Dyrobes

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DyRoBeS© Manual | Rotor

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Introduction

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 <u>Dynamics of Rotor</u> <u>Bearing Systems</u>.

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_x) 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: <u>Aerodynamic Cross Coupling Calculation</u>, <u>Rolling Element Bearing Stiffness Calculation</u>, <u>Liquid Annular</u> Seal Dynamic Coefficients, Gas Laby Seal Dynamic Coefficients, Squeeze Film Damper Stiffness and Damping Calculation, <u>Mass/Inertia Properties Calculation</u>, <u>Rotor</u> <u>Orbit Analysis</u>, <u>Balancing Calculation</u>, and <u>Auto-Balancing Analysis</u>

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 **<u>DyRoBeS©</u>BePerf**.

Getting Started

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.

	File Nam	e			
🛃 DyRoBeS_Rotor - U	ntitled			1	
<u>Project Model Analysis</u>	PostProcessor	<u>T</u> ools	⊻iew	<u>H</u> elp	
Do	cument Wir	ndow			
J For Help, press F1				NUM	
				<i>,</i>	
🛃 DyRoBeS_Rotor - C:	\DyRoBeS\Ex	ample\	Comp	es	- 🗆 🗵
Project Model Analysis	P <u>o</u> stProcessor	<u>T</u> ools	⊻iew	<u>H</u> elp	
□ 🚄 🤋 😢					
					< → ₩₩₩₽
For Help, press F1				NUM	

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

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 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**.

🛃 DyRoBeS_Rotor - C:\DyRo	BeS\Ex	ample	Wirc	raftEngine.rot	
Project Model Analysis PostPi	ocessor	Tools	View	Help	
Project Model Analysis PostPr New Open Close Save As Different Units Print Print Preview Print Setup Print to File Graphic Preferences Settings File Converter 1 AircraftEngine.rot 2 Fluid_Coupling2A.rot 3 AirCompressor_Nonlinear.rot 4 UTRC_NonLinearModel_1.rot 5 TurboCharger.rot 6 CompressorTrain.rot 7 NASA_NAG_Example.rot 8 AirCompressor Linear.rot	Ctrl+N Ctrl+O Ctrl+P		••••	Help	
Exit					
		_			

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.

Graphic Settings

Many graphic settings can be preset and saved as a preferences files (*.rpf) The program will automatically start with a default preferences file (MyPreferences.rpf) If file MyPreferences.rpf does not exist, a set of commonly used default setttings will be applied. This dialog allows you to Open a references file, Change the settings, Save the settings into a file, or, Reset the settings back to the default settings used by ETI. FileName: C:\DyRoBeS\MyPreferences.rpf Comment: Default Settings Model Display Settings Model Display Colors Post-Processor Graph Settings Post-Processor Graph Labels Post-Processor Graph Colors <u>O</u>pen <u>S</u>ave <u>R</u>eset <u>C</u>lose

Model Display Settings

Model Graphic Settings	
 ✓ Station Number ✓ Station Symbol ✓ Unbalance Mag & Angle 	CG Symbol OK Fill Element Color
Bearing Position Description	Line Size: 1 Description/Header Display
No. of Description: 3 Header/Description Font Font: Arial	C Left © Center C Right
Arial	Size: Auto Scale
Scale Factor Station Symbol: 1	Disk Circle: 1
Station Number: 1 Force Length: 1	Disk Length: 1 Support Size: 1
Bearing Length: 1 Intershaft Brg Width: 1	Constraint Size: 1 CG Symbol Size: 1

Model Display Colors

Color	Preference Rotor Element Material Color		Component Color	Click on any RGB value to change color
	Click to Select 🔺	Components	Click to Select	
1	RGB(000,192,192)	Shaft C.L.	RGB(192.064.192)	Color 🛛 🖓 🔀
2	RGB(000,255,128)	Station Symbol	RGB(192,255,000)	Basic colors:
3	RGB(000,128,255)	Disk	RGB(128,064,064)	
4	RGB(128,128,255)	Unbalance	RGB(255,000,000)	
5	RGB(052,128,128)	Bearing	RGB(192,064,064)	
6	RGB(052,128,000)	Intershaft Bearin	RGB(255,128,255)	
7	RGB(000,156,192)	Support	RGB(224,058,000)	
8	RGB(064,192,128)	User's Element	RGB(255,255,128)	
9	RGB(102,222,102)	Axial Forces	RGB(255,000,000)	
10	RGB(156,192,128)	Static Loads	RGB(255,000,000)	
11	RGB(156,128,192)	Time Forcing	RGB(000,000,255)	
12	RGB(156,255,128)	Constraints	RGB(128,000,128)	
13	RGB(156,156,128)	Torsion Rigid Lin	k RGB(255,000,000)	<u>C</u> ustom colors:
14	RGB(128,056,128)	Disk Inside		
15	RGB(128,192,128)	Foundation	RGB(000,000,255)	
16	RGB(128,000,128)	CG Symbol	RGB(255,000,000)	
17	RGB(128,000,156)			Define Custom Colory St
18	RGB(128,000,255)	M	ore Disks OK	Define Custom Colors >>
				OK Cancel

Post-Processor Graph Settings

Post-Processor Default Graphic Settings
Lateral Vibration Response
Displacement Multiplier: 1 🔽 🔽 Peak-Peak
Velocity Multiplier: 1
Acceleration Multiplier: 1
Probe Orientation (deg) - 1(X): 0 2(Y): 90
Label Amplification Factor: 🔽 X (1) 🔽 Y (2) 🔽 Mark Half-Power
Bode Plot - Phase display: C Academic 🕥 [Instrument (ADRE)]
Polar Plot - No. of Speed Labels: 3 on: 🔽 Probe 1 🔽 Probe 2
Decimal Places for the Graphics Title
Amplitude: auto default 💌 Phase: 0 💌
Torsional and Axial Vibration
Torsional Displacement Multiplier: 1
Torsional Velocity Multiplier: 1 OK
Axial Displacement Multiplier: 1

Post-Processor Graph Labels

Post-Processor Default X-	Y Label Settings		2
	These labels can be overwritter	n under the Options-Settings in each g	raph OK
Time Axis:	Rotational Speed (rpm) Time (sec)	Torsional Vibration Freq. Interference Diagram:	Torsional Natural Frequencies (rpm)
	Critical Speeds (rpm)	Station Displacement: Station Velocity:	Displacement (rad) Velocity (rad/sec)
X axis: Whirl Speed Map - Y axis:	Bearing Stiffness Damped Natural Frequencies (rpm)	Element Deflection: Vibratory Torque:	Element Deflection (rad) Vibratory Torque
Bode Plot - Amplitude: Phase:	Amplitudes Phases (Deg)	Vibratory Stress: Excitation Torque:	Vibratory Torsional Stress Excitation Torque
Response Displacement: Velocity:	Displacement Velocity	Driving and Load Torques:	Driving and Load Torques
Acceleration: Bearing/Support Forces:	Acceleration Transmitted Force	Axial Vibration Freq. Interference Diagram: Station Displacement:	Axial Natural Frequencies (rpm) Axial Displacement
Element Shear Forces: Bending Moments:	Element Shear Forces Element Bending Moments	Vibratory Force: Excitation Forces:	Vibratory Force
FFT Frequency axis: Stability Parameter: 📀	,		J

Post-Processor Graph Colors

ost-Pro	cessor Default Graph	ic Setting	2
	Color	Size 🔺	X-Y Labels Font
1	RGR(255.000.255)	3	Font: Arial 👻 Style: Regular 👻
2	RGB(255,000,000)	3	
3	RGB(128,000,128)	3	Arial Size: Auto Scale 💌
4	RGB(128,000,000)	3	
5	RGB(224,118,000)	3	Color number assignment
6	RGB(192,000,128)	3	3D Rotor Deflection: 1 2D Mode Shapes: 1
7	RGB(192,000,000)	3	
8	RGB(128,000,064)	3	3D Orbit Color: 131 3D Phase Mark: 141
9	RGB(128,064,000)	3	2D Orbit Color: 21 2D Timing Mark: 131
10	RGB(255,128,064)	3	2D Orbit Color: 21 2D Timing Mark: 131
11	RGB(000,000,128)	3	Critical Speed Map: 21 Bearing Stiffness: 40
12	RGB(000,128,000)	3	
13	RGB(000,128,255)	3	Frequency Curves: 21 Excitation Lines: 42
14	RGB(000,000,192)	3	Response X: 21 Response Y: 22
15	RGB(000,192,000)	3	Hesponse X. [27] Hesponse I. [22]
16	RGB(000,192,255)	3	Operating Speeds: 96 General Curves: 21
17	RGB(064,064,255)	3	
18	RGB(000,255,000)	3	Amplification Factor (AF) Markers: 89
19	RGB(000,128,255)	3	
20	RGB(000,000,255)	3	Increment Mode Shape Color: 💿 None 🕤 Shaft 🔿 Mode
21	RGB(255,000,255)	2	
22	RGB(255,000,000)	2	Size > 0 ==> Solid Line
23	RGB(128,000,128)	2	Size = 0 ==> DashDotDot OK
24	RGB(128,000,000)	2	Size < 0 ==> DashDot
25	RGB(224,118,000)	2 .	·

See also File Extension.

Description of File Name Extension

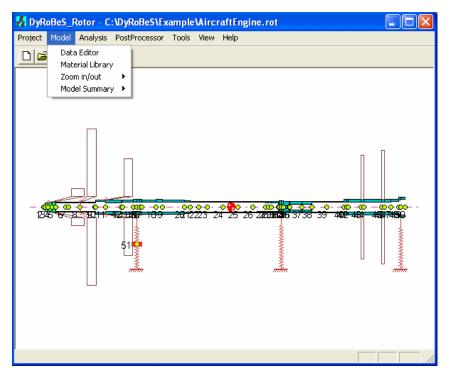
After the analysis is performed, additional files will be created for the post processing. The description of the file extension is listed below for reference.

Analysis NameASCII TEXTBinary (Post-Process)Input FileROTLateral AnalysisOU0Model SummaryOU0Static Deflection& Bearing LoadsOU1Critical Speed AnalysisOU2CS, CSE, CSXCritical Speed MapOU3Critical Speed MapOU3Whirl Speed/Stability AnalysisOU4Ws, WSMSteady Synchronous ResponseOU5Linear SystemsUR, URFSteady Synchronous ResponseOU6Non-Linear SystemsOU7Time Transient AnalysisOU7Steady State Harmonic ExcitationOU8Including non-synchronousOU9Steady Maneuver Load AnalysisOU9Steady Maneuver Load AnalysisOT0Notel SummaryOT0Natural Frequency AnalysisOT1OTA, OTXSteady Forced ResponseOT2Startup Transient – Time DependentExcitationsOT3Startup Transient – Size AnalysisModel SummaryOA0Natural Frequency AnalysisOT4OTD, OTZSpeed Dependent ExcitationsAxial AnalysisModel SummaryOA0Natural Frequency AnalysisOA1OAA, OAXSteady Forced ResponseOA2OAB, OAY			
Lateral AnalysisOU0NoneModel SummaryOU0NoneStatic Deflection & Bearing LoadsOU1ST, STXCritical Speed AnalysisOU2CS, CSE, CSXCritical Speed MapOU3CSMWhirl Speed/Stability AnalysisOU4WS, WSMSteady Synchronous ResponseOU5UR, URFLinear SystemsOU6UR, URFSteady Synchronous ResponseOU6UR, URFNon-Linear SystemsOU7TR, TRF, FCSSteady Synchronous ResponseOU8VR, VRFIme Transient AnalysisOU7TR, TRF, FCSSteady State Harmonic Excitation Including non-synchronous Linear SystemsOU9SM, SMXTorsional AnalysisOU1OT0NoneNatural Frequency AnalysisOT1OTA, OTXSteady Forced ResponseOT2OTB, OTYTime Transient – Time Dependent ExcitationsOT3OTC, OTZStartup Transient – Speed Dependent ExcitationsOT4OTD, OTZModel SummaryOA0NAUNAU	Analysis Name	ASCII TEXT	Binary (Post-Process)
Model SummaryOU0NoneStatic Deflection& Bearing LoadsOU1ST, STXCritical Speed AnalysisOU2CS, CSE, CSXCritical Speed MapOU3CSMWhirl Speed/Stability AnalysisOU4WS, WSMSteady Synchronous ResponseOU5UR, URFLinear SystemsOU6UR, URFNon-Linear SystemsOU7TR, TRF, FCSTime Transient AnalysisOU7TR, TRF, FCSSteady State Harmonic Excitation Including non-synchronous Linear SystemsOU9SM, SMXTorsional AnalysisOU1NoneNatural Frequency AnalysisOT1OTA, OTXSteady Forced Response ExcitationsOT3OTC, OTZTime Transient – Time Dependent ExcitationsOT4OTD, OTZModel SummaryOT4OTD, OTZStartup Transient – Time Dependent ExcitationsOT4OTD, OTZModel SummaryOT4OTD, OTZNatural Frequency AnalysisOT4OTD, OTZStartup Transient – Time Dependent ExcitationsOT4OTD, OTZModel SummaryOA0Natural Frequency Analysis	Input File	ROT	
Static Deflection& Bearing LoadsOU1ST, STXCritical Speed AnalysisOU2CS, CSE, CSXCritical Speed MapOU3CSMWhirl Speed/Stability AnalysisOU4WS, WSMSteady Synchronous ResponseOU5UR, URFLinear SystemsOU6UR, URFSteady Synchronous ResponseOU6UR, URFNon-Linear SystemsOU7TR, TRF, FCSSteady State Harmonic ExcitationOU8VR, VRFIncluding non-synchronousOU9SM, SMXTorsional AnalysisOU1NoneNatural Frequency AnalysisOT1OTA, OTXStady Forced ResponseOT2OTB, OTYTime Transient – Time DependentOT3OTC, OTZStartup Transient –OT4OTD, OTZStartup Transient –OT4OTD, OTZModel SummaryOA0Natural Frequency AnalysisStartup Transient –OT4OTD, OTZStartup Transient –OT4OTD, OTZStartup Transient –OA0NAU	Lateral Analysis		
Critical Speed AnalysisOU2CS, CSE, CSXCritical Speed MapOU3CSMWhirl Speed/Stability AnalysisOU4WS, WSMSteady Synchronous Response Linear SystemsOU5UR, URFSteady Synchronous Response Non-Linear SystemsOU6UR, URFTime Transient AnalysisOU7TR, TRF, FCSSteady State Harmonic Excitation Including non-synchronous Linear SystemsOU8VR, VRFSteady Maneuver Load AnalysisOU9SM, SMXTorsional AnalysisOT0NoneNatural Frequency AnalysisOT1OTA, OTXStatup Transient – Time Dependent ExcitationsOT3OTC, OTZStartup Transient – Speed Dependent ExcitationsOT4OTD, OTZModel SummaryOA0OA0OA0	Model Summary	OUO	None
Critical Speed MapOU3CSMWhirl Speed/Stability AnalysisOU4WS, WSMSteady Synchronous Response Linear SystemsOU5UR, URFSteady Synchronous Response Non-Linear SystemsOU6UR, URFTime Transient AnalysisOU7TR, TRF, FCSSteady State Harmonic Excitation Including non-synchronous Linear SystemsOU8VR, VRFSteady Maneuver Load AnalysisOU9SM, SMXTorsional AnalysisOT0NoneNatural Frequency AnalysisOT1OTA, OTXSteady Forced ResponseOT2OTB, OTYTime Transient – Time Dependent ExcitationsOT3OTC, OTZStartup Transient – Speed Dependent ExcitationsOT4OTD, OTZModel SummaryOA0OA0	Static Deflection & Bearing Loads	OU1	
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Non-Linear SystemsOU7TR, TRF, FCSTime Transient AnalysisOU7TR, TRF, FCSSteady State Harmonic Excitation Including non-synchronous Linear SystemsOU8VR, VRFSteady Maneuver Load AnalysisOU9SM, SMXTorsional AnalysisOU9SM, SMXModel SummaryOT0NoneNatural Frequency AnalysisOT1OTA, OTXSteady Forced ResponseOT2OTB, OTYTime Transient – Time Dependent ExcitationsOT4OTD, OTZStartup Transient – Speed Dependent ExcitationsOT4OTD, OTZModel SummaryOA0Natural Frequency Analysis			
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Steady State Harmonic Excitation Including non-synchronous Linear SystemsOU8VR, VRFSteady Maneuver Load AnalysisOU9SM, SMXTorsional AnalysisOU9SM, SMXModel SummaryOT0NoneNatural Frequency AnalysisOT1OTA, OTXSteady Forced ResponseOT2OTB, OTYTime Transient – Time Dependent ExcitationsOT3OTC, OTZStartup Transient – Speed Dependent ExcitationsOT4OTD, OTZAxial AnalysisOA0OA0			
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Linear SystemsImage: Constraint of the systemsSteady Maneuver Load AnalysisOU9SM, SMXTorsional AnalysisOU9SM, SMXModel SummaryOT0NoneNatural Frequency AnalysisOT1OTA, OTXSteady Forced ResponseOT2OTB, OTYTime Transient – Time DependentOT3OTC, OTZStartup Transient –OT4OTD, OTZSpeed Dependent ExcitationsOT4OTD, OTZModel SummaryOA0Natural Frequency AnalysisNatural Frequency AnalysisOA1OAA, OAX	Steady State Harmonic Excitation	OU8	VR, VRF
Steady Maneuver Load Analysis OU9 SM, SMX Torsional Analysis OT0 None Model Summary OT0 None Natural Frequency Analysis OT1 OTA, OTX Steady Forced Response OT2 OTB, OTY Time Transient – Time Dependent OT3 OTC, OTZ Startup Transient – OT4 OTD, OTZ Speed Dependent Excitations OT4 OTD, OTZ Model Summary OA0 Natural Frequency Analysis	Including non-synchronous		
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Model Summary OT0 None Natural Frequency Analysis OT1 OTA, OTX Steady Forced Response OT2 OTB, OTY Time Transient – Time Dependent OT3 OTC, OTZ Startup Transient – OT4 OTD, OTZ Speed Dependent Excitations OT4 OTD, OTZ Model Summary OA0 OA0	Steady Maneuver Load Analysis	OU9	SM, SMX
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 OA0 OA0 Natural Frequency Analysis OA1 OAA, OAX			
Steady Forced Response OT2 OTB, OTY Time Transient – Time Dependent OT3 OTC, OTZ Excitations OT4 OTD, OTZ Startup Transient – OT4 OTD, OTZ Speed Dependent Excitations Axial Analysis OA0 Model Summary OA1 OAA, OAX	Model Summary		None
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Excitations OT4 OTD, OTZ Speed Dependent Excitations OT4 OTD, OTZ Axial Analysis OA0 OA0 Natural Frequency Analysis OA1 OAA, OAX	Steady Forced Response	OT2	OTB, OTY
Startup Transient – OT4 OTD, OTZ Speed Dependent Excitations Axial Analysis OA0 Model Summary OA0 OAA, OAX	Time Transient – Time Dependent	OT3	OTC, OTZ
Speed Dependent Excitations Analysis Axial Analysis OAD Model Summary OAD Natural Frequency Analysis OA1	Excitations		
Axial Analysis OAD Model Summary OAD Natural Frequency Analysis OA1 OAA, OAX		OT4	OTD, OTZ
Model Summary OA0 Natural Frequency Analysis OA1 OAA, OAX	Speed Dependent Excitations		
Natural Frequency Analysis OA1 OAA, OAX			
	Model Summary	OAO	
Steady Forced Response OA2 OAB, OAY	Natural Frequency Analysis	OA1	0AA, 0AX
	Steady Forced Response	OA2	OAB, OAY

Model

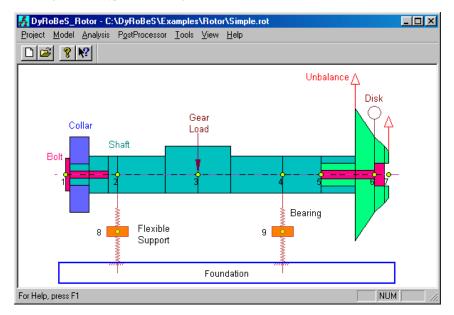
Model

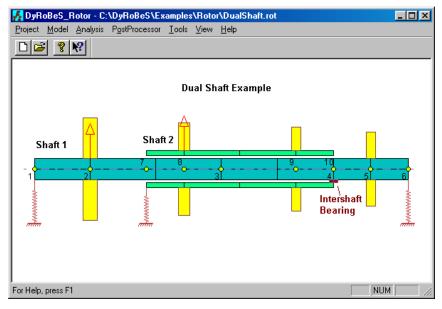
Under the Model menu, you can build a new rotor bearing system model or modify/edit your existing model by clicking the <u>Data Editor</u>. The usage of the Data Editor will be explained in Modeling and Data Editor session. You can also create and modify your <u>Material Library</u>. <u>Zoom in/out</u>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

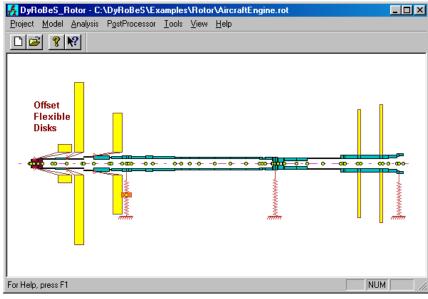


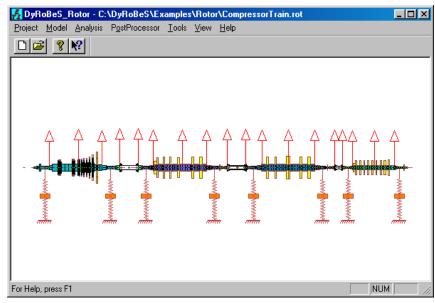
Typical Rotor Configurations

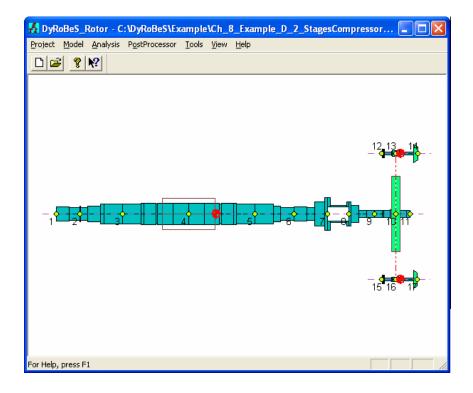
Followings are some typical rotor configurations:











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.

Material Library		
List: Typical Steel		•
Material: Typical Steel		
Units: English 💌		Update
Density: 0.283	Lb/in^3	
Elastic Modulus (E): 2.9E+007	psi	<u>C</u> lose
Shear Modulus (G): 1.1154E+007	psi	
G=E/(2(1+nu)) where nu = Poisson's rati	0	

Data Editor

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 $\begin{pmatrix} \theta_x, \theta_y \end{pmatrix}$ coordinates at each finite element station. For torsional vibration, the motion of each finite element station is described by a rotational displacement $\begin{pmatrix} \theta_x \end{pmatrix}$ 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.

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
System Units: (2) - Engineering English (s, in, Lbf, Lbm)
Description Context
1 DyRoBeS Example - Lateral Vibration
2 Aircraft engine example taken from CRITSPD manual
3 Flexible offset disks
4 Critical speed analysis
5 Critical speeds: 5450, 13164, 16491, 22785, 35480 rpm
20
21
Insert Row Delete Row
Save Save As Close Help

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

Rotor Bearing System Data	
	lance Bearings Supports Foundation User's Elements aft Bow Time Forcing Harmonic Excitation <mark>Torsional/Axia </mark>
Torsional and/or Axial Data	Torsional Time Dependent Excitations
Connectivity	Excitations in Equations
Non-Linear Coupling	Excitations from files
Modal Damping	Torsional Startup Transient Torques
Steady State	Driving Torque
Excitation	Load Torque
	Save Save As Close Help

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

<u>Unbalance</u>

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

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)

14-Future implementation

Flexible Supports

Foundation

User's Elements

Axial Force & Torque

Static Loads

Constraints

Misalignments

Shaft Bows

Time Forcing Functions

Harmonic Excitations

Torsional/Axial Data

Connectivity

Modal Damping

Non-Linear Connection/Coupling

Steady State Excitation

Time Dependent Torsional Excitations

Driving Torque

Load Torque

Engine Excitation

Reciprocating Excitation

Gear Mesh Data

Model Summary

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				01111 - 4
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		2.2.11		- U
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/m² (Pa)	N/mm ² (MPa)
Shaft Elements			1	
Length, Diameter	in	in	m	mm
Disks				
Mass	Lbf-s²/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			1 -	-
Imbalance (<i>me</i>)	Lbf-s ²	oz-in	kg-m	kg-mm
Angle	degree	degree	degree	degree
Flexible Supports				-
Mass	Lbf-s²/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	Lbf	Lbf	N	N
Moment	Lbf-in	Lbf-in	N-m	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	1	1	1	1	1
Pass Filter Active					
Magnetic Bearings					
Proportional Gain	Lbf/in	Lbf/in	N/m	N/mm	\top
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。	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/s²	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)	N = 4.448222 * Lbf
		1N = 1kg * 1m/s ²	N = 9.8066 * kgf
			kgf = 0.4535924 * Lbf
Moment	Lbf-in	N-m	N-m = 0.1129846 * Lbf-in
Mass	Lbf-s²/in	kg = N-s²/m	kg = 0.4535924 * Lbm
			kg = 175.1266 * Lbf-s²/in
			Lbm = 386.088 * Lbf-s ² /in
			Lbm = 2.20462 * kg
Density	Lbf-s²/in4	kg/m ³	kg/m ³ = 2.767990E+04 * Lbm/in ³
			kg/m ³ = 1.068688E+07 * Lbf-s ² /in ⁴
			g/cm³ = 1 * g /cc
			g/cm ³ = 2.767990E+01 * Lbm/in ³
			Lbm/in ³ = 0.0361273 * g/cm ³
Inertia	Lbf-s²-in	kg-m²	kg-m ² = 0.1129846 * Lbf-s ² -in
Moduli	Lbf/in ² (psi)	N/m² (Pa)	Pa = 6.894757E+03 * psi
		kN/ m² (kPa)	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	۰F	°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)	m³/hour	m ³ /hour = 0.2271 * gpm
			liter/min = 3.785412 * gpm
			liter/min = 16.66667 * m ³ /hour
Power	hp	kWatt	kVVatt = 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.

Rotor Bearing System Data	\mathbf{X}
Axial Forces Static Loads Constraints Misalignments Shaft Bow Time Forcing Harmonic Excitation Units/Description Material Shaft Elements Disks Unbalance Bearings Supports Foundation	Torsional/Axial User's Elements
System Units: (2) - Engineering English (s, in, Lbf, Lbm) (0) - Consistent set of units of the user's choice (1) - Consistent English (s, in, Lbf, Lbfs^2/in) DyRoB (2) - Engineering English (s, in, Lbf, Lbm) (4) - Engineering Metric units (s, mm, N, kg) (4) - Engineering Metric units (s, mm, N, kg) (4) - Engineering Metric units (s, mm, N, kg) (4) - Engineering Metric units (s, mm, N, kg) (5) - Contical speed analysis	
5 Critical speeds: 5450, 13164, 16491, 22785, 35480 rpm 6 7 7 8 9 10	
11 12 13 14 15 16	
17 17 18 19 20 21	
Insert Row Delete Row	
Save Save As Close	Help

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 <u>Project</u> menu. The material property number is used in the <u>Shaft Elements</u> 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-s²/in⁴ = 7832 kg/m³ = 0.283 Lbm/in³

Elastic Modulus (E) = 29.0E+06 Lbf/in² = 20.0E+10 N/m² (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

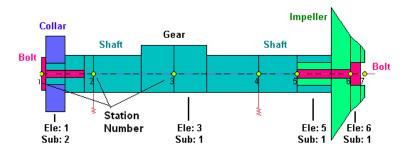
Rotor Bea	aring System Data				
		Constraints Misalignmer Shaft Elements Disks Material: Typical Ste	Unbalance Bear	ne Forcing Harmonic Excitation ings Supports Foundation	Torsional/Axial User's Elements
	Mass Density	Elastic Modulus	Shear Modulus	Comments	
1	0.283	2.9E+07	1.1154E+07	Typical Steel	
2	0.161	1.55E+07	6.5E+06	Titaniun	-
3	0.1	1E+07	3.8E+06	Aluminum	
4					
5					
6					
7					
8					
9					
10					
11					
12 13					
13					
14					
16					
17					
18					
19				···· ¢····	
20					
21					•
Insert	Row Delete Row		Ur	nit:(2) - Density: Lbm/in^3, Modulu:	s: Lbf/in^2 (psi)
				· · · · · · · · · · · · · · · · · · ·	
			Save	Save As Close	Help
			<u>0</u> 440		

Shaft Elements

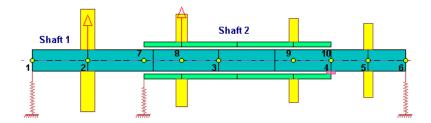
The shafts of the rotor system are numbered consecutively from 1 to Ns and the finite element stations of the model are numbered consecutively starting with 1 at the left end of shaft 1 and continuing to the last station at the right end of shaft Ns. The shafts are made up of Elements with the numbering 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).

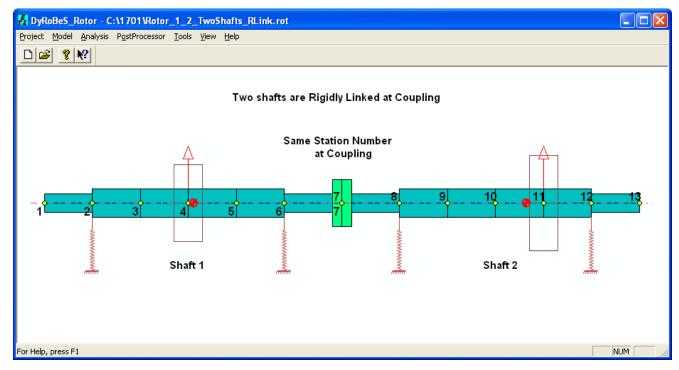
Note that an element does not exist for the last station of each shaft. For multi-shaft systems, the station numbers are consecutive and the element numbers are not continuous in general. However, in Ver 17. it allows a rigid link between two shafts with the same rotor station and rotor speeds. That 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:

kial Fo nits/D	irces escript		ic Loa Mater		Constraints 1 Shaft Elements	Misalignments Disks l	Shaft Bow Jnbalance	Time Forcin Bearings 9		c Excitation Torsior oundation User's E	
Shafi	: 1 of	1		Start	ing Station #:	1		Add Shaft	Del Shaft	Previous Ne	st
Spee	d Ratio	o: 1		A	ial Distance:	0	Y Distance:	0		Import *.xls Export	t *.xls
Comr	nent:	Rotor	Assem	bly, uni	; it: English syste	m		,			_
	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	-
	rt Row		elete F		ReNumber	Copy & Paste			U	nit:(1) - Length, Diame	eter: in

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

Ro	tor B	earing	g Syst	tem D	ata						
	Axial Fo Units/E	orces)escript		ic Loa Mater		Constraints 1 Shaft Elements					Excitation Torsional/Axial
	Shaf	t 2 of	2		Star	ting Station #: [7 🔽	R Link	Add Shaft	Del Shaft	Previous Next
	Spee	ed Ratio	r 1		A:	kial Distance: 🛛	31	Y Distance:	0		Import *.xls Export *.xls
	Com	ment:				-					
		Ele	Sub	Mat	Lev	Length	Mass ID	Mass OD	Stiff ID	Stiff OD	Comments 🔺
	1	7	1	2	0	1	0	5	0	0	
	2	7	2	1	0	5	0	2	0	0	
	3	8	1	1	0	5	0	3	0	0	
	4	9	1	1	0	5	0	3	0	0	
	5	10	1	1	0	5	0	3	0	0	
	6	1 11	1	1	0	5	0	3	0	0	
	7	12	1	1	0	5	0	2	0	0	
	8										
	9										
	10										
	11										
	12										
	13										
	14										
	15										
	16										
	17										
	18										
	19										
	20										-
	20						1			1	
	Inse	rt Row	D	elete F	low	ReNumber	Copy & Paste			Ur	nit:(2) - Length, Diameter: in
								<u>S</u> a	ive (Gave <u>A</u> s	<u>C</u> lose <u>H</u> elp

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 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).

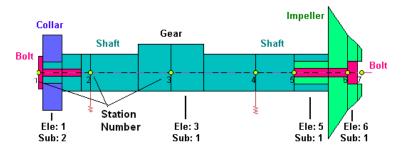
15.Comments: Description for this entry.

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:



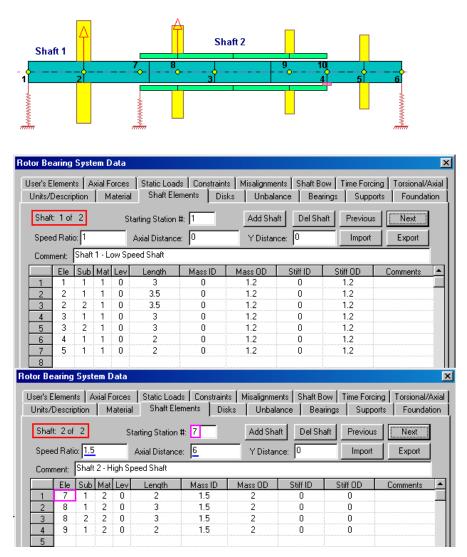
	Ele	Sub	Mat	Lev	Length	Mass ID	Mass OD	Stiff ID	Stiff OD	Comments	
1	1	1	3	0	0.25	0	1.75	0	0	Bolt Head	
2	1	2	3	0	1	0	0.375	0	0	Bolt	
3	1	2	1	1	1	0.375	1.125	0	0	Shaft	
4	1	2	4	2	1	1.125	4	0	0	Thrust Collar	
5	1	3	3	0	1	0	0.375	0	0	Bolt	
6	1	3	1	1	1	0.375	2	0	0	Shaft	
7	1	4	1	0	0.5	0	2	0	0	Shaft	
8	2	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

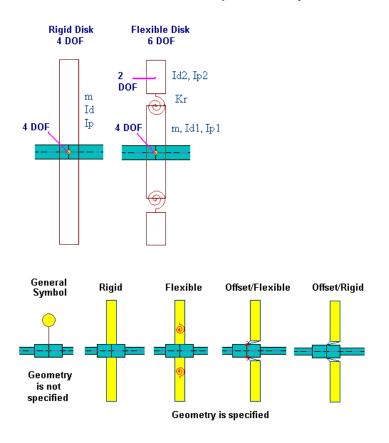
A dual shafts model and the associated shaft data are shown below:



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 \langle 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:



	Туре	Stn	Mass	Dia.Inertia	Polar Inertia	SkewX	SkewY	Length	ID	OD	Density	Offset	Kr-flex	It-flex	Ip-flex	Comments
	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
;	Flexible	11	13.7	124	202	0	0	1.2	3	13	0	3	5E+06	0	0	Flexible & Offset Disk
	Rigid	43	0	0	0	0	0	0.356	2.2	13.938	0.283	0.178	0	0	0	Rigid & Offset Disk
	Rigid	45	0	0	0	0	0	0.345	2.2	15.197	0.283	0.1725	0	0	0	Rigid & Offset Disk
	Rigid	1	0.25	0	0	0	0	0.1	0	0.5	0	0	0	0	0	
·	Rigid	2	0.25	0	0	0	0	0.1	0	0.5	0	0	0	0	0	
	Rigid	3	0.25	0	0	0	0	0.1	0	0.5	0	0	0	0	0	
	Rigid	4	0.25	0	0	0	0	0.1	0	0.5	0	0	0	0	0	
)																
2																
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Rotor Bearing System Data

	Туре	Stn	Mass	Dia.Inertia	Polar Inertia	SkewX	Shauly	Length	ID	OD	Density
1	Flexible	3	3.714	30.2	40.73		0	1.7	3	5	
2	Flexible	8	19.6	261	413	0	0	1.2	3	20.95	ů O
3	Flexible	11	13.7	124	202	Õ	Ō	1.2	3	13	Ō
4	Rigid	43	0	0	0	Ō	Ō	0.356	2.2	13.938	0.283
5	Rigid	45	Ū	Ū	Ū	Ō	Ō	0.345	2.2	15.197	0.283
6	Rigid	1	0.25	0	0	Ū	Ō	0.1	0	0.5	0
7	Rigid	2	0.25	0	0	Ū	Ō	0.1	Ū	0.5	Ō
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											
19											
20											
21									Sc	roll Bar	

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: Rigid or Flexible. 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 <u>Shaft Bow</u>. 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 <u>Tools</u> 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 <u>Shaft Elements</u>), 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.

