEA). In addition to bearing analysis, the program also performs lubricant properties analysis and oil flow calculation. The acronym, ***DyRoBeS***©, denotes **Dy**namics of **Ro**tor **Be**aring **S**ystems.

The governing equation for pressure distribution in a fluid film journal bearing is incompressible Reynolds equation which is derived from the Navier-Stokes equation, as expressed below. The fluid film forces acting on the journal are determined by application of boundary conditions and integration of pressure distribution. It is an iterative process until the convergence criterion is satisfied. Once the static equilibrium is found, the bearing static performance, such as bearing eccentricity ratio, attitude angle, minimum film thickness, maximum film pressure, frictional power loss, oil flow rate, etc., can be easily determined. Under dynamic conditions, the journal is oscillating with small amplitudes around the static equilibrium position. The eight bearing dynamic coefficients (stiffness and damping) are obtained by solving the perturbed pressure equations.

$$\frac{\partial}{\partial x}\left(\frac{1}{G_x}\frac{h^3}{\mu}\frac{\partial P}{\partial x}\right)+\frac{\partial}{\partial y}\left(\frac{1}{G_y}\frac{h^3}{\mu}\frac{\partial P}{\partial y}\right)=\frac{U_x}{2}\frac{\partial h}{\partial x}+\frac{U_y}{2}\frac{\partial h}{\partial y}+\frac{\partial h}{\partial t}$$

where x is in the axial direction and y is in the circumferential direction. Gx and Gy called the turbulent flow coefficients are the correctional terms of viscosity caused by the turbulent diffusion:

$$G_x = 12 + 0.0043\,\mathrm{Re}^{0.96} \quad \text{Axial direction}$$

$$G_y = 12 + 0.0136\,\mathrm{Re}^{0.90} \quad \text{Circumferential direction}$$

$$\mathrm{Re} = \frac{\rho U h}{\mu} \quad \text{Local Reynolds number}$$

For laminar flow, Gx = Gy = 12. A critical parameter affected by turbulence is the shear stress acting on the shaft.

$$\tau_s = C_f\frac{\mu U}{h}+\frac{h}{2}\frac{\partial P}{\partial y}$$

$$C_f = 1 + 0.0012\,\mathrm{Re}^{0.94}$$

where Cf is the turbulent Couette shear stress factor. For laminar flow, Cf = 1.

The boundary conditions in the axial coordinate are that the pressure is ambient at the edges of the bearing pad. The Swift-Stieber or Reynolds boundary conditions are applied in the circumferential coordinate. Film cavitation is considered and the transition boundary curve to the film rupture is determined by iteration.

The governing equation for pressure distribution in a gas/air lubricated journal bearing is compressible Reynolds equation.

$$\frac{\partial}{\partial x}\left(\frac{Ph^3}{12\mu}\frac{\partial P}{\partial x}\right)+\frac{\partial}{\partial y}\left(\frac{Ph^3}{12\mu}\frac{\partial P}{\partial y}\right)=\frac{U}{2}\frac{\partial(Ph)}{\partial x}+\frac{\partial(Ph)}{\partial t}$$

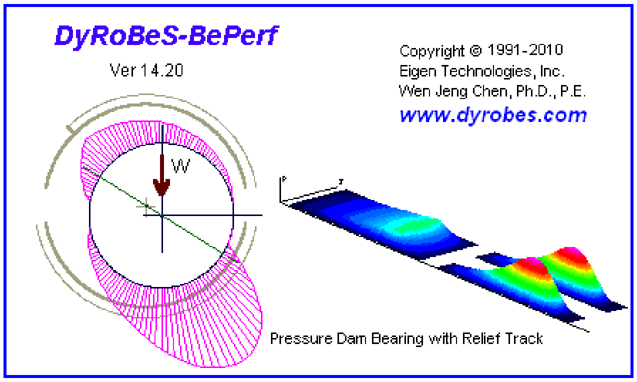
This compressible Reynolds equation is more difficult to analyze due to the existence of the pressure (*P*) in each terms compared with the incompressible flow, which makes the problem non-linear. Weak formulation based on variational principle is applied for generating the finite element model for the boundary value problems. Since this is a nonlinear problem, Newton-Raphson's iterative scheme is utilized to solve the pressure increment, or pressure correction.

The solutions techniques for the incompressible and compressible Reynolds equation are discussed in the book – **Introduction to Dynamics of Rotor-Bearing Systems** by W. J. Chen and E. J. Gunter, 2005.

The ***DyRoBeS***© Bearing program consists of seven primary modules:

1. Fixed Lobe Journal Bearings
2. Tilting Pad and Flexural Pad Journal Bearings

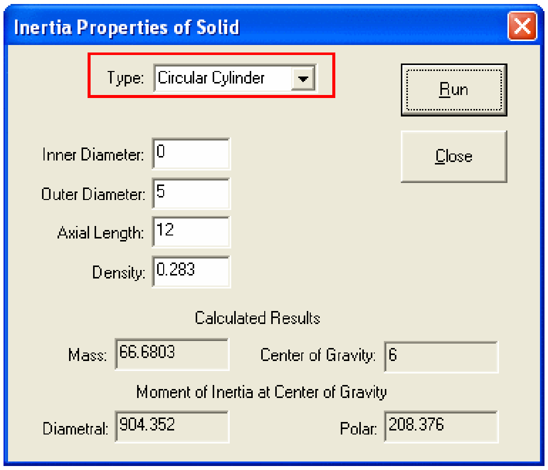
3. Floating Ring Bearings
4. Gas Bearings
5. Thrust Bearings
6. Hydrostatic-Hybrid Bearings
7. Lubricants
8. Flow Calculation

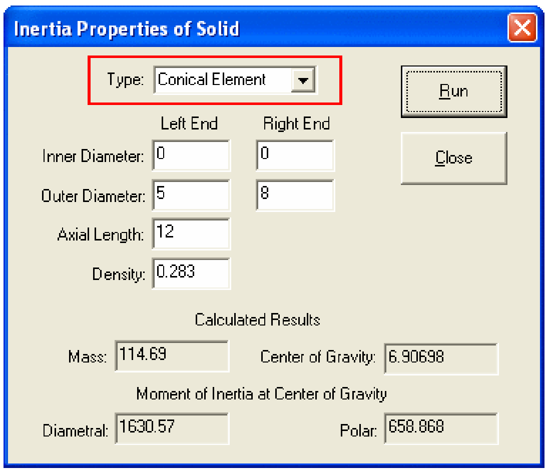


See Full DyRoBeS©_BePerf Manual at dyrobes.com

Mass/Inertia Properties of a Homogeneous Element

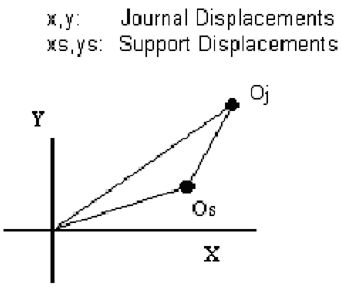
The mass/inertia properties of a homogeneous element can be calculated by using this option. The axis-symmetric element can be either circular cylinder or conical (tapered) element. The inputs are element geometry and density. The outputs are the mass/inertia properties of the element and the center of mass location. These values are required in disk properties.

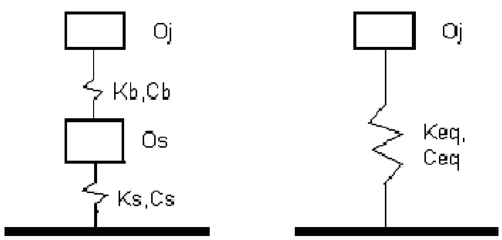




Reduction of Flexible Supports

Consider a harmonic motion with a frequency of ω. The 4 DOF journal-bearing-support system can be reduced to be a 2 DOF journal-bearing system with the reduced equivalent bearing-support dynamic coefficients. Normally, the shaft rotational speed is selected to be the reduction frequency. Thus, the reduced bearing coefficients are called the **synchronously** reduced coefficients. This reduction procedure is commonly used to demonstrate the effects of the flexible support. However, **it is not advisable to perform this reduction in the analysis of complete rotor-bearing systems due to the single harmonic motion assumptions**. The reduced coefficients are frequency dependent and can be used in the steady state synchronous response analysis, but not suitable for the whirl speed and stability analysis and the transient analysis.





For the harmonic motions, it is convenient and desirable to introduce the impedance notation in the complex form as:

$$Z = K + j\omega C$$

The forces acting on the journal are:

$$\begin{aligned}
\begin{Bmatrix} F_x \\ F_y \end{Bmatrix} &= -\begin{bmatrix} K_{xx} & K_{xy} \\ K_{yx} & K_{yy} \end{bmatrix}_b \begin{Bmatrix} x - x_s \\ y - y_s \end{Bmatrix} - \begin{bmatrix} C_{xx} & C_{xy} \\ C_{yx} & C_{yy} \end{bmatrix}_b \begin{Bmatrix} \dot{x} - \dot{x}_s \\ \dot{y} - \dot{y}_s \end{Bmatrix} \\
&= -\begin{bmatrix} Z_{xx} & Z_{xy} \\ Z_{yx} & Z_{yy} \end{bmatrix}_b \begin{Bmatrix} x - x_s \\ y - y_s \end{Bmatrix} = -\mathbf{Z}_b(\mathbf{x} - \mathbf{x}_s) \\
&= -\begin{bmatrix} K_{xx} & K_{xy} \\ K_{yx} & K_{yy} \end{bmatrix}_{eq} \begin{Bmatrix} x \\ y \end{Bmatrix} - \begin{bmatrix} C_{xx} & C_{xy} \\ C_{yx} & C_{yy} \end{bmatrix}_{eq} \begin{Bmatrix} \dot{x} \\ \dot{y} \end{Bmatrix} \\
&= -\begin{bmatrix} Z_{xx} & Z_{xy} \\ Z_{yx} & Z_{yy} \end{bmatrix}_{eq} \begin{Bmatrix} x \\ y \end{Bmatrix} = -\mathbf{Z}_{eq}\mathbf{x}
\end{aligned}$$

The motion of the support can be described by the following equations of motion:

$$\begin{bmatrix} M_{xx} & M_{xy} \\ M_{yx} & M_{yy} \end{bmatrix}_s \begin{Bmatrix} \ddot{x}_s \\ \ddot{y}_s \end{Bmatrix} + \begin{bmatrix} C_{xx} & C_{xy} \\ C_{yx} & C_{yy} \end{bmatrix}_s \begin{Bmatrix} \dot{x}_s \\ \dot{y}_s \end{Bmatrix} + \begin{bmatrix} K_{xx} & K_{xy} \\ K_{yx} & K_{yy} \end{bmatrix}_s \begin{Bmatrix} x_s \\ y_s \end{Bmatrix} = -\begin{Bmatrix} F_x \\ F_y \end{Bmatrix}$$

