MADYN 2000 Versions 3.1 to 3.3

This document describes the new features in MADYN 2000 since the introduction of version 3.0, i.e. of versions 3.1 to 3.3.

The main enhancements are the introduction of nonlinear fluid film bearing characteristics (since version 3.1) and the capability to run eigenvalue analyses with **d**ynamic **b**earing **s**upports DBS, where the support characteristics are defined by general coupled transfer functions (since version 3.2). The main purpose of version 3.3 is to ensure the compatibility of MADYN 2000 with the operating system WINDOWS VISTA.

In order to use the transient analysis with nonlinear fluid film bearings an extended licence is necessary.

1. Nonlinear Fluid Film Bearing Characteristics

Nonlinear analyses considering the nonlinear fluid film characteristics are of interest in the following cases:

- The linear system is unstable
- The bearings are highly dynamically loaded

In case of a linearly unstable system the limit cycle of the vibration can be of interest. The limit cycle is characterised by a condition where dynamic forces are in equilibrium. Normally such equilibrium exists at a high vibration level. In some cases the limit cycle vibration can be limited to an acceptable level by design measures. For an assessment it is necessary to calculate it.

High dynamic loads exceeding the linear characteristics of a bearing can be caused by large unbalances (due to a blade loss for example). They can also occur in gear bearings at start up or shut down due to high torque fluctuations.

In the following it is described how to create the nonlinear fluid film bearing characteristic and how to start a transient analysis considering it.

Furthermore the results of three examples are shown, where nonlinear characteristics play an important role:

1.) A simple shaft with cylindrical fluid film bearings, which becomes linearly unstable
2.) A steam turbine with tilting pad bearings, which is subject to a sudden high unbalance
3.) A turbo charger with floating ring bearings, which is linearly unstable

Two general remarks concerning nonlinear analyses:

- They require considerably more analysis time than linear analyses.
- They are numerically more complex and sensitive than linear analyses.

Although proven methods are used to solve the nonlinear system equations (MATLAB Runge Kutta Solvers, bimodal reduction of the system equations) further improvements may be possible. Thanks to the implementation of nonlinear analyses in the user friendly environment of MADYN 2000 it is possible to easily set them up, hence they will be done much more frequent with a variety of systems. Experience will thus grow faster than in the past and will be used for further development.

1.1 Creation of the Nonlinear Fluid Film Bearing Characteristic and Start of a Transient Analysis

The GUI including the controls to create the nonlinear bearing characteristic is shown in figure 1.1.1. The nonlinear characteristic describe a displacement dependent force (by means of the Sommerfeld number “So” and a direction “alpha”) and a damping force (by means of dimensionless damping coefficients “beta”) as a function of the shaft journal position, which is described by a relative displacement “eps” in relation to the clearance and an angle “gamma” (see figure 1.1.2). The analysis of the characteristic is available for all analysis types (ALP3T_T=c_ad, ALP3T_DIN, ALP3T_T=v_ad). Additionally the option “2-phases” can be activated. It means that 2-phase flow will be considered in the cavitation zones of the bearing. This yields smoother fields for the nonlinear characteristic, which is more favourable for the time integration of the nonlinear system of equations. It is therefore highly recommended to apply this option.

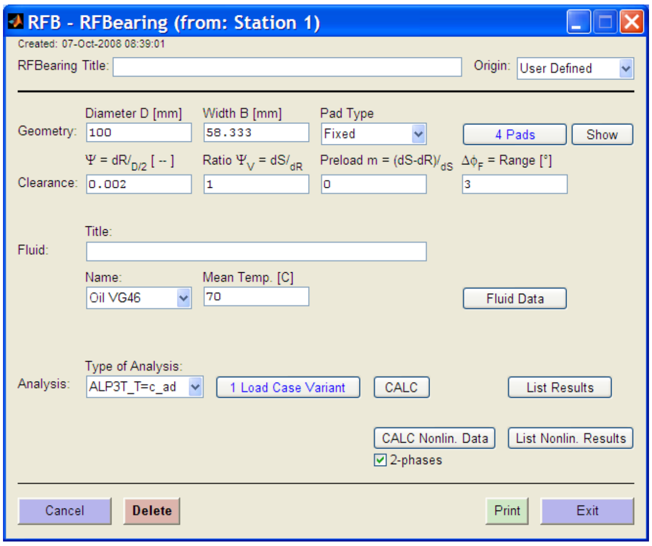
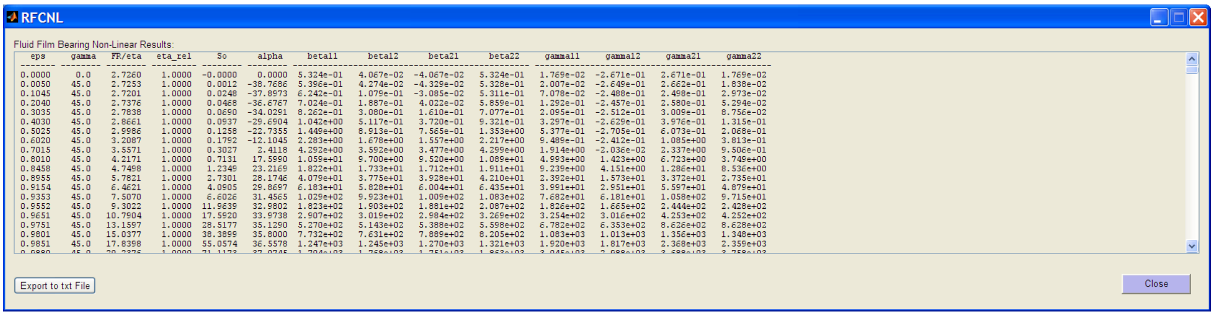


Figure 1.1.1: GUI for fluid film bearings with the new buttons to create the nonlinear characteristic



eps	gamma	FR/eta	eta_rel	So	alpha	beta11	beta12	beta21	beta22	gamma11	gamma12	gamma21	gamma22
0.0000	0.0	2.7260	1.0000	-0.0000	0.0000	5.324e-01	4.067e-02	-4.067e-02	5.324e-01	1.769e-02	-2.671e-01	2.671e-01	1.769e-02
0.0050	45.0	2.7253	1.0000	0.0012	-38.7686	5.396e-01	4.274e-02	-4.329e-02	5.328e-01	2.007e-02	-2.649e-01	2.662e-01	1.838e-02
0.1045	45.0	2.7201	1.0000	0.0248	-37.8973	6.242e-01	1.079e-01	-3.085e-02	5.311e-01	7.078e-02	-2.488e-01	2.498e-01	2.973e-02
0.2040	45.0	2.7376	1.0000	0.0468	-36.6767	7.024e-01	1.887e-01	4.022e-02	5.859e-01	1.292e-01	-2.457e-01	2.580e-01	5.294e-02
0.3035	45.0	2.7838	1.0000	0.0690	-34.0291	8.262e-01	3.080e-01	1.610e-01	7.077e-01	2.095e-01	-2.512e-01	3.009e-01	8.756e-02
0.4030	45.0	2.8661	1.0000	0.0937	-29.6904	1.042e+00	5.117e-01	3.720e-01	9.321e-01	3.297e-01	-2.629e-01	3.976e-01	1.315e-01
0.5025	45.0	2.9986	1.0000	0.1258	-22.7355	1.449e+00	8.913e-01	7.565e-01	1.353e+00	5.377e-01	-2.705e-01	6.073e-01	2.068e-01
0.6020	45.0	3.2087	1.0000	0.1792	-12.1045	2.283e+00	1.678e+00	1.557e+00	2.217e+00	9.489e-01	-2.412e-01	1.085e+00	3.813e-01
0.7015	45.0	3.5571	1.0000	0.3027	2.4118	4.292e+00	3.592e+00	3.477e+00	4.299e+00	1.914e+00	-2.036e-02	2.337e+00	9.506e-01
0.8010	45.0	4.2171	1.0000	0.7131	17.5990	1.059e+01	9.700e+00	9.520e+00	1.089e+01	4.993e+00	1.423e+00	6.723e+00	3.749e+00
0.8458	45.0	4.7498	1.0000	1.2349	23.2169	1.822e+01	1.733e+01	1.712e+01	1.911e+01	9.239e+00	4.151e+00	1.286e+01	8.536e+00
0.8955	45.0	5.7821	1.0000	2.7301	28.1746	4.079e+01	3.775e+01	3.928e+01	4.210e+01	2.392e+01	1.573e+01	3.372e+01	2.735e+01
0.9154	45.0	6.4621	1.0000	4.0905	29.8697	6.183e+01	5.828e+01	6.004e+01	6.435e+01	3.991e+01	2.951e+01	5.597e+01	4.879e+01
0.9353	45.0	7.5070	1.0000	6.6026	31.4565	1.029e+02	9.923e+01	1.009e+02	1.083e+02	7.682e+01	6.181e+01	1.058e+02	9.715e+01
0.9552	45.0	9.3022	1.0000	11.9639	32.9802	1.823e+02	1.903e+02	1.881e+02	2.087e+02	1.826e+02	1.665e+02	2.444e+02	2.428e+02
0.9651	45.0	10.7904	1.0000	17.5920	33.9738	2.907e+02	3.019e+02	2.984e+02	3.269e+02	3.254e+02	3.016e+02	4.253e+02	4.252e+02
0.9751	45.0	13.1597	1.0000	28.5177	35.1290	5.270e+02	5.143e+02	5.388e+02	5.598e+02	6.782e+02	6.353e+02	8.626e+02	8.628e+02
0.9801	45.0	15.0377	1.0000	38.3899	35.8000	7.732e+02	7.631e+02	7.889e+02	8.205e+02	1.083e+03	1.013e+03	1.356e+03	1.348e+03
0.9851	45.0	17.8398	1.0000	55.0574	36.5578	1.247e+03	1.245e+03	1.270e+03	1.321e+03	1.920e+03	1.817e+03	2.368e+03	2.359e+03

Figure 1.1.2: List of the nonlinear force characteristic

The GUI to start the transient analysis is shown in figure 1.1.3. In case nonlinear characteristics of RFB or FRB exist the check box “Nonlinear RFB/FRB” can be activated. Activation opens a window with a list of all nonlinear bearings, from which those to be considered can be selected.

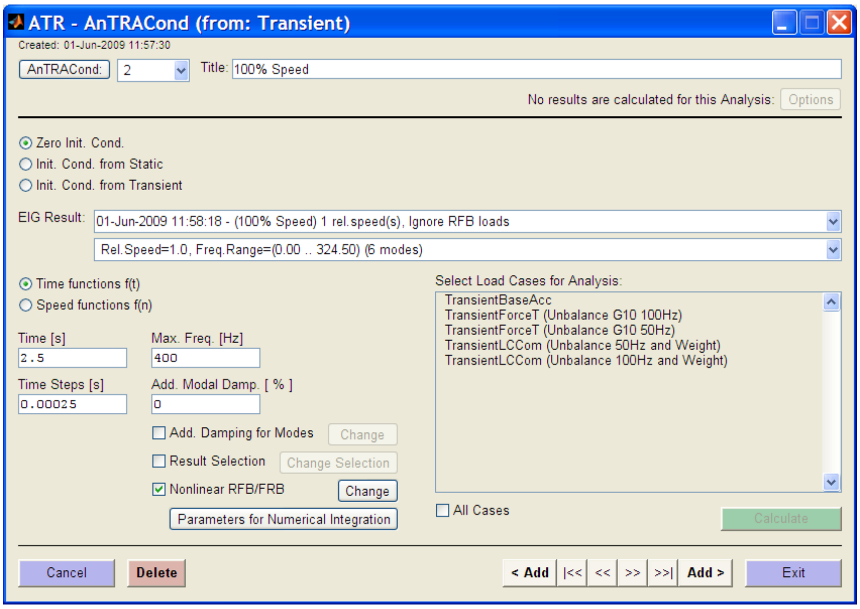


Figure 1.1.3: GUI to start transient analyses including new controls for the nonlinear RFB/FRB

The new button “Parameters for Numerical integration” opens the window in figure 1.1.4 for the definition of parameters for the integration of the nonlinear system.