 $1-\mbox{Constant}$ 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

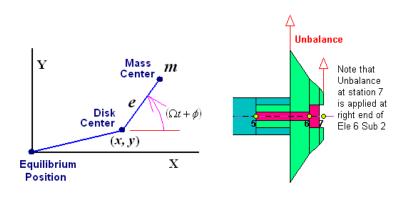
```
1 oz-in = 28.35 g-in
```

```
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)
```



Ro	tor Be	aring	Syster	n Data						
	Axial For Units/De		Static on M	Loads aterial	Constraints Mis Shaft Elements		Shaft Bow Time Fo balance Bearings		nonic Excitation Torsiona Foundation User's Ele	
									Import *.xls Export *.xls	:
		Ele	Sub	Туре	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	
	3									
	4									
	5									
	6									
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						:i		:		
	Inser	t Row	D	elete R	w	Unit:(1) - Type	0 (1): Mass (Magnet) Unbalance, .	Amp: Lbf-s^2 (Lbf), Phase: d	eg
							Save	Save <u>A</u> s	<u>C</u> lose <u>H</u>	elp

Bearings

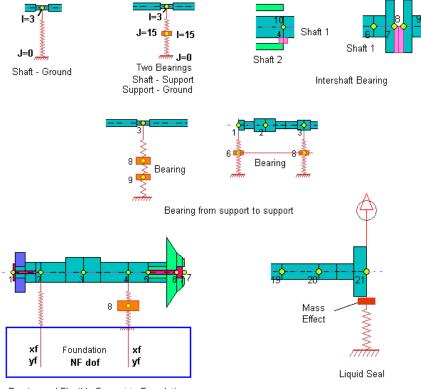
All the bearing/damper/seals/support forces acting on, or interacting between, the rotating assemblies and non-rotating structures fall into this category. Bearing is an interconnection component, which connects two finite element stations (station I and station J). The bearing can be of any type, such 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.

Bearin	ng:3of8			🗌 Fou	ndation	Add Brg	Del Brg	Pre	evious	Next
Statio	n I: 36	J: 146				Angle: 0	-	Imp	ort *.xls Exp	ort *.xls
Ту	pe: 1-Spe	eed Dependent Be	aring				-			
Comme		ne Bearing					-			
				I	<u> </u>		_	-		
_	rpm 1000	2.74113E+006	Kxy	Сух	Kyy	Cxx 18074.4	Сху	О	Суу 30094.4	Kaa
1	1000		0	U 0	5.015E+006	18074.4	U	U	30094.4 20649.4	0
2	1500 2000	2.46252E+006 2.35238E+006	0	0	4.36801E+006 4.03263E+006	10619.5	0	U 0	20649.4 15996.3	0
3	2500	2.35238E+006	0	÷	4.03263E+006	9245	0		13288.4	0
4			.	0		9245 8347.72	.	0		0
5	3000	2.33403E+006	0	0	3.72709E+006		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	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										
•										•

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.



Bearing and Flexible Support to Foundation

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:

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	🔽 Fou	ndation Add Brg	Del Brg Previous	Next
Station I:	11	X: 25 Y: 26 Rx:	28 Ry: 29 Angle: 0	_	
Туре:	0-Linear Constant Bearing				
Comment:	ent: Station 11 connects to the foundation DOF at 25, 26, 28, and 29				
Translational Bearing Properties					
Кж:	3500000	Кху: О	Сж. 2000	Cxy: 0	
Куж:	0	Куу: 1800000	Сух: 0	Суу: 1000	
Rotational Bearing Properties					
Kaa:	760000	Kab: 0	Caa: 0	Cab: 0	
Kba:	0	Кы: 510000	Cba: 0	Cbb: 0	
Unit(2) - Kt: Lbf/in, Ct: Lbf-s/in					
Save Save AsEloseElo					

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.

Rotor Bearing System Da	a
Axial Forces Static Loads Units/Description Materia	Constraints Misalignments Shaft Bow Time Forcing Harmonic Excitation Torsional/Axial Shaft Elements Disks Unbalance Bearings Supports Foundation User's Elements
Bearing: 3 of 3	Foundation Add Brg Del Brg Previous Next
Station I: 11	X: 25 Y: 26 Rx: 0 Ry: 0 Angle: 0
Type: 0-Linear Consta	nt Bearing
Comment: Station 11 conr	ects to the foundation DDF at 25 and 26
	Translational Bearing Properties
Кж: 3500000	Кжу: 0 Сжж. 2000 Сжу. 0
Куж: 0	Куу: 1800000 Сух: 0 Суу: 1000
	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
	<u>S</u> ave Save <u>A</u> s <u>C</u> lose <u>H</u> elp

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)

Rotor Bearin	ng System Dat	1					
Axial Forces Units/Descrip	Static Loads Static Loads	Constraints Misalignn Shaft Elements Disk			-	nic Excitation Foundation	Torsional/Axial User's Elements
Bearing: 3	3 of 3	Fou	ndation	Add Brg	Del Brg	Previous	Next
Station I:	11 J: 0			Angle: 0	-		
Туре:	0-Linear Constar	nt Bearing		•	1		
Comment:	0-Linear Constar 1-Speed Depend						
	2-Bearing Coeffic	cients From Data File					
	3-Pseudo Bearin 4-Squeeze Film [g (Aero, Seal, Fluid Interfere) amper	ence)				
Kuu-	5-Plain Journal B	earing (Short Bearing - Pi fil near Isotropic Bearing (Rolli			Cxy: 0		
	7-AMB - Linear F	'ID & Filter	ng Liomenta)		· · · ·		
Куж:	8-AMB - Nonline 9-Floating Ring B				Суу: 100	0	
	10-General NonL 11-Liguid Annula	.inear Bearing/Damper r Seal					
	12 Multi Lobe Be	arings - Identical Lobe (2-D arings - From BePerf Data I					
Kaa:	13-Multi-Lobe Be 14-Your Type No		-lie		Cab: 0		
Kba:	0	Кыр: О	Cba:	0	Сьр: 0		
	,				,		
						nit (2) - Kit I bf	/in, Ct: Lbf-s/in
				Save	Save <u>A</u> s	<u>C</u> lose	<u>H</u> elp

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:

$$\begin{cases} F_{x} \\ F_{y} \\ M_{x} \\ M_{y} \end{cases} = - \begin{bmatrix} K_{xx} & K_{xy} & 0 & 0 \\ K_{yx} & K_{yy} & 0 & 0 \\ 0 & 0 & K_{\alpha\alpha} & K_{\alpha\beta} \\ 0 & 0 & K_{\beta\alpha} & K_{\beta\beta} \end{bmatrix} \begin{cases} x \\ y \\ \theta_{x} \\ \theta_{y} \end{cases} - \begin{bmatrix} C_{xx} & C_{xy} & 0 & 0 \\ C_{yx} & C_{yy} & 0 & 0 \\ 0 & 0 & C_{\alpha\alpha} & C_{\alpha\beta} \\ 0 & 0 & C_{\beta\alpha} & C_{\beta\beta} \end{bmatrix} \begin{cases} \dot{x} \\ \dot{y} \\ \dot{\theta}_{x} \\ \dot{\theta}_{y} \end{cases}$$

or in matrix form:

where

 $Q = -Kq - C\dot{q}$

Kxx Cxx	Kxy Cxv	Kyx Cvx	Kyy Cw	Translational stiffnesses Translational dampings
Kaa	Καβ	Κβα	Κββ	Rotational stiffnesses
Caa	Cαβ	Cβα	Сββ	Rotational dampings

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

$\left[\mathbf{Q}_{i}\right]$	к	$-\mathbf{K} \left[\mathbf{q}_i \right]$	C	$-\mathbf{C}\left[\dot{\mathbf{q}}_{i}\right]$
$\left[\mathbf{Q}_{j}\right]^{=-}$	$-\mathbf{K}$	$\begin{bmatrix} -\mathbf{K} \\ \mathbf{K} \end{bmatrix} \begin{bmatrix} \mathbf{q}_i \\ \mathbf{q}_j \end{bmatrix}.$	C	C ∬q́j∫

Rotor Bearing System Data				
Axial Forces Static Loads Units/Description Material	Constraints Misalignmen Shaft Elements Disks		g Harmonic Excitation Supports Foundation	Torsional/Axial User's Elements
Bearing: 1 of 2	🔲 Found	ation Add Brg	Del Brg Previous	Next
Station I: 5 J: 20		Angle: 0		
Type: 0-Linear Constan	t Bearing	-		
Comment: Linear Bearing wi	th Constant Bearing Propertie	3		
	Translational Bea	ing Properties		
Кжк: 1940000	Kxy: 0	Схх: 1432	Cxy: 0	
Куж 0	Куу: 1080000	Суж: 0	Суу: 879	
	Rotational Bea	ring Properties		
Kaa: 0	Kab: 0	Caa: 0	Cab: 0	
Kba: 0	Кыс 0	Cba: 0	Сьр: 0	
		Unit:(2) - Kt:	Lbf/in, Ct: Lbf-s/in; Kr: Ll	bf-in, Cr: Lbf-in-s
		<u>S</u> ave S	Save <u>A</u> s <u>C</u> lose	<u>H</u> elp

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

Rotor Be	aring Sy	stem Data								X
Axial For Units/De	ces St escription	· · · ·	straints ft Element	Misalignm s Disk:		v Time For Bearings	cing Ha Supports	rmonic Ex : Foun		sional/Axial s Elements
Bearin	ıg:3of8			🗌 Four	ndation	Add Brg	Del Brg	Pre	evious I	Vext
Statio	n I: 36	J: 146				Angle: 0	_	Imp	ort *.xls Exp	ort *.xls
Ту	pe: 1-Spe	eed Dependent Be	aring				-			
Comme	ent: Turbi	ne Bearing								
	rpm	Kxx I	Кху	Кух	Куу	Cxx	Сху	Сух	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										-
•		· · · · · · · · · · · · · · · · · · ·			Scro	ll Bar				•
Inser	t Row	Delete Row						Unit:(2)	- Kt: Lbf/in, Ct:	Lbf-s/in
						Save	Save <u>A</u> s			<u>H</u> elp

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 the number of 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:

92	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.

Rotor Bearing System Data	
Axial Forces Static Loads Constraints Misalignmer	nts 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 🛛 🗖 Found	lation Add Brg Del Brg Previous Next
Station I: 2 J: 0	Angle: 0
Type: 2-Bearing Coefficients From Data File	
Comment: Bearing at collar end	
FileName: C:\DyRoBeS\Example\AirCompressor_Brg1.br	rg Browse
	<u>i</u>
	Unit:(2) - Kt: Lbf/in, Ct: Lbf-s/in
	<u>S</u> ave Save <u>A</u> s <u>C</u> lose <u>H</u> elp

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 <u>Fluid Elements</u>.

Example: Aerodynamic Cross-Coupling Q

Rotor Bearing System Data		X .
Axial Forces Static Loads Units/Description Material	Constraints Misalignments Shaft Bo Shaft Elements Disks Unbalance	
Bearing: 3 of 3	Foundation	Add Brg Del Brg Previous Next
Station I: 12 J: 0		Angle: 0
Type: 3-Pseudo Bearing) (Aero, Seal, Fluid Interference)	<u> </u>
Comment: Aerodynamic Cros	ss-Coupling Q	
	Translational Bearing Properties	
Кжк 0	Кжу: 572 Сжж.	0 Cxy: 0
Куж: -572	Куу: 0 Суж. 1	0 Cyy: 0
	Rotational Bearing Properties	s
Kaa: 0	Kab: 0 Caa: 0	0 Cab: 0
Kba: 0	Kbb: 0 Cba: 0	0 Сыр: 0
	Fluid Mass Dynamic Cplg	
	No fluid mass added, Click Fluid Button	above to add fluid mass
		Unit(2) - Kt: Lbf/in, Ct: Lbf-s/in; Kr: Lbf-in, Cr: Lbf-in-s
		Save Save As Close Help

Example: Structure Vibration in Fluid

Rotor Bearing System Data	
	w Time Forcing Harmonic Excitation Torsional/Axial Bearings Supports Foundation User's Elements
Bearing: 4 of 9 🗖 Foundation	Add Brg Del Brg Previous Next
Station I: 5 J: 13	Angle: 0
Type: 3-Pseudo Bearing (Aero, Seal, Fluid Interference)	<u> </u>
Comment: Water between Rotor and Stator	
Translational Bearing Properties	
Кжк: 0 Кжу: 0 Сжк: 2	0 Cxy: 0
Куж: 0 Куу: 0 Суж: 0	Суу: 20
Rotational Bearing Properties	
Kaa: 0 Kab: 0 Caa: 0	Cab: 0
Kba: 0 Kbb: 0 Cba: 0	Сыр: 0
Fluid Mass Dynamic Cplg	Click this for mass inputs
M11= 3.11, M22= 3.11, M33= 19.46, M44= 19	3.46, M13= -5.37, M24= -5.37
	Unit:(2) - Kt. Lbf/in, Ct. Lbf-s/in; Kr. Lbf-in, Cr. Lbf-in-s
	Save Save As Close Help
Dialog	
This added mass (dynamic coupling) is due to the dynamic response of two	structure points <u>C</u> lose
connected by a constrained mass of fluid. The gap between structure is big enough such that the conventional bearin	ng theory does not apply
The points represent the finite element stations of the rotor-bearing system.	g moor account apply.

Only the translational DOFs are considered. The coupling is between stations I and J, not directions x and y.

Mx_IJ (M13) = Mx_ JI (M31)

Mx_J (M33): 19.46

My_IJ (M24) = My_JI (M42)

Unit:(2) - Mass: Lbm

My_J (M44): 19.46

See also Bearings, Linear bearing, and Liquid Seals.

Again, if station J is zero, stations I is connected to the ground,

My_I (M22): 3.11

My_JI (M42): -5.37

Mx_I (M11): 3.11

Mx_JI (M31): -5.37

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_{i} = 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	$\frac{2\muRL^3\boldsymbol{s}\boldsymbol{\omega}}{\mathrm{C}^3\big(\mathbf{l}-\boldsymbol{s}^2\big)^2}$	$\frac{\mu R L^3 \pi}{2C^3(1-\varepsilon^2)^{\frac{3}{2}}}$
	2π film		0	$\frac{\mu R L^3 \pi}{C^3 (1-\varepsilon^2)^{3/2}}$
	π film	Pure Radial Squeeze Motion	0	$\frac{\mu R L^3 \pi \left(2 \varepsilon^2 + 1\right)}{2 C^3 \left(1 - \varepsilon^2\right)^{5/2}}$
	2π film		0	$\frac{\mu R L^3 \pi \left(2 \varepsilon^2 + 1\right)}{C^3 \left(1 - \varepsilon^2\right)^{5/2}}$
Long Bearing	π film	Circular Synchronous Precession	$\frac{24\muR^3Ls\omega}{C^3(2+s^2)(1-s^2)}$	$\frac{12\muR^3L\pi}{C^3(2+\varepsilon^2)(1-\varepsilon^2)^{1/2}}$
	2π film		0	$\frac{24\muR^3L\pi}{C^3(2+\sigma^2)(1-\sigma^2)^{\frac{1}{2}}}$

Where R = damper radius

L = damper axial length

C = radial clearance

- $\omega = whirl speed$
- 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	X
Axial Forces Static Loads Constraints Misalignments Shaft Bow Time Forcing Harmonic Excitation Torsional/A	xial
Units/Description Material Shaft Elements Disks Unbalance Bearings Supports Foundation User's Eleme	
Bearing: 1 of 1 Foundation Add Brg Del Brg Previous Next	
Station I: J: D	
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) Conneting and Decision K. Marco C. Marten	7
Unit:(4) - Geometry: mm, Viscosity: centiPoise, K: N/mm, C: N-s/mm	ן ני
<u>S</u> ave Save <u>A</u> s <u>C</u> lose <u>H</u> elp	
Rotor Bearing System Data	X
Rotor Bearing System Data Axial Forces Static Loads Constraints Misalignments Shaft Bow Time Forcing Harmonic Excitation Torsional/A Units/Description Material Shaft Elements Disks Unbalance Bearings Supports Foundation User's Elements	
Axial Forces Static Loads Constraints Misalignments Shaft Bow Time Forcing Harmonic Excitation Torsional/A Units/Description Material Shaft Elements Disks Unbalance Bearings Supports Foundation User's Eleme	
Axial Forces Static Loads Constraints Misalignments Shaft Bow Time Forcing Harmonic Excitation Torsional/A Units/Description Material Shaft Elements Disks Unbalance Bearings Supports Foundation User's Elements Bearing: 1 of 2 Foundation Add Brg Del Brg Previous Next	
Axial Forces Static Loads Constraints Misalignments Shaft Bow Time Forcing Harmonic Excitation Torsional/A Units/Description Material Shaft Elements Disks Unbalance Bearings Supports Foundation User's Eleme	
Axial Forces Static Loads Constraints Misalignments Shaft Bow Time Forcing Harmonic Excitation Torsional/A 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 Image: Station I Image: Station I	
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Axial Forces Static Loads Constraints Misalignments Shaft Bow Time Forcing Harmonic Excitation Torsional/A Units/Description Material Shaft Elements Disks Unbalance Bearings Supports Foundation User's Eleme 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 0il Viscosity: 1.9e-07 Damper Model: Short Bearing - Circular Synchronous Precession - pi film	nts

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
Short Bearing - Circular Synchronous Precession - pi film Centering Spring P Short Bearing - Circular Synchronous Precession - 2pi film Short Bearing - Pure Radial Squeeze Motion - 2pi film Short Bearing - Pure Radial Squeeze Motion - 2pi film Stiffnes Long Bearing - Circular Synchronous Precession - 2pi film Stiffnes Long Bearing - Circular Synchronous Precession - 2pi film
Unit (2) - Geometry: in, Viscosity: Reyn (Lbf-s/in^2), K: Lbf/in, C: Lbf-s/in
<u>Save</u> Save <u>A</u> s <u>C</u> lose <u>H</u> elp

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 <u>Type 12-Multi-Lobe hydrodynamic bearing (input data in Rotor)</u> or <u>Type 13-Multi-Lobe hydrodynamic bearing (input data in BePerf)</u>.

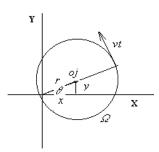
Rotor Bearing System Data		X
Axial Forces Static Loads Constrain Units/Description Material Shaft Ele		Shaft Bow Time Forcing Harmonic Excitation Torsional/Axial Inbalance Bearings Supports Foundation User's Elements
Bearing: 2 of 2	E Foundation	Add Brg Del Brg Previous Next
Station I: 6 J: 0		
Type: 5-Plain Journal Bearing (Sho	rt Bearing - Pi film)	•
Comment: NonLinear Analysis, Example	Chapter 7 Example 1	
Journal/Damper Diameter:	4	Axial Length: 1
Radial Clearance:	0.002	Oil Viscosity: 1e-006
		Unit:(2) - Geometry: in, Viscosity: Reyn (Lbf-s/in^2)
		Save Save As Close Help

See also Bearings, Multi-Lobe Hydrodynamic Bearing

6-Generalized non-linear isotropic 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:

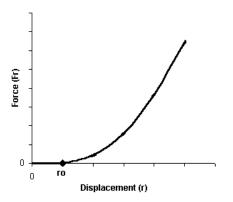
$$-F_{r} = \mu(-F_{r}) \operatorname{sign}(v_{r})$$

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

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

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)

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

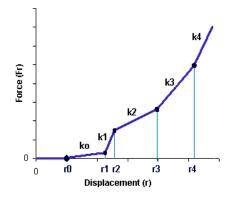
No forces are acting on the shaft when shaft vibration is smaller than the clearance.

When
$$r_0 \leq r$$

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

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 $r_0 \leq \! r < \! r_1$

$$\begin{split} F_r &= k_0 \left(r - r_0 \right) \\ F_t &= \mu_0 \ F_r \end{split}$$

When $r_1 \leq r < r_2$

$$\begin{split} F_r &= k_0 \left(r_1 - r_0 \right) + k_1 \left(r - r_1 \right) \\ F_t &= \mu_1 \ F_r \end{split}$$

When $r_2 \leq r < r_3$

$$\begin{split} F_r &= k_0 \left(r_1 - r_0\right) + k_1 \left(r_2 - r_1\right) + k_2 \left(r - r_2\right) \\ F_t &= \mu_2 \ F_r \end{split}$$

When $r_3 \leq r < r_4$

$$\begin{split} F_r &= k_0 \left(r_1 - r_0 \right) + k_1 \left(r_2 - r_1 \right) + k_2 \left(r_3 - r_2 \right) + k_3 (r - r_3) \\ F_t &= \mu_3 \; F_r \end{split}$$

and so on.

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

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: 6 J: 0
Type: 6-General Non-linear Isotropic Bearing (Rolling Elements)
Comment: Rotor Drop Simulation
Model: Piecewise Curve - 2 (bi-linear) Shaft Diameter: 2
ro: 0.015 ko: 1.0e06 co: 0 fo: 0.20
r1: 0.01 k1: 3.0e06 c1: 0 f1: 0.25
, , , , , ,
Gap Linear Stiffness Damping Friction Coefficient
Unit:(2) - r. in, Force: Lbf, K: Lbf/in, C: 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
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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: 6 J: 0 Type: 6-General Non-linear Isotropic Bearing (Rolling Elements) Comment: Rotor Drop Simulation Modet: Piecewise Curve - 2 (bilinear) Continuous Force-Displacement Curve ro: 0 Piecewise Curve - 2 (bilinear) Piecewise Curve - 2 (bilinear) Continuous Force-Displacement Curve ro: 0 Piecewise Curve - 2 (bilinear) Piecewise Curve - 2 (bilinear) Continuous Force-Displacement Curve ro: 0 Piecewise Curve - 2 (bilinear) Continuous Force-Displacement Curve ro: 0 Piecewise Curve - 2 (bilinear) Continuous Force-Displacement Curve ro: 0 Piecewise Curve - 2 (bilinear) Continuous Force-Displacement Curve ro: 0 Piecewise Curve - 2 (bilinear) Continuous Force-Displacement Curve ro: 0 Piecewise Curve - 2 (bilinear) Continuous Force-Displacement Curve ro: 0 Piecewise Curve - 2 (bilinear) Continuous Force-Displacement Curve ro: 0 Piecewise Curve - 2 (bilinear) Continuous Force-Displacement Curve ro: 0 Piecewise Curve - 2 (bilinear) Continuous Force-Displacement Curve ro: 0 Piecewise Curve - 3 (bilinear) Continuous Force-Displacement Curve ro: 0 Piecewise Curve - 3 (bilinear) Continuous Force-Displacement Curve ro: 0 Piecewise Curve - 3 (bilinear) Continuous Force-Displacement Curve ro: 0 Piecewise Curve - 3 (bilinear) Continuous Force-Displacement Curve ro: 0 Piecewise Curve - 3 (bilinear) Continuous Force-Displacement Curve ro: 0 Piecewise Curve - 3 (bilinear) Continuous Force-Displacement Curve Fielewise Curve - 3 (bilinear) Continuous Force-Displacement Curve Fielewise Curve - 3 (bilinear) Fielewise Curve - 3 (biline
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: 6 J: 0 Type: 6-General Non-linear Isotropic Bearing (Rolling Elements) Comment: Rotor Drop Simulation Modet: Piecewise Curve - 2 (bilinear) Continuous Force-Displacement Curve ro: 0 Piecewise Curve - 2 (bilinear) Piecewise Curve - 2 (bilinear) Continuous Force-Displacement Curve ro: 0 Piecewise Curve - 2 (bilinear) Piecewise Curve - 2 (bilinear) Continuous Force-Displacement Curve ro: 0 Piecewise Curve - 2 (bilinear) Continuous Force-Displacement Curve ro: 0 Piecewise Curve - 2 (bilinear) Continuous Force-Displacement Curve ro: 0 Piecewise Curve - 2 (bilinear) Continuous Force-Displacement Curve ro: 0 Piecewise Curve - 2 (bilinear) Continuous Force-Displacement Curve ro: 0 Piecewise Curve - 2 (bilinear) Continuous Force-Displacement Curve ro: 0 Piecewise Curve - 2 (bilinear) Continuous Force-Displacement Curve ro: 0 Piecewise Curve - 2 (bilinear) Continuous Force-Displacement Curve ro: 0 Piecewise Curve - 2 (bilinear) Continuous Force-Displacement Curve ro: 0 Piecewise Curve - 3 (bilinear) Continuous Force-Displacement Curve ro: 0 Piecewise Curve - 3 (bilinear) Continuous Force-Displacement Curve ro: 0 Piecewise Curve - 3 (bilinear) Continuous Force-Displacement Curve ro: 0 Piecewise Curve - 3 (bilinear) Continuous Force-Displacement Curve ro: 0 Piecewise Curve - 3 (bilinear) Continuous Force-Displacement Curve ro: 0 Piecewise Curve - 3 (bilinear) Continuous Force-Displacement Curve Fielewise Curve - 3 (bilinear) Continuous Force-Displacement Curve Fielewise Curve - 3 (bilinear) Fielewise Curve - 3 (biline
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: 6 J: 0 Type: 6-General Non-linear Isotropic Bearing (Rolling Elements) Comment: Rotor Drop Simulation Model: Flecewise Curve - 2 (bi-linear) ro: 0 Flecewise Curve - 2 (bi-linear) r1: 0 Flecewise Curve - 3 (ti-linear) r1: 0 Flecewise Curve - 4 (quad-linear) r1: 0 Flecewise Curve - 4 (quad-linear)

See also Bearings.

7-Active magnetic bearing (Linear Model) / 8-Active magnetic bearing (Nonlinear Model)

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:

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

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:

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

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

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

The force in the magnetic bearing is:

$$F = F_{o} \cdot \left[\left(\frac{i_{1}}{h_{1}} \right)^{2} - \left(\frac{i_{2}}{h_{2}} \right)^{2} \right]$$

where

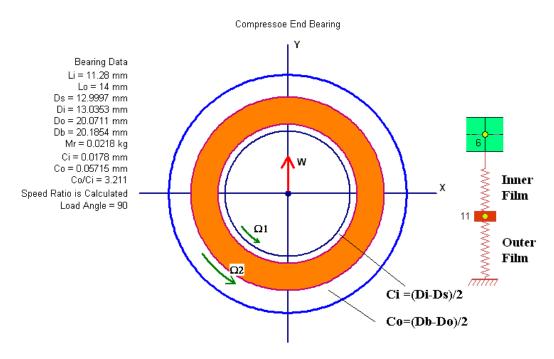
$$h_1 = gap - x_b$$
$$h_2 = gap + x_b$$

Rotor Bearing System Data
Axial Forces Static Loads Constraints Misalignments Shaft Bow Time Forcing Harmonic Excitation Torsional/Axial
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: 2 J: 0 Angle: 0 Sensor Station
Type: 7-AMB - Linear PID & Filter
Comment: Note: Sensor location is not the same as bearing location in this case
X - Direction Y - Direction
Proportional Gain: 3000 3000
Integral Gain: 10 10
Derivative Gain: 10 10
Amplifier Cut-Off Freq: 10000 10000
Unit:(1) - Gp: Lbf/in, Gi: Lbf/(in-s), Gd: Lbf-s/in, Freq: Hz
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: 2 of 2 🔽 Foundation Add Brg Del Brg Previous Next
Station I: 4 J: 0 Angle: 0 Sensor Station
Type: 8-AMB - Nonlinear Transient
Type: 8-AMB - Nonlinear Transient Comment: Nonlinear Transient Analysis X-Direction Y-Direction
Type: 8-AMB - Nonlinear Transient I:
Type: 8-AMB - Nonlinear Transient Comment: Nonlinear Transient Analysis X-Direction Y-Direction
Type: 8-AMB - Nonlinear Transient Comment: Nonlinear Transient Analysis X-Direction Y-Direction Proportional Gain: 399.4 399.4 Air Gap: 0.015
Type: 8-AMB - Nonlinear Transient I: I: <thi:< th=""> <thi:< th=""> I: I:</thi:<></thi:<>
Type: 8-AMB - Nonlinear Transient I: 4 J: 0 Comment: Nonlinear Transient Analysis Y-Direction Y-Direction Proportional Gain: 339.4 339.4 Air Gap: 0.015 Integral Gain: 50000 Current Limit: 5 5 Derivative Gain: 0.5524 0.5524 Bias Current (+): 2 2 Force Constant: 0.001018 Bais Current (-): 2 2
Type: BAMB - Nonlinear Transient I: 4 J: 0 Comment: Nonlinear Transient Analysis X-Direction Y-Direction X-Direction Y-Direction Proportional Gain: 399.4 Air Gap: 0.015 0.015 Integral Gain: 50000 50000 Current Limit: 5 5 Derivative Gain: 0.5524 0.5524 Bias Current (+): 2 2 Force Constant: 0.001018 Bais Current (-): 2 2 Failure at Time (sec.): 0 Zero means N0 failure
Type: 8-AMB - Nonlinear Transient I: 4 J: 0 Comment: Nonlinear Transient Analysis X-Direction Y-Direction Y-Direction Proportional Gain: 339.4 339.4 Air Gap: 0.015 0.015 Integral Gain: 50000 50000 Current Limit: 5 5 Derivative Gain: 0.5524 0.5524 Bias Current (+): 2 2 Force Constant: 0.001018 Bais Current (-): 2 2
I: 4 J: 0 I: 4 J: 0 Comment: Nonlinear Transient Analysis X-Direction Y-Direction Y-Direction Proportional Gain: 399.4 Air Gap: 0.015 Integral Gain: 50000 Current Limit: 5 5 Derivative Gain: 0.5524 0.5524 Bias Current (+): 2 2 Force Constant: 0.001018 Bais Current (-): 2 2 Failure at Time (sec.): 0 Zero means N0 failure
Type: BAMB - Nonlinear Transient I: 4 J: 0 Comment: Nonlinear Transient Analysis X-Direction Y-Direction X-Direction Y-Direction Proportional Gain: 399.4 Air Gap: 0.015 0.015 Integral Gain: 50000 50000 Current Limit: 5 5 Derivative Gain: 0.5524 0.5524 Bias Current (+): 2 2 Force Constant: 0.001018 Bais Current (-): 2 2 Failure at Time (sec.): 0 Zero means N0 failure

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.



Rotor Bearin	ng System Data		×
Axial Forces Units/Descri			
Bearing: 1	l of 2	Add Brg Del Brg Previous Next	
Station I:	10 J: 21 K: 0		
Туре:	9-Floating Ring Bearing/Damper	_	
Comment:	Compressor end		
Г	Floating Ring Data		
	Mass mr: 0.037	Shaft Diameter Ds: 0.4332	
	Inner Length Li: 0.42	Bearing Diameter Db: 0.7525	
	Outer Length Lo: 0.45	Inner Film Viscosity: 1e-006	
	Inner Diameter Di: 0.434	Inner Film Viscosity: 1e-006 Outer Film Viscosity: 2e-006 Bing/Shaft Speed Ratio: 0.2	
	Outer Diameter Do: 0.75	Ring/Shaft Speed Ratio: 0.2	
	0.0004, Co: 0.00125, Co/Ci = 3.125, Max. Es	J	
CI=	0.0004, C0. 0.00123, C0/C1 - 0.123, Max. 23		
		Unit:(2) - Geometry: in, Viscosity: Reyn, M: Lbm	
		Save Save As Close Help	

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

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 Add Brg Del Brg Previous Next
Station I: 10 J: 21 K: 23 A support data is required
Type: 9-Floating Ring Bearing/Damper
Comment: Compressor end with flexible bearing housing
Floating Ring Data
Mass mr: 0.037 Shaft Diameter Ds: 0.4332
Inner Length Li: 0.42 Bearing Diameter Db: 0.7525
Outer Length Lo: 0.45 Inner Film Viscosity: 1e-006 21 Ring
Inner Diameter Di: 0.434 Outer Film Viscosity: 2e-006 23 Bearing
Outer Diameter Do: 0.75 Ring/Shaft Speed Ratio: 0.2
Ci= 0.0004, Co: 0.00125, Co/Ci = 3.125, Max. Estimated Speed Ratio: 0.219373
1 - 0.0004, CO. 0.00123, CO/CI - 0.123, Max. Estimated Speed Hallo. 0.213373
Unit:(2) - Geometry: in, Viscosity: Reyn, M: Lbm
Save Save As Close Help

Rotor Bearing System Data
Axial Forces Static Loads Constraints Misalignments Shaft Bow Time Forcing Harmonic Excitation Torsional/Axia
Axia Folces State Loads Constraints misaignments State bow Time Folcing Hamonic Excitation Tostonal/Axia Units/Description Material Shaft Elements Disks Unbalance Bearings Supports Foundation User's Elements
Support: 1 of 1 Add Delete Previous Next
Station I: 23
Comment: Flexible Bearing Housing
<u>xx xy yx yy</u> M 1 0 0 1
C 5 0 0 5
K 1E+008 0 0 1E+008
Description loss 4 France
Damping Input Format C - Damping Coefficient C Zeta - Damping Factor Zeta-X:
C = Zeta * 2 * SQRT(M * K), Typical Zeta = 0.0001 - 0.02 Zeta-Y: 0.00491228
Unit:(2) - M: Lbm, C: Lbf-s/in, K: Lbf/in
Save As Close Help
Rotor Bearing System Data
Axial Forces Static Loads Constraints Misalignments Shaft Bow Time Forcing Harmonics Torsional/Axial Units/Description Material Shaft Elements Disks Unbalance Bearings Supports Foundation User's Elements
Units/Description Material Shaft Elements Disks Unbalance Bearings Supports Foundation User's Elements
Units/Description Material Shaft Elements Disks Unbalance Bearings Supports Foundation User's Elements Bearing: 1 of 2
Units/Description Material Shaft Elements Disks Unbalance Bearings Supports Foundation User's Elements Bearing: 1 of 2 Add Brg Del Brg Previous Next Station I: 6 J: 11 K: 0
Units/Description Material Shaft Elements Disks Unbalance Bearings Supports Foundation User's Elements Bearing: 1 of 2
Units/Description Material Shaft Elements Disks Unbalance Bearings Supports Foundation User's Elements Bearing: 1 of 2 Add Brg Del Brg Previous Next Station I: 6 J: 11 K: 0 Type: 9-Floating Ring Bearing/Damper Comment: min-max
Units/Description Material Shaft Elements Disks Unbalance Bearings Supports Foundation User's Elements Bearing: 1 of 2 Add Brg Del Brg Previous Next Station I: 6 J: 11 K: 0 Type: 9-Floating Ring Bearing/Damper Comment: min-max
Units/Description Material Shaft Elements Disks Unbalance Bearings Supports Foundation User's Elements Bearing: 1 of 2 Add Brg Del Brg Previous Next Station I: 6 J: 11 K: 0 Type: 9:Floating Ring Bearing/Damper Comment: min-max Floating Ring Data Mass mr: 0.0218 Shaft Diameter Ds: 12.9997 Inner Length Li: 11.28 Bearing Diameter Db: 20.1854 Outer Length Lo: 14 Inner Film Viscosity: 10 Speed
Units/Description Material Shaft Elements Disks Unbalance Bearings Supports Foundation User's Elements Bearing: 1 of 2 Add Brg Del Brg Previous Next Station I: 6 J: 11 K: 0 Type: 9-Floating Ring Bearing/Damper Comment: min-max Floating Ring Data Mass mr: 0.0218 Shaft Diameter Ds: 12.9997 Inner Length Li: 11.28 Bearing Diameter Db: 20.1854 14
Units/Description Material Shaft Elements Disks Unbalance Bearings Supports Foundation User's Elements Bearing: 1 of 2 Add Brg Del Brg Previous Next Station I: 6 J: 11 K: 0 Type: 9-Floating Ring Bearing/Damper Comment: min-max Floating Ring Data Mass mr: 0.0218 Shaft Diameter Ds: 12.9997 Inner Length Li: 11.28 Bearing Diameter Db: 20.1854 Outer Length Lo: 14 Inner Film Viscosity: 10 Speed Depedent Depedent Depede
Units/Description Material Shaft Elements Disks Unbalance Bearings Supports Foundation User's Elements Bearing: 1 of 2 Add Brg Del Brg Previous Next Station I: 6 J: 11 K: 0 Type: 9-Floating Ring Bearing/Damper Comment: min-max Floating Ring Data Mass mr: 0.0218 Inner Length Li: 11.28 Outer Length Li: 11.28 Outer Length Lo: 14 Inner Diameter Di: 13.0353 Outer Diameter Di: 20.0711 Ring/Shaft Speed Ratio: 0.35
Units/Description Material Shaft Elements Disks Unbalance Bearings Supports Foundation User's Elements Bearing: 1 of 2 Add Brg Del Brg Previous Next Station I: 6 J: 11 K: 0 Type: 9-Floating Ring Bearing/Damper Comment: min-max Floating Ring Data Mass mr: 0.0218 Inner Length L: 11.28 Bearing Diameter Ds: 12.9997 Bearing Diameter Db: 20.1854 Inner Film Viscosity: 10 Depedent Variables Yes
Units/Description Material Shaft Elements Disks Unbalance Bearings Supports Foundation User's Elements Bearing: 1 of 2 Add Brg Del Brg Previous Next Station I: 6 J: 11 K: 0 Type: 9-Floating Ring Bearing/Damper Comment: min-max Floating Ring Data Mass mr: 0.0218 Inner Length Li: 11.28 Outer Length Li: 11.28 Outer Length Lo: 14 Inner Diameter Di: 13.0353 Outer Diameter Di: 20.0711 Ring/Shaft Speed Ratio: 0.35
Units/Description Material Shaft Elements Disks Unbalance Bearings Supports Foundation User's Elements Bearing: 1 of 2 Add Brg Del Brg Previous Next Station I: 6 J: 11 K: 0 Type: 9-Floating Ring Bearing/Damper Comment: min-max Floating Ring Data Mass mr: 0.0218 Inner Length Li: 11.28 Outer Length Li: 11.28 Outer Length Lo: 14 Inner Diameter Di: 13.0353 Outer Diameter Di: 20.0711 Ring/Shaft Speed Ratio: 0.35
Units/Description Material Shaft Elements Disks Unbalance Bearings Supports Foundation User's Elements Bearing: 1 of 2 Add Brg Del Brg Previous Next Station I: 6 J: 11 K: 0 Type: 9-Floating Ring Bearing/Damper Comment: min-max Floating Ring Data Mass mr: 0.0218 Inner Length Li: 11.28 Outer Length Lo: 14 Inner Diameter Di: 13.0353 Outer Diameter Di: 13.0353 Outer Diameter Do: 20.0711 Ring/Shaft Speed Ratio: 0.35

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.

e	I D	NR-	
Sheer	l Depend	enr va	arianies
- poor	a popolito		

X

🔽 Speed Dependent Variables 🛛 You may enter the data in the following table, or Input data from BePerf results

-Speed Dependent Variables-

• From Table • From BePerf

	BPM	Inner Viscosity	Outer Viscosity	Inner Clearance	Outer Clearance	Speed Ratio	4
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	-

Speed Dependent Variables

🔽 Speed Dependent Variables 🛛 You may enter the data in the following table, or Input data from BePerf results

Speed	Depend	lent V	ariabl	es

C From Table	• From BePerf	Browse
C From Table	From BePerf	DIOWS

	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	-

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 X + Kyx2 X^2 + Kyx3 X^3 + ... + Kyxn X^n + Kyy1 Y + Kyy2 Y^2 + Kyy3 Y^3 + ... + Kyyn Y^n + Cyx1 Vx + 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:

Units/Description Material Shaft Elements Disks Unbalance Bearings Supports Foundation User's Elements									
Bearin	g:1of1		F F	oundation	Add Brg	Del Brg	Previous	Next	
Station I: 1 J: 0									
Тур	e: 10-Gener	al NonLinear Be	aring/Damper			-			
omme			ction only - SDO	F. Thomson, pp	. 391	_			
. [1		Сххі	Схуі		Kuui	Сухі		
1	<u>Kxxi</u> 1	Kxvi O	0.4		Kyxi O	<u>Kyyi</u> D		Ο	
2	0	0	0.4	0	0	0	0	0	
3	0.5	0	0	ů O	0	0	0	0	
4	0.0	0	0	0	Ū	Ū	0	Ū Ū	
				÷				···	
5 0									

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.