By introducing the dynamic stiffness notation

$$\hat{K} = K - \omega^2 M + j\omega C$$

$$\begin{bmatrix} \hat{K}_{xx} & \hat{K}_{xy} \\ \hat{K}_{yx} & \hat{K}_{yy} \end{bmatrix}_s \begin{Bmatrix} x_s \\ y_s \end{Bmatrix} = -\begin{Bmatrix} F_x \\ F_y \end{Bmatrix} \Rightarrow \hat{\mathbf{K}}_s\mathbf{x}_s = -\mathbf{F}$$

$$\begin{Bmatrix} x_s \\ y_s \end{Bmatrix} = -\begin{bmatrix} \hat{A}_{xx} & \hat{A}_{xy} \\ \hat{A}_{yx} & \hat{A}_{yy} \end{bmatrix}_s \begin{Bmatrix} F_x \\ F_y \end{Bmatrix} \Rightarrow \mathbf{x}_s = -\hat{\mathbf{A}}_s \mathbf{F}$$

Then we have

$$\mathbf{F} = -\left(\mathbf{I} + \mathbf{Z}_b \hat{\mathbf{A}}_s\right)^{-1} \mathbf{Z}_b \mathbf{x} = -\mathbf{Z}_{eq} \mathbf{x}$$

and

$$\mathbf{Z}_{eq} = \left(\mathbf{I} + \mathbf{Z}_b \hat{\mathbf{A}}_s\right)^{-1} \mathbf{Z}_b$$

For an isotropic and undamped system with the bearing stiffness of K_b, and the support mass and stiffness of M_s and K_s respectively,

We have

$$\hat{\mathbf{K}}_s = \begin{bmatrix} K_s - \omega^2 M_s & 0 \\ 0 & K_s - \omega^2 M_s \end{bmatrix}$$

$$\hat{\mathbf{A}}_s = \hat{\mathbf{K}}_s^{-1} = \begin{bmatrix} 1/(K_s - \omega^2 M_s) & 0 \\ 0 & 1/(K_s - \omega^2 M_s) \end{bmatrix}$$

$$\mathbf{Z}_b = \begin{bmatrix} K_b & 0 \\ 0 & K_b \end{bmatrix}$$

and

$$\mathbf{Z}_{eq} = \begin{bmatrix} \left(K_b\left(K_s - \omega^2 M_s\right)\right) / \left(K_b + K_s - \omega^2 M_s\right) & 0 \\ 0 & \left(K_b\left(K_s - \omega^2 M_s\right)\right) / \left(K_b + K_s - \omega^2 M_s\right) \end{bmatrix}$$

then

$$K_{eq} = \frac{K_b\left(K_s - \omega^2 M_s\right)}{K_b + K_s - \omega^2 M_s}$$

Several special cases are discussed below:

Case 1 - Zero Support Mass

When the support mass is very small, the effective stiffness is equivalent to two springs in series.

$$K_{eq} = \frac{K_b K_s}{K_b + K_s} = \frac{1}{1/K_b + 1/K_s}$$

Case 2 – High Support Stiffness

When $K_s \to \infty, \quad K_{eq} \to K_b$

Case 3 – High Support Inertia

When $\omega^2 M_s \to \infty, \quad K_{eq} \to K_b$

Case 4 – Support Resonant Condition, $\omega^2 = \dfrac{K_s}{M_s}$

When the support is in resonance with its stiffness, the effective stiffness goes to zero.

$$K_s - \omega^2 M_s = 0, \quad K_{eq} = 0$$

Case 5 – Tuned Vibration Absorber, $\omega^2 = \dfrac{K_b + K_s}{M_s}$

When the support mass is in resonance with the combination of bearing stiffness and support stiffness, the effective stiffness becomes infinite.

$$K_b + K_s - \omega^2 M_s = 0, \quad K_{eq} \to \infty$$

Case 6 - $\omega^2 > \dfrac{K_s}{M_s}$

When the synchronous frequency is higher than the support natural frequency, the effective stiffness may be negative.

$$\omega^2 > \frac{K_s}{M_s} \Rightarrow K_s - \omega^2 M_s < 0$$

Rotor Orbit Analysis

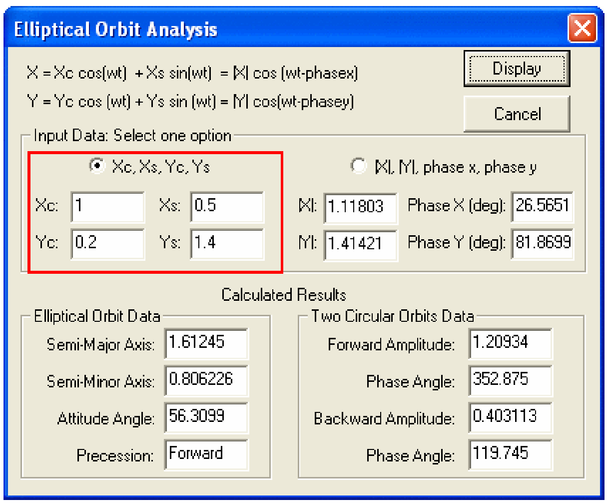
There are many types of excitations acting on a rotor system and they are frequently periodic, or can be approximated closely by the summation of periodic forces. The steady state response to a periodic excitation is a periodic motion. Any periodic motion can be represented by a series of harmonic motion. For a single harmonic motion, the rotor motion is an elliptical orbit, the total motion of a periodic excitation can be more complicated. For more information on the rotor orbit, please refer to the book by Chen and Gunter (2005), Chapter 2.

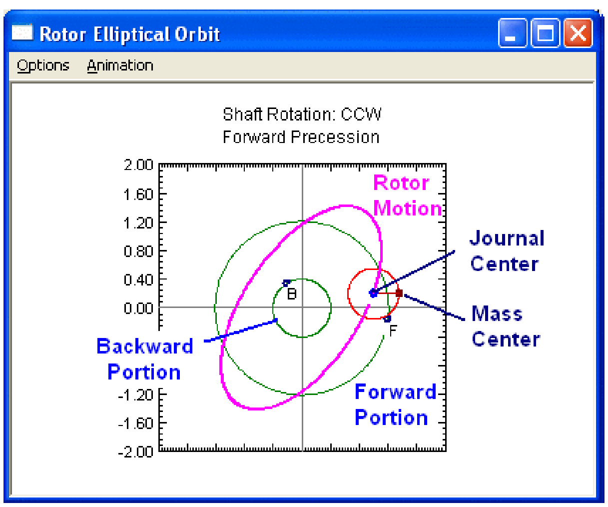
Elliptical Orbit

The rotor motion of most common interest is a harmonic motion with a whirl frequency. At each finite element station, the rotor translational motion has the form:

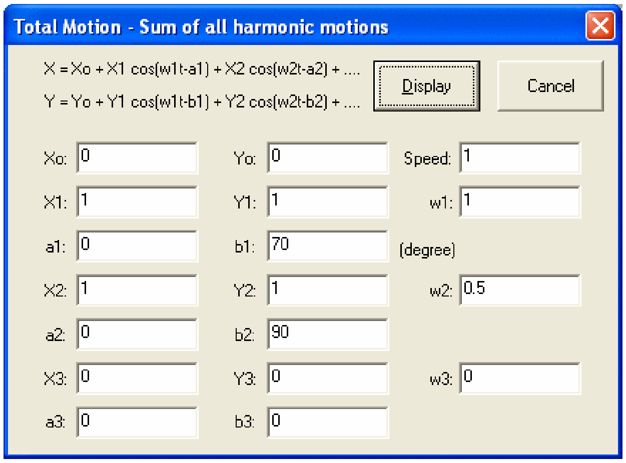
$$x(t) = x_c \cos\omega t + x_s \sin \omega t = |x| \cos(\omega t - \phi_x)$$
$$y(t) = y_c \cos\omega t + y_s \sin \omega t = |y| \cos(\omega t - \phi_y)$$

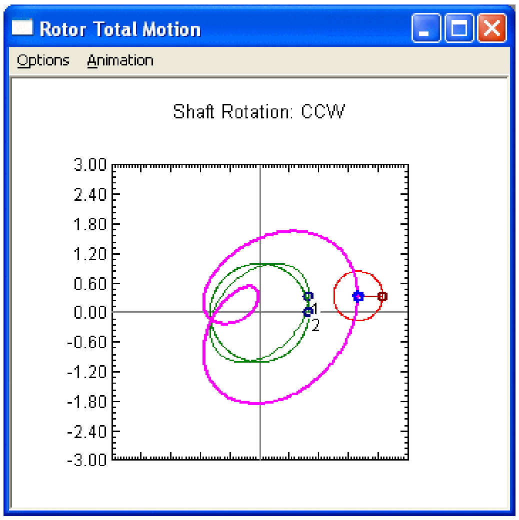
This represents an elliptical orbit. The elliptical orbit can be decomposed into two circular orbits: one is a forward circular orbit and the other is a backward circular orbit.





Total Motion


Total Motion - Sum of all harmonic motions
X = Xo + X1 cos(w1t-a1) + X2 cos(w2t-a2) + ...
Y = Yo + Y1 cos(w1t-b1) + Y2 cos(w2t-b2) + ...
Display
Cancel
Xo: 0
Yo: 0
Speed: 1
X1: 1
Y1: 1
w1: 1
a1: 0
b1: 70
(degree)
X2: 1
Y2: 1
w2: 0.5
a2: 0
b2: 90
X3: 0
Y3: 0
w3: 0
a3: 0
b3: 0


Rotor Total Motion
Options
Animation
Shaft Rotation: CCW
3.00
2.40
1.80
1.20
0.60
0.00
-0.60
-1.20
-1.80
-2.40
-3.00
1
2

Balancing Calculation

Influence coefficient method is used in the balancing calculation. The theory is based on two papers:

1. Tessarzik, J. M., Badgley, R. H., and Anderson, W. J., 1972, Flexible Rotor Balancing by the Exact Point-Speed Influence Coefficient Method, ASME Journal of Engineering for Industry, Feb., 1972, pp 148-158.
2. Lund, J. W. and Tonnesen, J., 1972, Analysis and Experiments on Multi-Plane Balancing of a Flexible Rotor, ASME Journal of Engineering for Industry, Feb., 1972, pp 233-242.

Since the least square method is used to solve the simultaneous equations, the Number of Speed Points times the Number of Measured Probes must be greater than or equal to the Number of Balancing Planes. (NsXNm >= Nb).

To use the influence coefficient method, no prior knowledge in rotor mode is required. However, trial weights are required to obtain the influence coefficients.

All the inputs are self-explanatory. They are briefly described below:

1. Number of Balancing Planes: Nb

The balancing planes are the planes along the rotor where the trial weights and balancing corrections are applied. Note that the trial weight can be **left-in** or **removed** after the trial run.

2. Number of Measured Probes: Nm

The measurement probes are where the vibrations are taken and recorded. The purpose of the balancing is to find the optimal balancing corrections at the balancing planes such that the vibrations at the measurement probes are minimized.

3. Number of Speeds/Cases: Ns

The number of speeds or cases allows for different speeds or cases, such as idle speed, full speed, full load, unloads, etc…

4. Runout Compensation

Runout can be included or excluded in this balancing calculation.

5. Comments

Up to 3 comment lines can be used to describe the system under study.