Two different solvers can be selected. The standard solver is the MATLAB solver ODE45. The alternative solver ODE23 is less sensitive to jumps in the derivatives of the function describing the nonlinearity.

In case the option “Store Intermediate Results During Integration” is selected, a maximum analysis time for the integration can be specified. This is useful, since the analysis time fore the nonlinear integration can be considerable.

Since curves of the damping coefficients versus displacement and angle can be quite rough at high relative displacements, a value for the displacement can be defined, after which the coefficients are linearly extrapolated. This may smooth the coefficients, but also distorts the characteristic to some extent. This option therefore must be used with care.

Please also refer to the documentation for more details about the parameters.

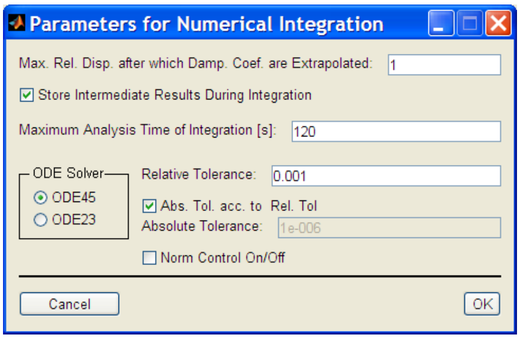


Figure 1.1.4: GUI with parameters for the numerical integration of the nonlinear system

1.2 Behaviour of a Simple Shaft with Cylindrical Bearings

In order to demonstrate limit cycles of an unstable system the behaviour of a simple shaft on cylindrical bearings is studied. The shaft is shown in figure 1.2.1, the bearing in figure 1.2.2. The oil quality is VG46 with a constant mean temperature of 70^{o}C.

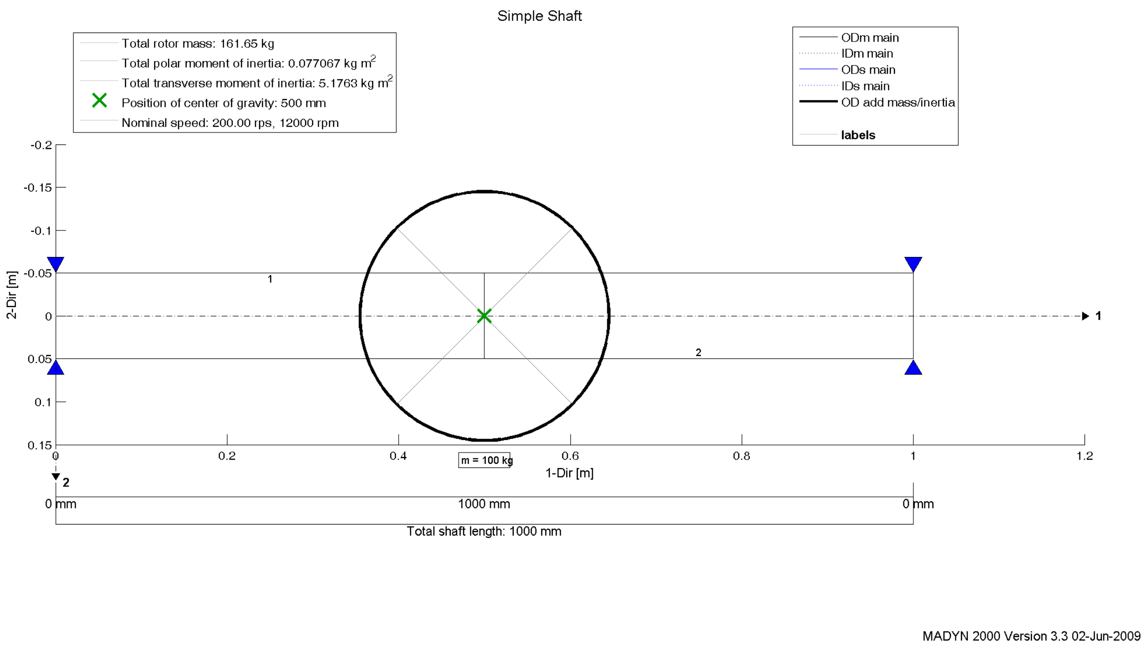


Figure 1.2.1: Simple shaft

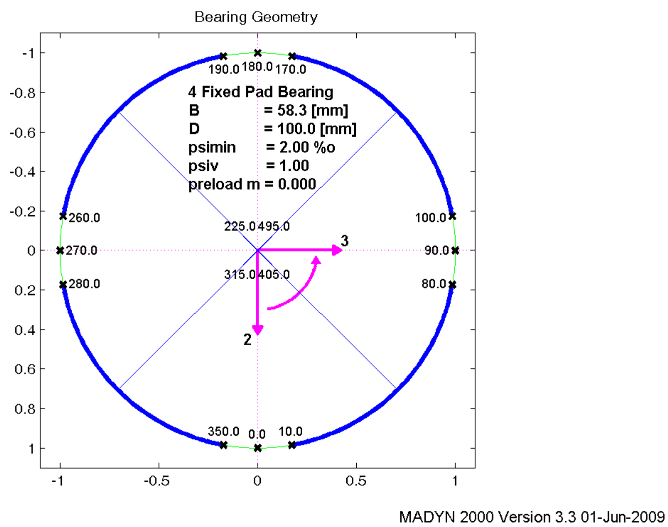


Figure 1.2.2: Bearing geometry

The Campbell diagram in figure 1.2.3 proves that the system is unstable above a speed of approximately 6’300rpm. The corresponding mode shapes at 12’000rpm (100%) are shown in figure 1.2.4. At the speed of the stability threshold the natural frequency is approximately 50% of the speed, i.e. corresponds to the frequency of the whirling oil in the lightly loaded cylindrical bearing. This is in accordance to the well known oil whirl phenomenon.

The transient response to the weight load with zero initial conditions has been calculated for the speeds of 6’000rpm (50%), 8’400rpm (70%), 12’000rpm (100%) and 15’600rpm (130%). At 50% speed the system is still stable, i.e. the vibration should stabilise at the static deformation. At the other speeds the system is unstable and should stabilise at a dynamic limit cycle.

The response at 50% speed is shown in figure 1.2.5. It proves the expected behaviour. The response at the other three speeds is shown in the figures 1.2.6 to 1.2.8 together with the orbits. The frequency and size of the limit cycle can clearly be recognised. The orbit radius increases from 25μm peak-peak to 100μm peak-peak and the frequency from 51 to 83 Hz with rising speed from 70% to 130%. Note, that further increasing the speed would further increase the frequency until it would stabilise at the frequency of the rigidly supported shaft (oil whip), which is at 96Hz for this example.