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
Station I: 21 J: 0
Type: 11-Liquid Annular Seal 👤
Comment: Impeller Seal
Method: Black & Jenssen 💽 Inlet Swirl Ratio: 0.5
Seal Length: 1.125 Fluid Density: 0.03607
Shaft Diameter: 10.25 Dynamic Viscosity: 1.52e-007
Radial Clearance: 0.01 Inlet Loss Factor: 0.1
Pressure Drop = dPo + dP1 * rpm + dP2 * rpm^2
dPo: 0 dP1: 0 dP2: 2.9155e-005
Nominal Operating Speed (rpm): 1760
Unit:(2) - Geometry: in, Viscosity: Reyn, Density: Lbm/in^3, Pressure: psi
Save Save As Close Help

See also Bearings, Liquid Annular Seal Dynamic Coefficients.

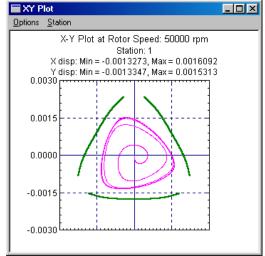
Multi-Lobe Hydrodynamic Bearing

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.

Rotor Bearing System Data								
	straints Misalignr it Elements Disk		Time Forcing Harm Bearings Supports	nonic Excitation Foundation L	Torsional/Axial Jser's Elements			
Bearing: 2 of 2	🗖 Fou	Indation Ad	d Brg Del Brg	Previous	Next			
Station I: 6 J: 0								
Type: 12-Multi-Lobe Bearings -	Identical Lobe (2-D) Reynolds Eq.)	-					
Comment: NonLinear Analysis, Exa	mple: Chapter 7 Ex	ample 1 - Use Type 12						
Length L:	1	Preload:	0					
Diameter D:	4	Offset:	0					
Brg Radial Cearance Cb:	0.002	No. of Lobes:	1					
Oil Dynamic Viscosity:	1e-006	Leading Edge:	0					
		Trailing Edge:	360					
		Lobe Arc (Theta Ang	gles) measured from +>	(axis				
		Unit:(2) - 0	Geometry: in, Viscosity:	Reyn (Lbf-s/in^2),	Angle: deg.			
		Sav	ve Save <u>A</u> s	<u>C</u> lose	<u>H</u> elp			

Rotor Bearing System Data					×
	traints Misalignn t Elements Disk		Time Forcing Bearings Suppo	(I	Torsional/Axial Jser's Elements
Bearing: 1 of 1	🗔 Fou	ndation Ad	d Brg Del E	Brg Previous	Next
Station I: 1 J: 0					
Type: 12-Multi-Lobe Bearings -	Identical Lobe (2-D	Reynolds Eq.)	-		
Comment: Hydrodynamic Journal B	earing, Standard 3	obe symmetric bearing), Chapter 6 Exam	ple A	
Length L:	1	Preload:	0.5	_	
Diameter D:	1.375	Offset:	0.5	-	
Brg Radial Cearance Cb:	0.00175	No. of Lobes:	3	-	
Dig Natilal Cearance Cb. Dil Dynamic Viscosity:	1.5e-006	Leading Edge:	100	-	
Oli Dynamic viscosity.	1.00 000		200	-	
		Trailing Edge: Lobe Arc (Theta And	1	m +X axis	
		20001110 (1110/01111)	gioo, mododiod no		
		.			
		Unit:(2) - (aeometry: in, Visco	osity: Reyn (Lbf-s/in^2),	Angle: deg.
		Sav	e Save,	<u>A</u> s <u>C</u> lose	<u>H</u> elp
		<u></u>			



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.

Rotor Bearing System Data	
Axial Forces Static Loads Constraints Misalignments Shaft Bow Time For Units/Description Material Shaft Elements Disks Unbalance Bearings	rcing Harmonic Excitation Torsional/Axial Supports Foundation User's Elements
Bearing: 1 of 1 Foundation Add Brg	Del Brg Previous Next
Station I: 1 J: 0	
Type: 13-Multi-Lobe Bearings - From BePerf Data File	-
Comment: 3 Lobe bearing from BePerf data file	
FileName: C:\DyRoBeS\Example\AirCompressor_Brg1.LDI	Browse
	Unit:(2) - Kt: Lbf/in, Ct: Lbf-s/in
	- Offic(2) - KC EDI7in, CC EDI-S7in
Save	Save <u>A</u> s <u>C</u> lose <u>H</u> elp

See also <u>Bearings</u> and See also <u>DyRoBeS©_BePerf</u>.

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

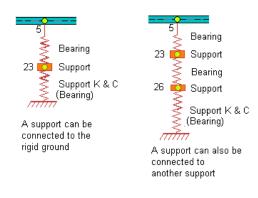
MxxMxyMyxMyy

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.



Rotor Bea	ring System Data				<u> </u>		
Axial Force Units/Des		straints Misalignments aft Elements Disks U		g Harmonic Excitation Supports Foundation	Torsional/Axial User's Elements		
Support: 1 of 2 Add Delete Previous Next							
Station	I: 23						
Comme	ent: 2 translational DOF for	each support station					
	xx	ΧŲ	γx	γų			
м	5	0	0	5			
С	1.97107	0	0	1.97107			
K	3E+006	0	0	3E+006			
	'amping Input Format [©] C - Damping Coefficien C = Zeta * 2 * SQRT(M * K				in, K: Lbf/in		
			Save	Save <u>A</u> s <u>C</u> lose	<u>H</u> elp		

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.

C:\DyRc	oBeS\Example\Large			****
	tion DOF = 15 *******	Foundation Moda	l Analysis ****	****
Mode	Damping Factor	Damping Coef.	Damped Freq.(H	R/S) Hz
	.15534E-01 .69470E-02 .80217E-02			988.81 2211.2 2553.3
15	.21968E-01	-965.22	43926.	6991.1
Rotor Bear	ring System Data			
Axial Force Units/Des Comment		Elements Disks Unt	Shaft Bow Time Forcing balance Bearings Sup on from NASTRAN - 8-15-20	pports Foundation User's Elements
FileName	: C;IDyRoBeS\Example\Lar	geTurbine\Foundation.fdn		Browse
Format		nclude the flexible foundation	n effect.	
	Data format in the FileNam NDOFF, IDamping (2 inte M matrix (Mij) (real [®] 8) K matrix (Kij) (real [®] 8)			
		depend on the value of IDa		
	NDOFF: Number of Degree IDamping: 0 - No Damping	es-Of-Freedom for the Found)	dation.	
	2 - Proportional	atrix, C is required Damping, C = a M + b K, a a amping factors are required	and b are required	
Foun	ndation will be included in the It is assumed to be r		l speed/stability analysis, and lysis and critical speed analy	-
			Uni	it:(2) - M: Lbm, C: Lbf-s/in, K: Lbf/in

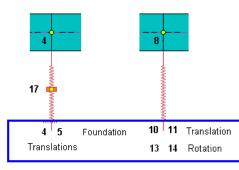
Example: The foundation has 15 degrees-of-freedom and has two connections to the rotor system. The first connection is from a support station 17 to the Foundation DOF 4 and 5. The second connection is from a rotor station 8 to the Foundation DOF 10 and 11 for the X and Y translational displacements,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

Close

<u>H</u>elp

Save <u>A</u>s

<u>S</u>ave



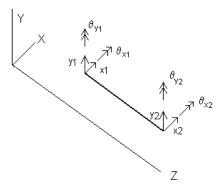
Rotor Bearin	ng System Dat	a			×
Axial Forces Units/Descri		Constraints Misalign Shaft Elements Disk		orcing Harmonic Excitation	Torsional/Axial User's Elements
Bearing: 3	3 of 3	🔽 Fou	Indation Add Brg	Del Brg Previous	Next
Station I:	17	X: 4 Y: 5 Rx:	0 Ry: 0 Angle: 0		
Туре:	0-Linear Constar	nt Bearing		•	
Comment:	Bearing connect	s support station 17 to Fou	ndation DOF 4 and 5		
		Translational E	earing Properties		
Кж	1.35e06	Кху: 0	Cxx: 2	Cxy: 0	
Кух:	0	Куу: 1.35е06	Сух: 0	Суу: 2	
		Rotational	Bearing Properties		
Kaa:	0	Kab: 0	Caa: 0	Cab: 0	
Kba:	0	кы: 0	Cba: 0	Сыр: 0	
				Unit:(2) - Kt: Lbf	/in Ct: Lbf-s/in
			<u>S</u> ave	Save <u>A</u> s <u>C</u> lose	<u>H</u> elp

0.01	Dear	ine C.	mtam D	ata														
	Force		<mark>/stem D</mark> itatic Loar		Const	rainte	1 M	fisalignmei	nto Í Sha	θ Ba		Time	For	ing Harmo	nio Evo	vitation	Lor	sional/Axia
		ription		- N-	Shaft				Unbal			learin	_			ation		's Elements
Be	earing:	4 of	4				C	🔽 Found	lation	[Ad	d Brg		Del Brg	Pre	vious		Next
St	ation l	: 8		X	10	Y:	11	Bx 1	3 Ry:	14	Ar	ngle:	0	- -				
	Туре	0-Li	near Cons	stant B	earing	,				,			-]				
Cor	mmen	Bea	ring conn	ects ro	otor st	ation	8 to F	oundation	DOF 10 a	and 1	1 (X.)	Y), ar	nd 10	3 and 14 (Rx,F	ly)			
						Т	ransla	ational Bea	aring Prope	erties								
	Kxx	: 1.35	e06	-	Kxy:	0			Cx	c [2			Ску: 0		_		
	Кух	: 0		_		1.3	5e06		Cy	c [0)			Суу: 2		_		
		,						-VID-						,				
	K	5.79	o07	_	K-L.	0	nu	ational be	aring Prop	_								
	Каа		eu/	_	Kab:			_		н 0 Г				Cab: 0		_		
	Kba	: 0			КЬЬ:	5.73	9e07		Cba	n 0	J			Сър: 0				
														U	nit(2) -	Kt: Lbf	7in. Ct.	Lbf-s/in
														U	nit:(2) -	Kt: Lbf	7in, Ct	Lbf-s/in
											<u>S</u> av	re		Ui Save <u>A</u> s		Kt: Lbf <u>C</u> lose	7/in, Ct:	Lbf-s/in <u>H</u> elp
											<u>S</u> av	'e					7 in, Ct:	
x	x	x :	x x	x	x	x	x	X	x	x	<u>S</u> av X	re X	x				7in, Ct:	
x	x	x :	x x	x	x	x	x	x	x	x	x x	x x	x				7in, Ct:	
x x	x x	x x	x x x x	x x	x x	x x	x x	x x	x x	x x	x x x x	x x x	x x	Save <u>A</u> s	<u> </u>		7 in, Ct:	
x x x	x x x	x : x : x 4	к х к х ,4 4, <u>-</u>	x x x	x x x	x x x	x x x	x x x	x x x	x x x	x x x x x	x x x x	x x x	Save <u>A</u> s	<u> </u>		7 in, Ct:	
x x x	x x x x	x : x : x 4 x 5	x x x x ,4 4,5 ,4 5,5	x x x x	x x x x	x x x x	x x x x	x x x x	x x x x	x x x x	x x x x x x	x x x x x	x x x x	Save <u>A</u> s	<u> </u>		7in, Ct:	
x x x x x	X X X X X	x : x 4 x 5 x 5	x x x x ,4 4, <u>5</u> , <u>4 5,5</u> x x	x x x x x x x	x x x x x	x x x x x x	x x x x x	x x x x x x	x x x x x x	x x x x x	x x x x x x x	x x x x x x	x x x x x	Save <u>A</u> s	<u> </u>		7 in, Ct:	
x x x x x x	x x x x	x : x 4 x 5 x : x :	x x x x ,4 4,5 ,4 5,5	x x x x	x x x x	x x x x	x x x x	x x x x	x x x x	x x x x	x x x x x x	x x x x x	x x x x	Save <u>A</u> s	<u> </u>		7 in, Ct:	
X X X X X X	X X X X X X	x : x 4 x 5 x : x : x :	x x ,4 4, <u>5</u> , <u>4 5,5</u> x x x x	x x x x x x x x	x x x x x x x	X X X X X X	x x x x x x	x x x x x x	x x x x x x x	x x x x x x	x x x x x x x x	x x x x x x x x	x x x x x	Save <u>A</u> s Conne Statior	<u>c</u> ect to n 17	Close	/in, Ct	
X X X X X X	X X X X X X	x 2 x 4 x 5 x 2 x 2 x 2 x 2 x 2 x 2 x 2 x 2 x 2 x 2	x x x x ,4 4, <u>-</u> ,4 5,5 x x x x x x	x x x x x x x x	x x x x x x x x	X X X X X X	X X X X X X	x x x x x x x	x x x x x x x	x x x x x x x	x x x x x x x x x x x	x x x x x x x x x x	x x x x x x	Save≜s Conne Station	ect to n 17	Close	/in, Ct	
x x x x x x x x	X X X X X X X	x : x 4 x 5 x : x : x : x : x :	x x x x ,4 4, <u>5</u> ,4 5,5 x x x x x x x x x x	x x x x x x x x x x x	x x x x x x x x x	X X X X X X X X	X X X X X X X	x x x x x x x x x	x x x x x x x x x x x	x x x x x x x x x	x x x x x x x x x x x x x	x x x x x x x x x x x x	x x x x x x x	Save <u>A</u> s Conne Statior	ect to n 17	Close	//in, Ct:	
X X X X X X X X	x x x x x x x x x	x 2 x 4 x 5 x 2 x 2 x 2 x 2 x 2 x 2 x 2 x 2 x 2 x 2	x x x x ,4 4, <u>-</u> ,4 5,5 x x x x x x x x x x x x x x x x	X X X X X X X X X X X	x x x x x x x x x x	X X X X X X X X X	x x x x x x x x x x	x x x x x x x x 10,10	x x x x x x x x x 10,11	x x x x x x x x x x x	x x x x x x x x x x x x x x	x x x x x x x x x x x x x x	x x x x x x x x x	Save≜s Conne Station	ect to n 17	Close	//in, Ct:	
	X X X X X X X X	x 2 x 4 x 5 x 2 x 2 x 2 x 2 x 2 x 2 x 2 x 2 x 2 x 2	x x x x ,4 4, <u>-</u> ,4 5,5 x x x x x x x x x x x x x x x x x x x	X X X X X X X X X X X X X	x x x x x x x x x x x	X X X X X X X X X X	X X X X X X X X X X	x x x x x x x 10,10 11,10 x	x x x x x x x x 10,11 11,11	X X X X X X X X X	x x x x x x x x x x x x x x x	x x x x x x x x x x x x x	* * * * *	Save≜s Conne Station	ect to n 17	Close	//in, Ct	
x x x x x x x x x x x	x x x x x x x x x	x : x 4 x 5 x : x : x : x : x : x : x : x :	x x x x ,4 4, <u>-</u> ,4 5,5 x x x x x x x x x x x x x x x x x x x	x x x x x x x x x x x x x	x x x x x x x x x x x	X X X X X X X X X X	X X X X X X X X X X	x x x x x x x 10,10 11,10 x	x x x x x x x x 10,11 11,11	x x x x x x x x x x x	x x x x x x x x x x x x x x x x	x x x x x x x x x x x x x x x	× × × × × × × × ×	Save≜s Conne Station	ect to n 17	Close	//in, Ct	

(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:

$m_{I/2}$	0	0	0		$\int \vec{x}_1$
0		0	0	for	$\left[\ddot{\theta}_{y1} \right]$
0	0	$\frac{m_R}{2}$	0	101	$\begin{vmatrix} \ddot{x}_2 \\ \ddot{\theta}_2 \end{vmatrix}$
0	0	0	$I_{d,R}/2$		$[\mathcal{O}_{y2}]$

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 <u>Rigid/Flexible Disks</u> tab as disks at both ends, and not entered here.

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

$\begin{bmatrix} K_{11} \end{bmatrix}$	K_{12}	K ₁₃	K ₁₄		$\begin{cases} x_1 \end{cases}$
K ₂₁	K ₂₂	K ₂₃	K ₂₄	for	$\left \theta_{y1} \right $
K ₃₁	K_{32}	K_{33}	K ₃₄	101	x_2
K_{41}	K_{42}	K_{43}	K ₄₄		$\left[\theta_{y2} \right]$

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 <u>Shaft Elements</u> data page should be set to **0** for User Supplied Elements.

Rotor Bearing System Data				Σ
Axial Forces Static Loads Constraints Units/Description Material Shaft Elerr		ft Bow Time Forcing an ce Bearings Su		Torsional/Axial User's Elements
User's Element: 1 of 1		Add De	lete Previous	Next
Element: 3 Sub-Element: 1	Length: 18.9			
Comment: Stiffness and Mass Model for	a User's Supplied Element			
1 - Left End	2 - Right End	Total	CG	
Mass: 2.8	3.1	5.9	9.93051	
Diametral Inertia: 68	77	670.523	@CG	
Stiffness Matrix (X-Z Plane): DOF: 1	- x1, 2 - theta y1, 3 - x2, 4	- theta y2		
K11	K12	K13	K14	
K11 1.2E+07	6E+06	-1.2E+07	6E+06	
K21 6E+06	4E+06	-6E+06	2E+06	
K31 -1.2E+07	-6E+06	1.2E+07	-6E+06	
K41 6E+06	2E+06	-6E+06	4E+06	
	Unit:(2) -	M: Lbm, Id: Lbm-in^2, K	(11: Lbf/in, K12: Lbf, K22	Lbf-in, etc.
		<u>S</u> ave Sa	ve <u>A</u> s <u>C</u> lose	<u>H</u> elp
Rotor Bearing System Data				
Axial Forces Static Loads Constraints	Misalignments Sha	ft Bow Time Forcing	Harmonic Excitation	Torsional/Axial

Jnits/D	escrip)	tion	Mater	ial	Shaft Elements	Disks	Unbalance	Bearings	Supports	Foundation	User's Elemer
Shaft: 1 of 1 Starting Station #: 1 Add Shaft Del Shaft Previous Next											
Speed Ratio: 1 Axial Distance: 0 Y Distance: 0 Import*.xls Export*.xls											
Comment: Element 3 Subelement 1 is an User's Element with Mat = 0											
	Ele	Sub	Mat	Lev	Length	Mass ID	Mass OD	Stiff ID	Stiff OD	Corr	iments 🔺
1	1	1	1	0	25	0	18.9	0	0		
2	2	1	1	0	9.4	0	25	0	0		
3	3	1	0	0	18.9	0	15	0	0	User's Ele	ment
4	4	1	1	0	9.4	0	25	0	0		
				0	25	Ω	18.9	···••	Ω		

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.

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 Harmonic Excitation Torsional/Axial Import *.xls Export *.xls											
	Stn:From	Stn:To	Force	Torque	Comments						
1	1	- 301.10 7	1000		Axial Force (Tension)						
2	· ·	•									
3											
4	_										
5	-										
7	-		····.	<u></u>							
8											
9											
10	_										
11	-										
13	-										
14											
15											
16	-										
17	-										
19											
20											
21											
	- 1		1		Unit:(2) - Forces: Lbf, Moments/Torque: Lbf-in						
Inse	ert Row	Delete Row			Onic.(2) * FOICES. EDF, MOMERICS/ FOIQUE: EDF/h						
				<u>S</u> ave	Save <u>A</u> s <u>C</u> lose <u>H</u> elp						

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.

Rotor Bearing System Data									
Units/D Axial Fo	escription prces St	Material Shaft atic Loads Const	Elements Disks raints Misalignments	N		Foundation User's Elerr monic Excitation Torsional/ Import *.xls Export *.xls			
	Stn	Fx	Fy	Mx		Comments	īΙ		
1	3	560	375	0	0	Helical Gear Load	-		
2									
3									
4									
5									
7			······¥						
8			3						
9									
10									
11 12									
13									
14									
15									
16									
17 18									
19						· · · · · · · · · · · · · · · · · · ·			
20									
21									
Inse	rt Row	Delete Row			Unit:(2) - Forc	es: Lbf, Moments/Torque: Lbf	-in		
				<u>S</u> ave	Save <u>A</u> s	<u>C</u> lose <u>H</u> e	lp		

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 <u>Connectivity</u> of <u>Torsional/Axial Data</u> 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.

Ro	tor Be	aring	System D	ata						×
	Units/De Axial For	· · .	n Materi Static Load		Elements aints Misa	Disks U alignments		rings Supporl me Forcing H	ts Foundation User's Eler armonic Excitation Torsional	
	x, y, theta x, and theta y: Fixed or None (0); Shear/Momnet: Release or None (0) Import *.xls Export *.xls									
		Stn	x	Ŷ	Theta x	Theta y	Shear	Moment	Comments]
	1	7	0	0	0	0	0	Release	Coupling	
	2	1	Fixed	Fixed	Fixed	Fixed	0	0	Clamped	
	3	10	Fixed	Fixed	0	0	0	0	Pinned	
	4	16	Fixed	Fixed	0	0	0	0	Pinned	
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		<u>A</u>	<mark>-}──-</mark> }	<u></u> ^		∽ç∰_?──	^ ?}	11 12 10	$\frac{2}{3}$	
		metric				Natural	Geomet	tric	Geometric	
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	Inser	t Row	Dele	te Row					Unit:(4) - None	
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							<u>S</u> ave	Save <u>A</u>	s <u>C</u> lose <u>H</u> e	ip

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

R	otor Be	aring S	ystem Data					×
	Units/D) Axial Fo			aft Elements Disk: constraints Misali	s Unbalance gnments Shaft B			
					[Catenary	Import *.xls Export *.xls	
		Stn	×	y	Theta x	Theta y	Comments	
	1	5	0	0.057	0	0		
	2	34	0	0.0072	0	0		
	3	85	0	0.107	0	0		
	4	91	0	0.1725	0	0		
	5							
	6							
	8							
	9							
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	11							
	12						••	
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							<u>.</u>	-
	Inser	t Row	Delete Row			Unit:(2	2) - Displacement: in, Slope: deg	ree
					<u></u>	ave Save 2	As <u>C</u> lose <u>H</u>	elp

Catenary button

Nat	ural (atenary	(Gravity Sag)	Calculation								
	The g locat The g is min	goal is to m iions where program wil nimized	inimize the mome the failure may o Il perform the optir	n, user needs to pr nt and shear force ccur during startup nization procedure - Initial guess valu	anı du to	d assoc ie to init find the	iated s ial grav optima	tresses at the ci vity sag. Il bearing elevat	oupling static	ons. and any	other	in
		Brg Stn	Initial Guess	Max. Allowable	٠	1		Coupling Stn	Moment	Force	Slope	
	1	5	0.005	0.2		1	1	37	1	0	0	
	2	34	0.005	0.2	_	1	2	69	1	0	0	
	3	85	0.1	0.2			3					
	4	91	0.1	0.2			4		- Weig	hing F	actor	
	5						5			-		
	6						6					-
	7	De	sign Varia	ables			7	0	bjectiv	'e		-
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	9						9		uncuo			-
	10						10					-
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	15				•		15					-
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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.

Rotor	Bearing Sys	stem Data		
	ts/Description al Forces │ Sta ┌─ Curve Fittir	atic Loads Constraints		balance Bearings Supports Foundation User's Elements haft Bow Time Forcing Harmonic Excitation Torsional/Axial
	C None	-	lynominal	Import *.xls Export *.xls
	Stn	×	Ŷ	Comments 🔺
	1 1	0.000000E+000	0.000000E+000	Bearing
	2 5	0.000000E+000	-1.000000E-003	Disk
	3 9	0.000000E+000	0.000000E+000	Bearing
	4 16	0.000000E+000	-1.500000E-003	Overhung Disk
	5			
	6			
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	20			
	21			▼
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_	Inself HOW	Delete Row		
				Save Save As Close Help

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 $t_1 \leq t < t_2$.

Type = 0 Exponentially Decay Force

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

where, $F_m = Par1$, $\lambda = Par2$, $\omega_{exe} = Par3$ (rpm), $\phi = Par4$ (degree) are the input parameters. By properly adjusting these parameters, the excitation can be in many other forms. For example:

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

Type = 1 Purely Harmonics

 $F = F_{o} \cos(\omega_{exo}(t - t_{1})) + F_{s} \sin(\omega_{exo}(t - t_{1}))$

where, $F_{e} = Par1$, $F_{s} = Par2$, $\omega_{exe} = Par3 (rpm)$. Fc and Fs are constant amplitude and ω_{exe} is the excitation frequency.

Type = 2 Step Constant

$$F = F_0$$

where, $F_0 = ParI$. 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

$$F = F_1$$
 at t_1 , and $F = F_2$ at t_2

where, $F_1 = Par1$, and $F_2 = Par2$

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, $F_0 = Par1$, $F_1 = Par2$, $F_2 = Par3$, $F_3 = Par4$

Type = 5 Purely Harmonics with speed dependent amplitude

$$F = F_{s} (\mathbf{n}\Omega)^{\mathbf{m}} \cos(\mathbf{n}\Omega(t-t_{1})) + F_{s} (\mathbf{n}\Omega)^{\mathbf{m}} \sin(\mathbf{n}\Omega(t-t_{1}))$$

where, $F_{c} = Parl$, $F_{s} = Par2$, m = Par3, n = Par4. *Fc* and *Fs* are constant amplitude and Ω is the rotational speed. In is the multiple of the rotational speed, m is the power of the (n Ω). To simulate the blade loss (unbalance), n=1 and m=2, and one needs two entries (one in × direction and one in y direction), such as:

in X-direction:	$F_{x} = \left F\right \Omega^{2}\cos(\Omega(t-t_{1}))$,Par1 = F , Par2 = 0, Par3 = 2, Par4=1
in Y-direction:	$F_y = \left F\right \Omega^2 \sin\left(\Omega(t-t_1)\right)$, 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:

 $\begin{array}{ll} \text{in X-direction:} & F_x = \left|F\right| \Omega^2 \, \cos(\Omega(t-t_1) + \phi) \\ \text{in Y-direction:} & F_y = \left|F\right| \Omega^2 \, \sin(\Omega(t-t_1) + \phi) \end{array}$

	escripti		m Dat Material	5	lements [Disks Unb	alance Bear	inas Í Supi	ports Fou	ndation 🗍 User's Ele	mer
Axial Fo	· · .	_	Loads	1				ne Forcing	Harmonic E		1/Ax
indiri o	1000	0.000	. 20000	1 contourd					i i dimonio E		
	The ex	citation	n is appl	ied when t	1 <= t < t2.	Press <f1> for</f1>	more detail				
	Stn	Dir	Туре	Start (t1)	Stop (t2)	Par 1	Par 2	Par 3	Par 4	Comments	•
1	1	1	3	0	0.2	0	100	0	0	Linear Increase	
2	1	1	3	0.2	0.4	100	0	0	0	Linear Decrease	
3	1	1	2	0	0.1	100	0	0	0	Step Constant	
4	1	1	3	0.1	0.2	100	0	0	0	Linear Decrease	
5	5	1	1	0	0.1	0.01	0	3600	0	Cosine force	
6	8	0	5	0.01	100000	0.000129504	0	2	1	Unbalance Force	
7	8	1	5	0.01	100000	0.000129504	0	2	1	Sum of 12 & 13	
8	8	2	5	0.01	100000	0	0.000129504	2	1	= line 10	
9											
10	Lin	e 1 &	Line 2	! form a tr	iangular e	excitation	\sim				
11											
12	Lin	e 3 &	Line 4	form a st	tep const	ant then line	arly decrease	ed force		<u> </u>	
13					1						
14	Lin	e 5 is	a han	monic forc	e						
15											
16	Lin	e 6 is	an un	balance fe	orce (dir =	: 0)					
17											
18	Lin	e 7 ar	nd Line	e 8 combi	nation for	ms the same	e unbalance f	orce as Lir	ne 6		
19											
20											
21											•
							haib (2) Easy I be	orl b f-in la			
Inse	rt Row	(Delete	Row]		L) MIC. (2) - FIN. LUI	01 201 11, 20	mda: 17s, Wi	exc: cpm, Phase: deg	jree

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

	rces S	Static Lo	bads	Constraints		nments Shaf isks Unbala	5 C	ıs Supj		undation	Torsional/Axial User's Elements ort *.xls
		0.1	T	1-0.4		1-0.4	Distance		· · ·		
1 13 20	Ele 8	Sub 1	Type O	Left A 0		Left Ang. 0	Right Amp. 0.8	. H	ight Ang. 0	Comm oz-in	
Inse	ert Row	D	elete F	łow	Unit(2	2) - Type () (1):	Mass (Magnet)	Unbalar	nce, Amp: (oz-in (Lbf), l	Phase: deg
							Save	Save <u>A</u>	ls	<u>C</u> lose	<u>H</u> elp
	earing	Syster	n Dat	a							Þ
	escriptio rces 9	n Ma Static Lo	iterial bads 1	Shaft Elen Constraints	: Misaligr		t Bow Time F				User's Elements Torsional/Axial
	rces Stn	n Ma Static Lo citation Dir 1	iterial bads 1 is appli Type	Shaft Elen Constraints ied when t' Start (t1)	Misaligr 1 <= t < t2. Stop (t2)	nments Shaf . Press <f1> f Par 1</f1>	t Bow Time From Time	orcing Par 3	Harmonic Par 4	Excitation	Torsional/Axial
	escriptio rces 5 The ex Stn 8	n Ma Static Lo citation Dir 1	iterial bads is appli <u>Type </u> 5	Shaft Elen Constraints ied when t	Misaligr 1 <= t < t2. <u>Stop (t2)</u> 100000	nments Shaf . Press <f1> f</f1>	t Bow <u>Time F</u> or more detail	orcing	Harmonic	Excitation	Torsional/Axial
Axial Fo	rescriptio rces S The ex Stn 8 Stn	n Ma Static Lo citation Dir Option Dir	iterial bads is appli Type 5 1 1: U Type	Shaft Elen Constraints ied when t' <u>Start (t1)</u> 0.01 se Dir == Start (t1)	Misaligr 1 <= t < t2. Stop (t2) 100000 = 0 Stop (t2)	nments Shaf Press <f1> f Par 1 0.000129504 Par 1</f1>	t Bow Time From Time	Par 3 2 Par 3	Harmonic Par 4 1 Par 4	Excitation Comr 0.8/386.	Torsional/Axial
Axial Fo	rces Str The ex Stn Stn Stn 8	n Ma Static Lo citation Dir 0 Option Dir 1	iterial bads is appli <u>Type</u> 5 1 1: Us Type 5	Shaft Elen Constraints ied when t' <u>Start (t1)</u> 0.01 se Dir == <u>Start (t1)</u> 0.01	: Misaligr 1 <= t < t2. <u>Stop (t2)</u> 100000 = 0 <u>Stop (t2)</u> 100000	Press <f1> f Press <f1> f Par 1 0.000129504 Par 1 0.000129504</f1></f1>	t Bow Time F or more detail Par 2 0 Par 2 0	Par 3 2 Par 3 2 Par 3 2	Harmonic Par 4 1 Par 4 1 1 1	Excitation Comr 0.8/386 Comn oz-in/16	Torsional/Axial
Axial Fo	escriptio rces 5 The ex Stn 8 Stn 8 8 8 8	n Ma Static Lo citation Dir 0 Option Dir 1 1 2	terial) bads is appli Type 5 1 1: U Type 5 5 5	Shaft Elen Constraints ied when t' <u>Start (t1)</u> 0.01 se Dir == Start (t1)	Misaligr 1 <= t < t2. <u>Stop (t2)</u> 100000 = 0 <u>Stop (t2)</u> 100000 100000	Press Shaf Par 1 0.000129504 Par 1 0.000129504 0.000129504 0	t Bow Time From Time	Par 3 2 Par 3	Harmonic Par 4 1 Par 4	Excitation Comr 0.8/386 Comn oz-in/16	Torsional/Axial
Axial Fo	escriptio rces 5 The ex Stn 8 Stn 8 8 8 8	n Ma Static Lo citation Dir O Option Dir 1 2 Option	terial) bads is appli Type 5 1 1: U Type 5 5 5	Shaft Elen Constraints ied when t' <u>Start (t1)</u> 0.01 se Dir == <u>Start (t1)</u> 0.01 se Dir ==	Misaligr 1 <= t < t2. <u>Stop (t2)</u> 100000 = 0 <u>Stop (t2)</u> 100000 100000	Press <f1> f Par 1 0.000129504 Par 1 0.000129504 0 2</f1>	t Bow Time F or more detail Par 2 0 Par 2 0	Par 3 2 Par 3 2 2 2	Harmonic Par 4 Par 4 1 1	Excitation Comr 0.8/386 Oz-in/16. oz-in/16.	Torsional/Axial nents nents /386.086 /386.086 /386.086 v

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 $|\mathbf{Q}|$ is the excitation amplitude and $\boldsymbol{\omega}$ is the excitation frequency in rad/sec for the above expression.