6. Shaft Rotation: CCW or CW

7. Phase: Lag or Lead

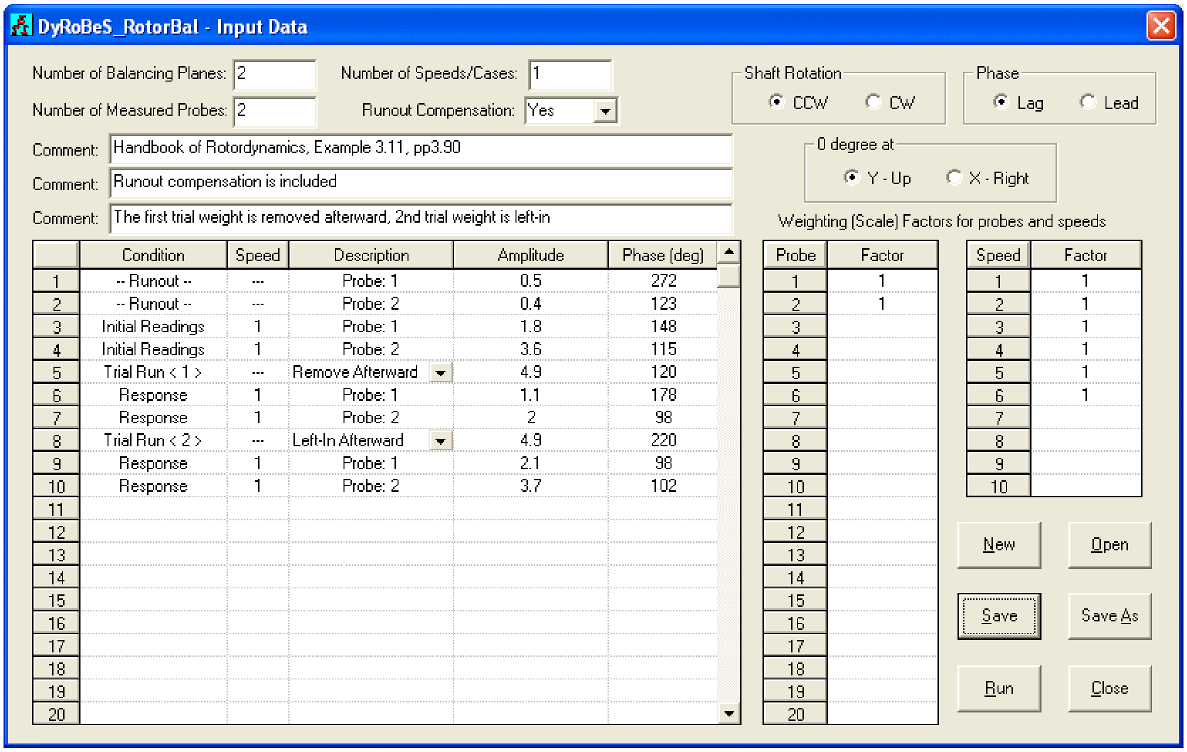
8. 0 degree: Up or Right.

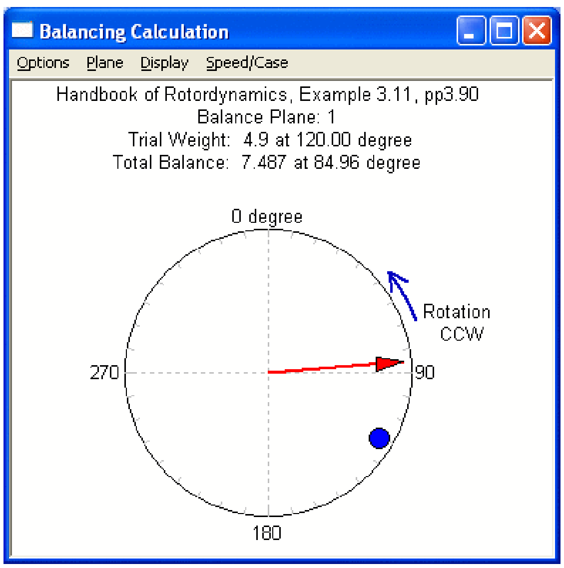
0 degree position defines the reference mark where all the angles (phases) are measured from.

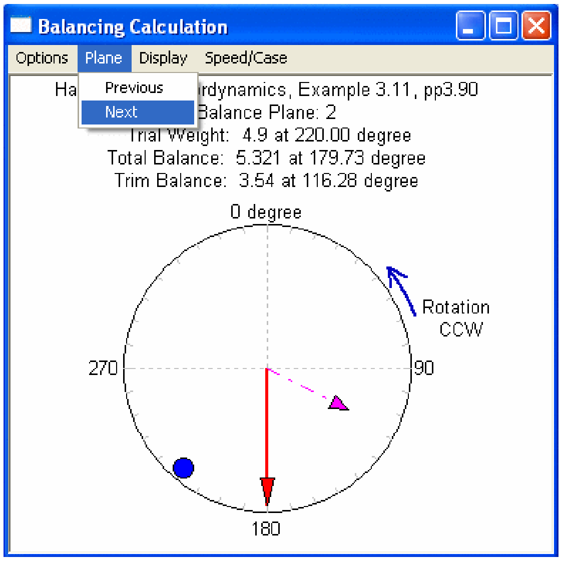
9. Weighting Factors

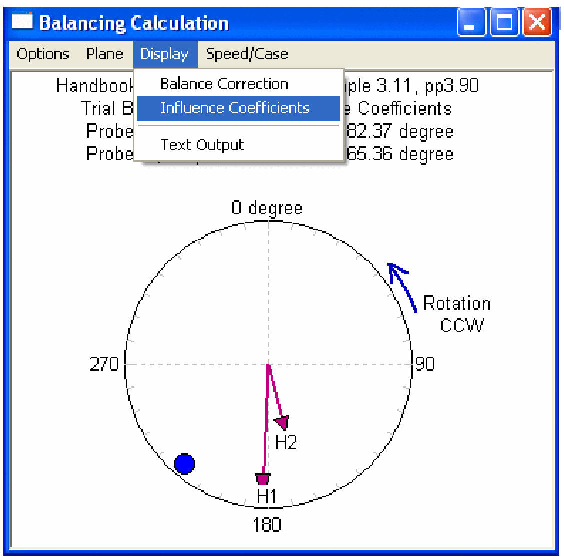
Weighting factor allows one to strengthen or weaken the data from the measurement probes or speeds. For example, one may use higher weighting factors for the probes where the critical components are located and/or speeds where the rotor will be operated most of the time. Weighting factor zero indicates that the specific probe data will not be included in the calculation.

Example 1:


DyRoBeS_RotorBal - Input Data
Number of Balancing Planes: 2
Number of Speeds/Cases: 1
Number of Measured Probes: 2
Runout Compensation: Yes
Shaft Rotation
CCW
CW
Phase
Lag
Lead
Comment: Handbook of Rotordynamics, Example 3.11, pp3.90
Comment: Runout compensation is included
Comment: The first trial weight is removed afterward, 2nd trial weight is left-in
0 degree at
Y - Up
X - Right
Weighting (Scale) Factors for probes and speeds
Condition | Speed | Description | Amplitude | Phase (deg)
1 -- Runout -- | --- | Probe: 1 | 0.5 | 272
2 -- Runout -- | --- | Probe: 2 | 0.4 | 123
3 Initial Readings | 1 | Probe: 1 | 1.8 | 148
4 Initial Readings | 1 | Probe: 2 | 3.6 | 115
5 Trial Run < 1 > | --- | Remove Afterward | 4.9 | 120
6 Response | 1 | Probe: 1 | 1.1 | 178
7 Response | 1 | Probe: 2 | 2 | 98
8 Trial Run < 2 > | --- | Left-In Afterward | 4.9 | 220
9 Response | 1 | Probe: 1 | 2.1 | 98
10 Response | 1 | Probe: 2 | 3.7 | 102
11 12 13 14 15 16 17 18 19 20
Probe | Factor
1 | 1
2 | 1
3 4 5 6 7 8 9 10 11 12 13 14 15 16 17 18 19 20
Speed | Factor
1 | 1
2 | 1
3 | 1
4 | 1
5 | 1
6 | 1
7 8 9 10
New
Open
Save
Save As
Run
Close


Balancing Calculation
Options Plane Display Speed/Case
Handbook of Rotordynamics, Example 3.11, pp3.90
Balance Plane: 1
Trial Weight: 4.9 at 120.00 degree
Total Balance: 7.487 at 84.96 degree
0 degree
Rotation
CCW
270
90
180






```
Handbook of Rotordynamics Example 3.11, pp 3.90
Runout Compensation is included
The first trial weight is removed afterward, 2nd trial weight is left-in

   *****  Number of Speeds or Cases    :    1
   *****  Number of Balancing Planes   :    2
   *****  Number of Measurement Probes:    2

   *****  Runout (slow-roll vectors)  *****
      Probe        Amplitude      Phase Angle
        1           0.50000         272.00
        2           0.40000         123.00

   ======  Initial Response (Without Trails)  ======
      Speed    Probe        Amplitude      Phase Angle
        1        1           1.8000         148.00
        1        2           3.6000         115.00

   **************  Trial Unbalance Run: 1  **************
      Plane        Amplitude      Phase Angle   Afterward
        1            4.9000         120.00       Remove

   ---------  Response to Trial Unbalance  ---------
      Speed    Probe        Amplitude      Phase Angle
        1        1           1.1000         178.00
        1        2           2.0000         98.000

   **************  Trial Unbalance Run: 2  **************
      Plane        Amplitude      Phase Angle   Afterward
        2            4.9000         220.00       Left-In

   ---------  Response to Trial Unbalance  ---------
```

```
    Speed    Probe       Amplitude      Phase Angle
      1        1          2.1000         98.000
      1        2          3.7000         102.00

 ***  Weighting Factors for probes and speeds ***
      Probe      Weighting Factor
        1             1.0000
        2             1.0000

      Speed      Weighting Factor
        1             1.0000

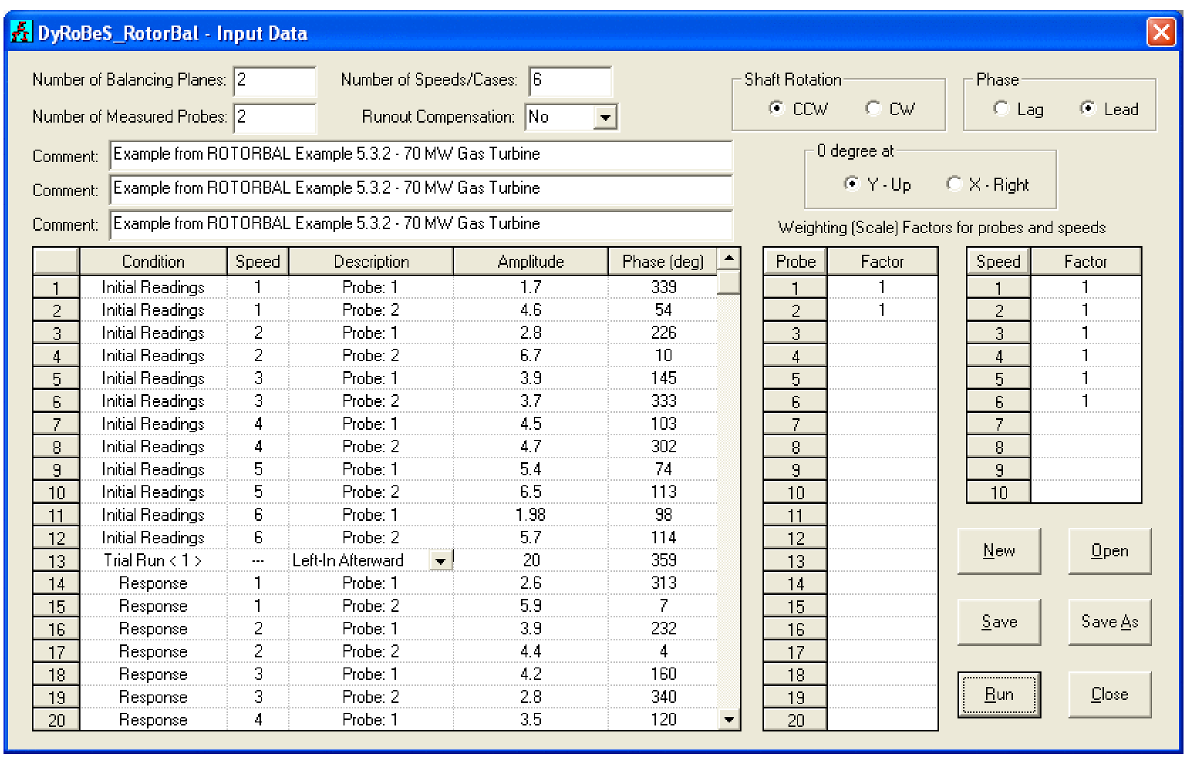
 ===================================================
 <<<<<<<<  Total Balance Correction  >>>>>>>>
   Correction Required to Balance the Rotor
    Plane No.     Amplitude      Phase Angle
       1           7.4873          84.957
       2           5.3209          179.73


 <<<<<<<<<  Trim Balance Correction  >>>>>>>>>>
   Correction Required if Trial Weight Left-in
   Trim Balance = Total Balance - Trial Weight
    Plane No.     Amplitude      Phase Angle
       2           3.5402          116.28
 ===================================================
 ___________________________________________________

 **********  The Influence Coefficients  ***********
                                 Influence Coef.
   Trial-Run  Speed  Probe   Amplitude     Phase
       1        1      1      0.20617      175.
       1        1      2      0.36446      194.
       2        1      1      0.34092      182.
       2        1      2      0.16986      165.
 ___________________________________________________

 =====================  Predicted Residual Response =======================
                        ----- WithOUT Runout -----   ------ With Runout ------
  Speed    Probe       Amplitude     Phase Angle   Amplitude      Phase Angle
    1        1          0.0000        0.0000       0.50000         272.00
    1        2          0.0000        0.0000       0.40000         123.00
 ==========================================================================
```