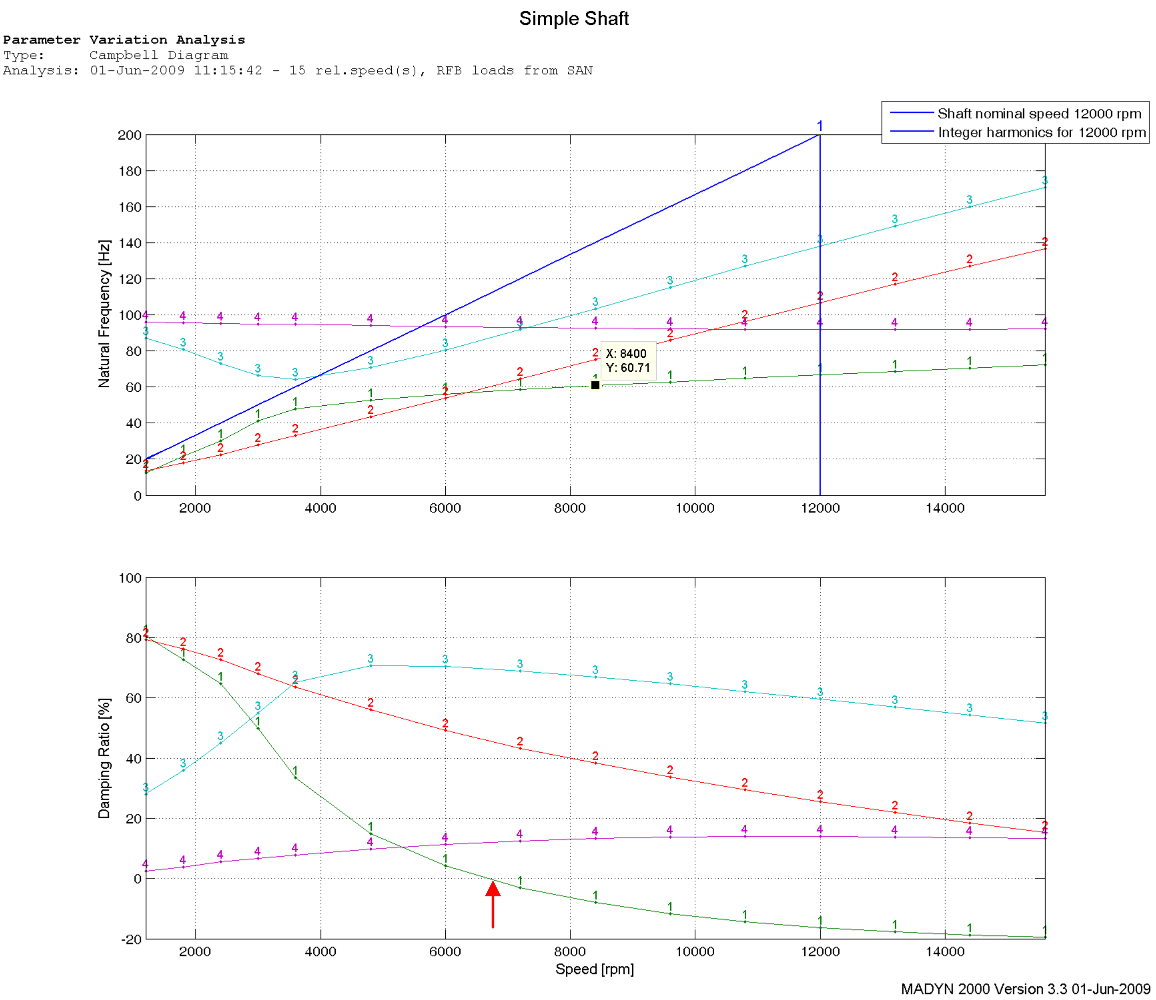


Figure 1.2.3: Campbell diagram of the simple shaft

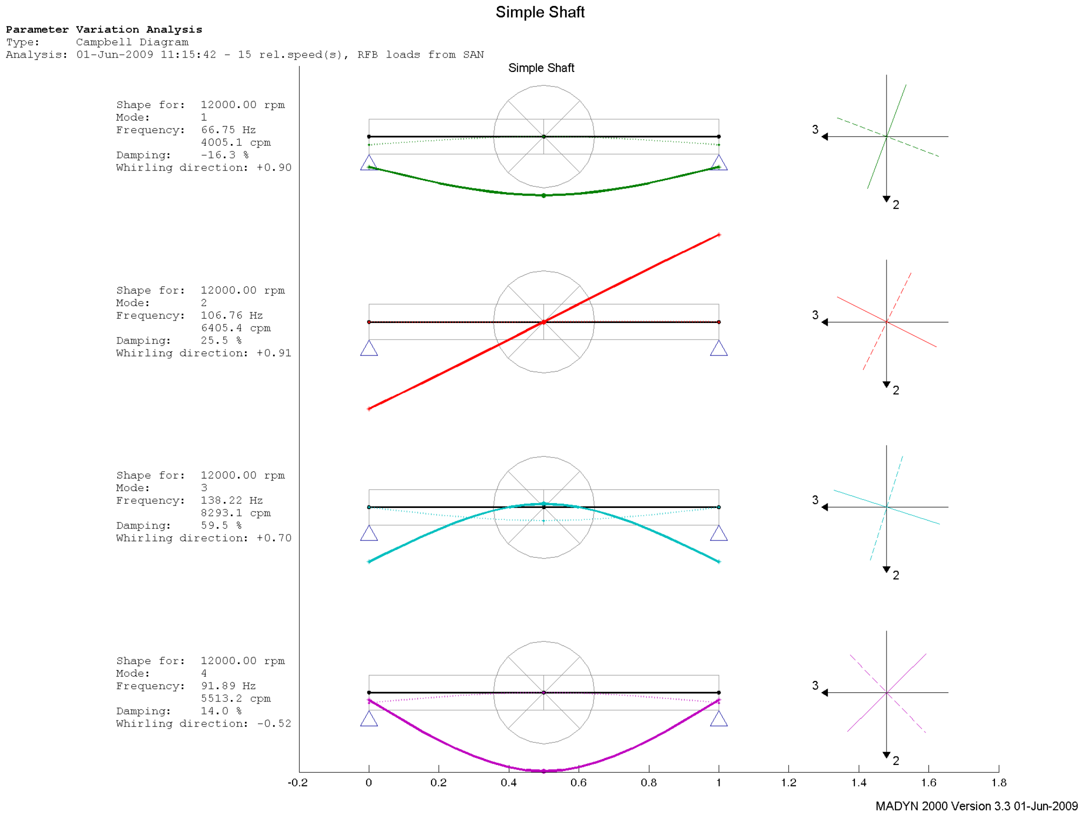


Figure 1.2.4: Mode shapes at 12'000rpm

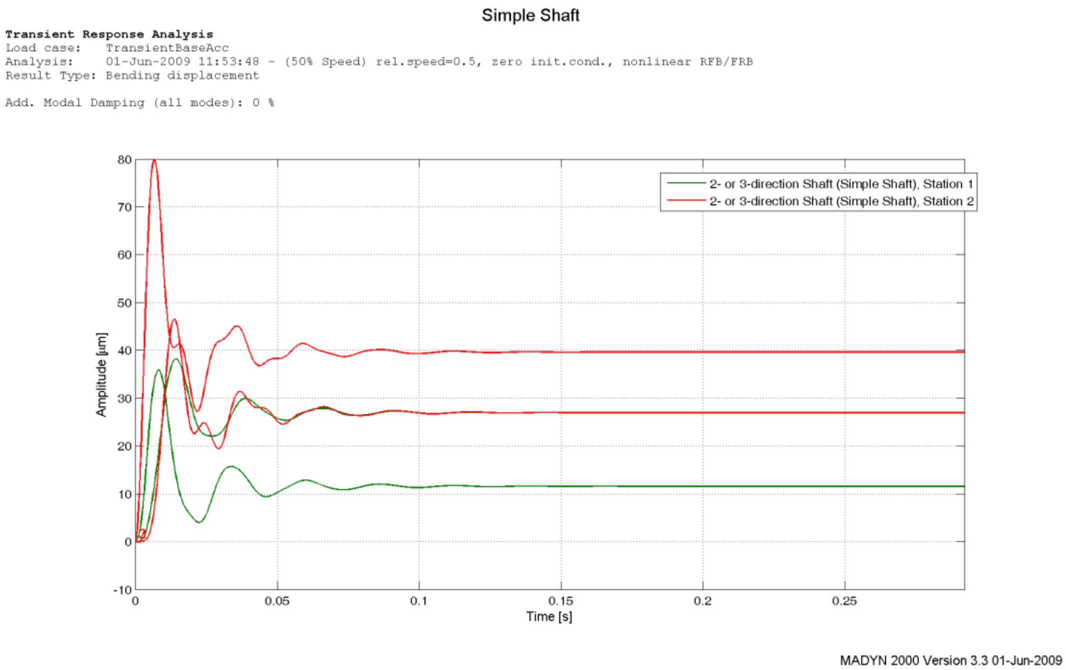
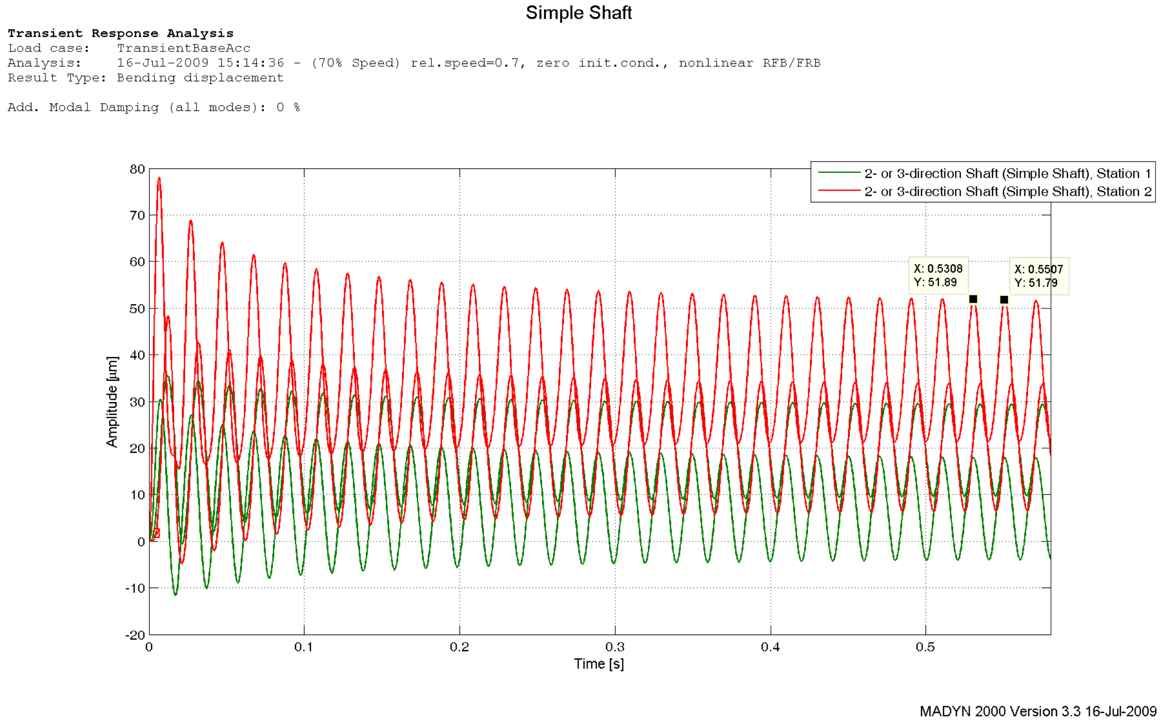


Figure 1.2.5: Transient response at 50% speed



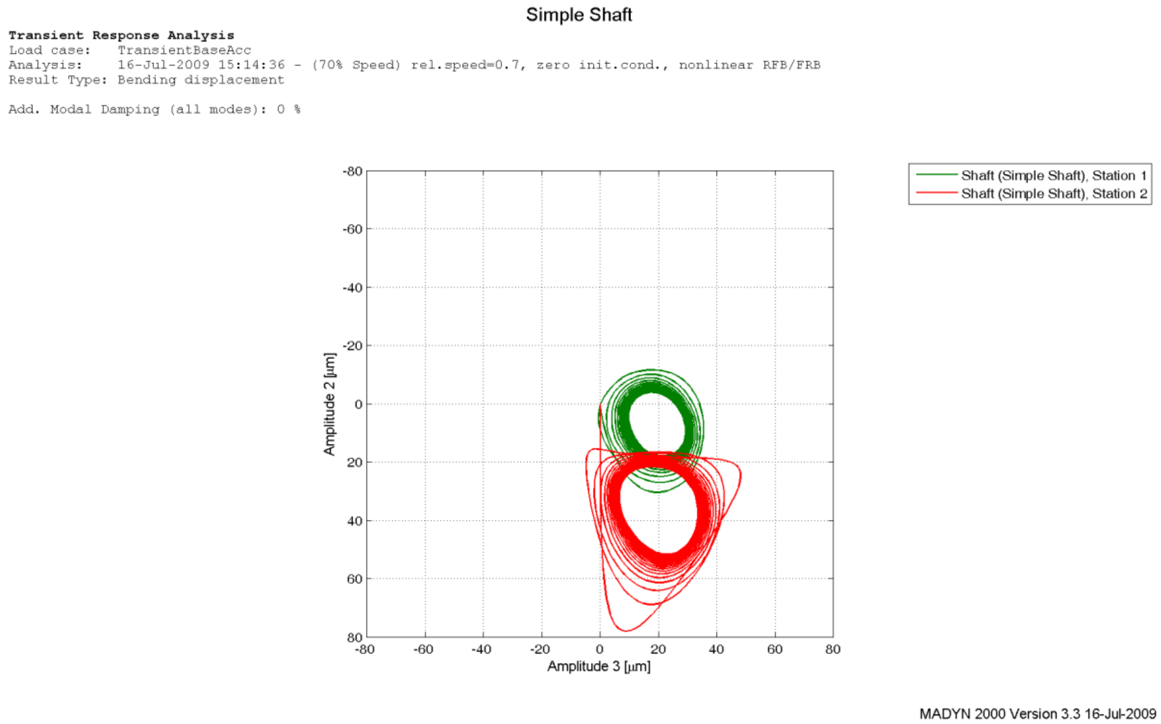
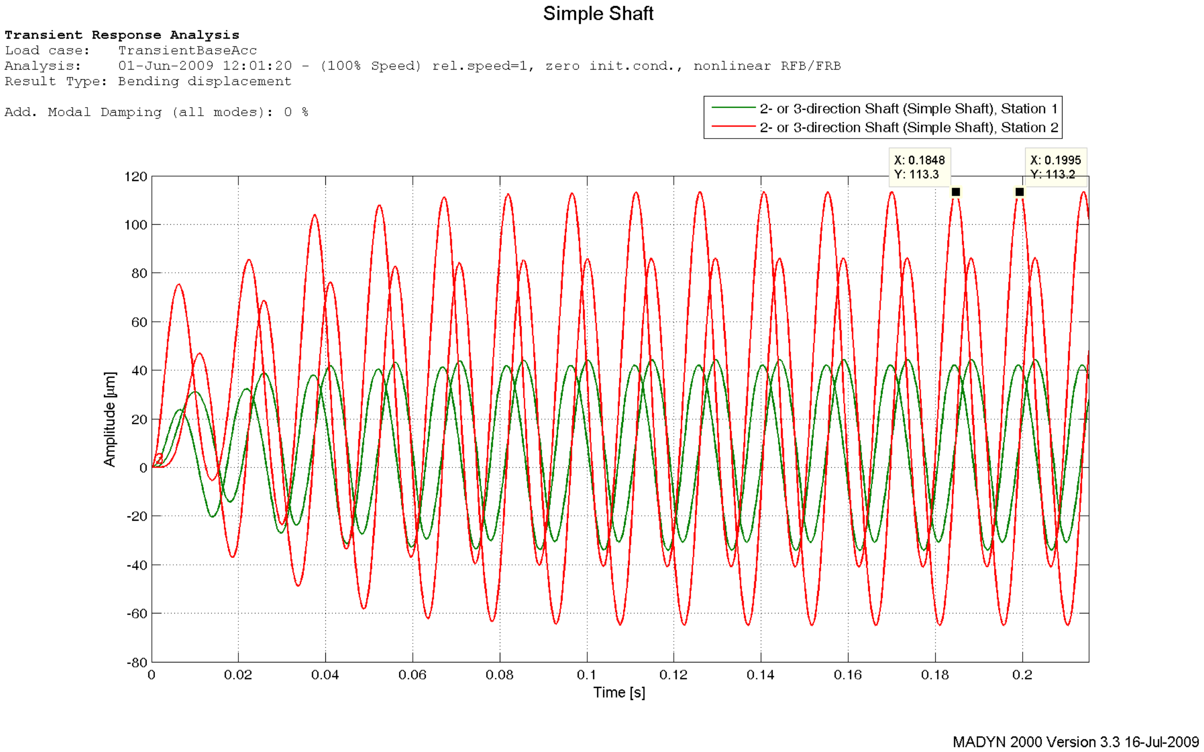