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

 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:

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

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:

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

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

Roto	r Bea	aring S	ystem	Data						X
	its/De al For	escription rces 9	Mat Static Lo	erial ads			balance Bearing Shaft Bow Time		ts Foundation armonic Excitatio	
	S	Steady Sta			xcitation: Excitation requency (cpm = wo			r Speed	T	
			W	p: 0	W1:	1	W2: 0			
			· ·	tude M o: 0	ultiplier (A = Ao + A1		wexc^2) A2: 1			
					tate Harmonic Excita ation frequency. A=					
		Ele	Sub	Dir	Left Amp.	Left Ang.	Right Amp.	Right Ang.	Comment	ts 🔺
	1	Ele 5	Sub 1	Dir O	Left Amp. 0.1	Left Ang. O	Right Amp. 0	Right Ang. 0	Commen Unbalance Forc	
	1 2 3									
	2 3 4									
	2 3 4 5									
	2 3 4									
	2 3 4 5 6 7 8									
	2 3 4 5 6 7 8 9	5								
	2 3 4 5 6 7 8 9 10	5			0.1					e

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

Roto	r Bea	ring S	ystem	Data							X
· · · · · · · · · · · · · · · · · · ·	its/Des ial Forc	cription es S	Mat Static Lo	erial bads	Shaft Elements Constraints Mis		oalance Bearin haft Bow Time		ts Foundatior armonic Excitatio		
	Sta	eady Sta	- Excit	ation Fr o: 0		o + w1 * rpm + v : 4	v2 * rpm^2) W2: 0	or Speed			
			Ad	o: 1 :eady SI	ultiplier (A = Ao + A A1 ate Harmonic Exci ation frequency. A	: 0 tation: Q = A * (0	A2: 0				
		Ele	Sub	Dir	Left Amp.	Left Ang.	Right Amp.	Right Ang.	Commer	nts 🔺	
	1 2 3 4 5 6 7 8 9 9 10	5			2500	0	0	0	4X excitation		
	Insert i	Row	Del	ete Rov	v			1	Unit:(2) - Amp:		
							Save	Save A	s Close	. I II	elp

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

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 Harmonic Excitation Torsional/Axial
Steady State Harmonic Excitation: Excitation Frequency varies at a Constant Rotor Speed Excitation Frequency (cpm) Start: 10000 Stop: 25000 Increment. 1000
Amplitude Multiplier (A = Ao + A1 * wexc + A2 * wexc^2) Ao: 1 A1: 0 A2: 0
Steady State Harmonic Excitation: Q = A * Q * cos (wexcT + phase) wexc = excitation frequency. A= Amplitude multiplier, omega = rotor speed
Ele Sub Dir Left Amp. Left Ang. Right Amp. Right Ang. Comments
1 5 1 2 765 0 0 Y direction only 2 3 4
Insert Row Delete Row Unit:(2) - Amp: Lbf, Phase: deg
Save Save As Close Help

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

$$\begin{split} \mathcal{Q}_x = & |\mathcal{Q}| \cdot \cos(\omega t + \alpha) \text{ in X direction, and} \\ \mathcal{Q}_y = & |\mathcal{Q}| \cdot \sin(\omega t + \alpha) \text{ in Y direction} \end{split}$$

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 <u>Connectivity</u>, <u>Modal Damping</u>, and <u>Steady State</u> <u>Excitation</u> are for torsional or axial vibration. The buttons for <u>Non-Linear Coupling</u>, the time dependent excitations in equations entered in <u>Torsional Excitations in Equation</u>, the time dependent excitations in data files entered in <u>Torsional Excitations in Data Files</u>, <u>Driving Torque</u>, and <u>Load Torque</u> are only for torsional transient analysis. <u>Engine</u> <u>Excitation</u> is the excitation inputs for the gas or diesel engines. <u>Reciprocating Excitation</u> 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.

Rotor Bearing System Da	ita	
Units/Description Materia Axial Forces Static Loa	- 1 v - v - v - v - v - v - v - v - v -	Unbalance Bearings Supports Foundation User's Elements ments Shaft Bow Time Forcing Harmonics Torsional/Axial
Torsiona	al and/or Axial Data	Torsional Time Dependent Excitations
	Linear Connectivity	Excitations from files
	Non-Linear Couplings/Connections	Torsional Startup Transient Torques
	Modal Damping	Load Torque
		Reciprocating Excitation (nX Harmonics) Engine Excitation
	Steady State Excitation (Single Harmonic)	Recip. Torque
Lateral-1	Torsional-Axial Geared Coupling	
	G	ear Mesh Data
		Save Save As <u>C</u> lose <u>H</u> elp

If you want to analyze the coupled Lateral-Torsional-Axial 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: Torsional or Axial 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: Rigid or Flexible 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	Stnl	StnJ	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
0	Torsional	8	0	Flexible Link	0	174400	Low P. Tuurbine
1	Axial	1	2	Flexible Link	100000	0	Axial Stiffness
2	Axial	15	0	Flexible Link	500000	0	Thrust Bearing
3							
4							
5							
6							
7							
8							

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.

Max 5	5 Non-Linear #	: 1		2		3		4	5		
	Station I:	11		25		0		0	0		ок
	Station J:	12	_	33		0		0	0		<u> </u>
Ra	ated Stiffness:	24700000	0	1000	0000000	0		0	0		Angle
Visc	ous Damping:	0	—	0		0	_	0			Degree
	mic Magnifier:	5.5	_	0		0	_	0			C Rad
Dyna	-		_	-		-	_	-	_		, nau
	Backlash:	0		0.000	15	0		0	0		
	Torque-1	Angle-1	Tor	rque-2	Angle-2	Torque-3	Angle-3	Torque-4	Angle-4	Torque-5	Angle-5 🔺
-											
1	0	0									
2	62500	0.017									
2	62500 125000	0.017 0.034		Гhe fi	rst one	is a rubb	er cou	pling wh	ich has	s torque	
2 3 4	62500	0.017							ich has	s torque	
2	62500 125000 250000	0.017 0.034 0.066	¢	curve	and Dy	/namic M	agnifie	r.			
2 3 4 5	62500 125000 250000 375000	0.017 0.034 0.066 0.098	¢	curve	and Dy		agnifie	r.			
2 3 4 5 6	62500 125000 250000 375000 500000	0.017 0.034 0.066 0.098 0.128	¢	curve	and Dy	/namic M	agnifie	r.			
2 3 4 5 6 7	62500 125000 250000 375000 500000	0.017 0.034 0.066 0.098 0.128	¢	curve	and Dy	/namic M	agnifie	r.			
2 3 4 5 6 7 8	62500 125000 250000 375000 500000	0.017 0.034 0.066 0.098 0.128	¢	curve	and Dy	/namic M	agnifie	r.			
2 3 4 5 6 7 8 9 10 11	62500 125000 250000 375000 500000	0.017 0.034 0.066 0.098 0.128	¢	curve	and Dy	/namic M	agnifie	r.			
2 3 4 5 6 7 8 9 10	62500 125000 250000 375000 500000	0.017 0.034 0.066 0.098 0.128	¢	curve	and Dy	/namic M	agnifie	r.			

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:

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

where

 $C_{\rm r}, J_{\rm r}, \omega_{\rm r}$ are modal damping, modal mass/inertia, and natural frequency

Modal De	ampings		D
Modal D	amping Factor or Dam	ping Ratio	<u>0</u> K
	Damping Factor	Comments	▲
1	0.015	1st mode - 1.5%	
2	0.021785	2nd mode - 2.1785%	_
3			
4			
5			
6			
7			
8			
9			
10			
11			
12			
13			
14			
15			•
<u>I</u> nsert I	Row <u>D</u> elete Ro	w Unit(1) - None,	Typical: 0.01-0.10

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 amplitude can be a constant, first order, or second order of the frequency (speed).

$$\begin{split} \omega_{exc} &= \left(C_0 + C_1 \ \Omega + C_2 \ \Omega^2\right) \times \ (2\pi / 60) \\ T &= \ \left(T_c \ \cos(\omega_{exc} t) + T_s \ \sin(\omega_{exc} t)\right) \times \ T_{multiplier} \\ T_{multiplier} &= \ M_0 + \ M_1 \ \Omega \ + \ M_2 \ \Omega^2 \end{split}$$

where Ω is the rotor speed (rpm)

 $\begin{array}{l} C_0, C_1, C_2; \ \ \mbox{Frequency coefficients}. \\ M_0, M_1, M_2; \ \ \mbox{Torque/force multiplier coefficients}. \end{array}$

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.

Torsio	nal/Axial S	teady S	tate Excitation		×
Torc	itation Freq.(rp que/Force Mu itation Freq = (Itiplier:	Mo: 0	C1: 5 M1: 0 rque/Force = Component	C2: 0 <u>O</u> K M2: 1 nt x (M0 + M1 x rpm + M2 x rpm^2)
	T/A	Stn	Cos Component	Sin Component	Comments 🔺
1	Torsional	1	307.82	0	Wexc= 5 x rpm, T1=0.1*Tp
2					
3					
4					
5					
6					
7					
8					
9					
10					
11					
12					•
<u>I</u> nse	ert Row	<u>D</u> elete Ro	w	U	nit:(1) - T: Torque: Lbf-in; A: Force: Lbf

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.

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

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:

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

xcitati	ions gi	ven in	Eqs., such	as Short C	ircuit Tor	que, etc.					٥
Freque	ncy (Hz)): 60	R	ated Torque ((Trated): 35	000				<u>0</u> K	
Type 0	: Torque	= Trateo	∃X (T0 exp(-a	i0 t) + T1 exp(-a1 t) sin(wt+	b1) + T2 exp(-	-a2 t) sin(2v				
	Stn	Туре	TO	aO	T1	a1	Ь1	T2	a2	Ь2	
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			2 phase sł They will n row is the	ypical 2000 nort circuit. ot occur at steady stat y state drivi	The secor the same f e driving to	nd row is th time. This rque and th	e 3 phas is just a l ie fourth i	e short circ illustration. row is the li	uit torque. The third pad torque.		
11 12 Insert	Row	Del	ete Row						Unit:(0) - Coi	nsistent U	▼ Inits

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:

00

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.

Torsional	excitation vs. time			×
The maxi Linear int Such as Time has The data	tion Number and Excitation File. The file contains 2 columns, 1-Time (sec) mum number of points at each excitation file is 2000. erpolation is used between points to allow sudden changes. a retangular shape excitation where the same time point has two different of to be in the increasing order, that is t (i) <= t (i+1) equal sign is allowed for starts from T0 to Tf. For a periodic excitation, the period starts from Ts. i.e. responsibility to ensure the excitation unit is consistent with the selected to	excitations. sudden char t. T0<=Ts <tf. unit system.</tf. 	uge in exc	<u>D</u> K sitation
Station	FileName (use Browse to select)		eriodic xcitation	T start
1	C:\DyRoBeS\Example\Torsion\TorData.dat	Browse	•	0
0		Browse		0
0		Browse		0
0		Browse		0
0		Browse		0

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.

$$N_{syn} = \frac{120 \times LineFrequency (Hz)}{No. of Poles}$$
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_{ose} 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:

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

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:

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

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

	% speed	% Tavq	% Tosc	▲
1	0	44	24.13	<u></u> K
2	10	44	24.84	
3	20	44.35	25.55	Motor Station: 1
4	30	44.71	26.26	Line Francisco -
5	40	44.35	27.68	Line Frequency: 50 Hz 💌
6	50	43.65	29.1	No. of Poles: 4
7	60	42.23	30.52	
8	70	40.81	31.94	Synchronous RPM: 1500
9	80	38.32	33.35	
0	90	34.99	34.77	Rated Torque: 26716
1	92.5	33.71	34.77	
2	95	32.29	33.35	Driver: Synchronous Motor
3	97.5	30.16	28.39	For Synchronous Motor
4	100	24.84	14.19	
5				Torque = Tavg + Tosc sin (Wexc t)
6				Wexc = twice the slip frequency
7				
8				
9				
20				-

For induction motor

Motor	Torque				×
	% speed	% Tavq	Tosc	•	
1	0	45	° % I ₀ 5.57		<u>o</u> k
2	10	48	11.32	$a_0 \square$	
3	20	52	% T ₁ 3.56		Motor Station: 1
4	30	57	38.5	a_1	
5	40	62	0	ø	Line Frequency: 60 Hz 💌
6	50	68	% T ₂ 1.44		No. of Poles: 6
7	60	75		a_2	
8	70	85		\$2.	Rated RPM: 1200
9	80	98	0		-
10	90	95	0		Rated Torque: 288866
11	100	28	0		D : Industion Mater
12					Driver: Induction Motor
13					For Synchronous Motor
14					Torque = Tavg + Tosc sin (Wexc t)
15					Wexc = twice the slip frequency
16					wexc = (wice the silp frequency
17					Course File Collins Course
18					Curve Fit: Spline Curve 🗨
20				_	
	1			•	
Inse	rt Row Delete	Row Import	*.xls <u>E</u> xport	*.xls	Unit:(1) - Torque: Lbf-in

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, %T₁, ω_1 , ϕ_1 , %T₂, ω_2 , ϕ_2

%speed – rpm in percentage of the rated speed. %Tavg, %T₁, %T₂ - 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.

2 10 48 5 20 0 0 3 20 52 5 20 0 0 4 30 57 5 20 0 0 5 40 62 5 20 0 0 6 50 68 5 20 0 0 7 60 75 5 20 0 0 8 70 85 5 20 0 0	0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0	0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0	<u>OK</u> Motor Station: 1 Line Frequency: No. of Poles:
3 20 52 5 20 0 0 4 30 57 5 20 0 0 5 40 62 5 20 0 0 6 50 68 5 20 0 0 7 60 75 5 20 0 0 8 70 85 5 20 0 0	0 0 0 0 0 0 0 0 0 0	0 0 0 0 0	Line Frequency:
4 30 57 5 20 0 0 5 40 62 5 20 0 0 6 50 68 5 20 0 0 7 60 75 5 20 0 0 8 70 85 5 20 0 0	0 0 0 0 0 0 0 0	0 0 0 0 0	Line Frequency:
5 40 62 5 20 0 0 6 50 68 5 20 0 0 7 60 75 5 20 0 0 8 70 85 5 20 0 0	0 0 0 0 0 0	0 0 0	
6 50 68 5 20 0 0 7 60 75 5 20 0 0 8 70 85 5 20 0 0	0 0 0 0	0	
7 60 75 5 20 0 0 8 70 85 5 20 0 0	0 0	0	No. of Poles:
8 70 85 5 20 0 0			
	0 0		
9 80 98 5 20 0	the second star because and the second starts	0	Rated RPM: 800
	0 0	0	
10 90 95 5 20 0	0 0	0	Rated Torque: 288866
11 100 28 5 20 0	0 0	0	
12			Driver: GE Power
13			For Synchronous Motor
14			
15			Torque = Tavg + Tosc sin (Wexc

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 <u>Shaft Elements</u> 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.

Rate	ed Torque: 28	6716		Up to 6 Load	l Torques can l	be applied	<u>0</u> K
St	ation Number:	9	12	15	0	0	0
[% 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.

Load T	orques							×
Rat	ed Torque: 🛛	38866		Up to 6 Load	Torques can l	oe applied	<u>0</u> K	
S	tation Number:	3	0	0	0	0	0	
	% 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								•
<u>I</u> nse	ert Row De	elete Row	Import *.xls	<u>Export</u> *.xls	:	Unit:(1) - Torque: Lbf	-in

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.

umber of Cylinders: 12 Rated Power: 1480 @ RPM: 1200 Stroke Cycle onnecting Rod Length: 18 Bore Dia: 9.375 Stroke: 8.5 C 2 4 eciprocating Weight (Piston+Rod) per cylinder: 66.5 Displacement: 7041 BMEP: 138.73 Gas Torque Multiplie: 1 Harmonic Coefficients of Tangential Pressure Import * Gas Torque Vs. RPM BMEP1: 100 BMEP2: 150 Export * Constant C Linear 2 2nd order 0 6 35.76 31.34 47.25 42 1.0 18.21 54.52 23.45 73 1.5 1.73 48.64 0.87 65.3 2.4 180 5 0 K ✓ 2.5 -6.69 27.37 -9.97 35.3 3.0 -6.78 20.04 -9.72 25.3 -6.69 27.37 -9.97 25.3 3.1 460 0 K ✓ - - - - - - - - - - - -	escription:	GE Waukesha	a Engine Mo	del L7042	:GSI - 60 de	g Vee										
Connecting Rod Length: 18 Bore Dia: 9.375 Stroke: 8.5 C 2 • 4 aciprocating Weight (Piston+Rod) per cylinde: 66.5 Displacement: 7041 BMEP: 138.73 Gas Torque Multiplie: 1 Harmonic Coefficients of Tangential Pressure Import * Gas Torque Vs. RPM • 2nd order BMEP1: 100 BMEP2: 150 Export * Cylinder Firing Angle Stn No. Condition • • 1.0 18.21 54.52 23.45 73. 1.R 0 4 0K • • 1.0 18.21 54.52 23.45 73. 1.R 0 4 0K • • 2.0 3.29 37.46 6.38 49. 2.4 180 5 0K • 3.5 -7.72 13.82 -10.88 17.2 13.8 10. 3.5 -7.72 13.82 -10.88 17.3 3.4 660 6 0K • • 5.0 -5.2 2.87 -6.97 2.2 5.0 -5.2 2.87 <td>mber of f</td> <td>ulinders: 12</td> <td>_</td> <td>Bated Po</td> <td>wer: 1480</td> <td> @</td> <td>вем: 1200</td> <td> Stro</td> <td>oke Cycle</td> <td><u>0</u>K</td>	mber of f	ulinders: 12	_	Bated Po	wer: 1480	@	вем: 1200	Stro	oke Cycle	<u>0</u> K						
Order Bore Dia: 9.375 Stroke: 8.5 aciprocating Weight (Piston+Rod) per cylinder: 66.5 Displacement: 7041 BMEP: 138.73 Gas Torque Multiplier: 1 Harmonic Coefficients of Tangential Pressure Import * Gas Torque Vs. RPM © 2nd order 0rder BMEP1: 100 BMEP2: 150 Export * Constant © Linear © 2nd order 0rder BMEP1-Cos BMEP1: 150 Export * Quinder Firing Angle Stn No. Condition • 1.0 18.21 54.52 23.45 73.3 1.4 420 4 0K • 2.0 -3.29 37.46 -6.38 49.9 2.1 180 5 0K • 3.5 -7.72 13.82 -10.08 17.7 3.4 660 6 0K • - <td< td=""><td></td><td>symbols. [</td><td></td><td></td><td>men. price</td><td></td><td></td><td></td><td></td><td></td></td<>		symbols. [men. price											
Gas Torque Multiplie: 1 Harmonic Coefficients of Tangential Pressure Import * Gas Torque Vs. RPM BMEP1: 100 BMEP2: 150 Export * Order BMEP1: 100 BMEP1: 150 Export * Quinder Firing Angle Stn No. Condition • 1.5 3.7.34 47.25 42 1.R 0 4 0K< ▼	nnecting	Rod Length: 1	8	Bore D	ia.: 9.375	S	troke: 8.5		2 9 4							
BMEP1: 100 BMEP2: 150 Export ** Order Order BMEP1: 50 BMEP2: 50 BMEP1 Order BMEP1: 50 BMEP1: 50 BMEP2: 50 BMEP1 Order BMEP1: 50 BMEP1: 50 BMEP2: 50 BMEP2 Order BMEP1: 50 BMEP1: 50 BMEP1: 50 BMEP2: 50 BMEP2: 50 BMEP2: 50 BMEP1: 50 S 3.0 G <th <="" colspan="6" td=""><td>ciprocatir</td><td>ng Weight (Pisto</td><td>on+Rod) per</td><td>cylinder:</td><td>66.5</td><td>Di</td><td>splacement: 70</td><td>)41</td><td>BMEP: 138.7</td><td>3</td></th>	<td>ciprocatir</td> <td>ng Weight (Pisto</td> <td>on+Rod) per</td> <td>cylinder:</td> <td>66.5</td> <td>Di</td> <td>splacement: 70</td> <td>)41</td> <td>BMEP: 138.7</td> <td>3</td>						ciprocatir	ng Weight (Pisto	on+Rod) per	cylinder:	66.5	Di	splacement: 70)41	BMEP: 138.7	3
Order BMEP1-Cos BMEP1-Sin BMEP2-Cos BMEF1 Quinder Firing Angle Stn No. Condition • 0.5 35.76 31.34 47.25 42. 1.R 0 4 0K<	Gas Toro	que Multiplier:		_		F	larmonic Coeffici	ients of Tangenl	tial Pressure	Import *.xls						
Order BMEP1-Cos BMEP1-Sin BMEP2-Cos BMEF1 Quinder Firing Angle Stn No. Condition • 0.5 35.76 31.34 47.25 42. 1.R 0 4 0K<	с. т.					BMEP1:	100	BMEP2: 15	50	Export *.xls						
Uplinder Firing Angle Stn No. Condition ▲ 1.R 0 4 0K ▲ 1.L 420 4 0K<								,	_							
Stn No. Condition 10 18.21 54.52 23.45 73. 1-R 0 4 0K - 1.5 1.73 48.64 0.87 65.3 1-R 0 4 0K - 3.29 37.46 6.6.38 49.9 1-L 420 4 0K - 3.29 37.46 6.6.38 49.9 2-L 180 5 0K - 3.5 -7.72 13.82 -10.88 10.0 3-L 660 6 0K - 3.5 -7.72 13.82 -10.88 17.0 4-R 600 7 0K - 5.0 -5.2 2.87 -6.97 2.2 4-R 300 7 0K - 5.0 -5.2 2.87 -6.97 2.2 5-R 120 8 0K - 5.0 -3.79 0.18 5.0 -1.18 -0.02 -2.2 6-R	C Con	stant 🔿 l	.inear	2nd or	rder	Order	BMEP1-Cos	BMEP1-Sin	BMEP2-Cos	BMEP2-Sin						
Dylinder Firing Angle Stn No. Condition Image: Cond						0.5			47.25	42.63						
1-R 0 4 0K • 1-R 0 4 0K • 1-L 420 4 0K • 2-R 480 5 0K • 2-R 480 5 0K • 2-L 180 5 0K • 3-R 240 6 0K • 3-R 240 6 0K • 3-R 240 6 0K • 3-L 660 6 0K • 4-R 600 7 0K • 4-L 300 7 0K • 5-R 120 8 0K • 5-L 540 8 0K • 6-R 360 9 0K • 6-L 60 9 0K • 80 1.32 2.07 1.63 2.2 80 1.32 1.16 1.74 -0.35 2.2 80 1.32<			a. 11			1.0	18.21			73.45						
1-L 420 4 0K • 2-R 480 5 0K • 2-L 180 5 0K • 3-L 660 6 0K • 4-L 300 7 0K • 4-L 300 7 0K • 5-R 120 8 0K • 5-L 540 8 0K • 6-R 360 9 0K • 6-L 60 9 0K • 8.5 1.32 2.07 1.63 2.2 8.0 1.32 2.07 1.63 2.2 8.0 1.32 2.07 1.63 2.2 8.0 1.32 2.183 0.69 2.2 8.0 1.32 1.16 1.74 0.35 <td>·</td> <td></td> <td></td> <td></td> <td>tion</td> <td></td> <td></td> <td></td> <td></td> <td>65.73</td>	·				tion					65.73						
2-R 480 5 0K • 2-L 180 5 0K • 3-R 240 6 0K • 3-L 660 6 0K • 4-L 300 7 0K • 5-R 120 8 0K • 5-L 540 8 0K • 6-R 360 9 0K • 6-L 60 9 0K • 6-L 60 9 0K • 8.5 -1.32 -1.18 -4.02 8.0 -1.32 -2.07 -1.63 8.0 -1.32 -1.98 -1.14 9.0 -1.22 -1.83 -0.69 9.0 -1.22 -1.83 -0.69										49.37						
8.0 -1.32 -2.07 -1.63 -2. 8.5 -1.32 -1.98 -1.14 -2. 9.0 -1.22 -1.83 -0.69 -2. 9.5 -1.16 -1.74 -0.35 -2.					•	2.5	-6.69	27.37	-9.97	35.26						
8.0 -1.32 -2.07 -1.63 -2. 8.5 -1.32 -1.98 -1.14 -2. 9.0 -1.22 -1.83 -0.69 -2. 9.5 -1.16 -1.74 -0.35 -2.					-	3.0	-6.78	20.04	-9.72	25.31						
8.0 -1.32 -2.07 -1.63 -2. 8.5 -1.32 -1.98 -1.14 -2. 9.0 -1.22 -1.83 -0.69 -2. 9.5 -1.16 -1.74 -0.35 -2.					-	3.5	-7.72	13.82	-10.88	17.23						
8.0 -1.32 -2.07 -1.63 -2. 8.5 -1.32 -1.98 -1.14 -2. 9.0 -1.22 -1.83 -0.69 -2. 9.5 -1.16 -1.74 -0.35 -2.					-	4.0	-7.05	8.33	-9.63	10.33						
8.0 -1.32 -2.07 -1.63 -2. 8.5 -1.32 -1.98 -1.14 -2. 9.0 -1.22 -1.83 -0.69 -2. 9.5 -1.16 -1.74 -0.35 -2.	3-L				•	4.5	-6.05	4.94	-7.99	5.82						
8.0 -1.32 -2.07 -1.63 -2. 8.5 -1.32 -1.98 -1.14 -2. 9.0 -1.22 -1.83 -0.69 -2. 9.5 -1.16 -1.74 -0.35 -2.	4-R				-	5.0	-5.2	2.87	-6.97	2.79						
8.0 -1.32 -2.07 -1.63 -2. 8.5 -1.32 -1.98 -1.14 -2. 9.0 -1.22 -1.83 -0.69 -2. 9.5 -1.16 -1.74 -0.35 -2.					-	5.5	-4.83	1.2	-6.36	0.15						
8.0 -1.32 -2.07 -1.63 -2. 8.5 -1.32 -1.98 -1.14 -2. 9.0 -1.22 -1.83 -0.69 -2. 9.5 -1.16 -1.74 -0.35 -2.					-	6.0	-3.79	-0.18	-5.03	-1.66						
8.0 -1.32 -2.07 -1.63 -2. 8.5 -1.32 -1.98 -1.14 -2. 9.0 -1.22 -1.83 -0.69 -2. 9.5 -1.16 -1.74 -0.35 -2.	5-L				•	6.5	-2.93	-1.18	-4.02	-2.42						
8.0 -1.32 -2.07 -1.63 -2. 8.5 -1.32 -1.98 -1.14 -2. 9.0 -1.22 -1.83 -0.69 -2. 9.5 -1.16 -1.74 -0.35 -2.					-	7.0	-2.12	-1.79	-3.08	-2.71						
8.5 -1.32 -1.98 -1.14 -2. 9.0 -1.22 -1.83 -0.69 -2 9.5 -1.16 -1.74 -0.35 -2.	6-L	60	9	OK	• •	7.5	-1.62	-2.1	-2.36	-2.86						
9.0 -1.22 -1.83 -0.69 -2 9.5 -1.16 -1.74 -0.35 -2.						8.0	-1.32	-2.07	-1.63	-2.64						
9.5 -1.16 -1.74 -0.35 -2.						8.5	-1.32	-1.98	-1.14	-2.48						
						9.0	-1.22	-1.83	-0.69	-2.3						
10.0 -0.94 -1.65 -0.16 -1.						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.						10.5	-0.63	-1.61	-0.1	-1.75						
11.0 -0.37 -1.38 -0.07 -1.						11.0	-0.37	-1.38	-0.07	-1.55						
11.5 -0.26 -1.16 -0.11 -1.						11.5	-0.26	-1.16	-0.11	-1.42						
							-0.14	-0.95	-0.12	-1.31						

gine (Reo	ciprocating)	Torsiona	l Excita	tion					
escription:	GE Waukesha	Engine Mo	del L7042	2GSI - 60 de	eg Vee				ок
lumber of C	Cylinders: 12		Rated Pr	ower: 1480) @	RPM: 1200	Stro	ke Cycle	
Connecting	Rod Length: 18	8	Bore D	ia.: 9.375	s	troke: 8.5		2 🖲 4	
eciprocatir	ng Weight (Pisto	n+Rod) per	cylinder:	66.5	Di	splacement: 70	041	BMEP: 138.73	3
Gas Toro	que Multiplier: 1		_		н	armonic Coeffic	ients of Tangen	tial Pressure	Import *.xls
- Gas Toro	que Vs. RPM —				BMEP1:	100	BMEP2: 15	50	Export *.xls
C Con	stant 🔿 L	.inear	2nd o	rder	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
Cylinder	Firing Angle	Stn No.	Condi	tion 🔺	1.5	1.73	48.64	0.87	65.73
1-R	0	4	OK	-	2.0	-3.29	37.46	-6.38	49.37
1-L	420	4	OK	-	2.5	-6.69	27.37	-9.97	35.26
2-R	480	5	OK	-	3.0	-6.78	20.04	-9.72	25.31
2-L	180	5	OK	▼ ▼ ▼	3.5	-7.72	13.82	-10.88	17.23
3-R	240	6	OK	-	4.0	-7.05	8.33	-9.63	10.33
3-L	660	6	OK	-	4.5	-6.05	4.94	-7.99	5.82
4-R	600	7	OK	•	5.0	-5.2	2.87	-6.97	2.79
4-L	300	7	OK	-	5.5	-4.83	1.2	-6.36	0.15
5-R	120	8	OK	-	6.0	-3.79	-0.18	-5.03	-1.66
5-L	540	8	OK	-	6.5	-2.93	-1.18	-4.02	-2.42
6-R	360	9	Misfire	•	7.0	-2.12	-1.79	-3.08	-2.71
6-L	60	9	OK	• •	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.

	GE Waukesha				Vee				
umber of (Cylinders: 12		Rated Pow	ver: 1480	@	RPM: 1200	Stro	oke Cycle	<u>o</u> k
	Rod Length: 1	8	Bore Dia	_		troke: 8.5	— c	2 🖲 4	
onneeding	nod Eengal. J	•	DOIC DIG						
eciprocati	ng Weight (Pisto	n+Rod) per	cylinder: 6	6.5	Di	splacement: 70	41	вмер: 138.73	3
Gas Tor	que Multiplier: 1		_		н	armonic Coeffici	ents of Tangeni	ial Pressure	Import *.xls
					BMEP1:		BMEP2: 15		
-Gas Tor	que Vs. RPM—				DINET I.	1.00		_	Export *.xls
C Con	stant 🔿 L	.inear	2nd ord	er	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
Cylinder	Firing Angle	Stn No.	Conditio	on 📥	1.5	1.73	48.64	0.87	65.73
1-R	0	4	OK	-	2.0	-3.29	37.46	-6.38	49.37
1-L	420	4	OK	•	2.5	-6.69	27.37	-9.97	35.26
2-R	480	5	OK	-	3.0	-6.78	20.04	-9.72	25.31
2-L	180	5	OK	•	3.5	-7.72	13.82	-10.88	17.23
3-R	240	6	OK	-	4.0	-7.05	8.33	-9.63	10.33
3-L	660	6	OK		4.5	-6.05	4.94	-7.99	5.82
4-R	600	7	OK	• • •	5.0	-5.2	2.87	-6.97	2.79
4-L	300	7	OK	-	5.5	-4.83	1.2	-6.36	0.15
5-R	120	8	OK	•	6.0	-3.79	-0.18	-5.03	-1.66
5-L	540	8	OK	•	6.5	-2.93	-1.18	-4.02	-2.42
6-R	360	0	Misfire	-	7.0	-2.12	-1.79	-3.08	-2.71
6-L	60	9	OK	v v	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

Reciprocating Torsional Excitation

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

Re	ciproca	ing Torsiona	al Excitation		
I	Description	: DyRoBeS_Ro	otor: Reciprocatin	g Data	<u>o</u> k
[Cylinder	Crank Angle	Rotor Stn No.	FileName - Click to Browse	
	1	0	1	C:\DyRoBeS1600\Example\Throw-1.dat	
	2	180	2	C:\DyRoBeS1600\Example\Throw-2.dat	
	3				
	4				
	5				
	6				
	7				
	8				
	9				
	10 11				
	12				
ľ	12		<u>. </u>		
	Insert R	ow Delete F	Row Import	*.xis Export *.xis Unit(2) - Angle: deg., Torquue:	Lbf-in

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

2-128679207

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

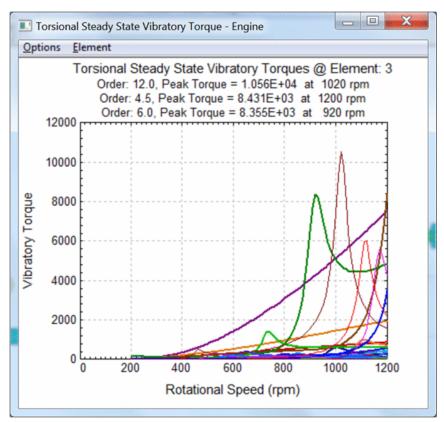
1 2423 -13467

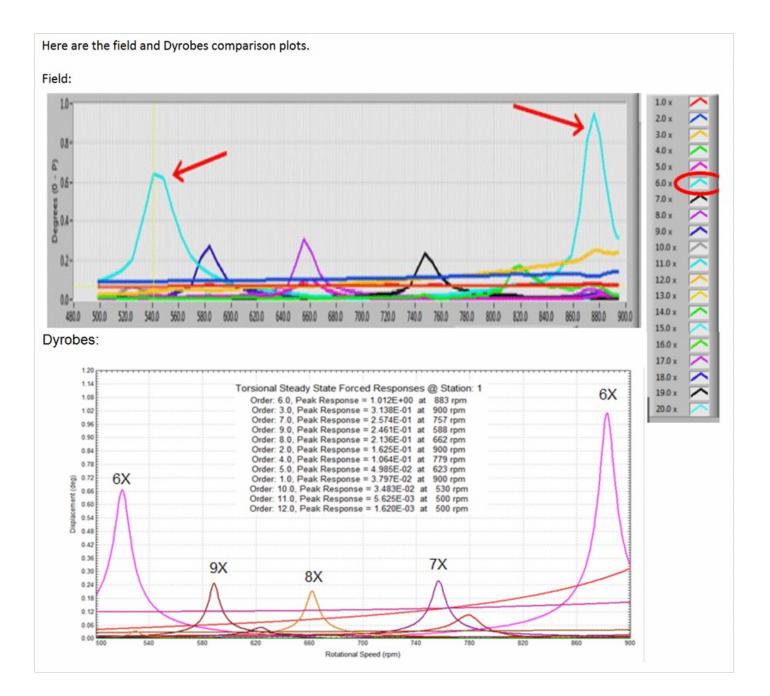
2-12867 9207

3 - 2973 5670

Other harmonics

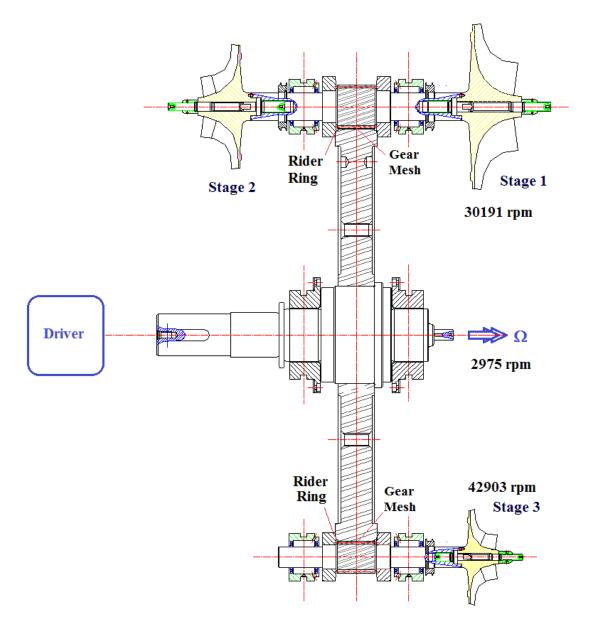
24 - 13.9 25.3





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).



tor Bearing System Data	
Jnits/Description Material Shaft Elements Disks Axial Forces Static Loads Constraints Misalignme	Unbalance Bearings Supports Foundation User's Elements Ints Shaft Bow Time Forcing Harmonics Torsional/Axial
Torsional and/or Axial Data Linear Connectivity Non-Linear Couplings/Connections Modal Damping	Torsional Time Dependent Excitations Excitations in Equations Excitations from files Torsional Startup Transient Torques Motor Driving Torque Load Torque
Steady State Excitation (Single Harmonic) Lateral-Torsional-Axial Geared Coupling Gear	Engine Excitation Recip. Torque
	Save Save As Close Help
eral-Torsional-Axial Gear Mesh Coupling	
GearMesh: 1 of 4	Add Delete Previous Next
Gear 1 Station I: 16 Pitch Diameter: 792.277	Oriving C Driven Gear Gear Data Pressure Angle: 20
Gear 2 Station J: 27 Pitch Diameter: 78.0711	Angular Position: 0

Gear Mesh Stiffness and Damping Matrices in (r't'a') coordinates

a'

0

0

1E+07

С

ľ,

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a'

Gear Mesh Data

Ľ,

0

0

0

Damping

ť

0

100

0

a'

0

0

100

Stiffness

ť

0

1E+09

0

<u>0</u>K

Thrust Collar (Rider Ring)-

Diameter: 0

Axial K: 0

Axial C: 0

Unit:(4) - Angle: deg., Length: mm, K:N/mm, C:N-s/mm

K r' ť a'

ľ,

0

0

0

Lateral-Torsional-Axial Gear Mesh Coupling		
GearMesh: 3 of 4 ⊏ Gear1	Add Delete P	revious Next <u>O</u> K
Station I: 16 Pitch Diameter: 792.277	Oriving Oriven Gear	Gear Data Pressure Angle: 20
Gear 2 Station J: 27 Pitch Diameter: 78.0711	Angular Position: 0	Helix Angle: 18
		Rider Ring Data
	ping Matrices in (r't'a') coordinates	Thrust Collar (Rider Ring)
Stiffness	Damping	
K r' t' a'	C r' ť a'	Diameter: 100.34
r' 0 0 0	r' 0 0 0	Axial K: 1E+08
۲ O O O	<u>к</u> 0 0 0	
a' 0 0 0	a' 0 0 0	Axial C: 100
	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.