Example 2:



DyRoBeS_RotorBal - Input Data

Number of Balancing Planes: 2
Number of Speeds/Cases: 6
Number of Measured Probes: 2
Runout Compensation: No
Shaft Rotation: CCW (selected), CW
Phase: Lag, Lead (selected)
0 degree at: Y - Up (selected), X - Right
Comment: Example from ROTORBAL Example 5.3.2 - 70 MW Gas Turbine
Comment: Example from ROTORBAL Example 5.3.2 - 70 MW Gas Turbine
Comment: Example from ROTORBAL Example 5.3.2 - 70 MW Gas Turbine

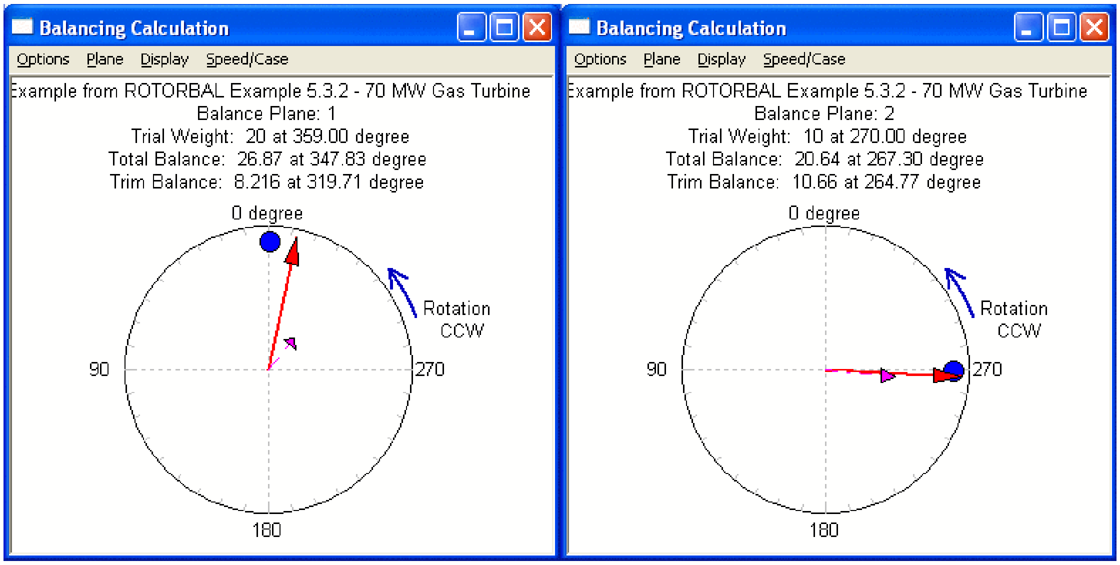
	Condition	Speed	Description	Amplitude	Phase (deg)
1	Initial Readings	1	Probe: 1	1.7	339
2	Initial Readings	1	Probe: 2	4.6	54
3	Initial Readings	2	Probe: 1	2.8	226
4	Initial Readings	2	Probe: 2	6.7	10
5	Initial Readings	3	Probe: 1	3.9	145
6	Initial Readings	3	Probe: 2	3.7	333
7	Initial Readings	4	Probe: 1	4.5	103
8	Initial Readings	4	Probe: 2	4.7	302
9	Initial Readings	5	Probe: 1	5.4	74
10	Initial Readings	5	Probe: 2	6.5	113
11	Initial Readings	6	Probe: 1	1.98	98
12	Initial Readings	6	Probe: 2	5.7	114
13	Trial Run < 1 >	---	Left-In Afterward	20	359
14	Response	1	Probe: 1	2.6	313
15	Response	1	Probe: 2	5.9	7
16	Response	2	Probe: 1	3.9	232
17	Response	2	Probe: 2	4.4	4
18	Response	3	Probe: 1	4.2	160
19	Response	3	Probe: 2	2.8	340
20	Response	4	Probe: 1	3.5	120

Weighting (Scale) Factors for probes and speeds

Probe	Factor
1	1
2	1
3	
4	
5	
6	
7	
8	
9	
10	
11	
12	
13	
14	
15	
16	
17	
18	
19	
20	

Speed	Factor
1	1
2	1
3	1
4	1
5	1
6	1
7	
8	
9	
10	

New | Open | Save | Save As | Run | Close




```
Example from ROTORBAL Example 5.3.2 - 70 MW Gas Turbine
2 probes at 6 speeds
No runout

   *****  Number of Speeds or Cases   :   6
   *****  Number of Balancing Planes  :   2
   *****  Number of Measurement Probes:   2

   *****  NO Runout Compensation

   ======  Initial Response (Without Trails)  ======
      Speed   Probe       Amplitude     Phase Angle
        1       1           1.7000        339.00
        1       2           4.6000        54.000
        2       1           2.8000        226.00
        2       2           6.7000        10.000
        3       1           3.9000        145.00
        3       2           3.7000        333.00
        4       1           4.5000        103.00
        4       2           4.7000        302.00
        5       1           5.4000        74.000
        5       2           6.5000        113.00
        6       1           1.9800        98.000
        6       2           5.7000        114.00

   **************  Trial Unbalance Run: 1  **************
      Plane       Amplitude      Phase Angle   Afterward
        1           20.000         359.00       Left-In

   ---------  Response to Trial Unbalance  ---------
      Speed   Probe       Amplitude     Phase Angle
        1       1           2.6000        313.00
        1       2           5.9000        7.0000
        2       1           3.9000        232.00
        2       2           4.4000        4.0000
        3       1           4.2000        160.00
        3       2           2.8000        340.00
        4       1           3.5000        120.00
        4       2           5.4000        325.00
        5       1           4.1000        73.000
        5       2           3.7000        97.000
        6       1           1.5000        141.00
        6       2           3.1000        99.000

   **************  Trial Unbalance Run: 2  **************
      Plane       Amplitude      Phase Angle   Afterward
        2           10.000         270.00       Left-In

   ---------  Response to Trial Unbalance  ---------
      Speed   Probe       Amplitude     Phase Angle
        1       1           1.8000        319.00
        1       2           4.3000        15.000
        2       1           3.1000        244.00
        2       2           3.4000        9.0000
        3       1           3.2000        107.00
        3       2           1.9000        329.00
        4       1           2.4000        122.00
```

```
      4       2          4.4000         330.00
      5       1          2.4000         61.000
      5       2          3.5000         101.00
      6       1          1.0200         170.00
      6       2          3.1100         104.00

 ***  Weighting Factors for probes and speeds ***
      Probe     Weighting Factor
        1            1.0000
        2            1.0000

      Speed     Weighting Factor
        1            1.0000
        2            1.0000
        3            1.0000
        4            1.0000
        5            1.0000
        6            1.0000

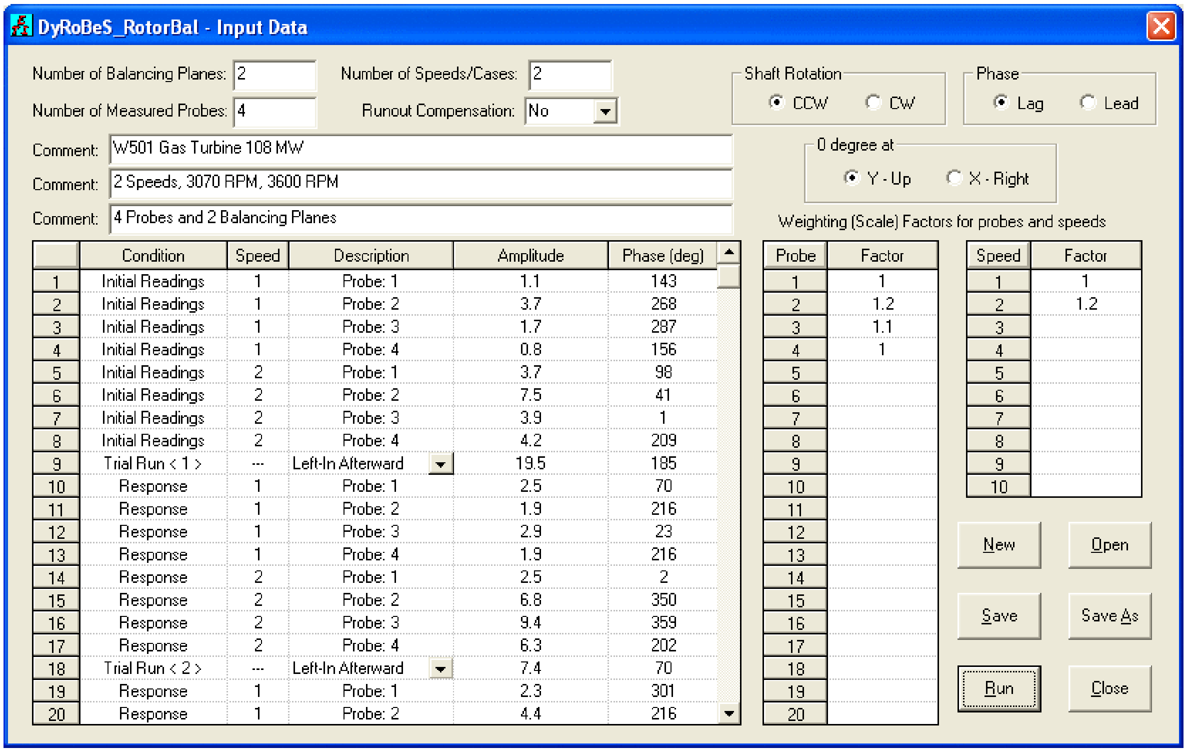
 ===================================================
 <<<<<<<<  Total Balance Correction  >>>>>>>>
   Correction Required to Balance the Rotor
    Plane No.    Amplitude     Phase Angle
       1          26.867         347.83
       2          20.639         267.30


 <<<<<<<<<  Trim Balance Correction  >>>>>>>>>>
   Correction Required if Trial Weight Left-in
   Trim Balance = Total Balance - Trial Weight
    Plane No.    Amplitude     Phase Angle
       1          8.2159         319.71
       2          10.660         264.77
 ===================================================
 ___________________________________________________

 **********  The Influence Coefficients  ***********
                              Influence Coef.
   Trial-Run  Speed  Probe   Amplitude    Phase
       1        1      1      0.65281E-01  279.
       1        1      2      0.21766      317.
       1        2      1      0.57655E-01  248.
       1        2      2      0.11846      202.
       1        3      1      0.54915E-01  228.
       1        3      2      0.49103E-01  134.
       1        4      1      0.77078E-01  242.
       1        4      2      0.10636       26.
       1        5      1      0.65130E-01  258.
       1        5      2      0.15575      313.
       1        6      1      0.67568E-01  230.
       1        6      2      0.14110      312.
       2        1      1      0.83143E-01  210.
       2        1      2      0.17475      257.
       2        2      1      0.10809      105.
       2        2      2      0.10554      258.
       2        3      1      0.34210      118.
       2        3      2      0.10027      271.
       2        4      1      0.11046       26.
       2        4      2      0.10867      214.
       2        5      1      0.18221      359.
       2        5      2      0.32108E-01  318.
       2        6      1      0.78362E-01   12.
       2        6      2      0.27106E-01  279.
 ___________________________________________________

======================= Predicted Residual Response ========================
                      ----- WithOUT Runout -----    ------ With Runout ------
   Speed    Probe      Amplitude       Phase Angle   Amplitude      Phase Angle
     1        1          1.0968          310.43        1.0968          310.43
     1        2          2.6067          358.29        2.6067          358.29
     2        1          2.8744          257.03        2.8744          257.03
     2        2          1.7963          40.778        1.7963          40.778
     3        1          4.8509          66.019        4.8509          66.019
     3        2         0.73525          317.09       0.73525          317.09
     4        1          1.6051          154.41        1.6051          154.41
     4        2          4.3282          341.01        4.3282          341.01
     5        1         0.55031          343.11       0.55031          343.11
     5        2          2.1098          114.00        2.1098          114.00
     6        1          1.6347          207.09        1.6347          207.09
     6        2          2.0997          118.93        2.0997          118.93
============================================================================
============================================================================
```