Figure 1.2.6: Transient response at 70% speed → limit cycle with a frequency of 51Hz



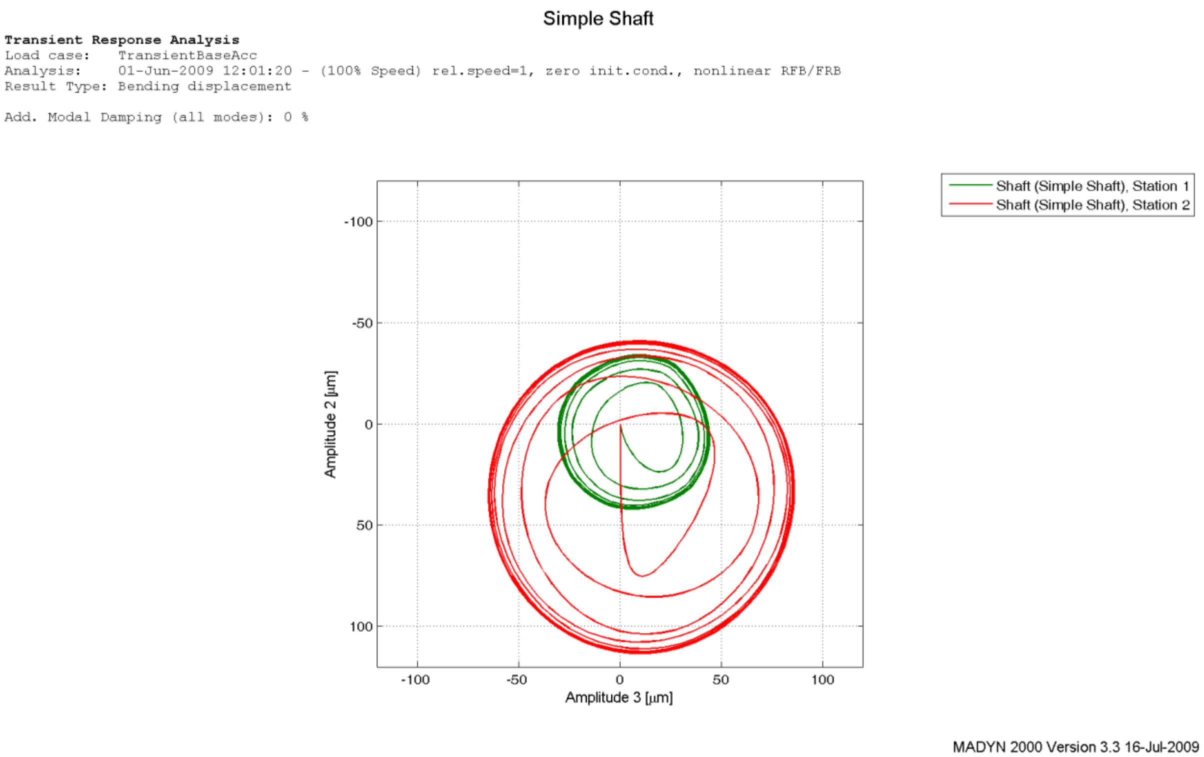
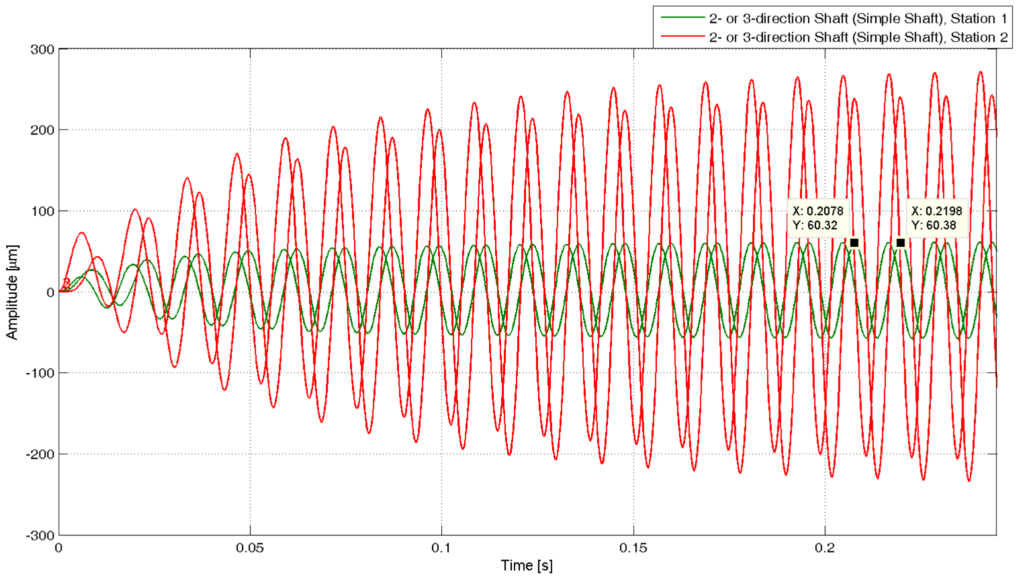


Figure 1.2.7: Transient response at 100% speed → limit cycle with a frequency of 68Hz

Simple Shaft

Transient Response Analysis
Load case: TransientBaseAcc
Analysis: 01-Jun-2009 12:09:42 - (130% Speed) rel.speed=1.3, zero init.cond., nonlinear RFB/FRB
Result Type: Bending displacement

Add. Modal Damping (all modes): 0 %

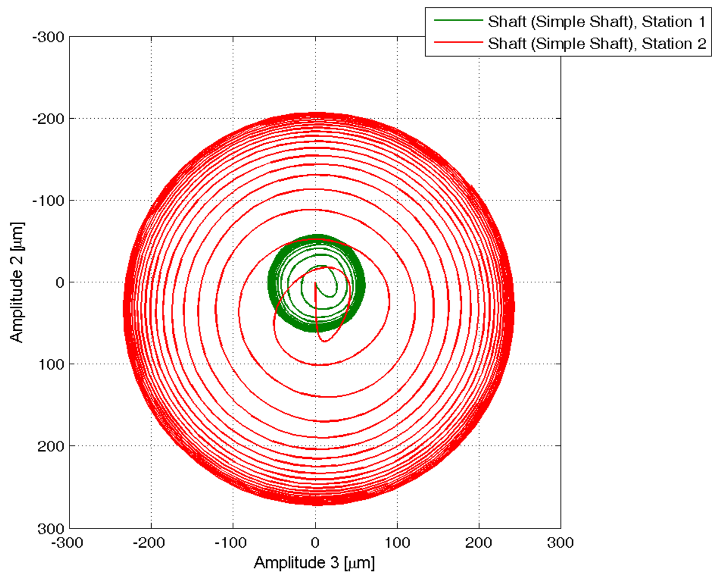


MADYN 2000 Version 3.3 01-Jun-2009

Simple Shaft

Transient Response Analysis
Load case: TransientBaseAcc
Analysis: 01-Jun-2009 12:09:42 - (130% Speed) rel.speed=1.3, zero init.cond., nonlinear RFB/FRB
Result Type: Bending displacement

Add. Modal Damping (all modes): 0 %



MADYN 2000 Version 3.3 01-Jun-2009

Figure 1.2.8: Transient response at 130% speed → limit cycle with a frequency of 83Hz

1.3 Behaviour of a Steam Turbine with Tilting Pad Bearings Subject to a Sudden High Unbalance

In the following the transient response of a steam turbine to a sudden high unbalance is studied. The system is shown in figure 1.3.1. The amplitudes of the unbalance force at a speed of 7’400rpm in direction 2 and 3 can be seen in figure 1.3.2. The shaft response in the bearings can be seen in figure 1.3.3. The analysis starts with initial conditions according to the weight deformation. The unbalance force is applied at the instant 0.1s. The response time history clearly contains higher harmonics of the fundamental frequency corresponding to the speed of 7’400rpm (123Hz).