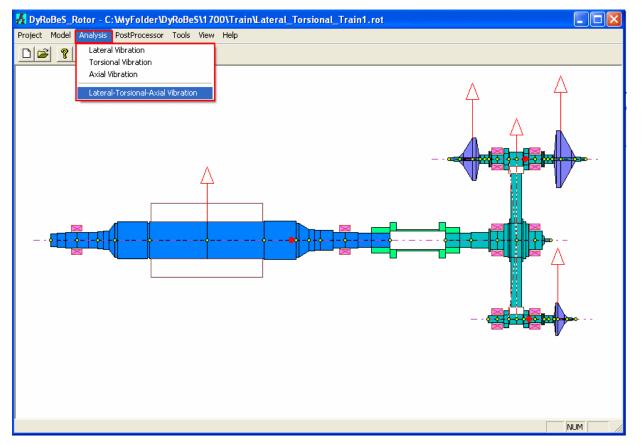
Analysis

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_x) 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.

🛃 DyRoBeS_F	Rotor - C	:WyFolderW	yRoBe	S\Exa	mples\AirCompressor_Linear.rot
Project Model	Analysis	PostProcessor	Tools	View	Help
02 3		l Vibration	1		
		nal Vibration	-		
	Axial \	'ibration]		

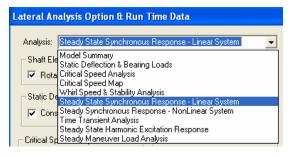
The run time data screen for the Lateral Vibration Analysis is shown below:

Lateral Analysis Option & Run Time Data				×
Analysis: Steady State Synchronous Response - Shaft Element Effects I Rotatory Inertia Shear Deformation Static Deflection Critical Sp Critical Sp Spin/Whit	Gyroscopic 🔽 Gz	Transient Analysis RPM: 35000 Constant Time-Start: 0 Ending: 0.05	StartUp / ShutDown t Speed: 35000 rpm Mass Unbalance Const. Unbalance	Gravity (g) X: 0 Y: -386.088 Z: 0
Spin/Whit Ratio: 1 Npts: 0	K - Min: 1000 Max: 1e+008 to be varied at All	Increment: 1e-006 Solution Method Newmark-beta Initial Cs: No	Shart Bow Disk Skew Gravity (X,Y) Gravity (Z) Static Loads Time Forcing Misalignment	None zero Gz Vertical Rotor
RPM-Starting: 10000 RPM-State Ending: 40000 Ending: Increment: 500 Increment No. of Modes: 6 Excitation	ding: 40000	Effects: Mass Unbalance Const. Unbalance Shaft Bow	Steady State Harmonic Excitation RPM-Starting: 10000 Ending: 40000 Increment: 100 Excitation Shaft: 1 ✓ All Shafts with same speed	Cancel
Steady Maneuvers (Base Constant Translational A Speed (RPM): 0 Acceleration - X: 1		te) Turn Rate - X: 0	Y: 0 Ref Pos	5: 0

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.

Analysis: Steady State Synchronous Shaft Element Effects Rotatory Inertia Shear Def		- Transient Analysis- RPM: 35000	StartUp / ShutDown	Gravity (g)
Static Deflection Constrained Bearing Stations Critical Speed Analysis	Critical Speed Map Spin/Whirl Ratio: 1 Bearing K - Min: 1000	Time-Start: 0 Ending: 0.05 Increment: 1e-006	Mass Unbalance	Y: -386.088 Z: 0 None zero Gz Vertical Rotor
Spin/Whirl Ratio: 1 No. of Modes: 5 Stiffness: Kxx	Npts: 0 Max: 1e+008 Stiffness to be varied at Bearings: All Steady State Synchronous Respon	Solution Method Newmark-beta Initial Cs: No	Gravity (Z) Gravi	Run
RPM-Starting: 10000 Ending: 40000 Increment: 500 No. of Modes: 6	RPM-Starting: 10000 Ending: 40000 Increment: 100 Excitation Shaft: 1	Effects: Mass Unbalance Const. Unbalance Shaft Bow Disk Skew	RPM-Starting:10000Ending:40000Increment:100Excitation Shaft:1	Cancel
	All Synchronized Shafts	Misalignment Rate) Turn Rate - X: 0	All Shafts with same speed	s: 0

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.

Analysis: Static Deflection & Bearing	Loads 🔹	- Transient Analysis-		Gravity (g)
Shaft Element Effects Image: Rotatory Inertia Image: Rotatory Content	ormation 🔽 Gyroscopic 🔲 Gz	RPM: 35000	StartUp / ShutDown	× 0
Static Deflection Constrained Bearing Stations Critical Speed Analysis Spin/Whirl Ratio: 1 No. of Modes: 5 Stiffness: Kxx ▼	Critical Speed Map Spin/Whirl Ratio: 1 Bearing K - Min: 1000 Npts: 0 Max: 1e+008 Stiffness to be varied at Bearings: All	Time-Start: 0 Ending: 0.05 Increment: 1e-006 Solution Method Newmark-beta Initial Cs: No	 ✓ Mass Unbalance Const. Unbalance Shaft Bow Disk Skew Gravity (X,Y) 	Y: -386.088 Z: 0 None zero Gz Vertical Rotor
Whirl Speed and Stability Analysis	Steady State Synchronous Response	e Analysis	Steady State Harmonic Excitation	Run
RPM-Starting: 10000 Ending: 40000 Increment: 500 No. of Modes: 6	Increment: 100	Effects: Mass Unbalance Const. Unbalance Shaft Bow Disk Skew	RPM-Starting: 10000 Ending: 40000 Increment: 100 Excitation Shaft: 1	Cancel
Steadu Maneuvers (Base Constant T		Misalignment	All Shafts with same speed	

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:

Analysis: Static Deflection & Bearing Loads Shaft Element Effects Image: Critical Speed Map Static Deflection Critical Speed Map Time-Start: 0 Time-Start: 0 Time-Start: 0 Static Deflection Y:
Static Deflection Critical Speed Map Y: 386.088
Critical Speed Analysis Spin/Whirl Ratio: 1 Ending: 0.05 Shaft Bow Z: 0 Critical Speed Analysis Bearing K · Min: 1000 Increment: 1e-006 Gravity (X,Y) None zero Gz
No. of Modes: 5 Stiffness to be varied at Stiffness to be varied at Bearings: All Whirl Speed and Stability Analysis State Synchronous Response Analysis
RPM-Starting: 10000 RPM-Starting: 10000 Effects: RPM-Starting: 10000 Ending: 40000 Ending: 40000 Image: Image:
Image: All Synchronized Shafts Image: Misalignment Image: All Shafts with same speed Steady Maneuvers (Base Constant Translational Acceleration and/or Turn Rate) Image: All Shafts with same speed Speed (RPM): 0 Acceleration -X: 0 Y: 0 Ref Pos: 0

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.

Lateral Analysis Option & Run T	īme Data			
Analysis: Critical Speed Analysis Shaft Element Effects Rotatory Inertia Shear Defe Static Deflection Constrained Bearing Stations Critical Speed Analysis Spin/Whirl Ratio: 1	Tritical Speed Map Spin/Whirl Ratio: Npts:	Transient Analysis RPM: 35000 Const Time-Start: 0 Ending: 0.05 Increment: 1e-006 Solution Method	Gravity (Z)	Gravity (g) X: 0 Y: -386.088 Z: 0 None zero Gz Vertical Rotor
No. of Modes: 5 Stiffness: Kxx -	Stiffness to be varied at Bearings: All Steady State Synchronous Response	Newmark-beta Initial Cs: No	Static Loads Time Forcing Misalignment Steady State Harmonic Excitation	Run
RPM-Starting: 10000 Ending: 40000 Increment: 500 No. of Modes: 6	RPM-Starting: 10000 Ending: 40000 Increment: 100 Excitation Shaft: 1 Image: All Synchronized Shafts Image: All Synchronized Shafts	Effects: Mass Unbalance Const. Unbalance Shaft Bow Disk Skew Misalignment	RPM-Starting: 10000 Ending: 40000 Increment: 100 Excitation Shaft: 1 I I	Cancel
	ranslational Acceeration and/or Turn R ration - X: 0 Y: 0	ate) Turn Rate - X: 0	Y: 0 Ref Po	s: 0

For sample outputs, click Critical Speed Analysis PostProcessor.

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..

nalysis: Critical Speed Map	•	- Transient Analysis-		Gravity (g)
Shaft Element Effects V Rotatory Inertia V Shear De	formation 🔽 Gyroscopic 🔲 Gz	RPM: 35000	StartUp / ShutDown	×: 0
Static Deflection Constrained Bearing Stations Critical Speed Analysis Spin/Whirl Ratio: 1 No. of Modes: 5 Stiffness: Kxx ▼	Critical Speed Map Spin/Whirl Ratio: 1 Bearing K - Min: 1000 Npts: 0 Max: 1e+008 Stiffness to be varied at Bearings: All	Time-Start: 0 Ending: 0.05 Increment: 1e-006 Solution Method Newmark-beta Initial Cs: No		Y: -386.088 Z: 0 None zero G Vertical Roto
Whirl Speed and Stability Analysis RPM-Starting: 10000 Ending: 40000 Increment: 500 No. of Modes: 6	Steady State Synchronous Respons RPM-Starting: 10000 Ending: 40000 Increment: 100 Excitation Shaft: 1	Effects: Effects: Mass Unbalance Const. Unbalance Shaft Bow Disk Skew Misalignment	Steady State Harmonic Excitation RPM-Starting: 10000 Ending: 40000 Increment: 100 Excitation Shaft: 1 All Shafts with same speed	Cancel

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:

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

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

where

 $\begin{array}{ll} \mathcal{E} & & \text{damping factor, or damping ratio} \\ \mathcal{S} &> 0 & & \text{Stable or damped system} \\ \mathcal{S} &= 0 & & \text{threshold of instability} \\ \mathcal{S} &< 0 & & \text{unstable system} \end{array}$

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

Analysis: Whirl Speed & Stability An Shaft Element Effects	<u> </u>	Transient Analysis RPM: 35000	StartUp / ShutDown	Gravity (g)
▼ Rotatory Inertia ▼ Shear Del Static Deflection ▼ Constrained Bearing Stations Critical Speed Analysis Spin/Whirl Ratio: 1 No. of Modes: 5 Stiffness: Kxx ▼	formation ♥ Gyroscopic Gz Critical Speed Map Spin/Whirl Ratio: 1 Bearing K - Min: 1000 Npts: 0 Max: 1e+008 Stiffness to be varied at Bearings: All	Consta Time-Start: 0 Ending: 0.05 Increment: 1e-006 Solution Method Newmark-beta Initial Cs: No	ant Speed: 35000 rpm Mass Unbalance Const. Unbalance Shaft Bow Disk Skew Gravity (X,Y) Gravity (Z) Static Loads Time Forcing Misalignment	X: 0 Y: -386.088 Z: 0 None zero G Vertical Roto
Whil Speed and Stability Analysis RPM-Starting: 10000 Ending: 40000 Increment: 500 No. of Modes: 6 Steady Maneuvers (Base Constant T	Excitation Shaft: 1	Effects: Mass Unbalance Const. Unbalance Shaft Bow Disk Skew Misalignment	Steady State Harmonic Excitation RPM-Starting: 10000 Ending: 40000 Increment: 100 Excitation Shaft: 1 ✓ All Shafts with same speed	Cancel

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.

Lateral Analysis Option & Run Time Data		<u> </u>
Analysis: Steady State Synchronous Response - Linear System Shaft El Model Summary Shaft El Static Deflection & Bearing Loads Image: Static Deflection & Bearing Loads Critical Speed Analysis Critical Speed Map Whill Speed & Stability Analysis Static D Steady State Synchronous Response - Linear System Image: Construction Steady State Synchronous Response - NonLinear System Time Transient Analysis Steady State Harmonic Excitation Response Critical Sp Steady Maneuver Load Analysis Spin/Whirl Ratio: No. of Modes: Stiffness: Kxx	Transient Analysis RPM: 35000 Constant Speed: 35000 rpm Time-Start: Mass Unbalance Ending: 0.05 Shaft Bow Increment: 1e-006 Solution Method Gravity (X,Y) Solution Method Gravity (Z) Newmark-beta Static Loads Initial Cs: No	Gravity (g) X: 0 Y: -386.088 Z: 0 None zero Gz Vertical Rotor
Ending: 40000 Ending: 40000 Increment: 500 Increment: 100 No. of Modes: 6 Excitation Shaft: 1	Effects: RPM-Starting: 10000 Mass Unbalance Ending: 40000 Const. Unbalance Increment: 100 Disk Skew Excitation Shaft: 1 Misalignment I All Shafts with same speed	Cancel

For sample outputs, click Steady State Response Plots.

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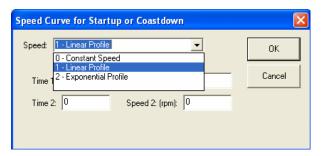
Time Transient Analysis (Time Domain)

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 <u>Speed Curvers</u> here.

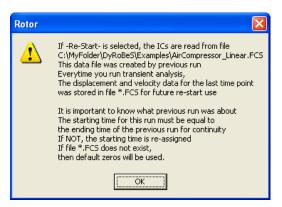
Lateral Analysis Option & Run	Time Data			X
Analysis: Time Transient Analysis	-	Transient Analysis-		Gravity (g)
Shaft Element Effects	formation 🔽 Gyroscopic 🔲 Gz	RPM: 35000	StartUp / ShutDown	×: 0
Static Deflection Constrained Bearing Stations Critical Speed Analysis	Critical Speed Map Spin/Whirl Ratio: 1 Bearing K - Min: 1000	Time-Start: 0 Ending: 0.05 Increment: 1e-006	✓ Mass Unbalance ✓ Const. Unbalance ✓ Shaft Bow ✓ Disk Skew	Y: -386.088 Z: 0 None zero Gz Vertical Rotor
Spin/Whirl Ratio: 1 No. of Modes: 5 Stiffness: Kxx	Npts: 0 Max: 1e+008 Stiffness to be varied at Bearings: All	Solution Method Newmark-beta Initial Cs: No		Bun
Whirl Speed and Stability Analysis	Steady State Synchronous Respo	nse Analysis	Steady State Harmonic Excitation	
RPM-Starting: 10000 Ending: 40000 Increment: 500	Ending: 40000	Effects: Mass Unbalance Const. Unbalance Shaft Bow	RPM-Starting: 10000 Ending: 40000 Increment: 100	Cancel
No. of Modes: 6	Excitation Shaft: 1	 Disk Skew Misalignment 	Excitation Shaft: 1	
	Franslational Acceeration and/or Turn leration - X: 0 Y: 772.1		Y: 0 Ref Po	s: 0



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

Rotor	
1	The default initial conditions (ICs) are zeros IF -Yes- is selected, the ICs are read from file C:\MYFolder\DyRoBeS\Examples\AirCompressor_Linear.ICS This data file must be in the same directory as C:\MYFolder\DyRoBeS\Examples\AirCompressor_Linear.ROT The ASCII data formats in *.ICS are Station No., X, Y, Xdot, Ydot repeat this line for all the non-zero ICs Example: MYRotorExample.ICS 1 0.0010 0 20 0 2 0.0011 0 30 0
	()

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.

Lateral Analysis Option & Run	Fime Data			
Analysis: Steady State Harmonic Ex Shaft Element Effects Image: The Analysis Static Deflection Image: Constrained Bearing Stations Critical Speed Analysis Spin/Whirl Ratio: No. of Modes: Stiffness: Kxx		Transient Analysis RPM: 35000 Const Time-Start: 0 Ending: 0.05 Increment: 1e-006 Solution Method Newmark-beta Initial Cs: No	Gravity (Z)	Gravity (g) X: 0 Y: -386.088 Z: 0 None zero Gz Vertical Rotor
- Whirl Speed and Stability Analysis	Steady State Synchronous Respon		Steady State Harmonic Excitation	Run
RPM-Starting:10000Ending:40000Increment:500No. of Modes:6	Ending: 40000	Effects: Mass Unbalance Const. Unbalance Shaft Bow Disk Skew Misalignment	RPM-Starting: 10000 Ending: 40000 Increment: 100 Excitation Shaft: 1	Cancel
· · ·	All Synchronized Shafts	late)	All Shafts with same speed	s: 0

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).

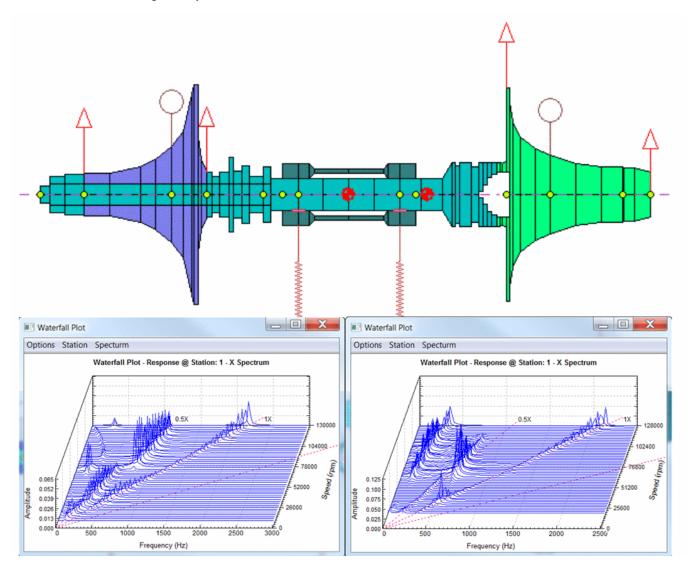
Turn rate: rad/sec.

Lateral Analysis Option & Run Time Data		
Analysis: Steady Maneuver Load Analysis	Transient Analysis RPM: 35000 StartUp / ShutDown	Gravity (g)
Static Deflection Critical Speed Map Image: Constrained Bearing Stations Spin/Whirl Ratio: Critical Speed Analysis Bearing K - Min:	Constant Speed: 35000 rpm Time-Start: Image: Const. Unbalance Ending: 0.05 Increment: 1e-006 Image: Const. Unbalance	Y: -386.088 Z: 0 None zero Gz Vertical Rotor
Spin/Whirl Ratio: 1 Npts: 0 Max: 1e+008 No. of Modes: 5 Stiffness to be varied at Stiffness: Kxx V	Solution Method Gravity (2) Newmark-beta Gravity (2) Very Static Loads Gravity (2) Static Loads Gravity (2) Misalignment	Run
Ending: 40000 Ending: 40000 V Increment: 500 Increment: 100 V	Effects: RPM-Starting: 10000 Mass Unbalance Ending: 40000 Const. Unbalance Increment: 100 Shaft Bow	Cancel
	Misalignment All Shafts with same speed ate)	: 0

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.



Lateral Analysis Option & Run 7	Time Data			
Analysis: 10-Time Transient Analysis Shaft Element Effects Rotatory Inertia Shear Def		Transient Analysis RPM: 4000 Const	Time Trequency Domain Tomain Trequency Domain	Gravity (g)
Static Deflection Constrained Bearing Stations Critical Speed Analysis Spin/Whirl Ratio: 1 No. of Modes: 3 Stiffness: Kxx 💌	Critical Speed Map Spin/Whirl Ratio: 1 Bearing K - Min: 1000 Npts: 0 Max: 1e+009 Stiffness to be varied at Bearings: All Allow Bearings in Series	Time-Start: 0 Ending: 0.5 Increment: 0.0001 Solution Met Wilson-thet Initial Cs: No		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 Respons RPM-Starting: 0 Ending: 0 Increment: 0 Excitation Shaft: 1	e Analysis Effects: Mass Unbalance Const. Unbalance Shaft Bow Disk Skew	Steady State Harmonic Excitation RPM-Starting: 0 Ending: 0 Increment: 0 Excitation Shaft: 1	Run
	All Synchronized Shafts ranslational Acceleration and/or Turn R eration - X: 0 Y:	Misalignment tate) Turn Rate - X: 0	Y: Ref Po	s: 0

Time Tra	ansient Ana	alysis - Mu	ltiple Speed	ls (Freque	ncy Domain)			
	ile: C:\MyFold		6\Exapmle_Tra	nsient\Examp	le_6_2_Water	fall.rot			Save & Close
I 2 3 4 5 6 7 8 9 10 11 12 13 14 15 16 17 18 19 20 21 22 23 23	RPM 1098 2197 3295 4029 5127 6226 6958 8057 9156 9888 10987 12085 13000 13916 15015	T (sec) 0.5 1	delta T 1E-05 1E-05 1E-05 1E-05 1E-05 1E-05 1E-05 1E-05 1E-05 1E-05 1E-05 1E-05 1E-05 1E-05 1E-05 1E-05 1E-05	Max Hz 25000 25000 25000 25000 25000 25000 25000 25000 25000 25000 25000 25000 25000 12500 12500	Delta H2 3.052	total n 50000 50000 50000 50000 50000 50000 50000 50000 50000 50000 50000 100000 100000	FFT N 16384 16384 16384 16384 16384 16384 16384 16384 16384 16384 16384 16384 16384 16384	pk track ▲ 0.01 0.00 0.01 0.01 0.01 0.01 0.01 0.01 0.01 0.01 0.01 0.01 0.01 0.01 0.01 0.01 0.01 0.01 0.01 0.01 0.01 0.01 0.01 0.01 0.01 0.01	Run Option Changed Files Rotor Speed (rpm) Start: 1098 End: 15015 Step: 732 Time (seconds) Time: 1 Step: 1e-005 Eill Table
Inser	t Row D	elete Row	4	lote: FFT delt	a freq = 1 / (N	dT). or dT = 1	I/(N delta fre	:q)	

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

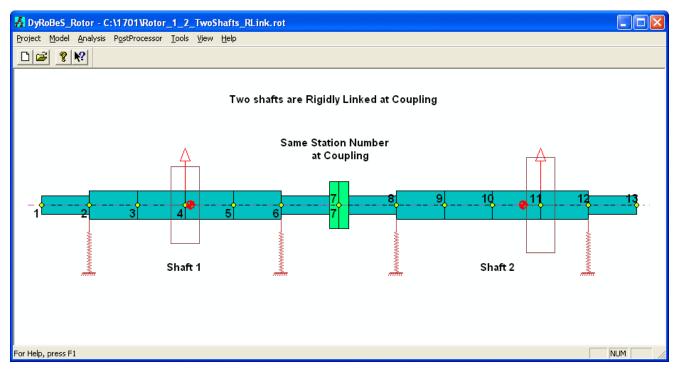
Natural Catenary (Gravity Sag) Analysis

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.

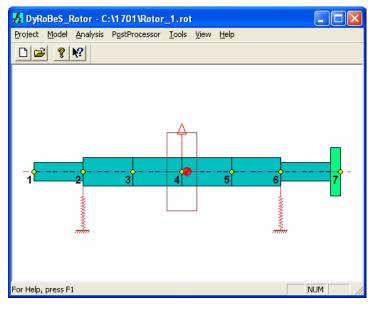
Caution must be taken when performing this task. Misalignment of the bearings will eliminate the high stress at or near couplings during the initial start up. However, this rotor bow also creates a synchronous excitation in addition to the mass unbalance excitation.

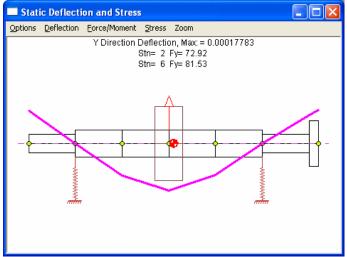
For each shaft supported by two bearings, the catenary curve can be obtained by the following procedure which will result in the rotor static weight of each rotor being supported extirely by its own bearings. An example is used here for illustration purposes:



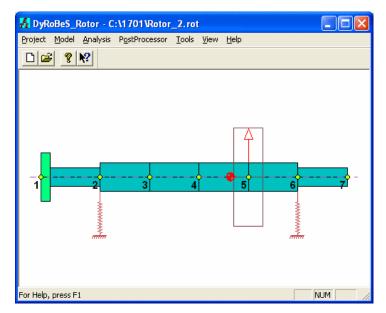
Step 1. Calculates the bearing reactions of the uncoupled rotor due to rotor weight only using Analysis Option 1 - Static Deflection and Bearing Loads.

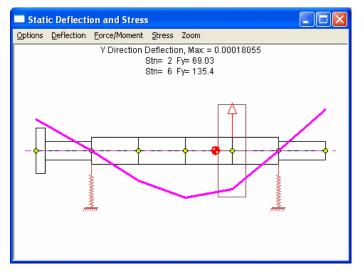
Rotor 1





Rotor 2

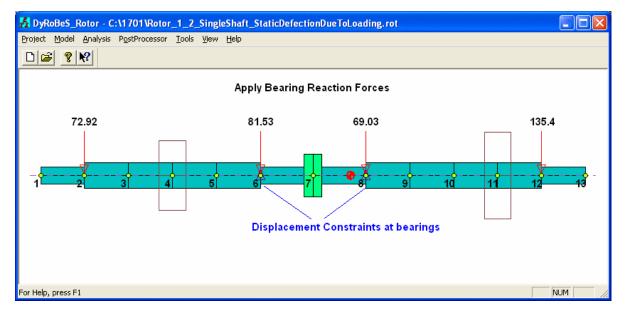


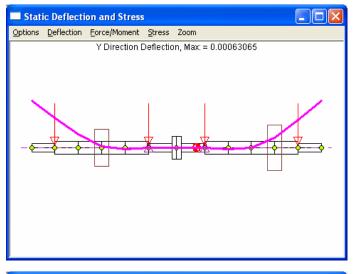


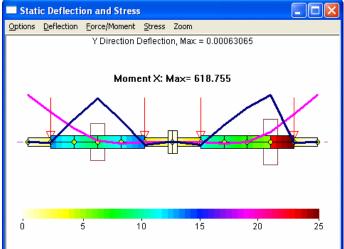
Note that the summation of 2 bearings reaction forces is the rotor static weight.

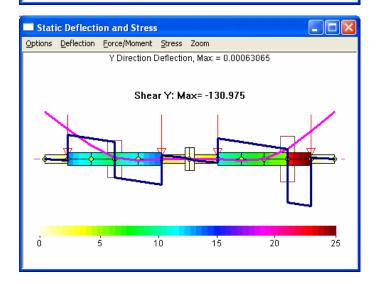
Step 2. Apply this bearing reaction forces on the combined rotor. Note that in this step, since the rotor is unconstrained, the solution is possible only after rigid body motion is eliminated. Since the rotor has translational and rotational motions for each plane, we will need at least two constraints. 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.



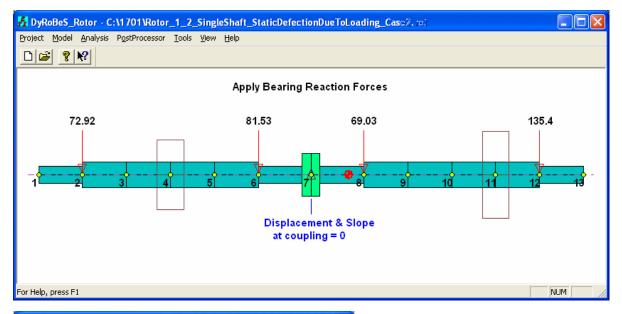


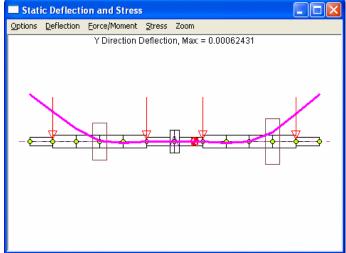


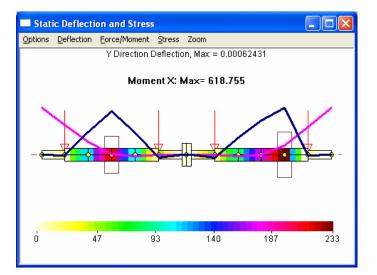


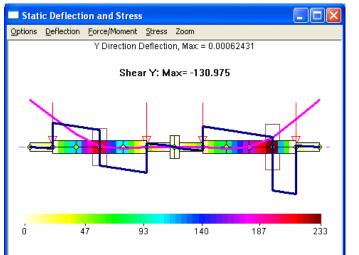
****	*******	******	Static Deflect	ions *******	************	r -
	Station	x	У	alpha	beta	
			63065 - 03	45000- 04		
		. 00000	.63065E-03	.45060E-04	.00000	
	2	.00000	.40469E-03	.45873E-04	.00000	
	3	.00000	.18150E-03	.39293E-04	.00000	
	4	.00000	.24521E-04	.20036E-04	.00000	
	5	.00000	19189E-04	.11092E-05	.00000	
	6	.00000	.00000	44766E-05	.00000	
	7	.00000	.63722E-05	36047E-09	.00000	
	8	.00000	.00000	.44784E-05	.00000	
	8	.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	
	13	.00000	.030572-05	J04//E-04	.00000	

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







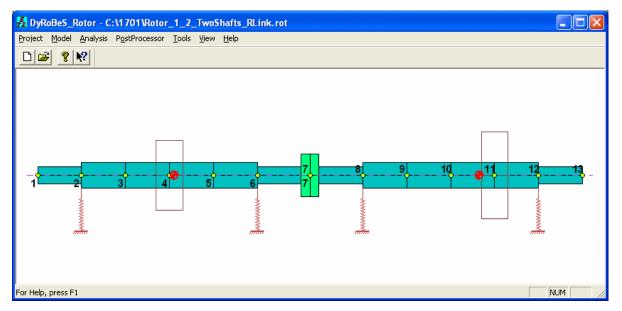


*****			Static Deflecti		
	Station	×	y y	alpha	beta
	1	.00000	.62431E-03	.45061E-04	.00000
	2	.00000	.39834E-03	.45874E-04	.00000
	3	.00000	.17515E-03	.39294E-04	.00000
	4	.00000	.18163E-04	.20037E-04	.00000
	_5	.00000	25552E-04	.11101E-05	.00000
	6	.00000	63678E-05	44756E-05	.00000
	7	.00000	.00000	.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.

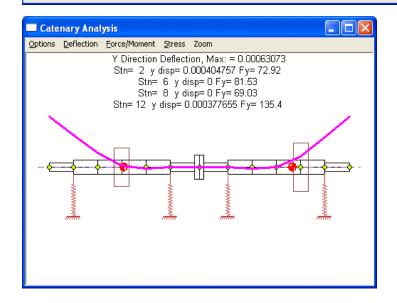


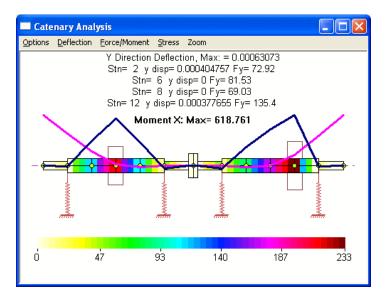
The initial guess values and objective are defined under Misalignment Tab – Catenary button as shown below. It shows that at bearing stations 2 & 12, the initial guess values are 0.0003 inches and maximum allowable values are 0.001 inches. At bearing stations 6 and 8, the initial values are zero and 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.

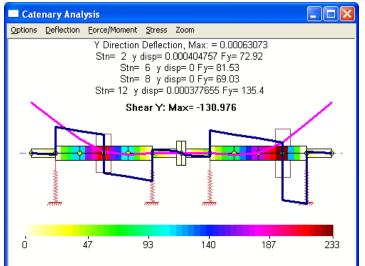
nits/Des xial Force	- <u>,</u>	Material Shaft : Loads Const	:Elements Disk traints Misalignr		nbalance Shaft Bo	- 1, - T		Foundation	- 1. j.	
						Catenary	Imp	oort *.xls	Export *.;	ds
atural (Catenary	(Gravity Sag)	Calculation							
ا به ا	a nahual	tenenu esteut-ti-		ervide H	e initial	unan uniturna (-l	(abiana) et b	onion station		
			n, user needs to pr nt and shear force							
			nt and snear rorce occur during startur				Jupling stati	uris, and any	omer	
			mization procedure		-		ions, such t	hat the object	tive functio	n
	nimized	perioriti die opii								
Com	manh Dut	loBeS_Rotor: Ca	tenery Data							_
COM	ment. Joyr	IUDES_NULUI. Ca	itenaly Data				Weigh	nting Fac	tors	
	Brg Stn	Initial Guess	Max. Allowable	•		Coupling Stn	Moment	Force	Slope	•
1	2	0.0003	0.001		1	7	1	1	0	
2	6	0	0		2					
3	8	0	0		3					
4	12	0.0003	0.001		4					
5					5					
6					6					
7					7					
8	· · · · · · · · · · · · · · · · · · ·				8					
	·				9					
					10					
9					11					
9 10					12					
9 10 11					13					
9 10 11 12					14					
9 10 11 12 13										-
9 10 11 12 13 14				-	15					
9 10 11 12 13				•	15		4			
9 10 11 12 13 14	Insert Ro	ow Delete	Row	•	15	Insert R	ow [)elete Row		

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 Shaft Ete 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 State Synchronous Response - Linear System 7 Cons 6 Steady Synchronous Response - NonLinear System 7 Time Transient Analysis - Time Domain 8 Steady Maneuver Load Analysis 10-Time Transient Analysis - Frequency Domain Spin/Write Stiffness to be varied at Bearings: All Stiffness: Kxx Void Modes: 5 Stiffness: Kxx Vhirl Speed and Stability Analysis Steady State Synchronous Response	Transient Analysis Time Frequency Omain RPM: 0 Domain Domain Constant Speed: 0 rpm X: 0 Time-Start: 0 ✓ Mass Unbalance Y: -386.088 Ending: 0 ✓ Shaft Bow Z: 0 Increment: 0 ✓ Shaft Bow Z: 0 Solution Method ✓ Gravity (Z) Vertical Rotor Vertical Rotor Wilson-theta ✓ Static Loads ✓ Time Forcing Initial Cs: None zero Gz Initial Cs: No ✓ Misalignment State Loads Vertical Rotor						
RPM-Starting: 0 RPM-Starting: 1000	Effects: RPM-Starting: 0 Run Mass Unbalance Ending: 0 Run Const. Unbalance Increment: 0						
No. of Modes: 4 Excitation Shaft: 1	Shaft Bow						
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							

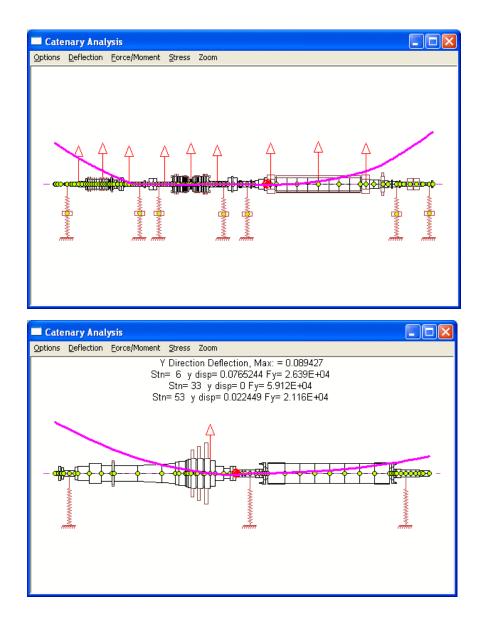






Rotor_1_2_TwoSI		otepad		
ile Fair i õlular Mew	Telb			
****************** Ca	tenary Analysis	and Bearing P	Elevation Calcu	lation *******
**********			traint Reaction	******
Station	Y-Disp	FУ		
2	.40476E-03	72,921		
6	.00000	81.534		
6 8 12	.00000	69.034		
12	.37766E-03	135.42		
		Static Dofloct		
Station	x	y 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
2	.00000	19174E-04 .00000	.11123E-05 44737E-05	.00000
7	.00000	.63627E-05	44/3/E-03 50157E-15	.00000
8	.00000	.00000	.44737E-05	.00000
1 2 3 4 5 6 7 8 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.