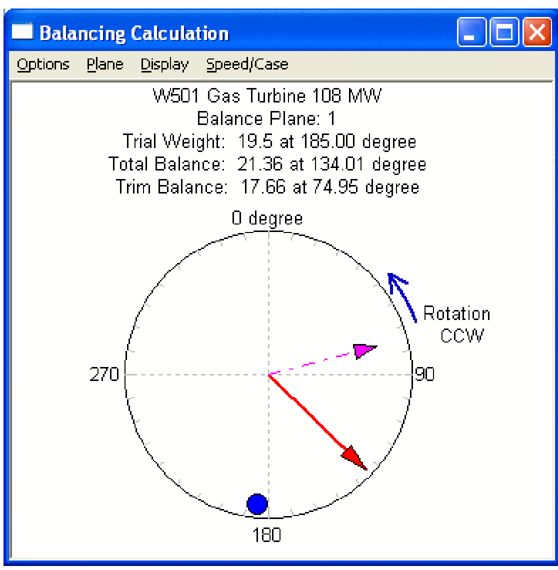
Example 3:

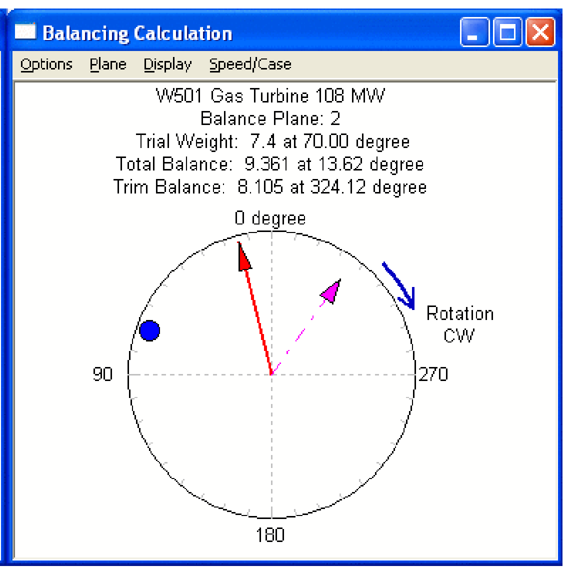


	Condition	Speed	Description	Amplitude	Phase (deg)
1	Initial Readings	1	Probe: 1	1.1	143
2	Initial Readings	1	Probe: 2	3.7	268
3	Initial Readings	1	Probe: 3	1.7	287
4	Initial Readings	1	Probe: 4	0.8	156
5	Initial Readings	2	Probe: 1	3.7	98
6	Initial Readings	2	Probe: 2	7.5	41
7	Initial Readings	2	Probe: 3	3.9	1
8	Initial Readings	2	Probe: 4	4.2	209
9	Trial Run < 1 >	---	Left-In Afterward	19.5	185
10	Response	1	Probe: 1	2.5	70
11	Response	1	Probe: 2	1.9	216
12	Response	1	Probe: 3	2.9	23
13	Response	1	Probe: 4	1.9	216
14	Response	2	Probe: 1	2.5	2
15	Response	2	Probe: 2	6.8	350
16	Response	2	Probe: 3	9.4	359
17	Response	2	Probe: 4	6.3	202
18	Trial Run < 2 >	---	Left-In Afterward	7.4	70
19	Response	1	Probe: 1	2.3	301
20	Response	1	Probe: 2	4.4	216

Probe	Factor
1	1
2	1.2
3	1.1
4	1
5	
6	
7	
8	
9	
10	
11	
12	
13	
14	
15	
16	
17	
18	
19	
20	

Speed	Factor
1	1
2	1.2
3	
4	
5	
6	
7	
8	
9	
10	






```
W501 Gas Turbine 108 MW
2 Speeds, 3070 RPM, 3600 RPM
4 Probes and 2 Balancing Planes

*****  Number of Speeds or Cases    :   2
*****  Number of Balancing Planes   :   2
*****  Number of Measurement Probes:    4

*****  NO Runout Compensation

======  Initial Response (Without Trails)  ======
   Speed    Probe         Amplitude      Phase Angle
     1        1             1.1000         143.00
     1        2             3.7000         268.00
     1        3             1.7000         287.00
     1        4            0.80000         156.00
     2        1             3.7000         98.000
     2        2             7.5000         41.000
     2        3             3.9000         1.0000
     2        4             4.2000         209.00
```

```
************** Trial Unbalance Run: 1 **************
   Plane      Amplitude     Phase Angle   Afterward
     1         19.500         185.00       Left-In

---------  Response to Trial Unbalance  ---------
   Speed   Probe      Amplitude     Phase Angle
     1       1          2.5000        70.000
     1       2          1.9000        216.00
     1       3          2.9000        23.000
     1       4          1.9000        216.00
     2       1          2.5000        2.0000
     2       2          6.8000        350.00
     2       3          9.4000        359.00
     2       4          6.3000        202.00

************** Trial Unbalance Run: 2 **************
   Plane      Amplitude     Phase Angle   Afterward
     2         7.4000         70.000       Left-In

---------  Response to Trial Unbalance  ---------
   Speed   Probe      Amplitude     Phase Angle
     1       1          2.3000        301.00
     1       2          4.4000        216.00
     1       3          2.5000        294.00
     1       4         0.80000        139.00
     2       1          1.7000        12.000
     2       2          5.6000        355.00
     2       3          5.3000        344.00
     2       4          3.8000        181.00

***  Weighting Factors for probes and speeds ***
     Probe     Weighting Factor
       1           1.0000
       2           1.2000
       3           1.1000
       4           1.0000

     Speed     Weighting Factor
       1           1.0000
       2           1.2000

===================================================
<<<<<<<<  Total Balance Correction  >>>>>>>>
  Correction Required to Balance the Rotor
   Plane No.    Amplitude     Phase Angle
      1          21.356         134.01
      2          9.3609         13.616


<<<<<<<<<  Trim Balance Correction  >>>>>>>>>>
  Correction Required if Trial Weight Left-in
  Trim Balance = Total Balance - Trial Weight
   Plane No.    Amplitude     Phase Angle
      1          17.665         74.947
      2          8.1048         324.12
===================================================
___________________________________________________

**********  The Influence Coefficients  ***********
  -- Before application of Weighting Factors --
                               Influence Coef.
  Trial-Run  Speed  Probe   Amplitude    Phase
      1        1      1      0.12406      219.
      1        1      2      0.15077      294.
      1        1      3      0.18008      227.
      1        1      4      0.84732E-01   56.
      1        2      1      0.23984      125.
      1        2      2      0.31737       95.
      1        2      3      0.28226      173.
      1        2      4      0.11241        3.
      2        1      1      0.58558      204.
      2        1      2      0.33784      146.
      2        1      3      0.51293      174.
      2        1      4      0.25519      350.
      2        2      1      0.11851       92.
      2        2      2      0.17773       78.
      2        2      3      0.60743      127.
      2        2      4      0.41498      338.
___________________________________________________

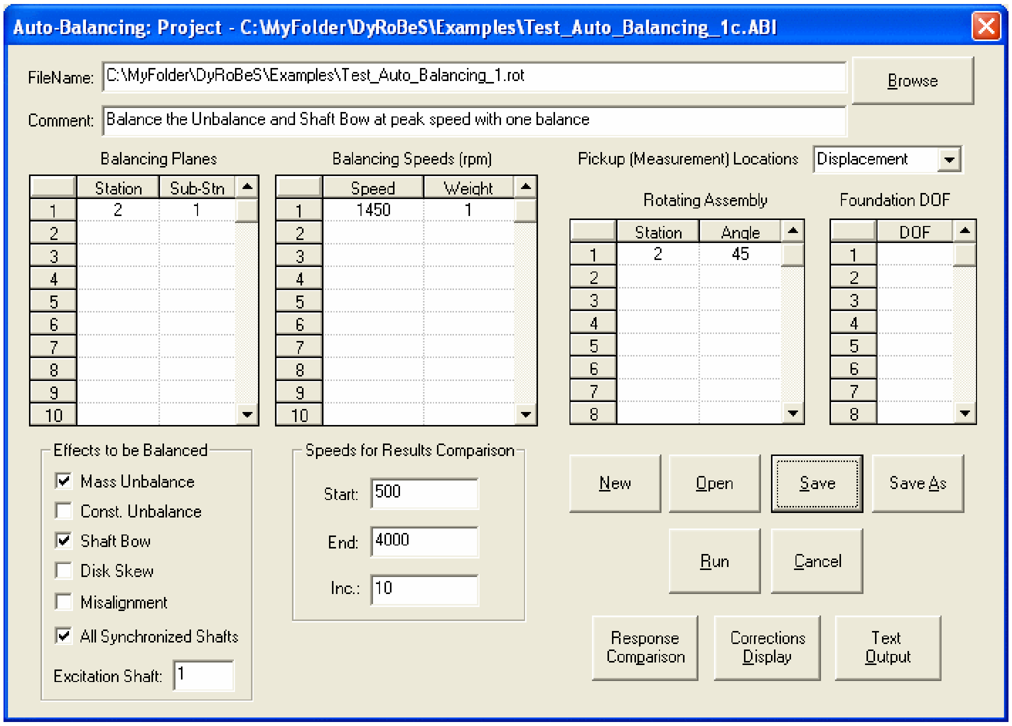
=======================  Predicted Residual Response ========================
                      ----- WithOUT Runout -----    ------ With Runout ------
```