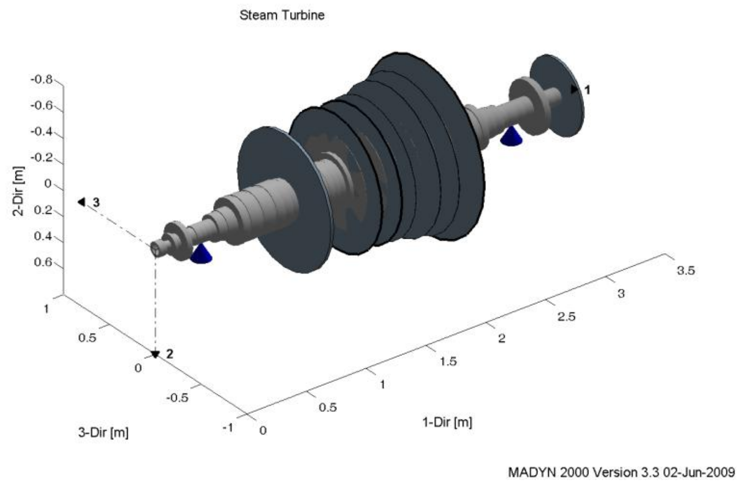


Figure 1.3.1: Steam turbine

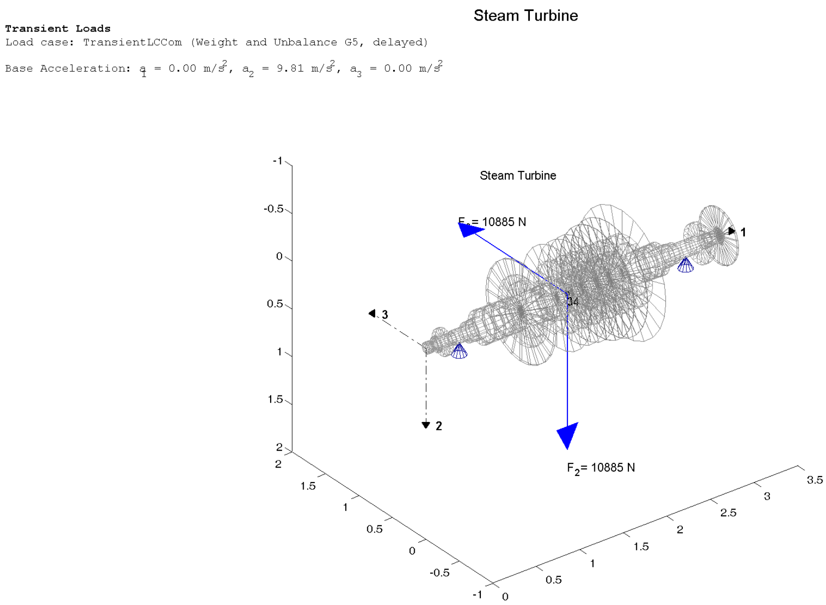
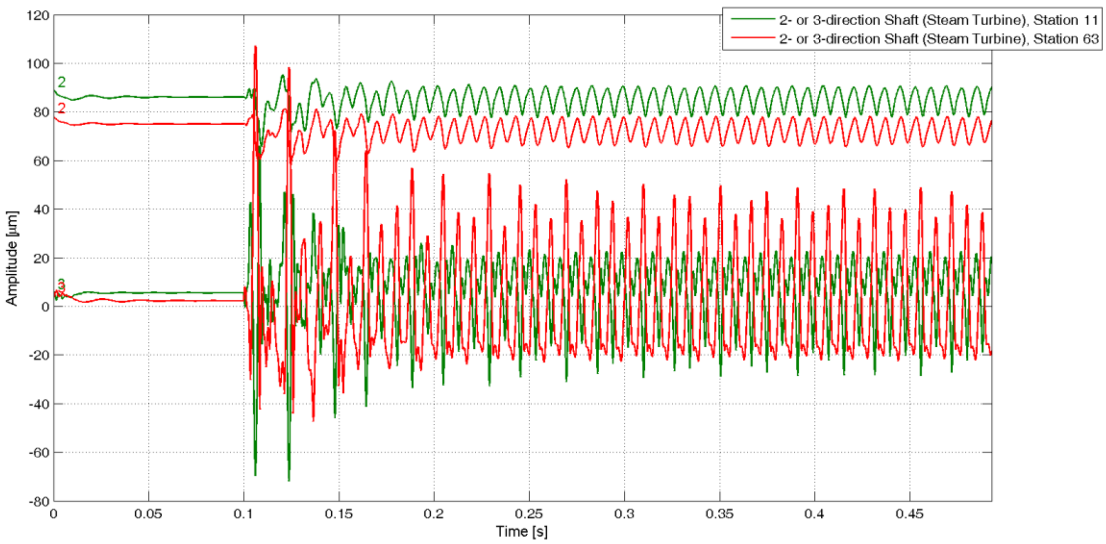


Figure 1.3.2: Unbalance force at 7’400rpm

Steam Turbine

Transient Response Analysis
Load case: TransientLCCom (Weight and Unbalance G5, delayed)
Analysis: 19-May-2009 17:58:44 - (Eig of free sytem 1000Hz, ODE23) rel.speed=1, init.cond. from SAN, nonlinear RFB/FRB
Result Type: Bending displacement

Add. Modal Damping (all modes): 2 %

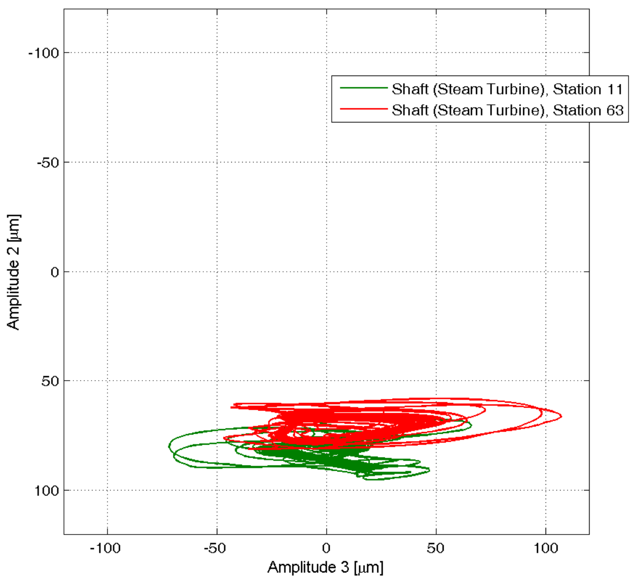


MADYN 2000 Version 3.3 01-Jun-2009

Steam Turbine

Transient Response Analysis
Load case: TransientLCCom (Weight and Unbalance G5, delayed)
Analysis: 19-May-2009 17:58:44 - (Eig of free sytem 1000Hz, ODE23) rel.speed=1, init.cond. from SAN, nonli
Result Type: Bending displacement

Add. Modal Damping (all modes): 2 %



MADYN 2000 Version 3.3 01-Jun-2009

Figure 1.3.3: Response in the bearings due to a sudden unbalance (radial bearing clearance 125µm)

1.4 Behaviour of a Turbocharger with Floating Ring Bearings

The turbocharger model of this example is shown in figure 1.4.1.

Its Campbell diagram can be seen in figure 1.4.2, the mode shapes at 100% speed in enclosure 1.4.3. It can be clearly seen that the system is unstable in the whole shown speed range from 20% to 100% speed. At low speeds mode 3 and 4 are unstable. They are characterised by large deflections of the ring and are caused by the inner oil whirl (whirling of the oil between the shaft and the floating ring). At higher speeds mode 1 and 2 are unstable. These two modes have practically no relative displacement between shaft and ring. Their instability is caused by the outer oil whirl.

The limit cycles at speeds of 20%, 50% and 100% in the form of the shaft vibrations at the bearings, the relative vibrations between shaft and ring (orbits and time histories) can be seen in the figures 1.4.4 to 1.4.9. The inner whirl with higher frequencies can best be seen by the relative vibrations.