Torsional Vibration Analysis

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.

🧏 DyR	🛃 DyRoBeS_Rotor - C:\WyFolder\DyRoBeS\Examples\Ch_8_Example_2a_SynchronousMotorCompressor.rot												
Project	Model	Analysis	PostProcessor	Tools	View	Help							
	; ?	Lateral Vibration Torsional Vibration Axial Vibration											

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

Torsional Vibration Run Time Data 🛛 🛛 🔀									
Analysis: Startup Transient Analysis (Speed Dependent Excitations) 💌 Run									
Damping Model		Cancel							
O Direct Damping	Modal Damping 🛛 🔿 E	oth							
Steady State Response	TimeTransient	Startup Transient							
Rotor Speed (rpm)	Time (sec)	Time (sec)							
Starting: 0	Starting: 0	Starting: 0							
Ending: 1500	Ending: 1500 Ending: 20 Ending: 18								
Increment: 10	Increment: 0.001	Increment: 0.0005							

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.

Torsional Vibration Run Time Data								
Analysis: Startup Transient Analysis (Speed Dependent Excitations) V								
Model Summary Damping Natural Frequencies and Modes Steady State Forced Response C Dire Transient Response by Time Dependent Excitations								
Startup Transient Ar	halysis (Speed Dependent Exc	stations						
Steady State Response	TimeTransient	Startup Transient						
Rotor Speed (rpm)	Time (sec)	Time (sec)						
Starting: 0	Starting: 0	Starting: 0						
Ending: 1500	Ending: 1500 Ending: 20 Ending: 18							
Increment: 10	Increment: 10 Increment: 0.001 Increment: 0.0005							

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 <u>Modal Damping</u>

Torsional Vibration Run Time Data 🛛 🛛 🔀								
Analysis: Startup Transient Analysis (Speed Dependent Excitations) 💌 🛛 🛛 🖉								
Damping Effects Included Cancel Cancel Viscous Damping Modal Damping C=K/(DmXw)								
Steady State Response Rotor Speed (rpm) Starting: 0 Ending: 1500 Increment: 5	TimeTransient Time (sec) Starting: 0 Ending: 1 Increment: 0.0001	Startup Transient Time (sec) Starting: 0 Ending: 18 Increment: 0.0005						

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 reference d 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	Ĵ	Jn ²
Stiffness	K	Kn²
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

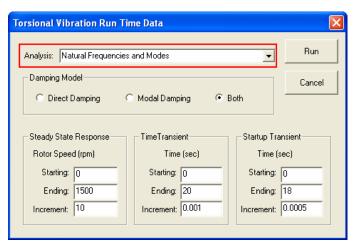
Steady State Forced Response Analysis

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 <u>Steady State Excitation</u>. 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-<u>Data Editor</u>.

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.

Torsional Vibration Run Time Data 🛛 🛛 🔀							
Analysis: Steady State Forced	Run						
Damping Model C Direct Damping C Modal Damping C Both Cancel							
Steady State Response Rotor Speed (rpm) Starting: 0 Ending: 1500 Increment: 10	TimeTransient Time (sec) Starting: 0 Ending: 20 Increment: 0.001	Ending:					

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 <u>Torsional Excitations in Equation</u>. The time dependent excitations in data files are entered in <u>Torsional Excitations in Data Files</u>. 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 <u>Modal Damping</u>.

Torsional Vibration Run Time Data 🛛 🛛 🕅							
Analysis: Transient Response by Time Dependent Excitations							
Damping Model C Direct Damping C Modal Damping C Both							
Steady State Response Rotor Speed (rpm) Starting: 0 Ending: 1500 Increment: 10	TimeTransient Time (sec) Starting: 0 Ending: 20 Increment: 0.001	Startup Transient Time (sec) Starting: 0 Ending: 18 Increment: 0.0005					

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 <u>Torsional</u> <u>Driving Torque</u> and <u>Load Torque</u>. 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 <u>Modal Damping</u>.

Torsional Vibration Run Time Data 🛛 🛛 🔀							
Analysis: Startup Transient Analysis (Speed Dependent Excitations) y 🛛 🛛 🔤							
Damping Model Cancel Cancel							
Steady State Response Rotor Speed (rpm) Starting: 0 Ending: 1500 Increment: 10	TimeTransient Time (sec) Starting: 0 Ending: 20 Increment: 0.001	- Startup Transient Time (sec) Starting: 0 Ending: 18 Increment: 0.0005					

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.

🛃 DyRoBeS_Rotor - C:\WyFolder\DyRoBeS\Examples\Turbo_Pump.rot							
Project Model	Analysis	PostProcessor	Tools	View	Help		
0 🖻 🤶	Lateral Vibration Torsional Vibration						
	Axial Vibration						
			-				

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

Axial Vibration Run Time Data	X
Analysis: Steady State Forced Response	Run
Damping Model	Cancel
Direct Damping C Modal Damping C Both	
Steady State Response	
Rotor Speed (rpm)	
Starting: 0	
Ending: 3550	
Increment: 10	

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.

Axial Vibration Run Time Data	
Analysis Steady State Forced Response Model Summary Damping Natural Frequencies and Modes Steady State Forced Response © Direct Damping © Modal Damping © Both Steady State Response Rotor Speed (rpm) Starting: 0 Ending: 3550 Increment: 10	Run Cancel

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

Axial Vibration Run Time Data	X
Analysis: Steady State Forced Response	Run
Damping Model © Direct Damping C Modal Damping C Both	Cancel
Steady State Response Rotor Speed (rpm) Starting: 0 Ending: 3550 Increment: 10	

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.

Axial Vibration Run Time Data	
Analysis: Natural Frequencies and Modes	Run
Damping Model	Cancel
O Direct Damping C Modal Damping C Both	
Steady State Response	
Rotor Speed (rpm)	
Starting: 0	
Ending: 3550	
Increment: 10	
,	

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 <u>Steady State Excitation</u>. It will be calculated only when the excitation exists. The excitation amplitude and frequency can be functions of rotor speed.

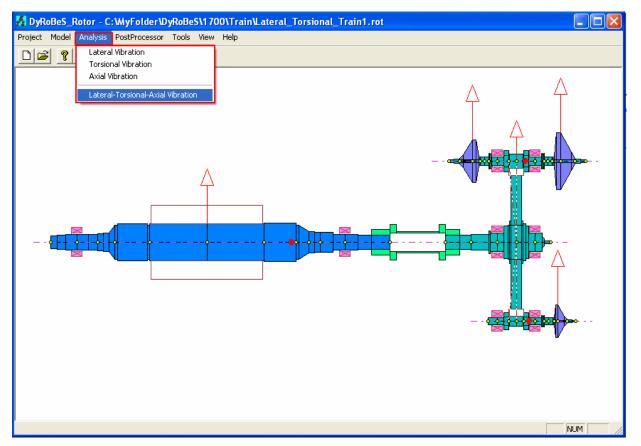
For forced response analysis, the modal damping can be specified if the direct damping is not readily available. Details on the damping model are explained in the <u>Modal</u> <u>Damping</u>. If modal damping is specified, then the vibratory force caused by the damping, i.e., the second term in the above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

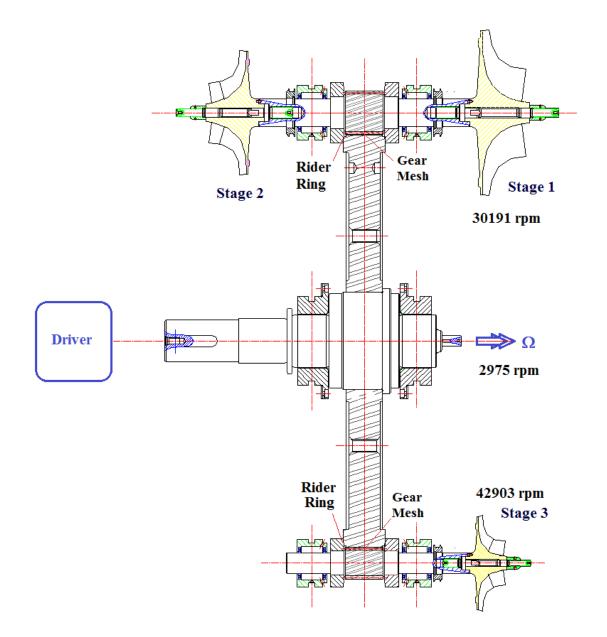
Axial Vibration Run Time Data	
Analysis: Steady State Forced Response	Run Cancel
Steady State Response Rotor Speed (rpm) Starting: 0 Ending: 3550 Increment: 10	

For sample outputs, click Torsional Steady State Response.

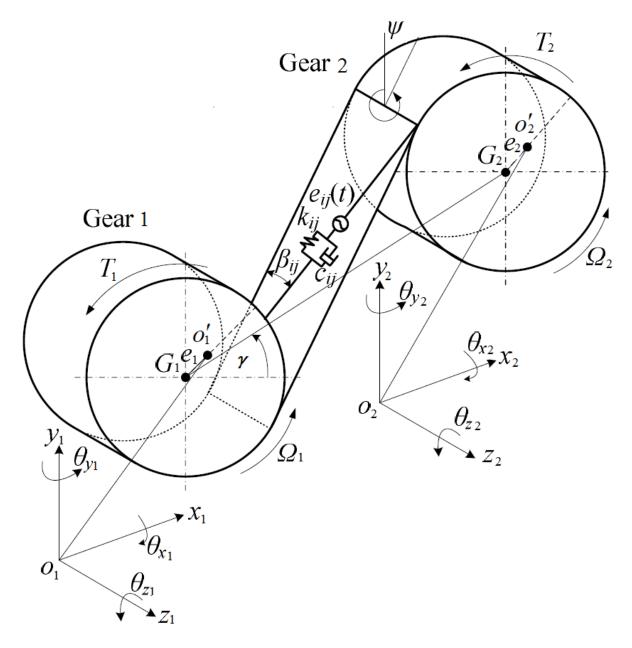
Coupled Lateral-Torsional-Axial Vibration Analysis

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 $(\theta x, \theta y, \theta z)$ 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.

🛃 DyRoBeS_Rotor - C:\BookII_LTCoupling\Example\Jida+rot	
Project Model Analysis PostProcessor Tools View Help	
Lateral Vibration Torsional Vibration Axial Vibration Lateral-Torsional-Axial Vibration	
Lateral-Torsional-Axial Geared Vibration Run Time Data Analysis: 1 · Whirl Speed & Stability Analysis Vibrations Included Convert Cross-Coupled Bearing Coefficients If Rotation is CW If Rotation is CW Convert Cross-Coupled Bearing Coefficients If Rotation is CW Whird Speed and Stability Analysis Steady State Synchronous Response Analysis RPM-Starting: 0 Ending: 25000 Increment: 10 Increment: 0 Increment: 10 Increment: 000 Increment: 100 Increment: 1000 Increment: 1000 <td< td=""><td></td></td<>	
NU	M //

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. <u>Animation</u> 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:

Lateral Vibration Analysis

Static Deflection

Critical Speed Mode Shapes

Critical Speed Energy Distribution

Critical Speed Modal Stress

Critical Speed Map

Whirl Speed Map

Stability Map

Root Locus Plot

3D Precessional Mode Shape

Bode Plot

<u>Polar Plot</u>

Elliptical Orbit Axes Plot

Responses at Multiple Stations

Shaft Response - 3D Animation

Displacement Orbit at Single Station

Bearing/Support Transmitted Forces

Element Shear Forces and Moments

Time Transient Shaft Response

Time Transient X-Y Plot

Transient Response vs. Time (or Speed)

FFT Spectrum

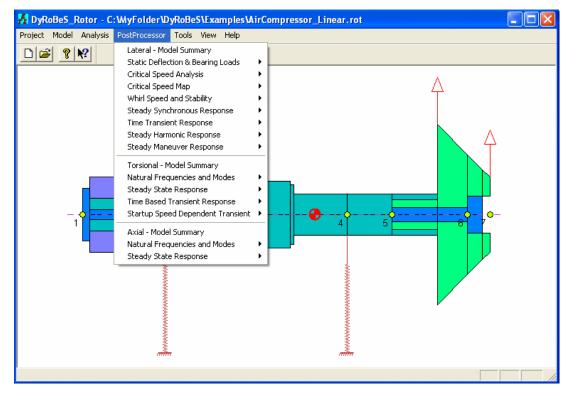
Bearing Reaction Forces

Time Transient Speed vs. Time

Steady Harmonic Response

Steady Maneuver Load Response

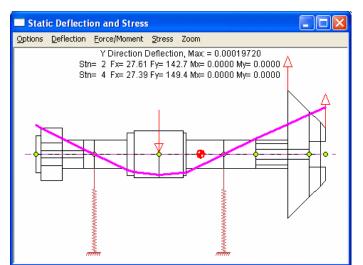
Torsional Vibration Analysis Torsional Frequency Interference Diagram Torsional Mode Shapes Torsional Modal Stress Torsional Steady State Response Torsional Steady State Element Vibratory Torque Torsional Steady State Element Vibratory Stress Torsional Steady State Excitations Torsional Transient Response (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.

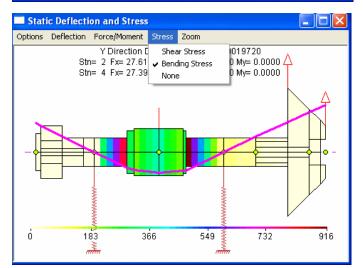


The line Control Data allows you to change the line thickness and color. The plotted curves on a laser printer or ink jet printer will not be as thick as the lines shown on the computer screen. Therefore, if a hardcopy from printers is needed, the line thickness should be increased. The lines may appear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

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.



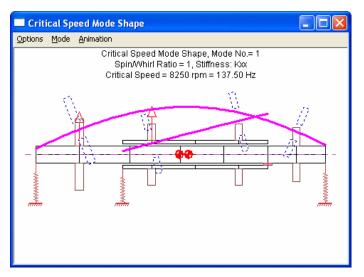


See also Lateral Vibration Analysis, Static Deflection and Bearing/Constraint Reactions Analysis.

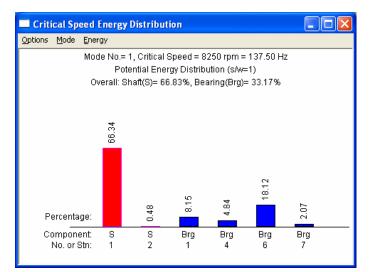
Critical Speed Mode Shape and Energy Distribution

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.

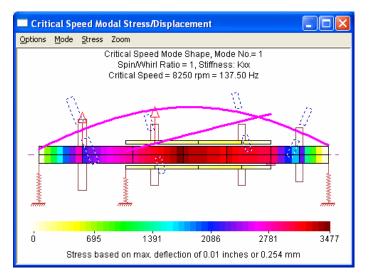
Mode Shape



Energy Distribution



Modal Stress



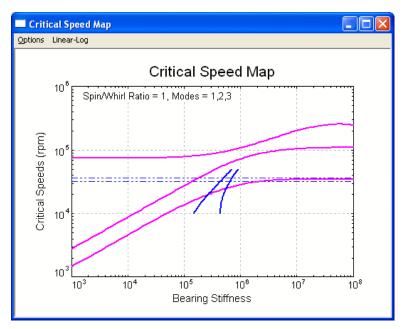
The above figure shows the forward synchronous critical speed mode shape of a two spool dual rotor at 8250 rpm. The bearing stiffnesses Kxx was selected in the run time data. The mode shape is relative displacement and normalized eigenvector which does not have dimension. The kinetic energy is divided 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 negative 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.

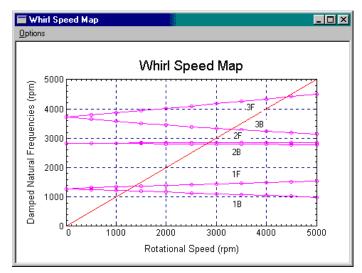
See also Lateral Vibration Analysis, Critical Speed Map Analysis.

Whirl Speed and Stability Results

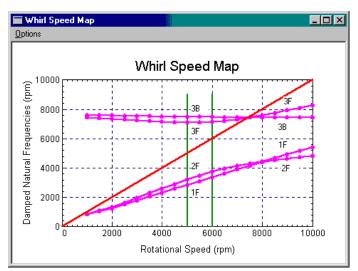
Whirl Speed Map is a linear plot of damped natural frequencies vs. shaft rotational speeds. The damped natural frequency is usually referred to as whirl speed. The damped critical speeds and any excitation resonance speeds are determined by noting the coincidence of the shaft speed with the system natural 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. The third mode now is a backward bending 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.

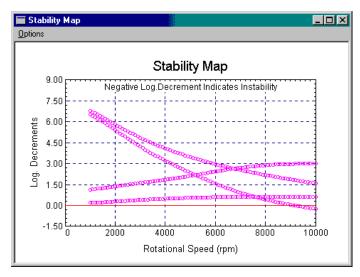


Stability Map

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.

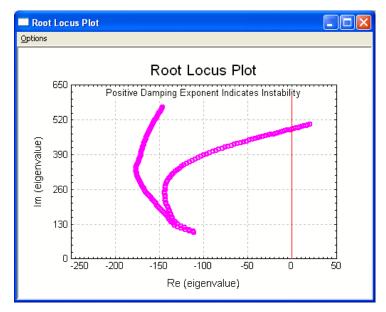
You can change the graph title, labels, number of modes, and many others by changing the default settings in the Setting dialog under Options menu.

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.



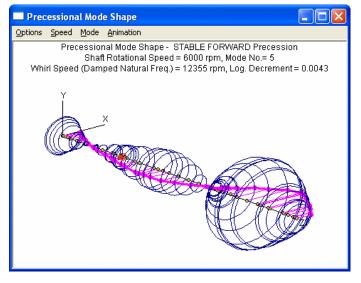
Root Locus Plot

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.

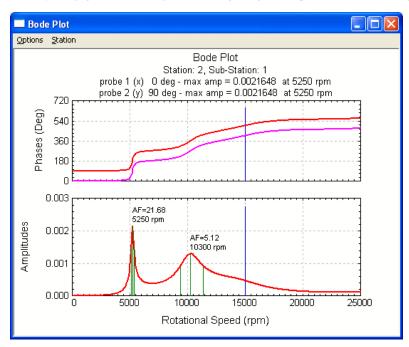


See also Lateral Vibration Analysis, Whirl Speed and Stability Analysis.

Steady State Synchronous Response Results

Bode Plot

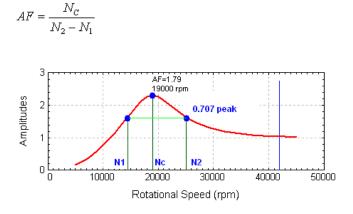
A Bode plot is a linear plot of amplitude and phase vs. shaft rotational speeds. You can change the graph title, labels, scale, number of divisions, probes angles and many others by changing the default settings in the **Setting** dialog under **Options** menu. Relative displacement can be specified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

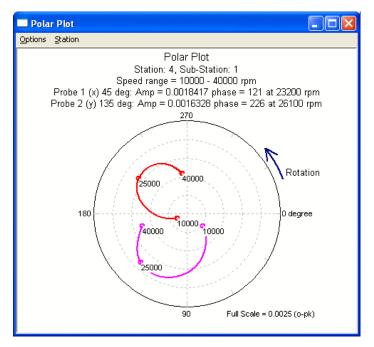
The program can allow you to label the Amplification Factors. The Amplification Factor (AF) is defined based on API Standards, such as API 611 for steam turbines or API 672 for geared compressors. The Amplification factor (AF) is defined as:



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

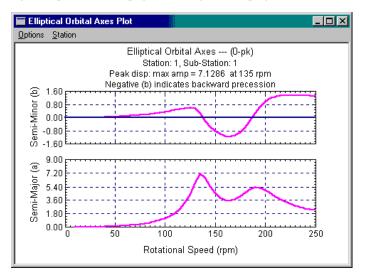
Polar Plot shows amplitude and phase in the polar coordinates over a speed range for a given finite element station. The shaft rotational direction is counterclockwise. You can change the graph title, labels, scale, number of divisions, probes angles and many others by changing the default settings in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





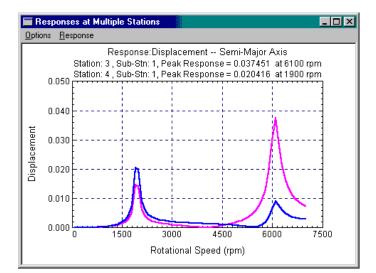
The Elliptical Orbit Axes Plot is a linear plot of semi-major axis amplitudes and semi-minor axis amplitudes vs. shaft rotational speeds. Positive semi-minor axis indicates the **Forward Precession** orbit.

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.



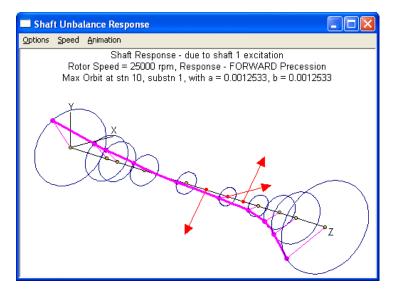
Responses at Multiple Stations

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.



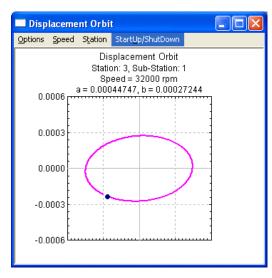
Shaft Response 3D Animation

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.



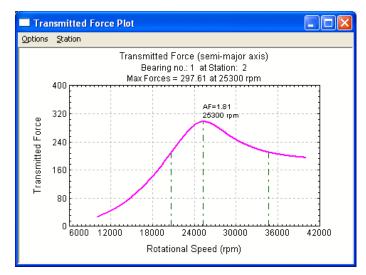
Displacement Orbit at Specified Speed and Station

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.



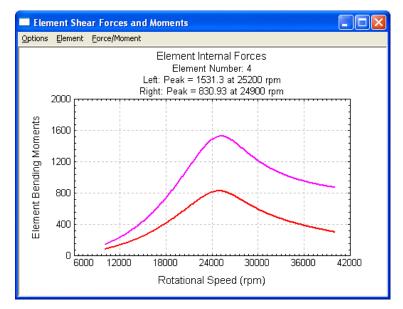
Bearing/Support Transmitted Forces

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.

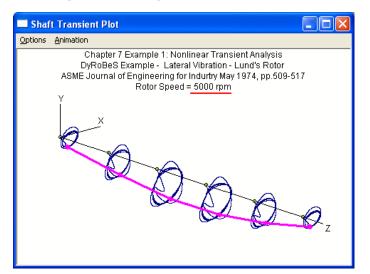


See also Lateral Vibration Analysis, Steady State Synchronous Response Analysis - Linear System, Steady State Synchronous Response Analysis - NonLinear System.

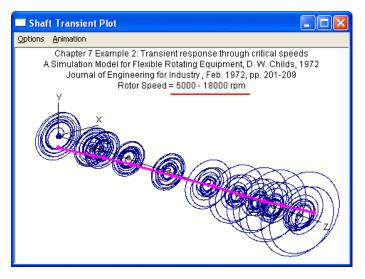
Time Transient Shaft Response

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:

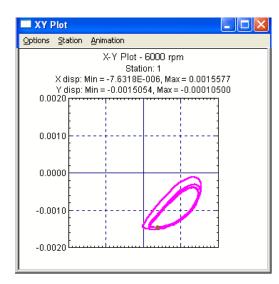


Time Transient X-Y Plot

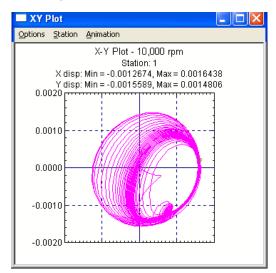
This plot displays the time transient orbit displacement. This option is useful in the investigation of the initial response of a rotor to a sudden excitation and also for the investigation of nonlinear effects such as caused by nonlinear bearings.

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**.

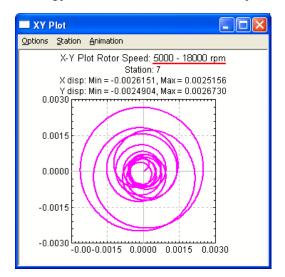
At 6000 rpm



At 10000 rpm



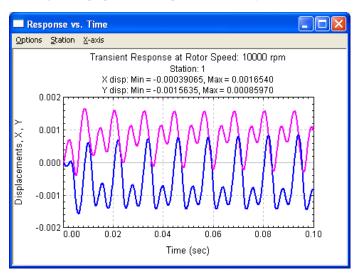
Following picture shows the rotor motion from 5000 rpm to 18000 rpm.



Transient Response vs. Time (Speed)

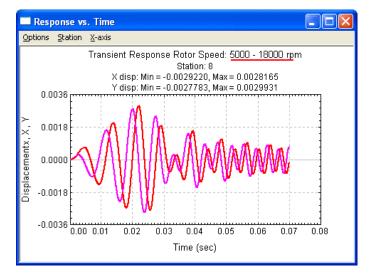
This plot displays the transient response vs. time. In the setting, you can display X displacement, Y displacement, and/or radial displacement. You can change the graph title, labels, scale, number of divisions and many others by changing the default settings in the **Setting** dialog under **Options** 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.

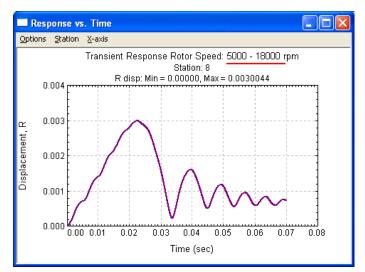


Followings are results for startup/shutdown transient analysis.

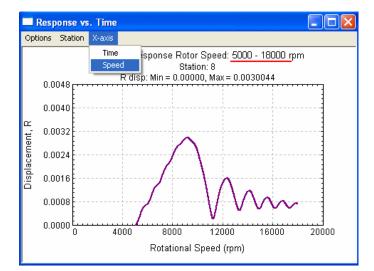
X, Y displacements vs. time



R displacement vs. time



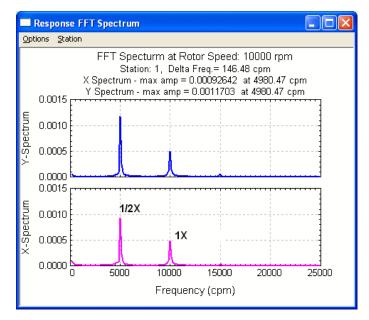
R displacement vs. speed



FFT Spectrum

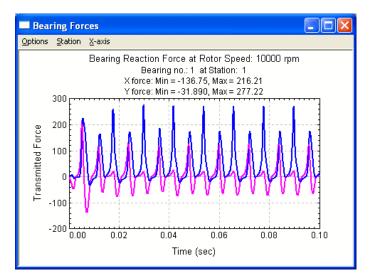
This plot displays the transient response in spectrum format. You can change the graph title, labels, scale, number of divisions and many others by changing the default settings in the **Setting** dialog under **Options** menu. In the default setting, the zero harmonic component (DC level) is not shown 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.

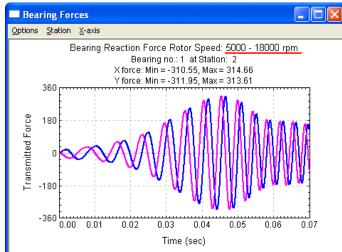
The following figure shows a rotor with a subsynchronous vibration component. The rotor is operated at 10,000 rpm, the subsynchronous vibration (4980 rpm) is larger than the synchronous vibration. This indicates that the rotor is operated at the unstable condition.

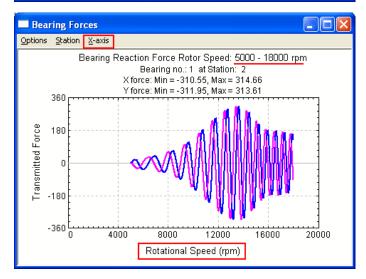


Transient Bearing Reaction Forces

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.

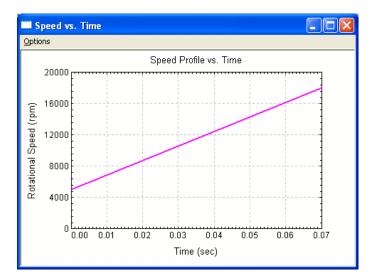






Transient Speed vs. Time

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

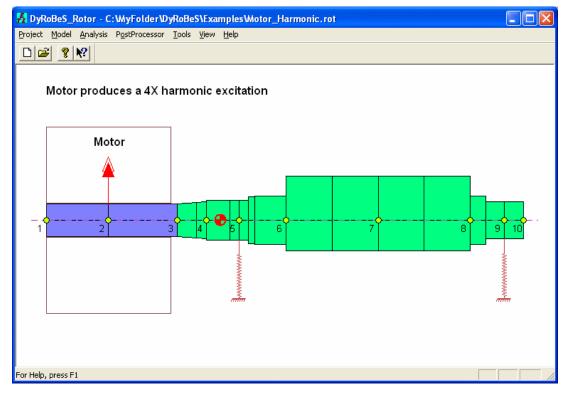


See also Lateral Vibration Analysis, Time Transient Analysis.

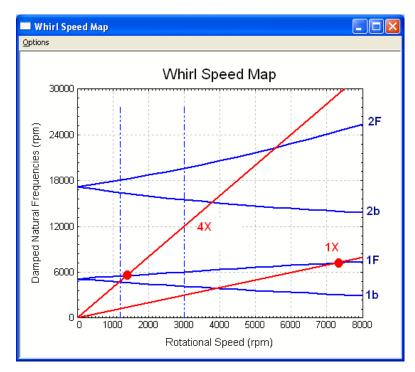
Steady Harmonic Response

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 ¹/₄. 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.

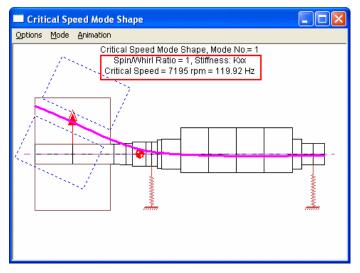
Rotor system

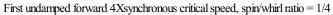


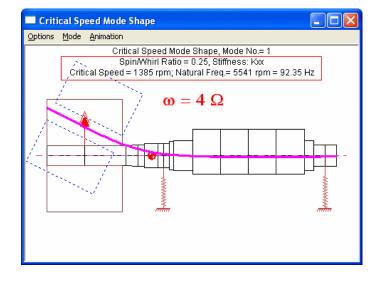
Whirl Speed Map



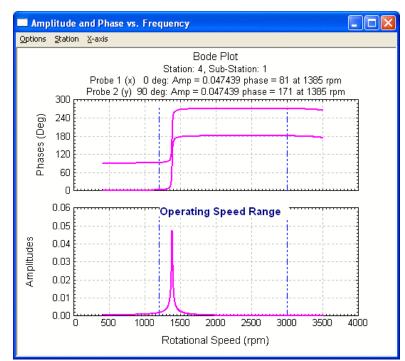
First undamped forward synchronous critical speed, spin/whirl ratio = 1



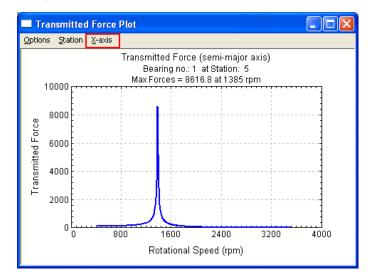




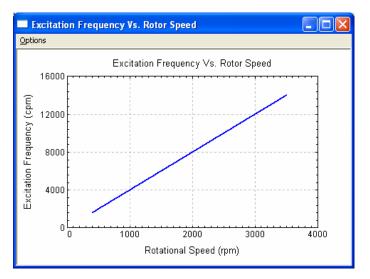
Response amplitude



Bearing force



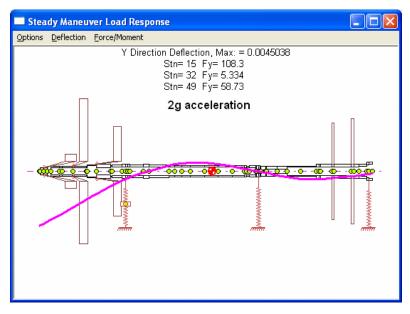
Excitation frequency vs. rotor speed



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.

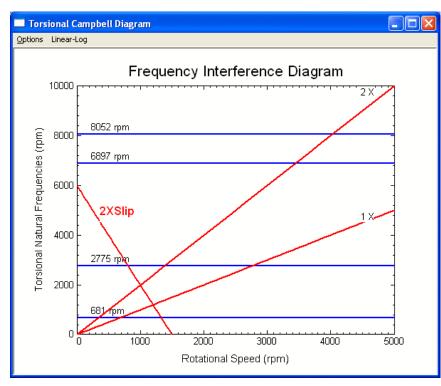


See also Lateral Vibration Analysis, Steady Maneuver Load Analysis.

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.

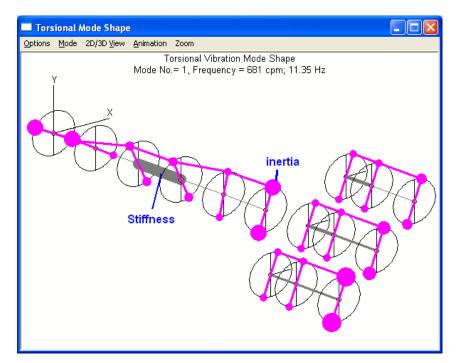


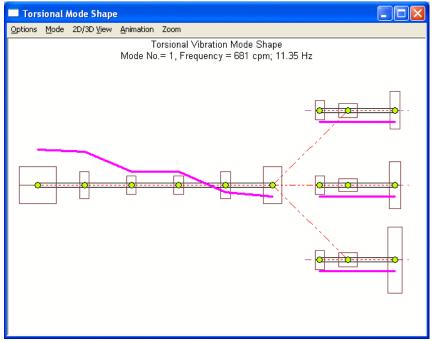
Note: for torsional analysis, the outputs can be referenced to the actual or equivalent system. For details on this conversion, click Torsional Vibration Analysis.

Torsional/Axial Mode Shapes

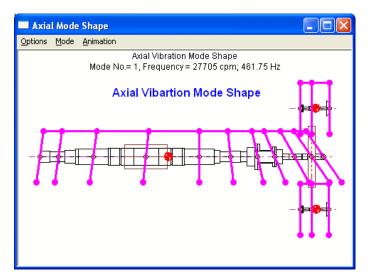
This option displays the torsional or axial vibration mode shape. You can change the graph title, labels, scale, and many others by changing the default settings in the **Setting** dialog under **Options** menu. **Animation** option allows you to animate the motion.