Speed	Probe	Amplitude	Phase Angle	Amplitude	Phase Angle
1	1	3.9669	49.648	3.9669	49.648
1	2	1.9065	348.46	1.9065	348.46
1	3	2.2628	79.442	2.2628	79.442
1	4	0.22558	305.47	0.22558	305.47
2	1	1.8109	9.5156	1.8109	9.5156
2	2	1.8740	232.93	1.8740	232.93
2	3	3.3169	159.79	3.3169	159.79
2	4	1.8632	30.989	1.8632	30.989

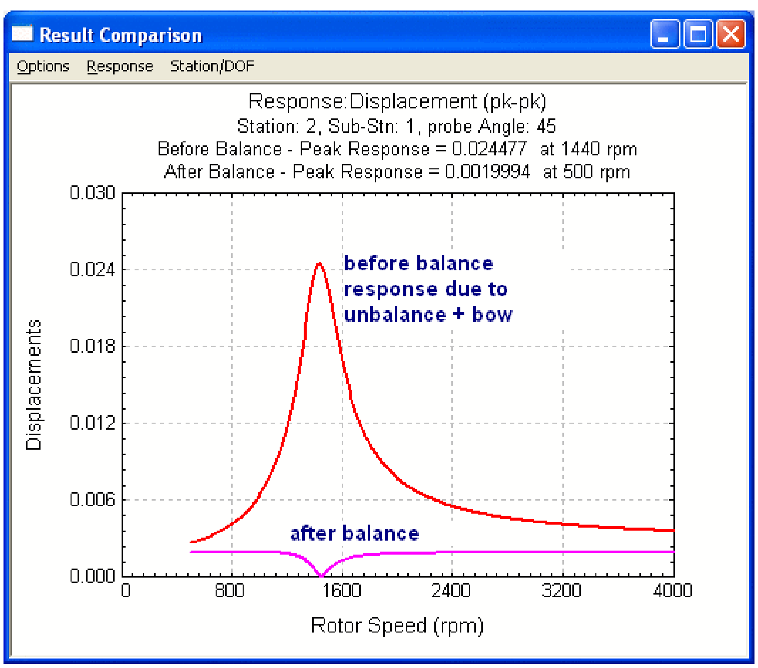
==
==

Auto-Balancing Calculation

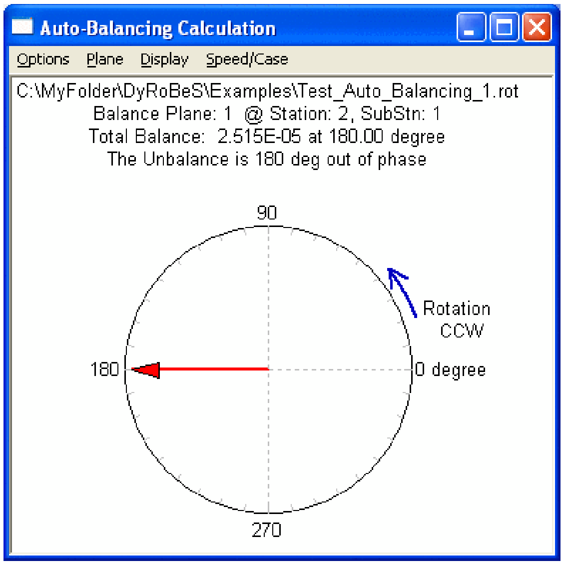
In contrast to the Influence coefficient method, the auto-balancing method requires full knowledge in the rotor-bearing model including all the harmonic excitations. An example is presented below. The rotor system has one mass unbalance and shaft bow at the center. Now, we will try to balance both excitations with one balance. The input screen is shown below. The inputs are self-explanatory.



The response comparison button displays the responses before and after the balancing.



The balance magnitude and angular location is shown below.



There are many examples are included in the DyRoBeS\Example directory.

See also Balancing Calculation.

Error Code Description

Note that the same error codes may be used in different Message

Error CodeMessageDescription

Error in reading Shaft Element Data

1001DATAINShaft number > max allowable number of shafts = 25

1002DATAINStarting station number <= 0

1003DATAINStarting station number > ending station number

1004DATAINStation number > max station number = 501

1005DATAINElement number > max element number = 500

1006DATAINSubelement number > max subelement number = 20

DATAINLayer number > max layer number = 10

Error in reading Rigid Disk Data

2001DATAINDisk station number > max station number = 501

Error in reading Unbalance Data

3001DATAINElement number > max element number = 100

3002DATAINSubelement number > max subelement number = 20

Error in reading Bearing Data

4001DATAINStation I > max station number = 501

4002DATAINStation J > max station number = 501

Error in reading Flexible Support Data

5001DATAINStation number > max station number = 501

Error in reading Axial Forces and Torques

6001DATAINStation number <= 0

6002DATAINStation number > max station number = 501

Error in reading Static Loads

7001DATAINStation number > max station number = 501

Error in reading Boundary Conditions

Shear/Moment Release

8001DATAINStation number <= 0

8002DATAINStation number > max station number = 501

Error in reading Geometric Misalignment Data

9002DATAINStation number > max station number = 501

Error in reading Shaft Bow Data

9019DATAINStation number > max station number = 501

Error in reading Time Forcing Function Data

9020DATAINStation number > max station number = 501

1001ELEMTXFail to perform subelement static condensation

1006ELEMTXFail to perform shear/moment release condensation

IUNITIOFILEFail to open/close files

1001PREPAR One of the following is zero

no. of shafts, no. of elements,

no. of materials, no. of degrees of freedom

1002LENGTHElement length <= 0

1003RADIUSInner Radius >= Outer Radius

3000BRGINTExceeds Max speed increment points (max 3000 points)

1001CSANALError in solving the Eigenvalues (Critical Speed Analysis)

1001CSPMAPError in solving the Eigenvalues (Critical Speed Map)

9001STANALFail to solve KX=F in Static Deflection,

Stiffness Matrix, K, is singular

5020SFANALNot enough memory for squeeze Film damper (SFD) analysis

5010SFDCCOStiffness matrix is singular in SFD analysis

5020SFDCCOConverged SFD orbits are questionable

5075SFDSOLFail to converge the solution in SFD analysis

1 TRANALNumber of bearings <=0

1001TRANALMass matrix is singular

-1 DIFFEQFail to perform time integration

-7EQAXB2Stiffness matrix is singular

1001URANALFail to solve kx=f in Unbalance Response Analysis

Matrix is singular

1001WSANALMass matrix is singular, density = 0?

2002WSANALFail to solve Eigenvalue problem

-1MODULAUnknown analysis type

1001ALLOCANot enough computer memory for allocation

1002DEALLOFail to de-allocate the computer memory

2001MAINPSNot enough computer memory for allocation

2002MAINPSFail to de-allocate the computer memory

3001MAINPSNot enough computer memory for allocation

3002MAINPSFail to de-allocate the computer memory

4001MAINPSNot enough computer memory for allocation

4002MAINPSFail to de-allocate the computer memory

5001MAINPSNot enough computer memory for allocation

5002MAINPSFail to de-allocate the computer memory

6002MAINSFNot enough computer memory for allocation

7001MAINPSNot enough computer memory for allocation

7002MAINPSFail to de-allocate the computer memory

2002MAINPSNon-Linear bearing exists in Linear Analysis

3001MAINPSNumber of bearings <=0,

In Critical Speed Map Analysis

2003Run>>>No forces specified in the Synchronous Response Analysis

6001NSF =0Number of Squeeze Film Dampers = 0

6001SFANALNumber of Squeeze Film Damper >10

The most common errors in the modeling and analysis are:

1. The mass matrix is singular. Check your material properties -density

2. The system is unconstrained. Check your bearings and supports.

3. Nonlinear bearings are provided, however, linear analysis is requested.

4. Missing element numbers in the model, use **ReNumbe**r button to correct this problem.

5. Units inconsistent

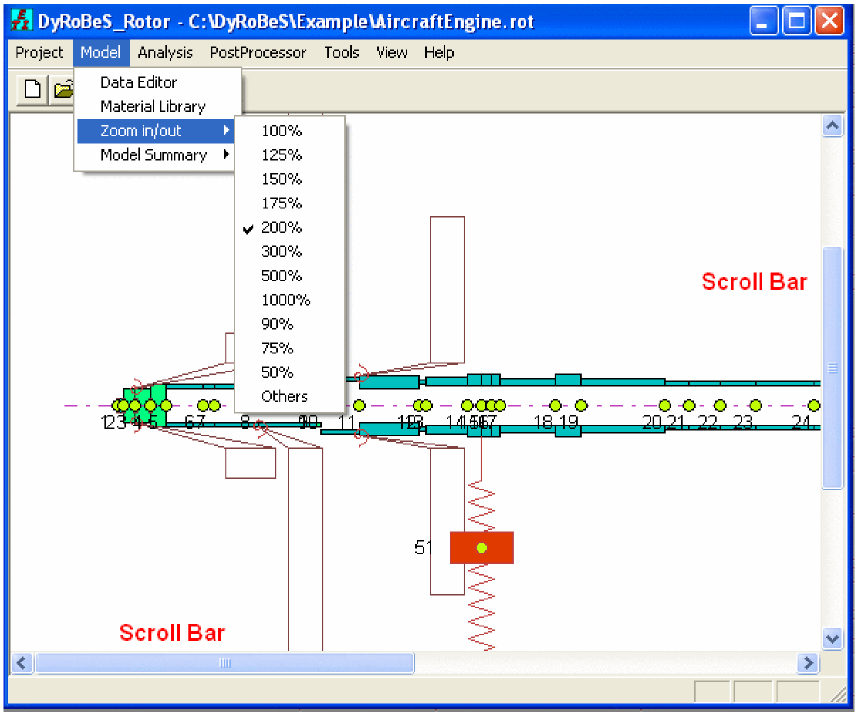
Always check the model using the Model Summary and check the equivalent rigid body properties with the anticipated values. Zoom in the model to see any missing elements or subelements.