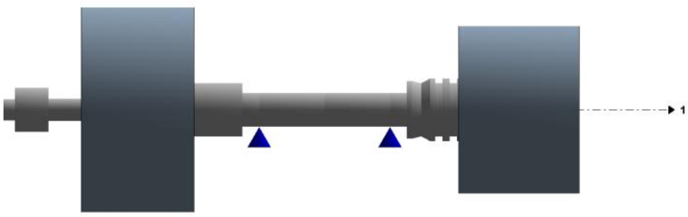


Figure 1.4.1: Model of the turbocharger

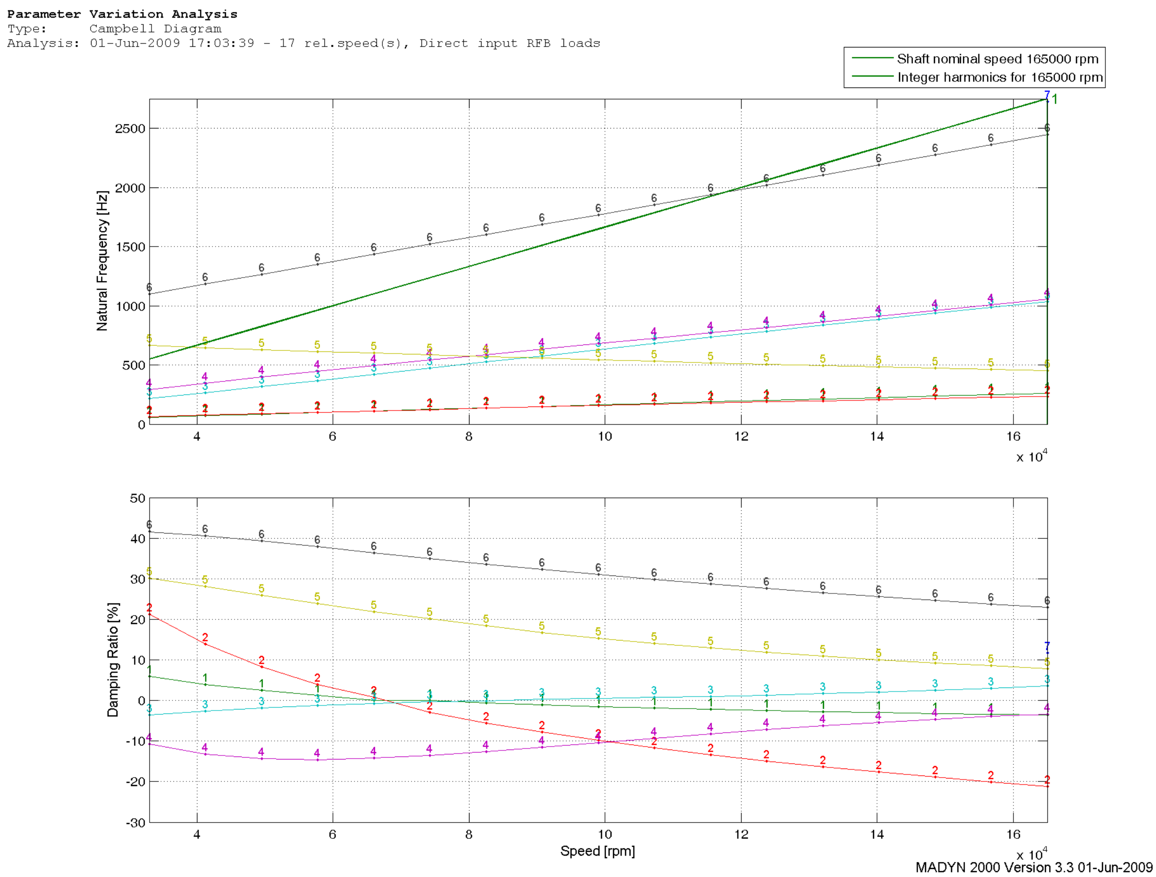


Figure 1.4.2: Campbell diagram

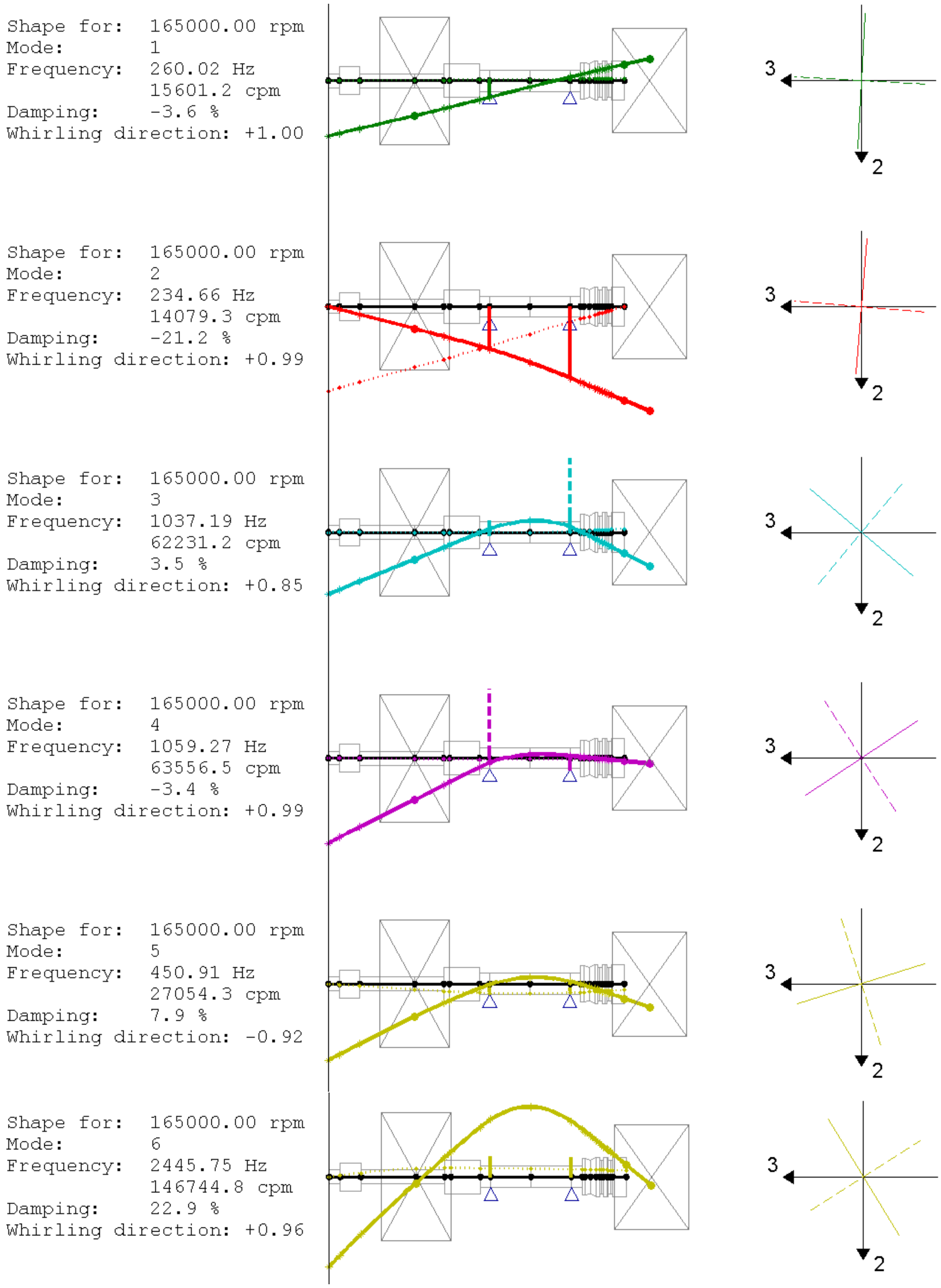
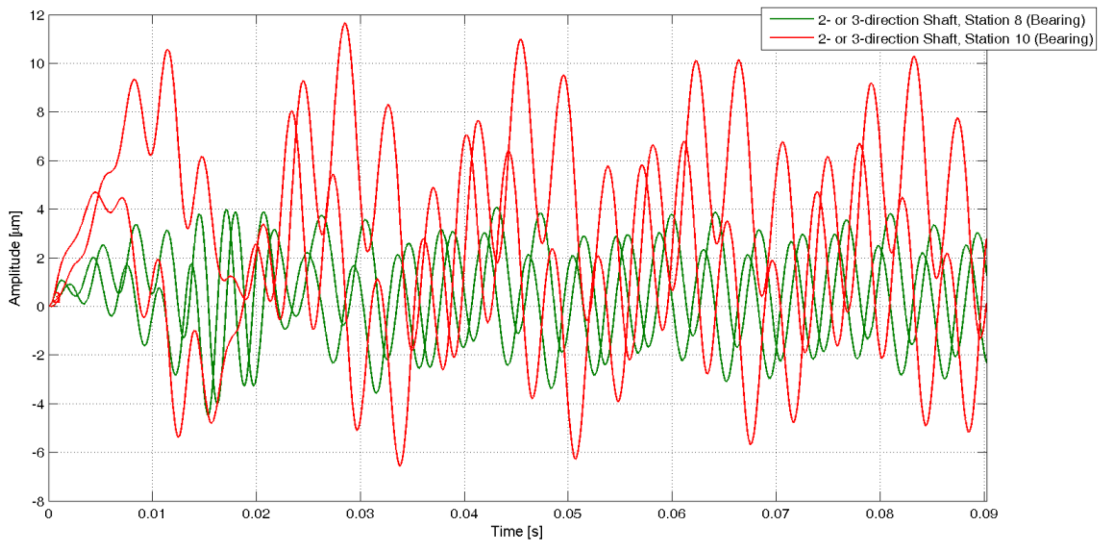


Figure 1.4.3: Campbell diagram

Transient Response Analysis
Load case: TransientBaseAcc (Weight)
Analysis: 28-May-2009 08:07:21 - rel.speed=0.2, zero init.cond., nonlinear RFB/FRB, fluids deact.
Result Type: Bending displacement

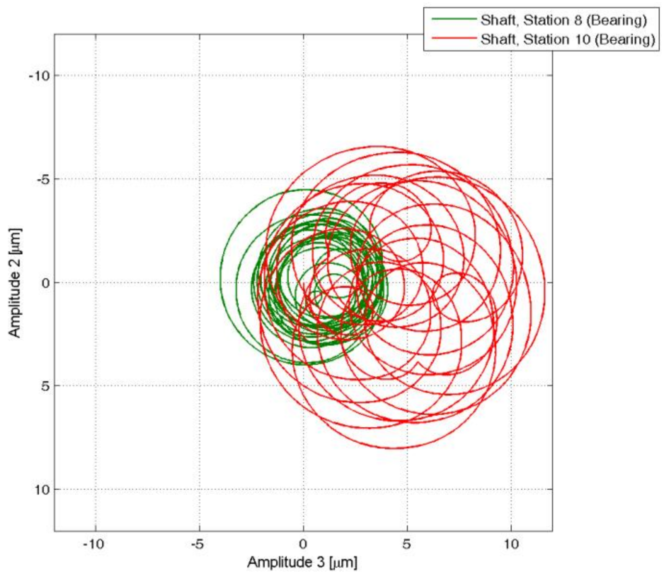
Add. Modal Damping (all modes): 0 %



MADYN 2000 Version 3.3 02-Jun-2009

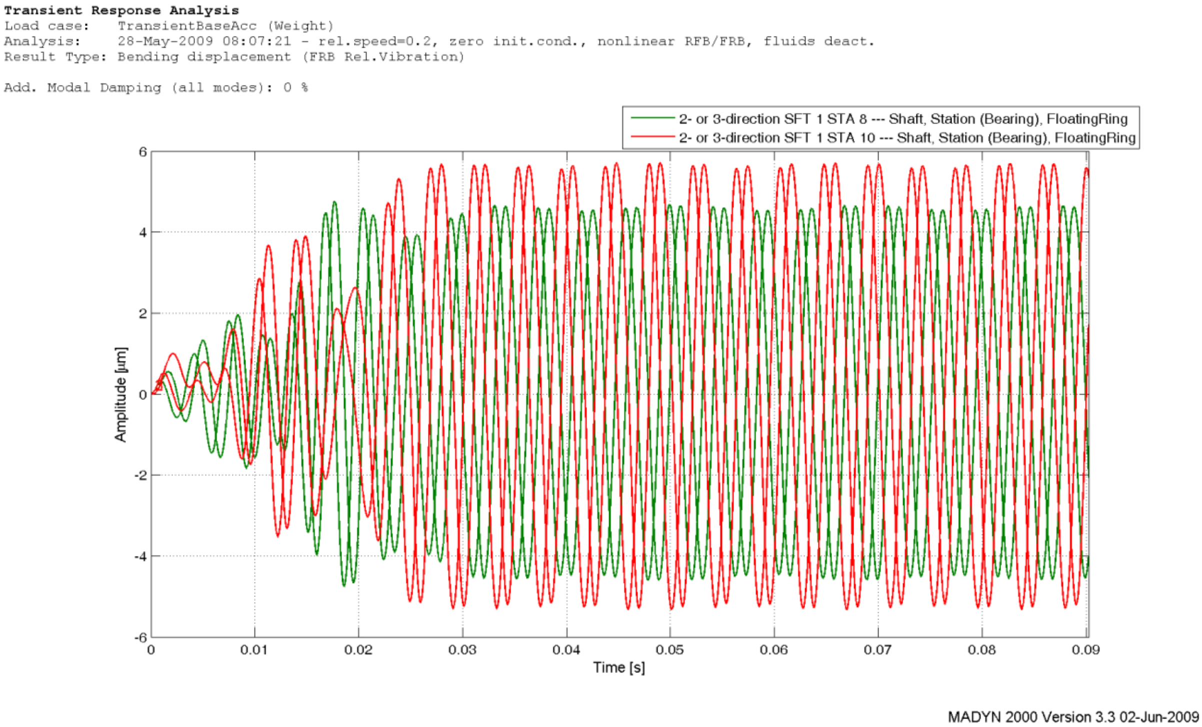
Transient Response Analysis
Load case: TransientBaseAcc (Weight)
Analysis: 28-May-2009 08:07:21 - rel.speed=0.2, zero init.cond., nonlinear RFB/FRB, fluids deact.
Result Type: Bending displacement

Add. Modal Damping (all modes): 0 %



MADYN 2000 Version 3.3 02-Jun-2009

Figure 1.4.4: Limit cycle at 20% speed, shaft vibration at the bearings



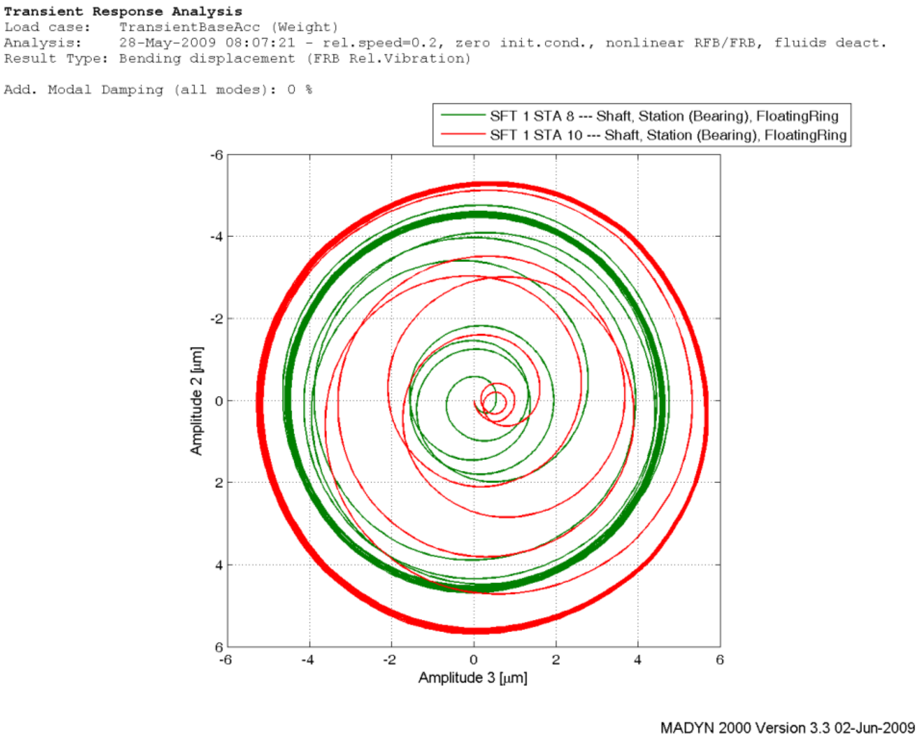
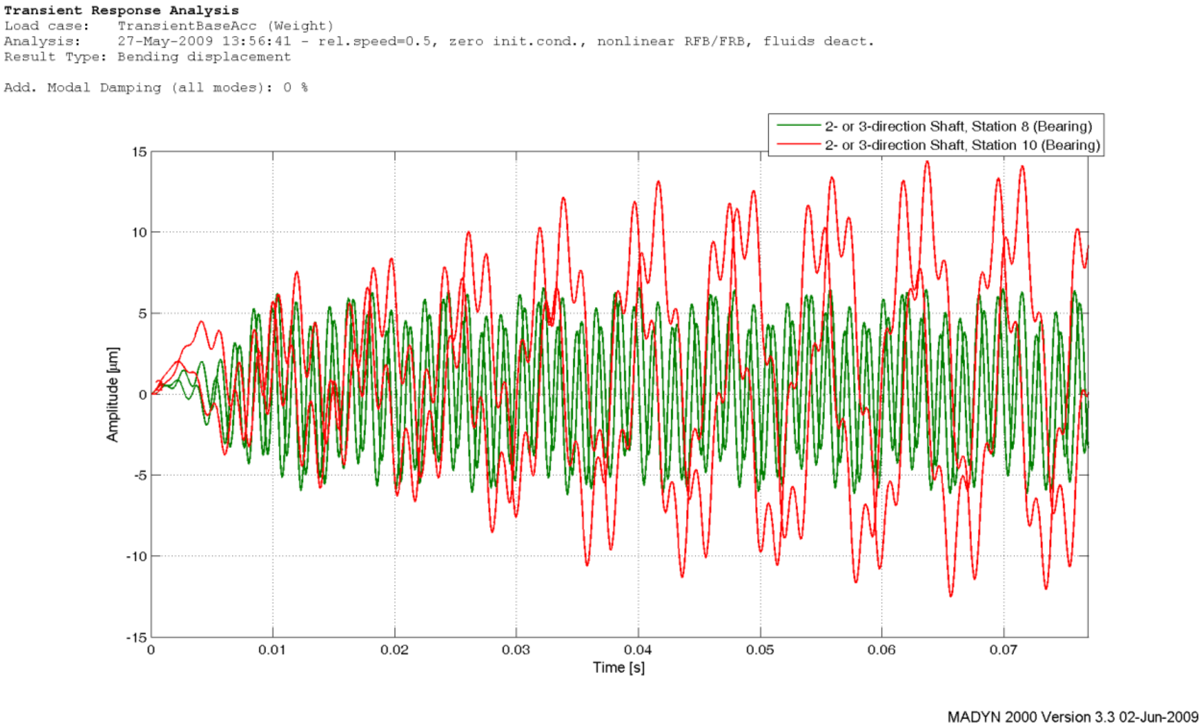