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



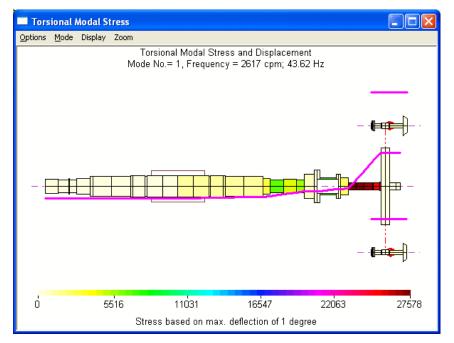


Axial vibration mode shape



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 <u>Torsional Vibration Analysis</u>. 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

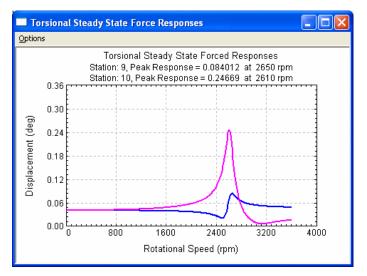
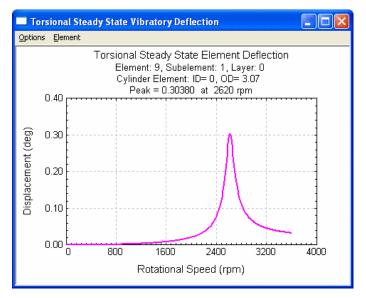


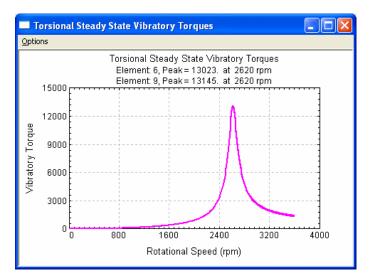
Figure shows the **Deflection** at Element 9 = abs(displacement 9 - displacement 10)



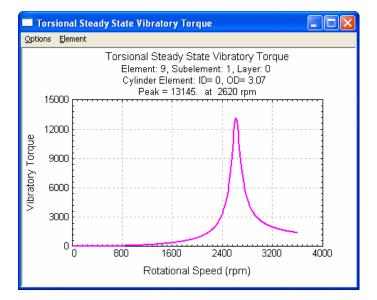
Torsional/Axial Vibratory Torque/Forces

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 <u>Shaft Elements</u> 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):



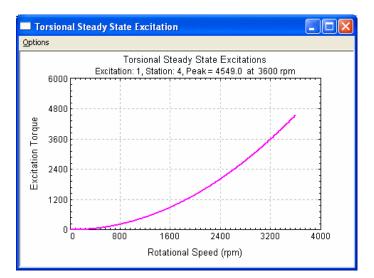
Vibratory Torque at Element 9, Subelement 1, Layer 0 (this option goes to the subelement level):



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 <u>Torsional Steady State Excitation</u>. 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.

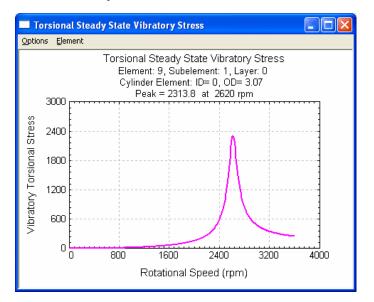
Excitation amplitude vs. speed



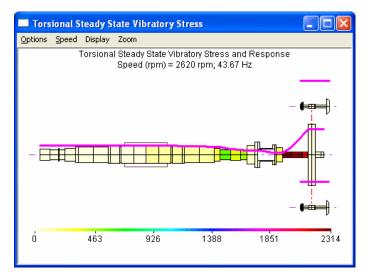
Torsional Vibratory Stress

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 <u>Shaft</u> <u>Elements</u> 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.

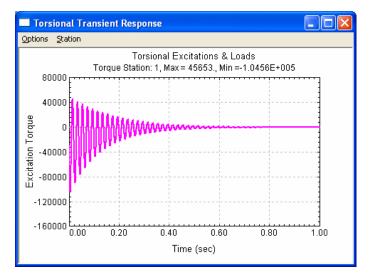
Torsional Transient Response – Time Dependent Excitations

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 <u>Torsional Excitations in Equations</u> and <u>Torsional Excitations in Data Files</u>. For speed dependent excitation, the driving and load torques are entered in <u>Torsional Driving Torque</u> and <u>Load Torque</u>.

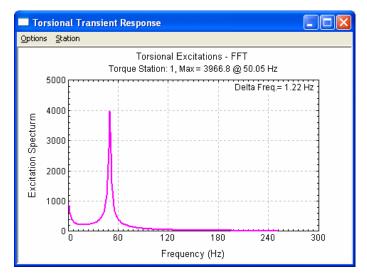
Consider a 4200 kW, 50 Hz, 4 Poles motor with a 3-phase fault, the short circuit excitation torque is shown below:

Torsional Short Circuit Applied Torque

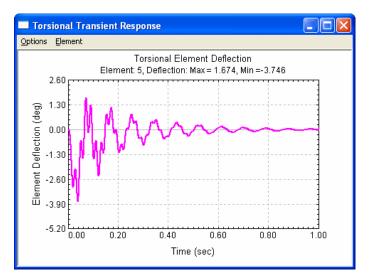
This option displays the torsional short circuit torque vs. time. Note that in a typical torsional short circuit excitation, there is a large initial transient that dies out exponentially.



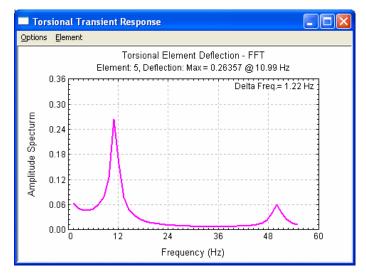
FFT of the excitation, it shows that the excitation is mainly the 50 Hz line frequency.



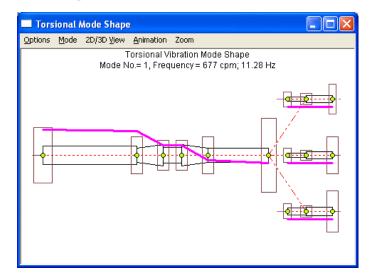
Torsional Short Circuit Transient Element Deflection

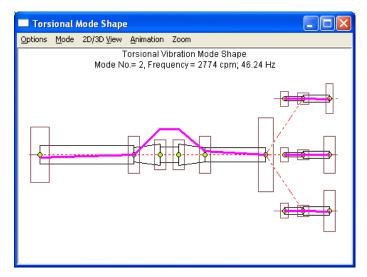


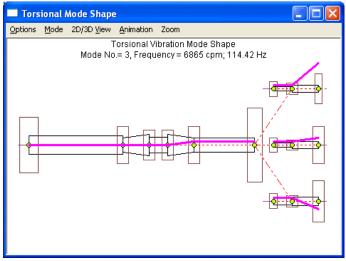
FFT of the response, it shows that the response has two major frequency components. One is 11 Hz, which is the system natural frequency and is excited by the initial conditions. The other one is 50 Hz which is the excitation frequency caused by the short circuit.



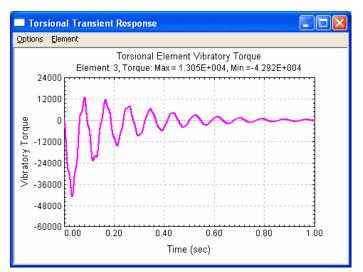
The mode shapes for the first three natural modes are shown below:



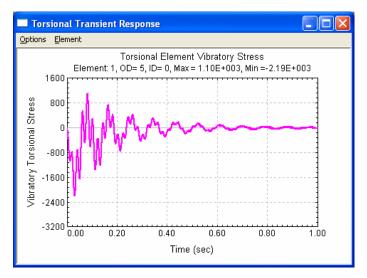




Torsional Short Circuit Transient Element Vibratory Torque



Torsional Short Circuit Transient Element Vibratory Stress



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

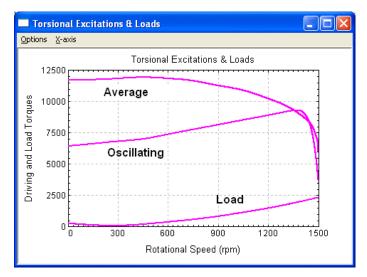
Torsional Transient Response – Speed Dependent Excitations

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 <u>Torsional Excitations in Equations</u> and <u>Torsional Excitations in Data Files</u>. For speed dependent excitation, the driving and load torques are entered in <u>Torsional Driving Torque</u> and <u>Load Torque</u>.

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.

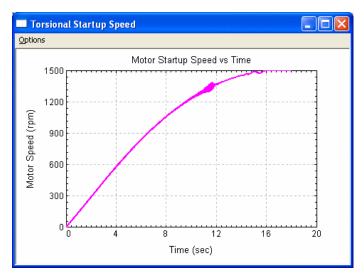
Torsional Startup Driving and Load Torques

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.



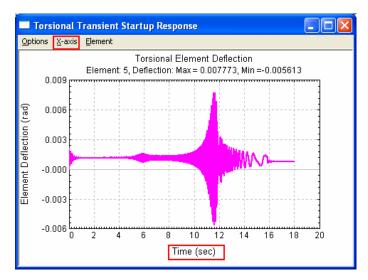
Torsional Startup Speed vs. Time

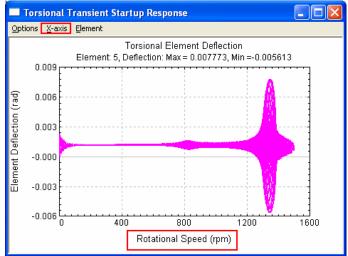
This option displays the torsional transient startup driver speed vs. time. You can change the graph title, labels, number of divisions and many others by changing the default settings in the Setting dialog under Options menu.



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:

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

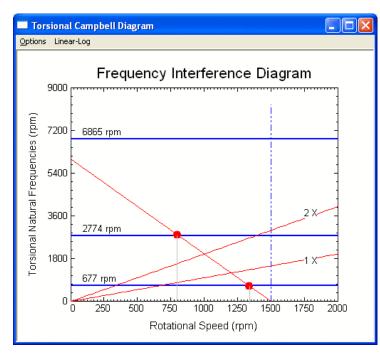
where

- \mathcal{N}_{cr} is the critical speed in rpm to be calculated.
- N_{syn} is the synchronous speed in rpm.
- ϖ_i is the system natural frequency that is less than 2X line frequency (rad/sec).
- $f_{\rm L}$ is the Line Frequency in Hz (50 or 60 Hz).

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

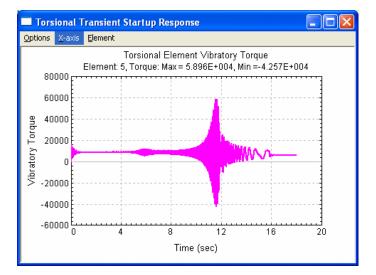
- 1. 70.89 rad/sec = 11.28 Hz = 677 rpm
- 2. 290.52 rad/sec = 46.24 Hz = 2774 rpm

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

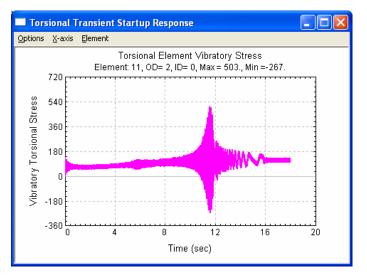


Torsional Startup Transient Element Vibratory Torque

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.

 Aerodynamic Cross Coupling

 Rolling Element Bearings

 Liquid Annular Seal Dynamic Coefficients

 Gas Laby Seal Dynamic Coefficients

 Squeeze Film Damper Design Tool

 Activate BePerf Program

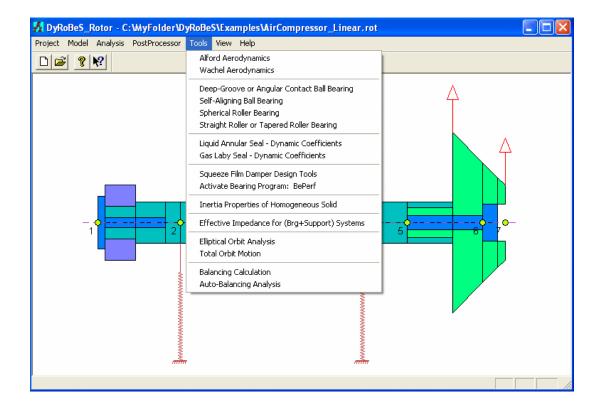
 Mass/Inertia Properties Calculation

 Reduction of Flexible Supports

 Rotor Elliptical and Total Orbit Analysis

 Balancing Calculation

 Auto Balancing Analysis



Aerodynamic Cross Coupling

Aerodynamic Cross Coupling

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

$$\begin{cases} F_{x} \\ F_{y} \end{cases} = - \begin{bmatrix} 0 & K_{xy} \\ K_{yx} & 0 \end{bmatrix} \begin{cases} x \\ y \end{cases}$$

Alford has proposed the cross-coupled stiffness:

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

where

 $K_{_{\mathbf{x}\mathbf{v}}}$ 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:

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

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
- ho_d density of fluid at discharge
- ho_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

$$\delta = 46.2E - 06 \sqrt[3]{\frac{F^2}{DZ^2 \cos^5 \alpha}}$$
$$K = 0.0325E06 \sqrt[3]{DFZ^2 \cos^5 \alpha}$$

K = 0.0325200 (D12 C

Self-Aligning Ball Bearings

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

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

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

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

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

where δ Radial Deflection

D Ball Diameter (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:

$$\begin{cases} -F_x \\ -F_y \end{cases} = \begin{bmatrix} K_d & k_o \\ -k_o & K_d \end{bmatrix} \begin{cases} x \\ y \end{cases} + \begin{bmatrix} C_d & c_o \\ -c_o & C_d \end{bmatrix} \begin{cases} \dot{x} \\ \dot{y} \end{cases} + \begin{bmatrix} m_d & 0 \\ 0 & m_d \end{bmatrix} \begin{cases} \ddot{x} \\ \ddot{y} \end{cases}$$

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:

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

where *ร*

is the entrance loss coefficient (typical value = 0.1) is the fluid density

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

 ρ

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 $\ \frac{1}{2} \left(\mathbf{l} + \boldsymbol{\xi} \right) \boldsymbol{\rho} \ \boldsymbol{V}^2$ defines the inlet pressure drop.

Since $\,\sigma\,$ 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:

$$\begin{split} K_{d} &= \mu_{3} \left(\mu_{0} - 0.25 \, \mu_{2} \Omega^{2} T^{2} \right) \\ k_{o} &= \mu_{3} \left(0.5 \, \mu_{1} \, \Omega T \right) \\ C_{d} &= \mu_{3} \, \, \mu_{1} \, T \\ c_{o} &= \mu_{3} \, \, \mu_{2} \, \Omega T^{2} \\ m_{d} &= \mu_{3} \, \, \mu_{2} \, T^{2} \end{split}$$

with

$$\begin{split} \mu_{0} &= \frac{(1+\xi)\sigma^{2}}{(1+\xi+2\sigma)^{2}} \\ \mu_{1} &= \frac{(1+\xi)^{2}\sigma + (1+\xi)(2.33+2\xi)\sigma^{2} + 3.33(1+\xi)\sigma^{3} + 1.33\sigma^{4}}{(1+\xi+2\sigma)^{3}} \\ \mu_{2} &= \frac{0.33(1+\xi)^{2}(2\xi-1)\sigma + (1+\xi)(1+2\xi)\sigma^{2} + 2(1+\xi)\sigma^{3} + 1.33\sigma^{4}}{(1+\xi+2\sigma)^{4}} \\ \mu_{3} &= \frac{\pi R\Delta P}{\lambda} \\ \mu_{3} &= \frac{\pi R\Delta P}{\lambda} \\ \lambda &= 0.079 \ R_{a}^{-0.25} \bigg[1 + \bigg(\frac{7R_{a}}{8R_{a}} \bigg)^{2} \bigg]^{0.375} \text{ Friction Loss Factor} \\ R_{a} &= \frac{2\rho VC}{\mu} \qquad \text{Axial Reynolds number} \\ R_{c} &= \frac{\rho R\Omega C}{\mu} \qquad \text{Circumferential Reynolds number} \\ \mu &= \text{ is the fluid viscosity} \\ T &= \frac{L}{V} \end{split}$$

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.

Liquid Annular Seal - Dynamic Coefficients 🛛 🔀										
Units: Me	tric	• Me	Method: Black & Jenssen (1970) 🗨							
Seal Length:	50	mm	Fluid Density:	979	Kg/m^3					
Shaft Diameter:	150	mm	Dynamic Viscosity:	0.414	CentiPoise					
Radial Clearance:	0.25	mm	Pressure Drop:	13.8	bars					
Speed (rpm):	1200		Inlet Loss Factor:	0.1						
Flow Rate =	0.0033684	m^3/s	Inlet Swirl Ratio:	0.5						
К = Кхх = Куу =	6.247E+006	N/m	С = Схх = Суу	= 29391	N-s/m					
k = Кху = -Кух =	1.8467E+006	N/m	с = Сху = -Сух :	= 1104.6	N-s/m					
M = Mxx = Myy =	8.7899	N-s^2/m	B	un	<u>C</u> lose					

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:

$$\begin{split} K_{d} &= \left(\frac{\pi R \triangle P}{\lambda}\right) \frac{2\sigma^{2}}{(1+\xi+2\sigma)} \left\{ 1.25 E - \frac{(\Omega T)^{2}}{4\sigma} \left\{ \frac{1}{2} \left(\frac{1}{6} + E\right) + \frac{2\nu_{0}}{a} \left[\left(E + \frac{1}{a^{2}}\right) (1-e^{-a}) - \left(\frac{1}{2} + \frac{1}{a}\right) e^{-a} \right] \right\} \right] \\ k_{e} &= \left(\frac{\pi R \triangle P}{\lambda}\right) \frac{\sigma^{2} \Omega T}{(1+\xi+2\sigma)} \left\{ \frac{E}{\sigma} + \frac{B}{2} \left(\frac{1}{6} + E\right) + \frac{2\nu_{0}}{a} \left\{ EB + \left(\frac{1}{\sigma} - \frac{B}{a}\right) \left[(1-e^{-a} \left(E + \frac{1}{2} + \frac{1}{a}\right) - 1 \right] \right\} \right\} \\ C_{d} &= \left(\frac{\pi R \triangle P}{\lambda}\right) \frac{2\sigma^{2} T}{(1+\xi+2\sigma)} \left[\frac{E}{\sigma} + \frac{B}{2} \left(\frac{1}{6} + E\right) \right] \\ c_{e} &= \left(\frac{\pi R \triangle P}{\lambda}\right) \frac{2\sigma \Omega T^{2}}{(1+\xi+2\sigma)} \left\{ \frac{1}{2} \left(\frac{1}{6} + E\right) + \frac{\nu_{0}}{a} \left[(1-e^{-a} \left(E + \frac{1}{2} + \frac{1}{a^{2}}\right) - \left(\frac{1}{2} + \frac{e^{-a}}{a}\right) \right] \right\} \\ m_{d} &= \left(\frac{\pi R \triangle P}{\lambda}\right) \frac{\sigma \left(\frac{1}{6} + E\right)}{(1+\xi+2\sigma)} \cdot T^{2} \end{split}$$

where

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

$$R_a = \frac{\rho \, VC}{\mu} \qquad {\rm Axial \ Reynolds \ number}$$
 (note: it is a half of the Reynolds number defined by Black)

$$R_{\sigma} = \frac{\rho \operatorname{R}\Omega C}{\mu}$$

$$b = \frac{R_{\sigma}}{R_{\sigma}} = \frac{V}{R\Omega}$$

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

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

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

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

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

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.

Liquid Annular S	eal - Dynamic	: Coefficie	nts				
Units: Me	tric	• Me	Method: Childs (1983)				
Seal Length:	43.2	mm	Fluid Density:	70.78	Kg/m^3		
Shaft Diameter:	79.8	mm	Dynamic Viscosity:	0.0116	CentiPoise		
Radial Clearance:	0.1397	mm	Pressure Drop:	149.2	bars		
Speed (rpm):	37360		Inlet Loss Factor:	0.1			
Flow Rate =	0.013517		Inlet Swirl Ratio:	0.5			
К = Кхх = Куу =	9.0192E+007	N/m	C = Cxx = Cyy	= 22628	N-s/m		
k = Kxy = -Kyx =	4.4264E+007	N/m	с = Сху = -Сух :	■ 3797.6	N-s/m		
М = Мхх = Муу =	0.97068	N-s^2/m	B	un	<u>C</u> lose		

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

Liquid Annular Se	eal - Dynamic	: Coefficier	its		×
Units: Me	tric	• Me	thod: Childs (1983)	•
Seal Length:	43.2	mm	Fluid Density:	70.78	Kg/m^3
Shaft Diameter:	79.8	mm	Dynamic Viscosity:	0.0116	CentiPoise
Radial Clearance:	0.1397	mm	Pressure Drop:	149.2	bars
Speed (rpm):	37360		Inlet Loss Factor:	0.1	
Flow Rate =	0.013517	m^3/s	Inlet Swirl Ratio:	0	
К = Кхх = Куу =	9.4649E+007	N/m	С = Схх = Суу	= 22628	N-s/m
k = Kxy = -Kyx =	1.8023E+007	N/m	с = Сху = -Сух	= 1519.2	N-s/m
М = Мхх = Муу =	0.97068	N-s^2/m	B	un	<u>C</u> lose

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:

Rotor Bearing System Data	
Notor Dearing System Data	
Axial Forces Static Loads Constraint	
Units/Description Material Shaft Eler	nents Disks Unbalance Bearings Supports Foundation User's Elements
Bearing: 1 of 1	Foundation Add Brg Del Brg Previous Next
Station I: 5 J: 0	
Type: 11-Liquid Annular Seal	<u>+</u>
Comment: Seal Example	
· · · · · · · · · · · · · · · · · · ·	
Method: Black & Jer	ssen 🔄 Inlet Swirl Ratio: 0.5
Seal Length: 1.2	Fluid Density: 9.325e-05
Shaft Diameter: 6	Dynamic Viscosity: 1.878407
Radial Clearance: 0.0075	Inlet Loss Factor: 0.1
Pressure Drop = dPo + dP1 *	rpm + dP2 * rpm^2
dPo: 500	dP1: 0 dP2: 0
Nominal Operating Speed (pm): 3600
	Unit:(1) - Geometry: in, Viscosity: Reyn, Density: Lbf-s^2/in^4, Pressure: psi
	<u>Save</u> Save <u>C</u> lose <u>H</u> elp

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

 $\frac{1}{2} \left(1+\xi\right) \rho V^2.$

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 15th 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.

Laby Seal Calculation - C:\WyFolder\DyRoBeS\Examples\Laby_Seal_Exam	ple.LSI								
Comment: Example									
Comment: For more complicated seal geometry and analysis, you may contact Dr. Kirk for additional consultation.									
Tooth Location: Tooth on Stator 🗾 Gas Molecular Weight:	43.86								
Number of Teeth 5 Compressibility:	0.7857								
Tooth Height: 0.0938 in Ratio of Specific Heat:	1.6								
Tooth toTooth Length: 0.0937 in Cp at Constant Pressure	0.37	BTU/LBM/DEG	-R						
Radial Clearance: 0.008 in Viscosity:	1.65e-005	LBM/FT/SEC							
Absolute Temperature	770	DEG-R							
Shaft Diameter: 5.75 in Upstream High Pressure:	1755	PSI							
First tooth for K, C calculation: 1 Low Pressure at Exit:									
Last tooth for K, C calculation: 5 Absolute Gas Velocity:	252	FT/SEC							
Analysis Type: Synchronous Whirl and Aero Q Calculation Inlet Swirt:	0.759		New						
First Critical Speed: 7400 RPM Multiple Case/Speed Rotor Speed:	10832	RPM	<u>O</u> pen						
Results for Selected Case, Use << or >> to Select			Save						
Case Number: <<- 1 +>> Effective Aero Q =		Lbf/in							
K = Kxx = Kyy = Lbf/in C = Cxx = Cyy =		Lbf-s/in	Save <u>A</u> s						
k = Kxy = -Kyx = Lbf/in c = Cxy = -Cyx =		Lbf-s/in	<u>R</u> un						
Flow Rate = Lbm/sec SCFM =									

1 1		Inlet Swirl	P high	P low	T-degR	Compressibilit	Cp/Cv	MW	Velocity	Viscosity	Ср	Nor
	10832	0.759	1755	938	770	0.7857	1.6	43.86	252	1.65E-05	0.37	7400
2 1	10000	0.62										
3 1	10832		1500	800								
4												
5												
6												
7												
3												
3												

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	$\frac{2\muRL^3s\omega}{C^3(1-s^2)^2}$	$\frac{\mu R L^3 \pi}{2C^3 (1-\varepsilon^2)^{3/2}}$
	2π film		0	$\frac{\mu R L^3 \pi}{C^3 (1-\boldsymbol{s}^2)^{3/2}}$
	π film	Pure Radial Squeeze Motion	0	$\frac{\mu R L^3 \pi \left(2 \varepsilon^2 + 1\right)}{2 C^3 \left(1 - \varepsilon^2\right)^{\frac{5}{2}}}$
	2π film		0	$\frac{\mu R L^3 \pi \left(2 \varepsilon^2 + 1\right)}{C^3 \left(1 - \varepsilon^2\right)^{5/2}}$
Long Bearing	π film	Circular Synchronous Precession	$\frac{24\muR^3Ls\omega}{C^3(2+s^2)(1-s^2)}$	$\frac{12\muR^3L\pi}{C^3(2+s^2)(1-s^2)^{\frac{1}{2}}}$
	2π film		0	$\frac{24\muR^3L\pi}{C^3(2+\varepsilon^2)(1-\varepsilon^2)^{\frac{1}{2}}}$

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 <u>dyrobes.com</u>

DyRoBeS©_**BePerf** computer program has been developed to analyze the <u>Be</u>aring steady state and dynamic <u>Perf</u>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 <u>Dy</u>namics of <u>Ro</u>tor <u>Bearing Systems</u>.

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:

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

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

$$\text{Re} = \frac{\rho Uh}{\mu} \quad \text{Local Reynolds number}$$

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

$$\tau_s = C_f \frac{\mu U}{h} + \frac{h}{2} \frac{\partial P}{\partial y}$$
$$C_f = 1 + 0.0012 \operatorname{Re}^{0.94}$$

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 Swiff-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.

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

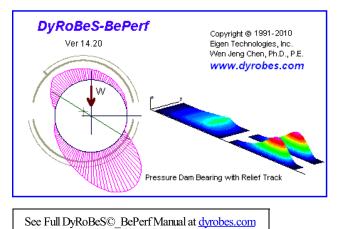
This compressible Reynolds equation is more difficult to analyze due to the existence of the pressure (P) in each terms compared with the incompressible flow, which makes the problem non-linear. Weak formulation based on variational principle is applied for generating the finite element model for the 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



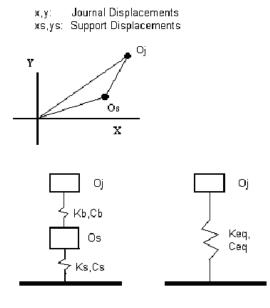
Mass/Inertia Properties of a Homogeneous Element

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.

Inertia Properties of Solid 🛛 🔀
Type: Circular Cylinder
Inner Diameter: 0
Mass: 66.6803 Center of Gravity: 6
Moment of Inertia at Center of Gravity Diametrat: 904.352 Polar: 208.376
Inertia Properties of Solid
Inertia Properties of Solid
Type: Conical Element Image: Element Left End Right End Inner Diameter: 0
Type: Conical Element Left End Right End Inner Diameter: 0 Outer Diameter: 5 Axial Length: 12
Type: Conical Element Image: Bun image
Type: Conical Element Image: Bun Left End Right End Inner Diameter: 0 0 Outer Diameter: 5 8 Axial Length: 12 12 Density: 0.283 Calculated Results

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:

$$\begin{cases} F_{x} \\ F_{y} \end{cases} = -\begin{bmatrix} K_{xx} & K_{xy} \\ K_{yx} & K_{yy} \end{bmatrix}_{b} \begin{cases} x - x_{s} \\ y - y_{s} \end{cases} - \begin{bmatrix} C_{xx} & C_{xy} \\ C_{yx} & C_{yy} \end{bmatrix}_{b} \begin{cases} \dot{x} - \dot{x}_{s} \\ \dot{y} - \dot{y}_{s} \end{cases}$$
$$= -\begin{bmatrix} Z_{xx} & Z_{xy} \\ Z_{yx} & Z_{yy} \end{bmatrix}_{b} \begin{cases} x - x_{s} \\ y - y_{s} \end{cases} = -\mathbf{Z}_{b} (\mathbf{x} - \mathbf{x}_{s})$$
$$= -\begin{bmatrix} K_{xx} & K_{xy} \\ K_{yx} & K_{yy} \end{bmatrix}_{eq} \begin{cases} x \\ y \end{bmatrix} - \begin{bmatrix} C_{xx} & C_{yy} \\ C_{yx} & C_{yy} \end{bmatrix}_{eq} \begin{cases} \dot{x} \\ \dot{y} \end{bmatrix}$$
$$= -\begin{bmatrix} Z_{xx} & Z_{xy} \\ K_{yx} & K_{yy} \end{bmatrix}_{eq} \begin{cases} x \\ y \end{bmatrix} = -\mathbf{Z}_{eq} \mathbf{x}$$

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

$$\begin{bmatrix} M_{xx} & M_{xy} \\ M_{yx} & M_{yy} \end{bmatrix}_{s} \begin{bmatrix} \ddot{x}_{s} \\ \ddot{y}_{s} \end{bmatrix} + \begin{bmatrix} C_{xx} & C_{xy} \\ C_{yx} & C_{yy} \end{bmatrix}_{s} \begin{bmatrix} \dot{x}_{s} \\ \dot{y}_{s} \end{bmatrix} + \begin{bmatrix} K_{xx} & K_{yy} \\ K_{yx} & K_{yy} \end{bmatrix}_{s} \begin{bmatrix} x_{s} \\ y_{s} \end{bmatrix} = - \begin{bmatrix} F_{x} \\ F_{y} \end{bmatrix}$$

By introducing the dynamic stiffness notation

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

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

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

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

$$\mathbf{Z}_{sq} = \left(\mathbf{I} + \mathbf{Z}_{b}\hat{\mathbf{A}}_{s}\right)^{-1}\mathbf{Z}_{b}$$

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

$$\mathbf{Z}_{eq} = \begin{bmatrix} \begin{pmatrix} K_b \begin{pmatrix} K_s - \omega^2 M_s \end{pmatrix} \end{pmatrix} \\ \begin{pmatrix} K_b + K_s - \omega^2 M_s \end{pmatrix} & \mathbf{0} \\ 0 & \begin{pmatrix} K_b \begin{pmatrix} K_s - \omega^2 M_s \end{pmatrix} \end{pmatrix} \\ \begin{pmatrix} K_b \begin{pmatrix} K_s - \omega^2 M_s \end{pmatrix} \end{pmatrix} \end{pmatrix} \\ \end{pmatrix}$$

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.

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

Case 2 - High Support Stiffness

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

Case 3 - High Support Inertia

When $\omega^2 M_s
ightarrow \infty$, $K_{sq}
ightarrow K_b$

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

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, $\omega^2 = \frac{K_b + K_s}{M_s}$

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

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 Elliptical and Total Orbit Analysis

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.

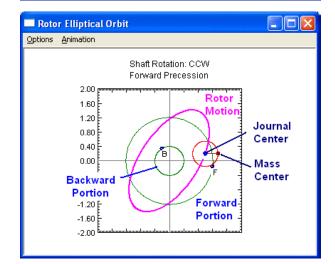
Elliptical Orbit

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:

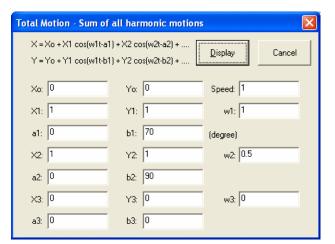
 $\begin{aligned} x(t) &= x_s \cos \alpha t + x_s \sin \alpha t = |x| \cos(\alpha t - \phi_s) \\ y(t) &= y_s \cos \alpha t + y_s \sin \alpha t = |y| \cos(\alpha t - \phi_s) \end{aligned}$

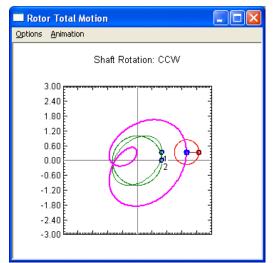
This represents an elliptical orbit. The elliptical orbit can be decomposed into two circular orbits: one is a forward circular orbit and the other is a backward circular orbit.

Elliptical Orbit Analysis									
X = Xc cos(wt) + Xs sin(wt) = XI cos (wt-phasex)									
Y = Yc cos (wt) + Ys sin (wt) = [Y cos	(wt-phasey) Cancel								
Input Data: Select one option									
Xc, Xs, Yc, Ys	🔘 KI, MI, phase x, phase y								
Xc: 1 Xs: 0.5	K: 1.11803 Phase X (deg): 26.5651								
Yc: 0.2 Ys: 1.4	M: 1.41421 Phase Y (deg): 81.8699								
Calculate	ed Results								
Elliptical Orbit Data	Two Circular Orbits Data								
Semi-Major Axis: 1.61245	Forward Amplitude: 1.20934								
Semi-Minor Axis: 0.806226	Phase Angle: 352.875								
Attitude Angle: 56.3099	Backward Amplitude: 0.403113								
Precession: Forward	Phase Angle: 119.745								



Total Motion





Balancing Calculation

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

- 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.
- 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.