Examples

A number of examples installed under **DyRoBeS\Examples** directory are included in the software package to demonstrate the program's features and capabilities. The user is encouraged to go through the examples and verify the results.

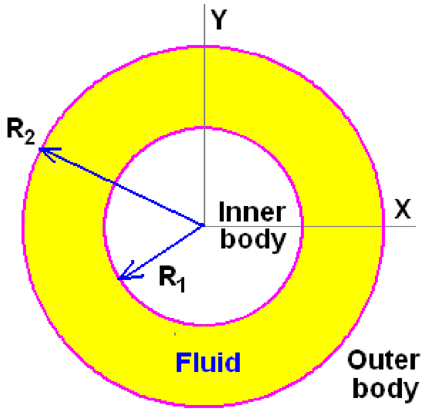
Zoom In/Out

The option allows you to zoom in/out your model and results in the screen. You will need to use scroll bars to scroll the figure when you zoom in your model.



Fluid Coupling – Added Mass

There are some applications in rotating machinery, such as pumps, the rotor is within another cylinder (pipe) and there is fluid filled in the annular space between two cylinders, as illustrated in the following figure.



When one structure vibrates, the adjacent structure also vibrates, as if there is a connection between these two structures. The coupling is based on the dynamic response of two points, or finite element stations at the centerlines of two cylinders, connected by the constrained fluid. The motion is assumed to be small and the fluid is incompressible. The two points are coupled by the added mass, often called fluid dynamic coupling. It is ideally treated as a pseudo bearing which connects two structure points with added mass matrix in the translational motion. Assuming the finite element station of the inner cylinder is station *i* with displacements of (x_i, y_i) and finite element station of the outer cylinder is station *j* with displacements of (x_i, y_i), the added dynamic force (mass matrix) has the form:

$$\left\{\begin{bmatrix} m_{11} & 0 & m_{13} & 0 \\ 0 & m_{22} & 0 & m_{24} \\ m_{31} & 0 & m_{33} & 0 \\ 0 & m_{42} & 0 & m_{44} \end{bmatrix}\begin{Bmatrix} \ddot{x}_i \\ \ddot{y}_i \\ \ddot{x}_j \\ \ddot{y}_j \end{Bmatrix}\right\}$$

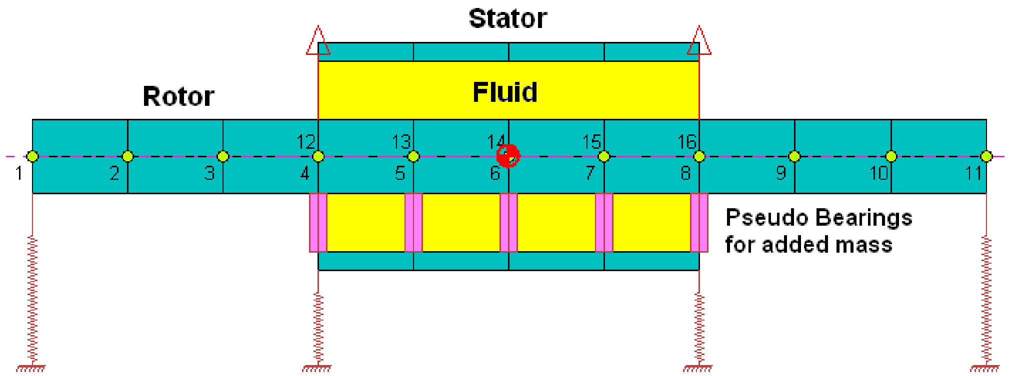
Assuming the inner cylinder has a radius of R_1 and the outer cylinder has a radius of R_2. The length of the cylinder is *L*, the density of the fluid is ρ. The values for ***m*** are:

$$m_{11} = m_{22} = \rho\pi L R_1^2\left(\frac{R_2^2 + R_1^2}{R_2^2 - R_1^2}\right) = \text{added mass of inner cylinder}$$

$$m_{33} = m_{44} = \rho\pi L R_2^2\left(\frac{R_2^2 + R_1^2}{R_2^2 - R_1^2}\right) = \text{added mass of outer cylinder}$$

$$m_{13} = m_{31} = m_{24} = m_{42} = -\rho\pi L\left(\frac{2R_1^2 R_2^2}{R_2^2 - R_1^2}\right) = \text{fluid coupling}$$

Example



As illustrated in the above figure, water is constrained between the rotor and stator. The rotor OD is 4 in (R1=2 in), the stator ID is 10 in (R2 = 5 in), and the total length of the fluid occupied area is 20 in. Water density is 62 Lb/ft^3. The added mass due to the fluid is modeled with 5 pseudo bearings, which connect the rotor and stator. The added masses for bearings at both ends (using L=2.5 in), stations 4-12 and stations 8-16 are:

$$m_{11} = m_{22} = \rho \pi L R_1^2 \left(\frac{R_2^2 + R_1^2}{R_2^2 - R_1^2} \right) = 1.5566 \text{ Lb}$$

$$m_{33} = m_{44} = \rho \pi L R_2^2 \left(\frac{R_2^2 + R_1^2}{R_2^2 - R_1^2} \right) = 9.7287 \text{ Lb}$$

$$m_{13} = m_{31} = m_{24} = m_{42} = -\rho \pi L \left(\frac{2 R_1^2 R_2^2}{R_2^2 - R_1^2} \right) = -2.6838 \text{ Lb}$$

The added masses for the 3 bearings inside (using L=5 in), stations 5-13, stations 6-14, and stations 7-15 are:

$$m_{11} = m_{22} = \rho \pi L R_1^2 \left(\frac{R_2^2 + R_1^2}{R_2^2 - R_1^2} \right) = 3.1132 \text{ Lb}$$

$$m_{33} = m_{44} = \rho \pi L R_2^2 \left(\frac{R_2^2 + R_1^2}{R_2^2 - R_1^2} \right) = 19.4575 \text{ Lb}$$

$$m_{13} = m_{31} = m_{24} = m_{42} = -\rho \pi L \left(\frac{2 R_1^2 R_2^2}{R_2^2 - R_1^2} \right) = -5.3676 \text{ Lb}$$

Small amount of damping, Cxx and Cyy, is also added in these pseudo bearings.

References:

1. Fritz, R. J., 1972, *The Effect of Liquids on the Dynamic Motions of Immersed Solids*, Journal of Engineering for Industry, Vol. 94, pp.167-173.

2. Blevins, R. D., 2001, *Flow-Induced Vibration*, Krieger Publishing Co.

Torsional Time Dependent Excitation given in data file

This option allows you to input the time dependent torsional excitations from the data files. The data stored in data files are ASCII free format file containing two columns, separated by spaces. One is time in seconds and the other is excitation as shown below:

0 0

0.001 50

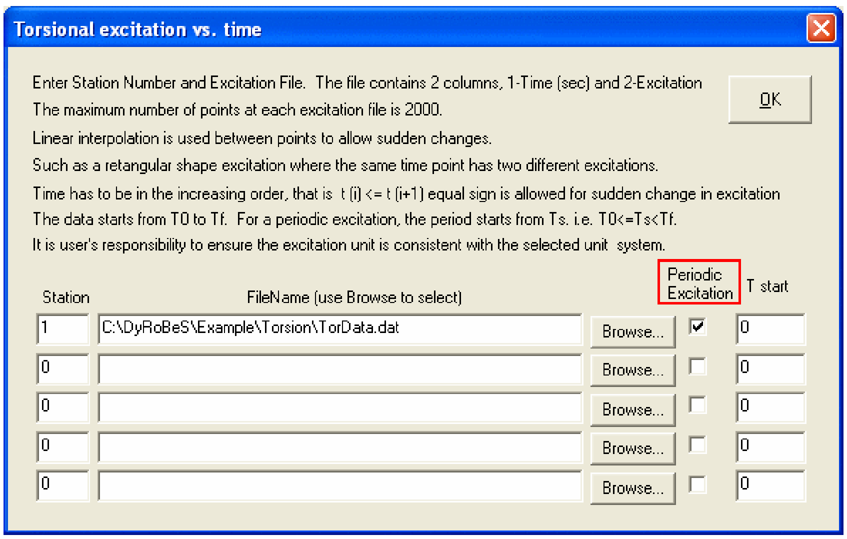
0.002 100

0.005 900

0.010 7000

Be careful to avoid any strange characters or unusual formats in this ASCII data file.

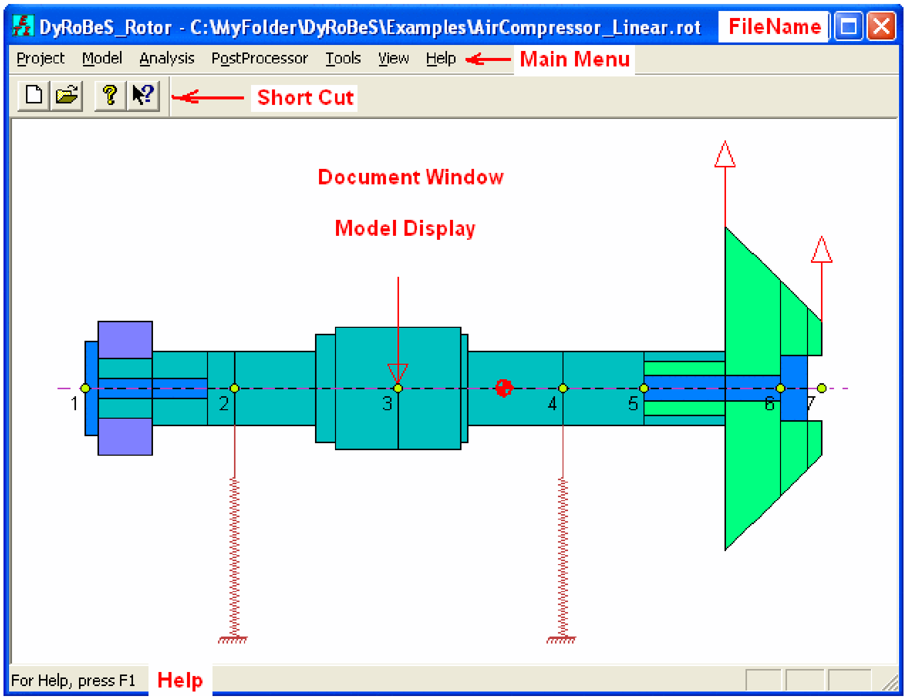
Linear interpolation will be used for sudden changes in excitation, such as step function, triangular shapes, etc… You may specify the starting time to include this excitation in the analysis. If the Periodic Excitation is checked, then the excitation will become periodic and it will repeat when the time is end. The maximum time (time at the last point) is the period. If the Periodic Excitation is NOT checked, the excitation will be a constant when the time exceeds the last point. No extrapolation will be used. Any time larger than the given time in the data file, the last point (torque) will be used when time is greater than the last point.



See other Torsional/Axial Data Torsional/Axial Data.

Main Menu

DyRoBeS-Rotor is a Windows-Based program. The Main Window is shown below.



For more information on each menu, click the following:

1 Project

2 Model

3 Analysis

4 PostProcessor

5 Tools

To jump into the specific topic, click the **GREEN** font with underline. This will bring you into the specific topic. If you cannot find the topic in the above list, you may use **index** to find the key word for your specific topic.