Figure 1.4.5: Limit cycle at 20% speed, relative vibration shaft ring



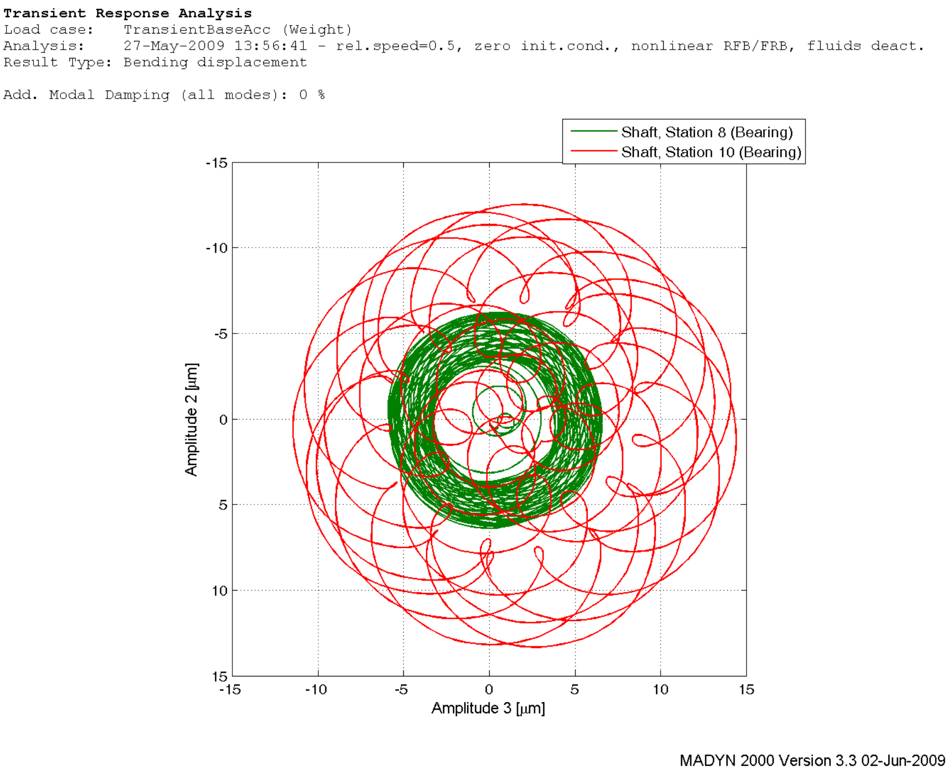
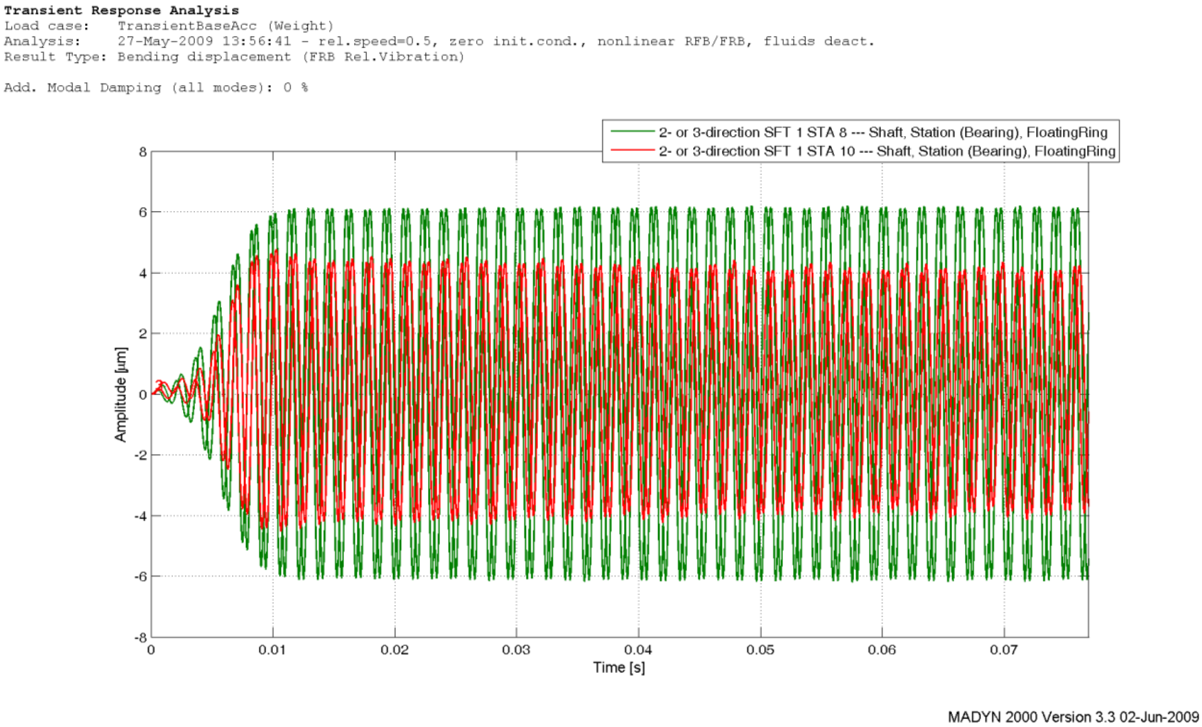


Figure 1.4.6: Limit cycle at 50% speed, shaft vibration at the bearings



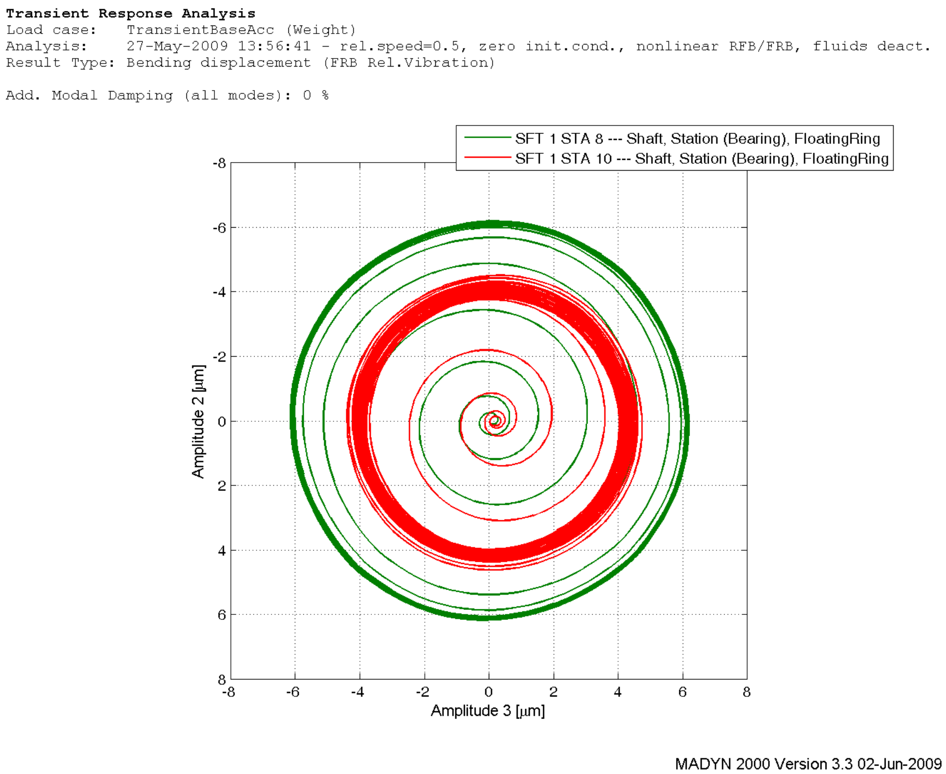
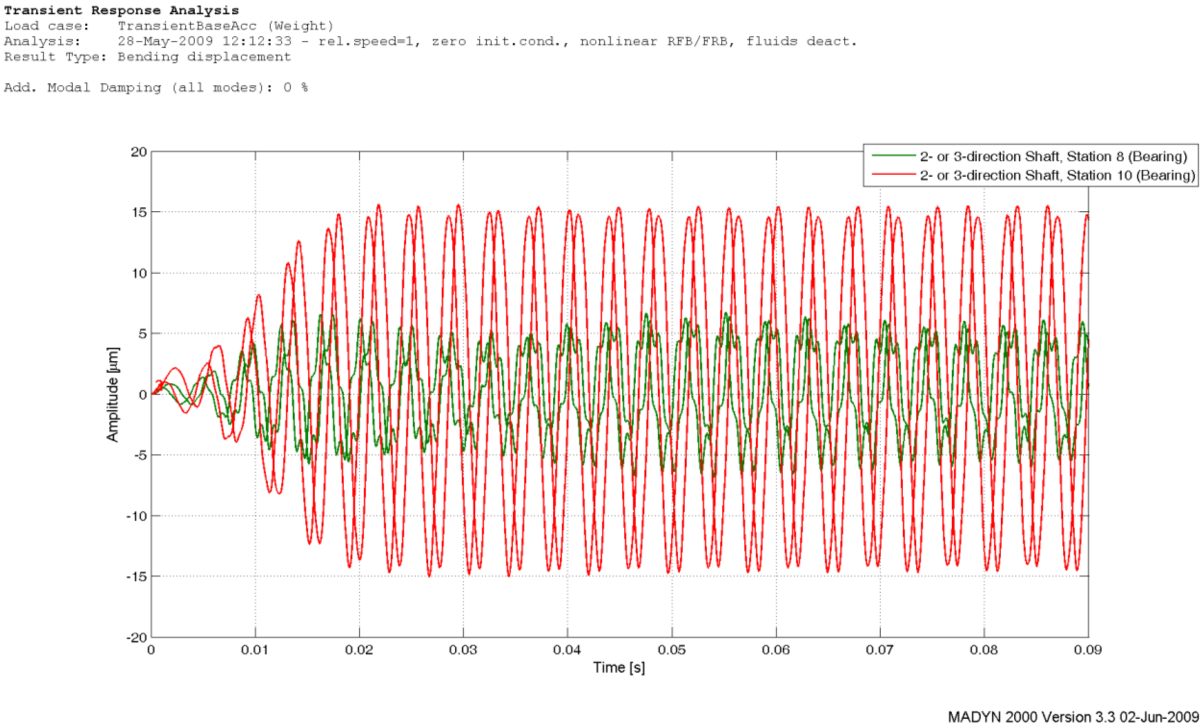