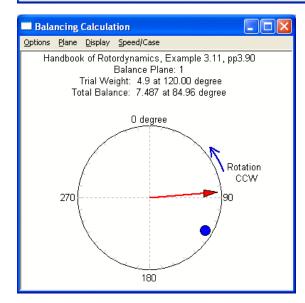
0 degree position defines the reference mark where all the angles (phases) are measured from

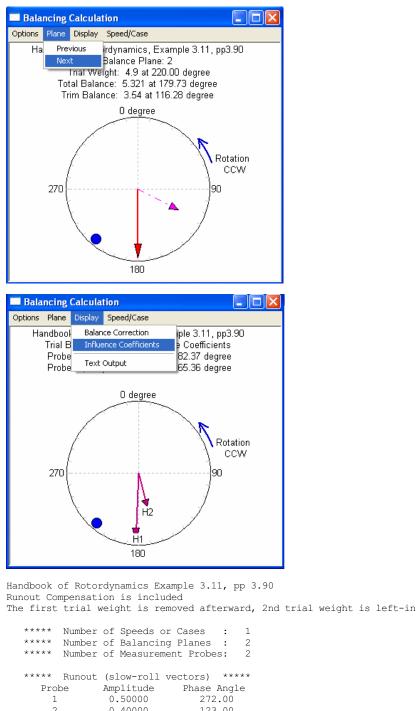
9. Weighting Factors

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:

umber	of Balancing Planes	: 2	Number of Speeds	/Cases: 1	_	⊢S	haft Rotatio	on	_	- Phase	
Number of Measured Probes: 2 Runout Compensation: Yes						• ccw	O CW		🖲 Lag	C Lead	
ambor		1			<u> </u>						
Comment: Handbook of Rotordynamics, Example 3.11, pp3.90								degree at			
ommen	t Runout compen	sation is i	ncluded			_		🖲 Y - Up	0	X - Right	
ommen	t: The first trial wei	ight is rem	oved afterward, 2nd trial w	eight is left-in			Weighti	ing (Scale) Fac	tors f	or probes an	d speeds
	Condition	Speed	Description	Amplitude	Phase (deg)		Probe	Factor		Speed	Factor
1	Runout		Probe: 1	0.5	272		1	1		1	1
2	Runout		Probe: 2	0.4	123		2	1		2	1
3	Initial Readings	1	Probe: 1	1.8	148		3			3	1
4	Initial Readings	1	Probe: 2	3.6	115		4			4	1
5	Trial Run < 1 >		Remove Afterward 💌	4.9	120		5			5	1
6	Response	1	Probe: 1	1.1	178		6			6	1
7	Response	1	Probe: 2	2	98		7			7	
8	Trial Run < 2 >		Left-In Afterward 🛛 💌	4.9	220		8			8	
9	Response	1	Probe: 1	2.1	98		9			9	
10	Response	1	Probe: 2	3.7	102		10			10	
11							11				
12							12			New	Open
13							13			<u>N</u> ew	<u>o</u> pen
14							14				
15							15				C
16							16			<u>S</u> ave	Save <u>A</u> :
17							17				
18							18				





 1
 0.50000
 2/2.00

 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

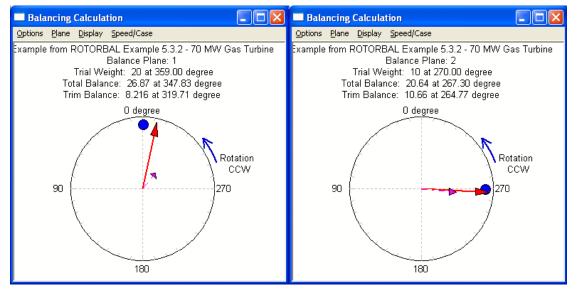
Plane Amplitude Phase Angle Afterward 1 4.9000 120.00 Remove ----- Response to Trial Unbalance ------Speed Probe Amplitude Phase Angle 1.1000 178.00 1 1 1 2 2.0000 98.000 Amplitude Phase Angle Afterward Plane 2 4.9000 220.00 Left-In

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

5	Speed 1 1	Probe 1 2	Ampli 2.1 3.7	000		se Angle 98.000 102.00		
***	Weight Probe 1 2		ors for hting Fa 1.0000 1.0000		and s	peeds ***	¢	
	Speed 1	Weig	hting Fa 1.0000	ctor				
Co		7.		lance t Phas 84		tor		
Co T:	orrectio		ed if Tr	ial Wei nce - T Phas	ght L	eft-in Weight		
***	*****	The Inf	luence C			*******	***	
T	rial-Rur	1		-		Phase		
	1 1	1	1 2	0.206		175. 194.		
	2	1	2	0.364		194.		
	2	1	2	0.169	86	165.		
						al Respor		
Sne	eed Pi	robe	Amplitu			t e Angle	Amplitu	Phase Angle
-	l I	1	0.000			0000	0.5000	272.00
	L	2	0.000			0000	0.4000	123.00

Example 2:

lumbe	r of Balancing Planes	: 2	Number of Speeds	:/Cases: 6		_ SI	haft Rotatio	n	Phase	
lumbe	r of Measured Probes	: 2	Runout Compet	nsation: No	•		⊙ CCW	C CW	C Lag	🔹 💽 Lead
	- Evample from Pi		_ Example 5.3.2 - 70 MW G	an Turbina		_	∩	degree at		_
onine						_		-	C 14 B 14	
omme	nt: Example from R	DTORBAL	- Example 5.3.2 - 70 MW G	ias Turbine				● Y・Up	◯ X - Right	
omme	nt: Example from R	DTORBAL	. Example 5.3.2 - 70 MW G	ias Turbine			Weightir	ng (Scale) Facto	ors for probes ar	nd speeds
	Condition	Speed	Description	Amplitude	Phase (deg)		Probe	Factor	Speed	Factor
1	Initial Readings	1	Probe: 1	1.7	339		1	1	1	1
2	Initial Readings	1	Probe: 2	4.6	54		2	1	2	1
3	Initial Readings	2	Probe: 1	2.8	226		3		3	1
4	Initial Readings	2	Probe: 2	6.7	10		4		4	1
5	Initial Readings	3	Probe: 1	3.9	145		5		5	1
6	Initial Readings	3	Probe: 2	3.7	333		6		6	1
7	Initial Readings	4	Probe: 1	4.5	103		7		7	
8	Initial Readings	4	Probe: 2	4.7	302		8		8	
9	Initial Readings	5	Probe: 1	5.4	74		9		9	
10	Initial Readings	5	Probe: 2	6.5	113		10		10	
11	Initial Readings	6	Probe: 1	1.98	98		11			
12	Initial Readings	6	Probe: 2	5.7	114		12			
13	Trial Run < 1 >		Left-In Afterward 🛛 🚽	20	359		13		<u>N</u> ew	<u>O</u> pen
14	Response	1	Probe: 1	2.6	313		14			
15	Response	1	Probe: 2	5.9	7		15			
16	Response	2	Probe: 1	3.9	232		16		<u>S</u> ave	Save <u>A</u> s
17	Response	2	Probe: 2	4.4	4		17			
18	Response	3	Probe: 1	4.2	160		18		·	
19	Response	3	Probe: 2	2.8	340		19		<u>R</u> un	<u>C</u> lose



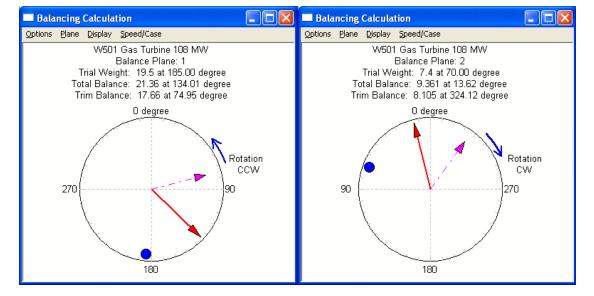
Example from ROTORBAL Example 5.3.2 - 70 MW Gas Turbine 2 probes at 6 speeds No runout

)	runout			
	****	Number of Sr	eeds or Cases	: 6
			lancing Planes	
			asurement Probe	
		Number of He	abarement 1100	
	****	NO Runout Co	mpensation	
			1	
		Initial Res	ponse (Without	Trails) ======
	Spee	ed 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 6	2	6.5000	113.00
	6	1 2	1.9800 5.7000	98.000 114.00
	0	Z	5.7000	114.00
	******	***** Tri	al Unbalance Ru	ın: 1 *************
	Plan		tude Phase	
	1	-		9.00 Left-In
			e to Trial Unba	lance
	Spee		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 2	4.2000	160.00
	3 4	2	2.8000 3.5000	340.00
	4	2	5.4000	120.00 325.00
	4 5	1	4.1000	73.000
	5	2	3.7000	97.000
	6	1	1.5000	141.00
	6	2	3.1000	99.000
	******	***** Tri	al Unbalance Ru	un: 2 ************
	Plan	ne Ampli	tude Phase	Angle Afterward
	2	10.	000 270	0.00 Left-In
			to Trial Unba	
	Spee		Amplitude	Phase Angle
	1	1	1.8000	319.00
	1	2	4.3000	15.000
	2	1 2	3.1000	244.00
	2 3	2	3.4000 3.2000	9.0000 107.00
	3	2	1.9000	329.00
	4	2	2.4000	122.00

	4 5 5	2 1 2	4.40	000	330.00 51.000		
	5 6 6	2 1 2	3.50 1.02 3.11	200	L01.00 L70.00 L04.00		
***	Weighti Probe 1 2		ors for p hting Fac 1.0000 1.0000	probes and sp ctor	peeds ***		
	Speed	Weig	hting Fac	ctor			
	1 2		1.0000 1.0000				
	3 4		1.0000				
	5		1.0000 1.0000				
	6		1.0000				
				rection >>>> lance the Rot		==	
	lane No.	Amp	litude	Phase Ang			
	1 2		.867 .639	347.83 267.30			
				rection >>>> ial Weight Le			
Tr	im Balan	ice = To	tal Balar	nce - Trial W	Veight		
P	lane No. 1		litude 2159	Phase Angl 319.71	Le		
	2		.660	264.77			
****	*****	The Inf	luence Co	Defficients		**	
Tr	ial-Run	Speed	Probe	Influence Amplitude			
	1 1	1 1	1 2	0.65281E-01 0.21766	279. 317.		
	1	2	1	0.57655E-01			
	1 1	2 3	2 1	0.11846 0.54915E-01	202.		
	1	3	2	0.49103E-01			
	1 1	4 4	1 2	0.77078E-01 0.10636	242. 26.		
	1	5	1	0.65130E-01			
	1 1	5 6	2 1	0.15575 0.67568E-01	313. 230.		
	1	6	2	0.14110	312.		
	2	1	1	0.83143E-03	210. 257.		
	2 2	1 2	2 1	0.17475 0.10809	105.		
	2	2	2	0.10554	258.		
	2 2	3 3	1 2	0.34210 0.10027	118. 271.		
		4	1 2	0.11046	26. 214.		
	2						
	2 2 2	4 5	1	0.10867 0.18221	359.		
	2 2 2	4 5 5	1 2	0.18221 0.32108E-03	359. 318.		
	2 2	4 5	1	0.18221	359. 318. 12.		
	2 2 2 2	4 5 5 6	1 2 1 2	0.18221 0.32108E-01 0.78362E-01 0.27106E-01	359. 318. 12. 279.		
	2 2 2 2	4 5 6 6	1 2 2 === Pred: W:	0.18221 0.32108E-01 0.78362E-01 0.27106E-01 icted Residua ithOUT Runout	359. 318. 12. 279.	With	
 Spe 1	2 2 2 2 2 2 	4 5 6 6	1 2 1 2 W: Amplitud	0.18221 0.32108E-01 0.78362E-01 0.27106E-01 icted Residua ithOUT Runout de Phase	359. 318. 12. 279. al Respons	With Amplitude	Phase An
1	2 2 2 2 2 2 eed Prc	4 5 6 6	1 2 1 2 W: Amplituc 1.0968 2.606	0.18221 0.32108E-01 0.78362E-01 0.27106E-01 icted Residua ithOUT Runout de Phase 3 310 7 358	359. 1 318. 1 12. 279. al Respons e Angle 0.43 3.29	With Amplitude 1.0968 2.6067	Phase An 310.43 358.29
1 1 2	2 2 2 2 2 2 2 2 2 2 2 2 2 2 2 2 2 2 2	4 5 6 6	1 2 1 2 W: Amplituc 1.0968 2.606 2.874	0.18221 0.32108E-07 0.78362E-07 0.27106E-07 icted Residua ithOUT Runout de Phase 3 310 7 358 4 25	359. 318. 12. 279. al Respons e Angle 0.43 0.29 7.03	With Amplitude 1.0968 2.6067 2.8744	Phase An 310.43 358.29 257.03
1 1 2 2 3	2 2 2 2 2 2 2 2 2 2 2 2 2 2 2 2 2 2 2	4 5 6 6	1 2 1 2 W: Amplituc 1.0966 2.8744 1.7963 4.8509	0.18221 0.32108E-02 0.78362E-02 0.27106E-02 icted Residua ithOUT Runout de Phase 3 310 7 356 4 225 3 40 9 66	359. 318. 12. 279. al Respons Angle 0.43 3.29 7.03 778 019	With Amplitude 1.0968 2.6067 2.8744 1.7963 4.8509	Phase An 310.43 358.29 257.03 40.778 66.019
1 1 2 2 3 3	2 2 2 2 2 2 2 2 2 2 2 2 2 2 2 2 2 2 2	4 5 6 6	1 2 1 2 W3 Amplituc 1.0968 2.606 2.8744 1.7963 4.8509 0.73525	0.18221 0.32108E-01 0.78362E-01 0.27106E-01 icted Residua ithOUT Runout de Phase 3 310 7 356 4 257 3 40 9 66 5 31	359. 318. 12. 279. al Respons Angle 43 3.29 7.03 778 019 7.09	With Amplitude 1.0968 2.6067 2.8744 1.7963 4.8509 0.73525	Phase An 310.43 358.29 257.03 40.778 66.019 317.09
1 1 2 2 3 3 4 4	2 2 2 2 2 2 2 2 2 2 2 2 2 2 2 2 2 2 2	4 5 6 6 	1 2 1 2 W: Amplituc 1.0966 2.8744 1.7963 4.8509	0.18221 0.32108E-01 0.78362E-01 0.27106E-01 icted Residua ithOUT Runout de Phase 3 31(7 358 4 257 3 40 9 666 5 317 1 154	359. 318. 12. 279. al Respons Angle 0.43 3.29 7.03 778 019	With Amplitude 1.0968 2.6067 2.8744 1.7963 4.8509	Phase An 310.43 358.29 257.03 40.778 66.019 317.09 154.41
1 1 2 3 3 4 4 5	2 2 2 2 2 2 2 2 2 2 2 2 2 2 2 2 2 2 2	4 5 6 6 	1 2 1 2 W: Amplitud 1.0968 2.606 ⁻ 2.8744 1.7963 4.8509 0.7352 ² 1.6055 4.3282 0.5503 ²	0.18221 0.32108E-01 0.78362E-01 0.27106E-01 icted Residual ithOUT Runout de Phase 3 310 7 358 4 255 3 40 9 66 5 311 1 152 2 342 1 343	359. 318. 12. 279. al Respons Angle 0.43 3.29 7.03 7.03 7.09 4.41 01 3.11	With Amplitude 1.0968 2.6067 2.8744 1.7963 4.8509 0.73525 1.6051 4.3282 0.55031	Phase An 310.43 358.29 257.03 40.778 66.019 317.09 154.41 341.01 343.11
1 1 2 2 3 3 4 4	2 2 2 2 2 2 2 2 2 2 2 2 2 2 2 2 2 2 2	4 5 6 6 	1 2 1 2 W: Amplituc 1.0968 2.606' 2.8744 1.7963 4.8500 0.73525 1.6055 4.3282	0.18221 0.32108E-01 0.78362E-01 0.27106E-01 icted Residua ithOUT Runout de Phase 3 310 7 355 4 225 3 40 9 66 5 31 ² 1 15 2 34 ² 1 34 ² 3 10 ⁴ 1 15 ⁴ 2 34 ² 1 34 ² 3 11 ⁴	359. 318. 12. 279. al Respons Angle 0.43 3.29 7.03 7.78 0.19 0.09 1.41 1.01	With Amplitude 1.0968 2.6067 2.8744 1.7963 4.8509 0.73525 1.6051 4.3282	Phase An 310.43 358.29

Example 3:

DyRo	BeS_RotorBal - I	nput Da	ıta								
umbe	r of Balancing Planes	: 2	Number of Speed:	s/Cases: 2	_	⊢ S	ihaft Rotati	on	F	'hase	
umbe	r of Measured Probes	s: 4	Runout Compe	nsation: No	•		⊙ CCW	C CW		🖲 Lag	C Lead
				1	_			degree at			
omme	ent: W501 Gas Turb	ine 108 M	W				-0	-			
omme	ent: 2 Speeds, 3070	RPM, 36	DO RPM			_		Y · Up	ΟX	- Right	
omme	ent: 4 Probes and 21	Balancing	Planes				Weight	ing (Scale) Fac	tors for p	robes an	d speeds
	Condition	Speed	Description	Amplitude	Phase (deg)		Probe	Factor	S	peed	Factor
1	Initial Readings	1	Probe: 1	1.1	143		1	1		1	1
2	Initial Readings	1	Probe: 2	3.7	268		2	1.2		2	1.2
3	Initial Readings	1	Probe: 3	1.7	287		3	1.1		3	
4	Initial Readings	1	Probe: 4	0.8	156		4	1		4	
5	Initial Readings	2	Probe: 1	3.7	98		5			5	
6	Initial Readings	2	Probe: 2	7.5	41		6			6	
7	Initial Readings	2	Probe: 3	3.9	1		7			7	
8	Initial Readings	2	Probe: 4	4.2	209		8			8	
9	Trial Run < 1 >		Left-In Afterward 🛛 🗨	19.5	185		9			9	
10	Response	1	Probe: 1	2.5	70		10			10	
11	Response	1	Probe: 2	1.9	216		11				
12	Response	1	Probe: 3	2.9	23		12			. 1	
13	Response	1	Probe: 4	1.9	216		13			New	<u>O</u> pen
14	Response	2	Probe: 1	2.5	2		14				
15	Response	2	Probe: 2	6.8	350		15			. 1	
16	Response	2	Probe: 3	9.4	359		16		1	<u>ave</u>	Save <u>A</u>
17	Response	2	Probe: 4	6.3	202		17				
18	Trial Run < 2 >		Left-In Afterward 🛛 💌	7.4	70		18		· · · · ·		
19	Response	1	Probe: 1	2.3	301		19			<u>R</u> un	<u>C</u> lose
20	Response	1	Probe: 2	4.4	216	-	20		<u></u>		



W501 Gas Turbine 108 MW 2 Speeds, 3070 RPM, 3600 RPM 4 Probes and 2 Balancing Planes

****	Number	of	Speeds or C	ases	:	2
****	Number	of	Balancing P	lanes	:	2
****	Number	of	Measurement	Probes	:	4

***** NO Runout Compensation

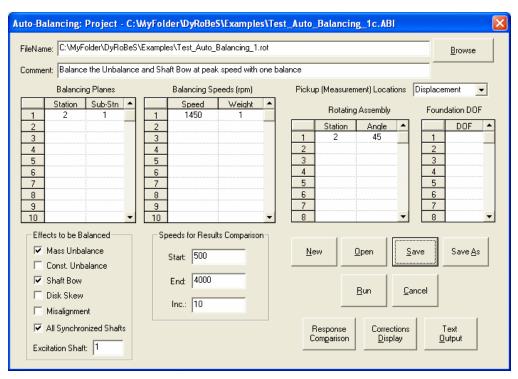
======	Initial Re	esponse (Without	Trails)	
Speed	d Probe	Amplitude	Phase	Angle
1	1	1.1000	143	.00
1	2	3.7000	268	.00
1	3	1.7000	287	.00
1	4	0.80000	156	.00
2	1	3.7000	98.	000
2	2	7.5000	41.	000
2	3	3.9000	1.0	000
2	4	4.2000	209	.00

	****** lane					**************************************
r	1	Ampli 19.	500	185.0		Left-In
		Response	to Tria	l Unbala	nce -	
S	peed	Probe	Ampli			Angle
	1	1		000		.000
	1 1	2 3		000		6.00
	1	4		000		.000 6.00
	2	1		000		0000
	2	2		000		0.00
	2 2	3 4		000		9.00 2.00
	Z	4	0.5	000	20	2.00
	******					**********
P	lane 2	Ampli 7.4	tude 000	Phase An 70.0	2	Afterward Left-In
	2		000			2010 11
		-		l Unbala		
S	peed 1	Probe 1		tude 000		Angle 1.00
	1	2		000		6.00
	1	3	2.5	000	29	4.00
	1	4	0.80			9.00
	2 2	1 2		000		.000 5.00
	2	2		000		4.00
	2	4		000		1.00
***	Weight Probe 1 2 3 4	-	ors for hting Fa 1.0000 1.2000 1.1000 1.0000	probes an ctor	nd spe	eds ***
	Speed	Weig	hting Fa	ctor		
	1 2		1.0000 1.2000			
Со	2 	on Requir o. Amp 21	1.2000 ance Cor	rection lance the	e Roto Angle .01	r
Co P <<<< Co Tr	2 	on Requir b. Amp 21 9. Trim Bal on Requir ince = To b. Amp 17	1.2000 ance Cor ed to Ba litude .356 3609 ance Cor ed if Tr	rection lance the Phase 134 13. rection rial Weigh nce - Tr	e Rotc Angle .01 616 >>>>> ht Lef ial We Angle 947	r >>>>> 't-in ight
Co P Co Tr P	2 <<<< T rrectic lane Nc 1 2 <</p constant of the second sec	on Requir b. Amp 21 9. Trim Bal on Requir ance = To b. Amp 17 8. The Inf	1.2000 ance Cor ed to Ba litude .356 3609 ance Cor ed if Tr tal Bala litude .665 1048 	rection lance the Phase 134 13. rection ial Weigh nce - Tr: Phase 74. 324 coefficient	e Roto Angle .01 616 >>>>> ht Lef ial We 947 .12 ====== nts *	r :in ight :
Co P Co Tr P	2 <<<< T rrectic lane Nc 1 2 <</p constant of the second sec	on Requir b. Amp 21 9. Trim Bal on Requir ance = To b. Amp 17 8. The Inf	1.2000 ance Cor ed to Ba litude .356 3609 ance Cor ed if Tr tal Bala litude .665 1048 	rrection lance the Phase 134 13. rrection ial Weigl nce - Tr: Phase 74. 324 coefficien Weighting	e Roto Angle .01 616 >>>>> ht Lef ial We 947 .12 ===== nts * g Fact	r
Co P Co Tr P	2 <<<< T rrectic lane Nc im Bala lane Nc 1 2 ****** Before	on Requir D. Amp 21 9. Trim Bal on Requir nnce = To D. Amp 17 8. The Inf e applica	1.2000 ance Cor ed to Ba litude .356 3609 ance Cor ed if Tr tal Bala litude .665 1048 luence C tion of	rection lance the Phase 134 13. rection ial Weigl nce - Tr: Phase 74. 324 coefficier Weightine Influe	e Roto Angle .01 616 >>>>> ht Lef ial We 947 .12 nts * g Fact ence C	r
Co P Co Tr P	2 <<<< T rrectic lane Nc 1 2 <</p constant of the second sec	on Requir D. Amp 21 9. Trim Bal on Requir nnce = To D. Amp 17 8. The Inf e applica	1.2000 ance Cor ed to Ba litude .356 3609 ance Cor ed if Tr tal Bala litude .665 1048 	rrection lance the Phase 134 13. rrection ial Weigl nce - Tr: Phase 74. 324 coefficien Weighting	e Rotc Angle .01 616 >>>>> ht Lef ial We Angle 947 .12 ====== nts * g Fact ence C de	r
Co P Co Tr P	2 <<<< T rrectic lane Nc 1 2 </	on Requir o. Amp 21 9. Trim Bal on Requir ance = To o. Amp 17 8. The Inf e applica 1 1	1.2000 ance Cor ed to Ba litude .356 3609 ance Cor ed if Tr tal Bala litude .665 1048 luence C tion of Probe 1 2	rection lance the Phase 134 13. rection Tial Weigh nce - Tr Phase 74. 324 coefficien Weightin Meightin Influc Amplituu 0.12400 0.1507	e Rotc Angle .01 616 >>>>> ht Lef ial We Angle 947 .12 ====== nts * g Fact ence C de 6 7	r
Co P Co Tr P	2 <<<< T rrectic lane Nc 1 2 <</ rrectic im Bala lane Nc 1 2 ****** Before ial-Run 1 1 1	n Requir n Requir 9. Trim Bal n Requir ince = To 0. Amp 17 8. The Inf e applica 1 1 1	1.2000 ance Cor ed to Ba litude .356 3609 ance Cor ed if Tr tal Bala litude .665 1048 uence Cor tion of Probe 1 2 3	rection lance the Phase 134 13. rection ial Weigl nce - Tr: Phase 74. 324 coefficien Weightine Influe Amplitue 0.1240 0.1507 0.1800	e Rotc Angle .01 616 >>>>> ht Lef ial We Angle 947 .12 nts * g Fact ence C de 6 7 8	r
Co P Co Tr P	2 <<<< T rrectic lane Nc 1 2 < rrectic im Bala lane Nc 1 2 	n Requir n Requir 9. Trim Bal n Requir nnce = To 0. Amp 17 8. The Inf e applica 1 1 1 1	1.2000 ance Cor ed to Ba litude .356 3609 ance Cor ed if Tr tal Bala litude .665 1048 uence C tion of Probe 1 2 3 4	rection lance the Phase 134 13. rection ial Weigl nce - Tr: Phase 74. 324 coefficiel Weighting Influe Amplitus 0.1240 0.1507 0.1800 0.8473	e Rotc Angle .01 616 >>>>> ht Lef ial We 947 .12 ====== nts * g Fact ence C de 6 7 8 22=01	r
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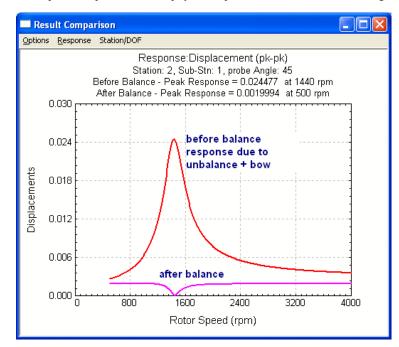
Speed	Probe	Amplitude	Phase Angle	Amplitude	Phase Angle
1	1	3.9669	49.648	3.9669	49.648
1	2	1.9065	348.46	1.9065	348.46
1	3	2.2628	79.442	2.2628	79.442
1	4	0.22558	305.47	0.22558	305.47
2	1	1.8109	9.5156	1.8109	9.5156
2	2	1.8740	232.93	1.8740	232.93
2	3	3.3169	159.79	3.3169	159.79
2	4	1.8632	30.989	1.8632	30.989
			==================		

Auto-Balancing Calculation

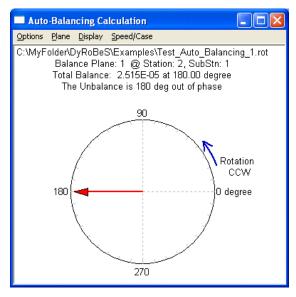
In contrast to the Influence coefficient method, the auto-balancing method requires full knowledge in the rotor-bearing model including all the harmonic excitations. An example is presented below. The rotor system has one mass unbalance and shaft bow at the center. Now, we will try to balance both excitations 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
6001 DATAINStation 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
$8001DATAINStation number \le 0$
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 -7EOAXB2Stiffness 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 2001 MAINPSNot 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 ReNumber button to correct this problem.
- 5. Units inconsistent

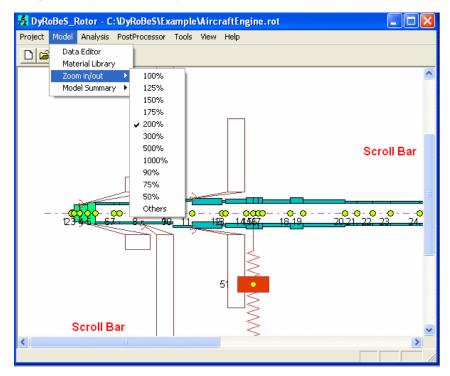
Always check the model using the Model Summary and check the equivalent rigid body properties with the anticipated values. Zoom in the model to see any missing elements or subelements.

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.

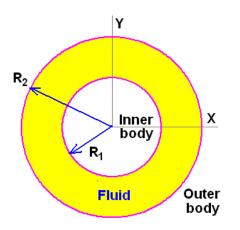
Zoom In/Out

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:

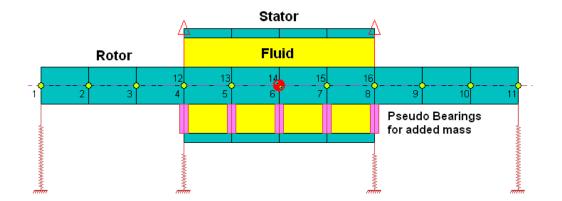
m_{11}	0	m_{13}		$\left\{ \ddot{x}_{i} \right\}$
ļo	m_{22}	0	m 24	\mathbf{y}_i
m_{31}	0	$m_{33}^{}$		$\left \ddot{x}_{j} \right $
0	m 42	0	m 44	$[\ddot{y}_i]$

Assuming the inner cylinder has a radius of R₁ and the outer cylinder has a radius of R₂. The length of the cylinder is L, the density of the fluid is p. The values for m are:

$$\begin{split} m_{11} &= m_{22} = \rho \pi L R_1^2 \left(\frac{R_2^2 + R_1^2}{R_2^2 - R_1^2} \right) = \text{added mass of inner cylinder} \\ m_{33} &= m_{44} = \rho \pi L R_2^2 \left(\frac{R_2^2 + R_1^2}{R_2^2 - R_1^2} \right) = \text{added mass of outer cylinder} \end{split}$$

$$m_{13} = m_{31} = m_{24} = m_{42} = -\rho\pi L \left(\frac{2R_1^2R_2^2}{R_2^2 - R_1^2}\right) = \text{fluid coupling}$$

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:

$$\begin{split} m_{11} &= m_{22} = \rho \pi L R_1^2 \left(\frac{R_2^2 + R_1^2}{R_2^2 - R_1^2} \right) = 1.5566 \text{ Lb} \\ m_{33} &= m_{44} = \rho \pi L R_2^2 \left(\frac{R_2^2 + R_1^2}{R_2^2 - R_1^2} \right) = 9.7287 \text{ Lb} \\ m_{13} &= m_{31} = m_{24} = m_{42} = -\rho \pi L \left(\frac{2 R_1^2 R_2^2}{R_2^2 - R_1^2} \right) = -2.6838 \text{ Lb} \end{split}$$

The added masses for the 3 bearings inside (using L=5 in), stations 5-13, stations 6-14, and stations 7-15 are:

$$\begin{split} m_{11} &= m_{22} = \rho \pi L R_1^2 \left(\frac{R_2^2 + R_1^2}{R_2^2 - R_1^2} \right) = 3.1132 \text{ Lb} \\ m_{33} &= m_{44} = \rho \pi L R_2^2 \left(\frac{R_2^2 + R_1^2}{R_2^2 - R_1^2} \right) = 19.4575 \text{ Lb} \\ m_{13} &= m_{31} = m_{24} = m_{42} = -\rho \pi L \left(\frac{2 R_1^2 R_2^2}{R_2^2 - R_1^2} \right) = -5.3676 \text{ Lb} \end{split}$$

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:

00

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.

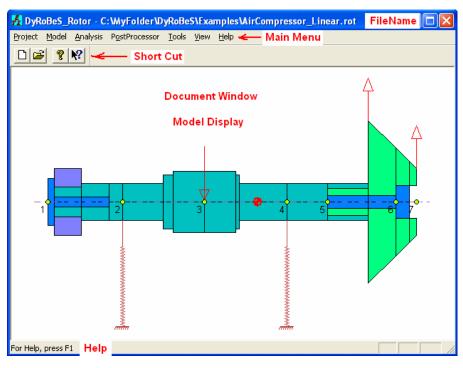
Linear interpolation will be used for sudden changes in excitation, such as step function, triangular shapes, etc... You may specify the starting time to include this excitation in the analysis. If the Periodic Excitation is checked, then the excitation will become periodic and it will repeat when the 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.

Torsional	excitation vs. time				×					
Enter Station Number and Excitation File. The file contains 2 columns, 1-Time (sec) and 2-Excitation The maximum number of points at each excitation file is 2000. Linear interpolation is used between points to allow sudden changes. Such as a retangular shape excitation where the same time point has two different excitations. Time has to be in the increasing order, that is t (i) <= t (i+1) equal sign is allowed for sudden change in excitation The data starts from T0 to Tf. For a periodic excitation, the period starts from Ts. i.e. T0<=Ts <tf. consistent="" ensure="" excitation="" is="" it="" responsibility="" selected="" system.<="" td="" the="" to="" unit="" user's="" with=""></tf.>										
Station	FileName (use Browse to select)		Periodic Excitatio	n ^{T start}						
1	C:\DyRoBeS\Example\Torsion\TorData.dat	Browse		0						
0		Browse		0						
0		Browse		0						
0		Browse		0						
0		Browse		0	1					

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 <u>Tools</u>

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

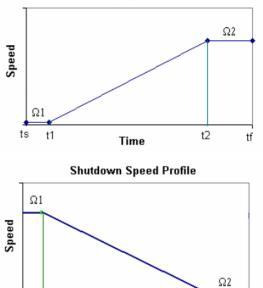
ts t1

Linear profile gives a constant acceleration during startup and deceleration during shutdown.

t2

tf

Startup Speed Profile



Time

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 $t \leq t_1$: Constant Speed

$$\begin{split} \ddot{\varphi}_0 &= 0 \\ \dot{\varphi}_0 &= \Omega_1 \\ \varphi_0 &= \Omega_1 t \end{split}$$

When $|t_1| \leq t \leq t_2$: Linear Acceleration or Deceleration

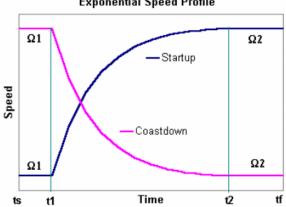
$$\begin{split} \ddot{\varphi}_1 &= \frac{\Omega_2 - \Omega_1}{t_2 - t_1} = \frac{\Delta \Omega}{\Delta t} \\ \dot{\varphi}_1 &= \Omega_1 + \ddot{\varphi}_1 \left(t - t_1 \right) = \left(\frac{\Omega_1 t_2 - \Omega_2 t_1}{t_2 - t_1} \right) + \left(\frac{\Omega_2 - \Omega_1}{t_2 - t_1} \right) t \\ \varphi_1 &= C_1 + \left(\frac{\Omega_1 t_2 - \Omega_2 t_1}{t_2 - t_1} \right) t + \frac{1}{2} \left(\frac{\Omega_2 - \Omega_1}{t_2 - t_1} \right) t^2 \\ C_1 &= \Omega_1 t_1 - \left(\frac{\Omega_1 t_2 - \Omega_2 t_1}{t_2 - t_1} \right) t_1 - \frac{1}{2} \left(\frac{\Omega_2 - \Omega_1}{t_2 - t_1} \right) t_1^2 \end{split}$$

When $t_2 \leq t$: Constant Speed

$$\begin{split} \ddot{\varphi}_2 &= 0\\ \dot{\varphi}_2 &= \Omega_2\\ \varphi_2 &= \Omega_2 t + C_2\\ C_2 &= \phi_1(t_2) - \Omega_2 t_2 \end{split}$$

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.



Exponential Speed Profile

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,

 $\Omega_2 = A\left(1-e^{-\lambda\left(t_2-t_1\right)}\right) + \Omega_1$

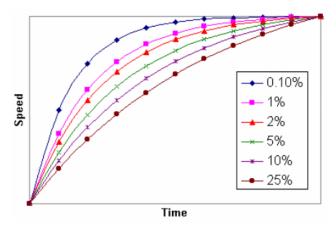
where $|A| \mathrm{and} |\mathcal{X}|$ can be related by a single parameter \mathcal{S} , let

$$\begin{split} & \Delta \Omega = \Omega_2 - \Omega_1 \\ & \Delta t = t_2 - t_1 \\ & A = (1 + \mathcal{S}) \Delta \Omega \end{split}$$

then we have

$$\lambda = -\frac{\ln\left(\frac{\delta}{1+\delta}\right)}{\Delta t}$$

 δ determines the shape of the curve, the following curves are for various $\,\delta$



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 :

$$\begin{split} \dot{\varphi} &= B \Big(1 - e^{-\lambda \left(t - t_1 \right)} \Big) + \Omega_1 \\ B &= -A \end{split}$$

When $t \leq t_1$: Constant Speed

$$\begin{split} \ddot{\varphi}_0 &= 0 \\ \dot{\varphi}_0 &= \Omega_1 \\ \varphi_0 &= \Omega_1 t \end{split}$$

When $t_1 \leq t \leq t_2$: Exponential profile

$$\begin{split} \ddot{\varphi}_1 &= A\lambda \, e^{-\lambda(t-t_1)} \\ \dot{\varphi}_1 &= A \big(1 - e^{-\lambda(t-t_1)} \big) + \Omega_1 \\ \varphi_1 &= C_1 + \big(A + \Omega_1 \big) t + \frac{A}{\lambda} \, e^{-\lambda(t-t_1)} \\ C_1 &= -At_1 - \frac{A}{\lambda} \end{split}$$

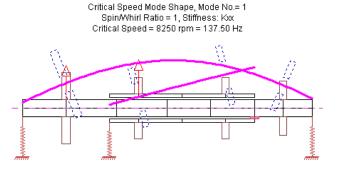
When $t_2 \leq t$: Constant Speed

$$\begin{split} \ddot{\varphi}_2 &= 0 \\ \dot{\varphi}_2 &= \Omega_2 \\ \varphi_2 &= \Omega_2 t + C_2 \\ C_2 &= \phi_1(t_2) - \Omega_2 t_2 \end{split}$$

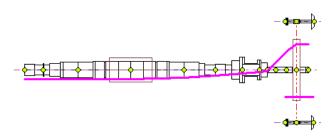
Animation

Animation

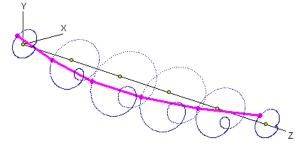
The Animation feature is implemented whenever there is a need in the postprocessor. You are encouraged to use this feature to visualize the rotor motor. Select **Play** to start the animation and select **Stop** to end the animation. **Timer** allows you to adjust the speed of the animation. You can even save this animation file for presentation purposes.



Torsional Vibration Mode Shape Mode No.= 1, Frequency = 2617 cpm; 43.62 Hz



Chapter 7 Example 1: Nonlinear Transient Analysis only DyRoBeS Example - Lateral Vibration - Lund's Rotor ASME Journal of Engineering for Indurtry May 1974, pp.509-517 Rotor Speed = 9200 rpm



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