Transient Analysis – Speed Profile

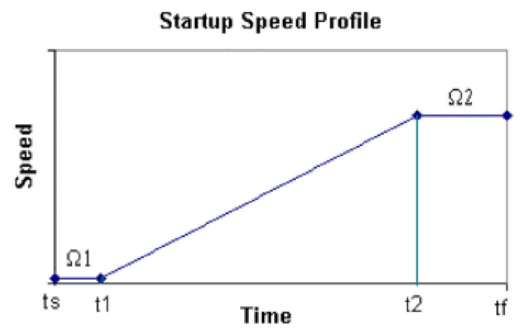
In many applications, there are needs to study the rotor motion during startup, shutdown, going through critical speeds, or rotor drop when the magnetic bearings failure occurred. In these situations, the angular velocity (speed) is no longer a constant and is a function of time. Two more terms introduced in the governing equations due to the speed variation are: circulatory matrix and forcing function. The details of the equations are documented in the DyRoBeS Theoretical Manual.

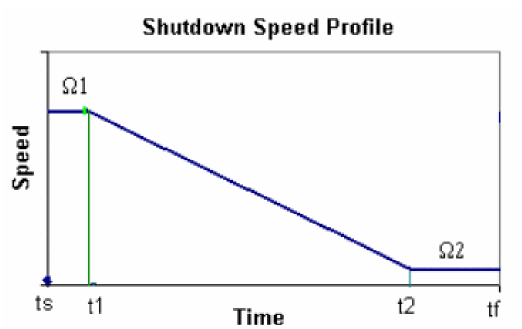
Transient Speed Profile

In DyRoBeS, two types of speed profiles can be considered.

1. Linear Profile

Linear profile gives a constant acceleration during startup and deceleration during shutdown.





Note that numerical time integration starts from ts (zero) and ends at tf. OMEGA-1 is the initial speed from ts to t1 and OMEGA-2 is the final speed after t2. The angular acceleration, velocity and displacement are:

When $t \le t_1$: Constant Speed

$$\ddot{\varphi}_0 = 0$$
$$\dot{\varphi}_0 = \Omega_1$$
$$\varphi_0 = \Omega_1 t$$

When $t_1 \le t \le t_2$: Linear Acceleration or Deceleration

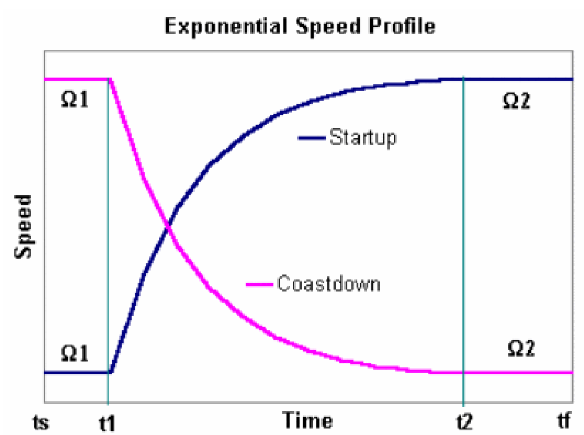
$$\ddot{\varphi}_1 = \frac{\Omega_2 - \Omega_1}{t_2 - t_1} = \frac{\Delta\Omega}{\Delta t}$$
$$\dot{\varphi}_1 = \Omega_1 + \ddot{\varphi}_1 (t - t_1) = \left(\frac{\Omega_1 t_2 - \Omega_2 t_1}{t_2 - t_1}\right) + \left(\frac{\Omega_2 - \Omega_1}{t_2 - t_1}\right) t$$
$$\varphi_1 = C_1 + \left(\frac{\Omega_1 t_2 - \Omega_2 t_1}{t_2 - t_1}\right) t + \frac{1}{2}\left(\frac{\Omega_2 - \Omega_1}{t_2 - t_1}\right) t^2$$
$$C_1 = \Omega_1 t_1 - \left(\frac{\Omega_1 t_2 - \Omega_2 t_1}{t_2 - t_1}\right) t_1 - \frac{1}{2}\left(\frac{\Omega_2 - \Omega_1}{t_2 - t_1}\right) t_1^2$$

When $t_2 \le t$: Constant Speed

$$\ddot{\varphi}_2 = 0$$
$$\dot{\varphi}_2 = \Omega_2$$
$$\varphi_2 = \Omega_2 t + C_2$$
$$C_2 = \phi_1(t_2) - \Omega_2 t_2$$

2. Exponential Profile

In some industrial application, the rotor speed increases fast initially during startup and then gradually increases the speed as the rotor speed approaches the operating speed.



The angular velocity can be expressed in the exponential form:

$$\dot{\varphi}(t) = A + B e^{-\lambda t}$$

Due to the characteristics of the exponent, the startup curve can be expressed as:

$$\dot{\varphi} = A\left(1 - e^{-\lambda(t-t_1)}\right) + \Omega_1$$

which satisfies the initial condition at t1 instant. At t2 instant,

$$\Omega_2 = A\left(1 - e^{-\lambda(t_2-t_1)}\right) + \Omega_1$$

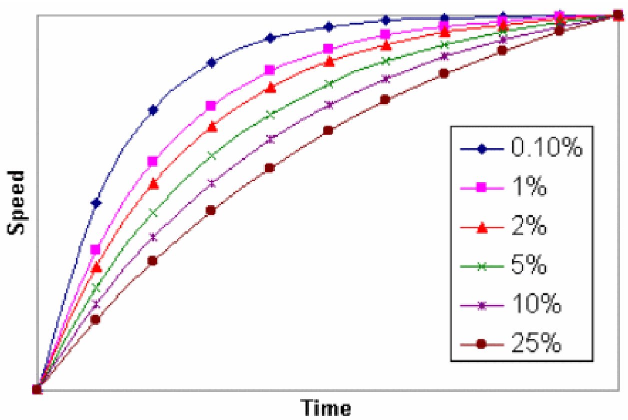
where A and λ can be related by a single parameter δ, let

$$\Delta\Omega = \Omega_2 - \Omega_1$$
$$\Delta t = t_2 - t_1$$
$$A = (1 + \delta)\Delta\Omega$$

then we have

$$\lambda = -\frac{\ln\left(\dfrac{\delta}{1+\delta}\right)}{\Delta t}$$

δ determines the shape of the curve, the following curves are for various δ



The value of 2% (0.02) was used in Lalanne and Ferraris. The larger delta value is, the straight the curve becomes. The coastdown curve is :

$$\dot{\varphi} = B\left(1 - e^{-\lambda(t-t_1)}\right) + \Omega_1$$
$$B = -A$$

When $t \le t_1$: Constant Speed

$$\ddot{\varphi}_0 = 0$$
$$\dot{\varphi}_0 = \Omega_1$$
$$\varphi_0 = \Omega_1 t$$

When $t_1 \le t \le t_2$: Exponential profile

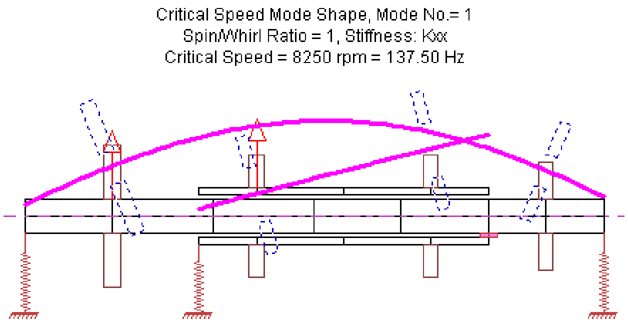
$$\ddot{\varphi}_1 = A\lambda e^{-\lambda(t-t_1)}$$
$$\dot{\varphi}_1 = A\left(1 - e^{-\lambda(t-t_1)}\right) + \Omega_1$$
$$\varphi_1 = C_1 + \left(A + \Omega_1\right)t + \frac{A}{\lambda} e^{-\lambda(t-t_1)}$$
$$C_1 = -At_1 - \frac{A}{\lambda}$$

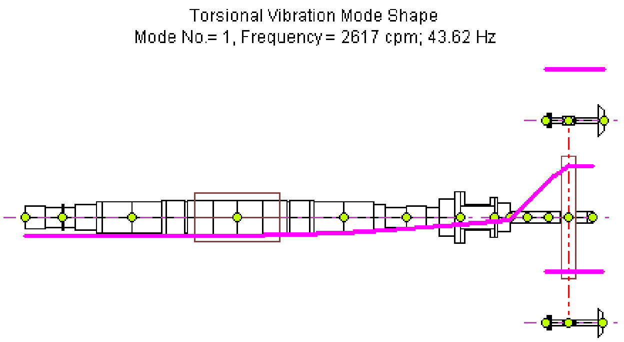
When $t_2 \le t$: Constant Speed

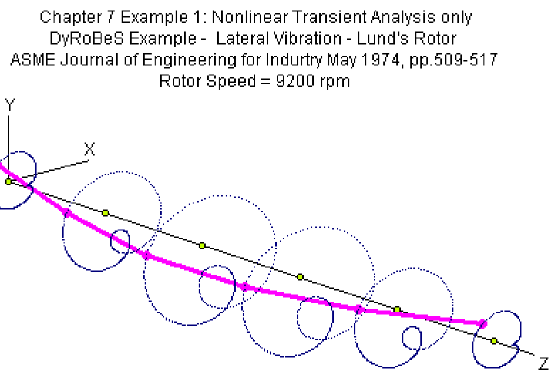
$$\ddot{\varphi}_2 = 0$$
$$\dot{\varphi}_2 = \Omega_2$$
$$\varphi_2 = \Omega_2 t + C_2$$
$$C_2 = \phi_1(t_2) - \Omega_2 t_2$$

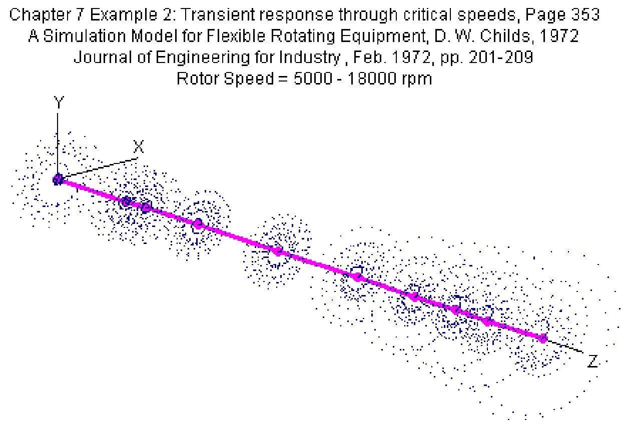
Animation

The Animation feature is implemented whenever there is a need in the postprocessor. You are encouraged to use this feature to visualize the rotor motor. Select **Play** to start the animation and select **Stop** to end the animation. **Timer** allows you to adjust the speed of the animation. You can even save this animation file for presentation purposes.








Chapter 7 Example 2: Transient response through critical speeds, Page 353
A Simulation Model for Flexible Rotating Equipment, D. W. Childs, 1972
Journal of Engineering for Industry , Feb. 1972, pp. 201-209
Rotor Speed = 5000 - 18000 rpm
Y
X
Z