Figure 1.4.7: Limit cycle at 50% speed, relative vibration shaft ring



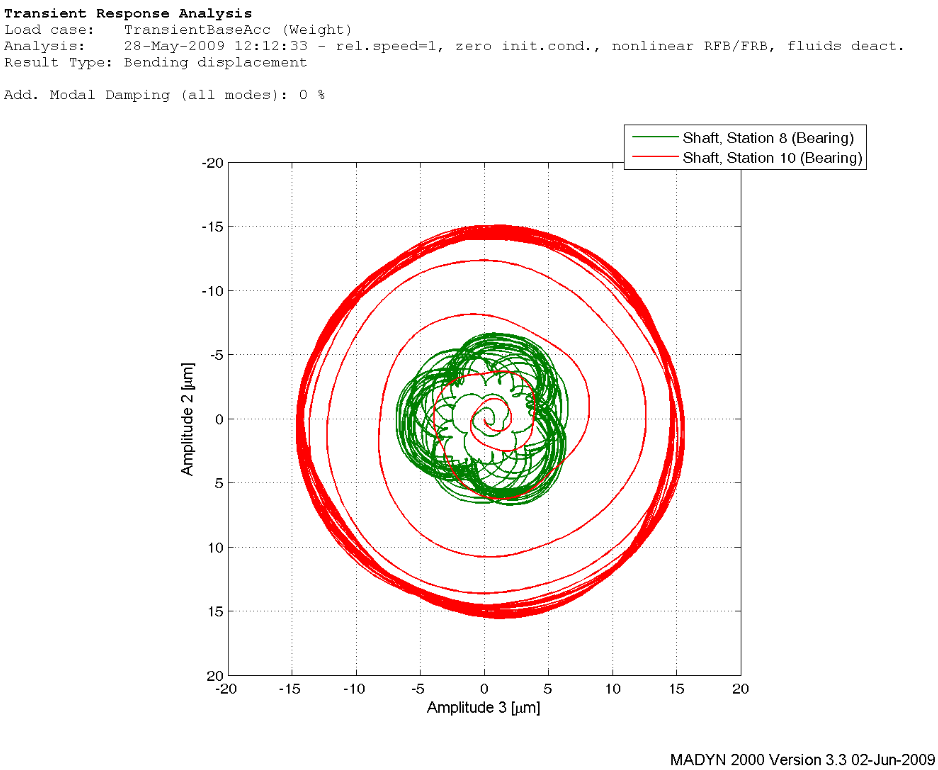
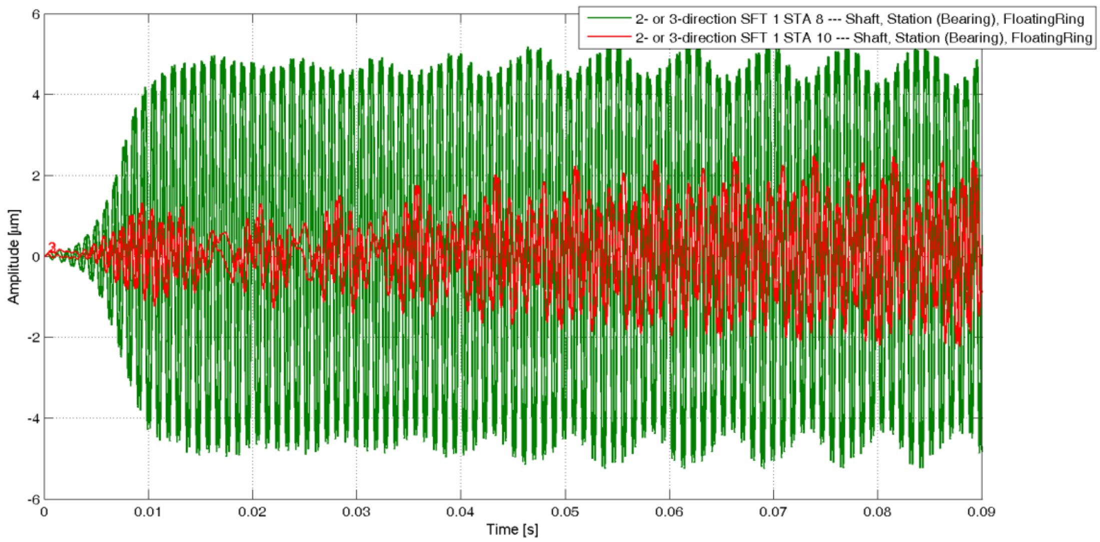


Figure 1.4.8: Limit cycle at 100% speed, shaft vibration at the bearings

Transient Response Analysis
Load case: TransientBaseAcc (Weight)
Analysis: 28-May-2009 12:12:33 - rel.speed=1, zero init.cond., nonlinear RFB/FRB, fluids deact.
Result Type: Bending displacement (FRB Rel.Vibration)

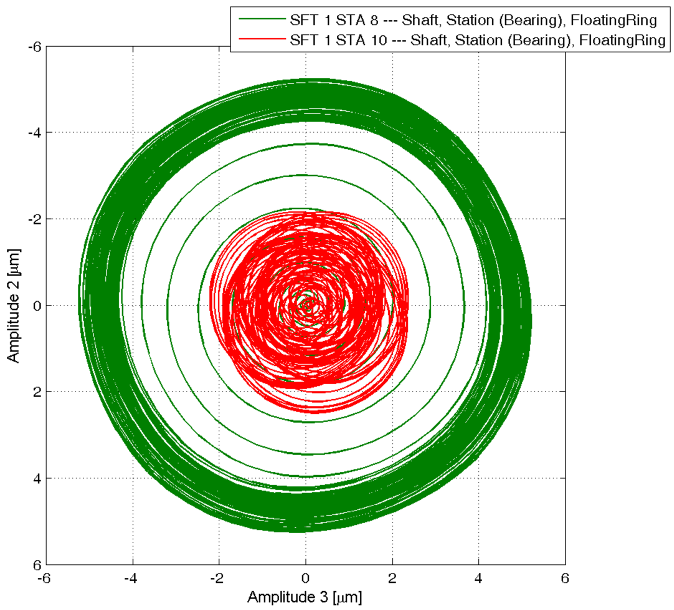
Add. Modal Damping (all modes): 0 %



MADYN 2000 Version 3.3 02-Jun-2009

Transient Response Analysis
Load case: TransientBaseAcc (Weight)
Analysis: 28-May-2009 12:12:33 - rel.speed=1, zero init.cond., nonlinear RFB/FRB, fluids deact.
Result Type: Bending displacement (FRB Rel.Vibration)

Add. Modal Damping (all modes): 0 %



MADYN 2000 Version 3.3 02-Jun-2009

Figure 1.4.9: Limit cycle at 100% speed, relative vibration shaft ring

2. General Dynamic Bearing Supports DBS

General dynamic bearing supports (DBS) are transfer functions (TFUs) characterising the bearing support properties. They are the displacement responses to a harmonic unit force excitation with different frequencies, i.e. dynamic flexibilities. The transfer functions may describe a single direction (excitation and response are at the same location and in the same direction) or may describe couplings (e.g. the response and excitation may have different location or direction). The transfer functions may stem from a general finite element model of a casing or from measurements.

In the following is described how to define DBS as well as the influence of DBS on a system.

2.1 Definition of a DBS

To define DBS in a system, SBS must be present. The SBS can be substituted by the transfer functions of DBS. The GUI to define DBS is invoked by the button “Dyn.Bear.Supports” in the system GUI (see figure 2.1.1). The DBS GUI can be seen in figure 2.1.2. A matrix with the rows and columns corresponding to the locations of SBS described by the shaft and station numbers is shown in the GUI.

The buttons with “+” indicate, that a TFU can be added at this location. Once more than one diagonal TFU is defined, it is also possible to define TFUs for couplings between locations and directions. In the GUI in figure 2.2 the situation is shown, where TFUs on the diagonal are defined in direction 2 at the stations 4 and 100 of shaft 1 as well as the off diagonal TFU for the coupling between station 4 and 100 in direction 2. Since the matrix of dynamic flexibilities for structures is symmetric only elements in the upper triangle can be defined.

The “+” buttons open a window to select a text file with the definition of a transfer function. The file must contain frequencies, amplitudes and phase angles or real and imaginary parts of the transfer function. The content of a file for the variant with amplitude and phase angle is shown below:

```
f              Ampl         Phase
0.0000        0.108231E-08   0.00000
5.4430        0.111723E-08 -0.205860
6.1511        0.115257E-08 -0.886225
6.2806        0.115640E-08  -4.34498
 …
```

Once a TFU is defined in this way, it can be used in harmonic response analyses HAR. For this purpose the check box “Consider DBS” must be activated in the HAR analysis GUI (see figure 2.1.3)

To plot the TFU a menu can be opened with a right mouse click in the TFU field of the DBS (see figure 2.1.4). The transfer functions can be plotted as amplitude and phase or real and imaginary part.

In order to use TFUs for eigenvalue analyses it is necessary to create numerator and denominator polynomials. The respective GUI is opened by the menu item “Edit TFU parameters” of the menu invoked by the right mouse click in the TFU field (see figure 2.1.4). The menu can be seen in figure 2.1.5. It allows defining the polynomial orders and a frequency range for the adaptation of the transfer function. The quality of the adaptation can be checked by comparison of the original and adapted curves (see figure 2.1.6, original blue, adapted red).

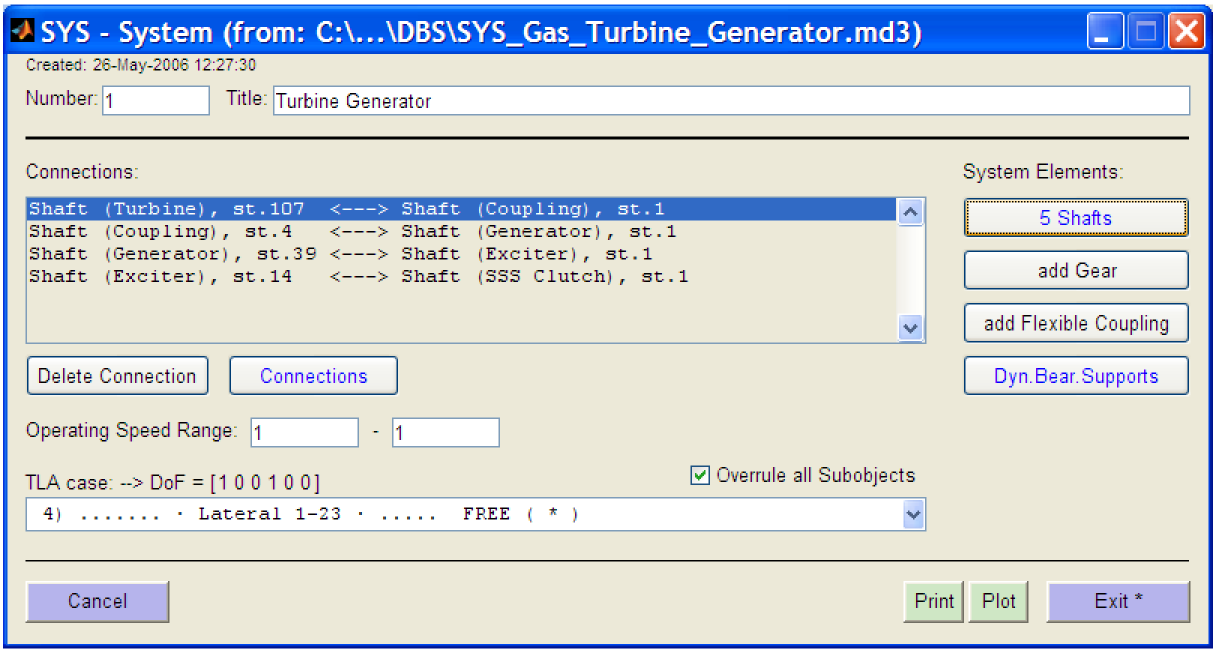


Figure 2.1.1: System GUI with the button to define dynamic bearing support

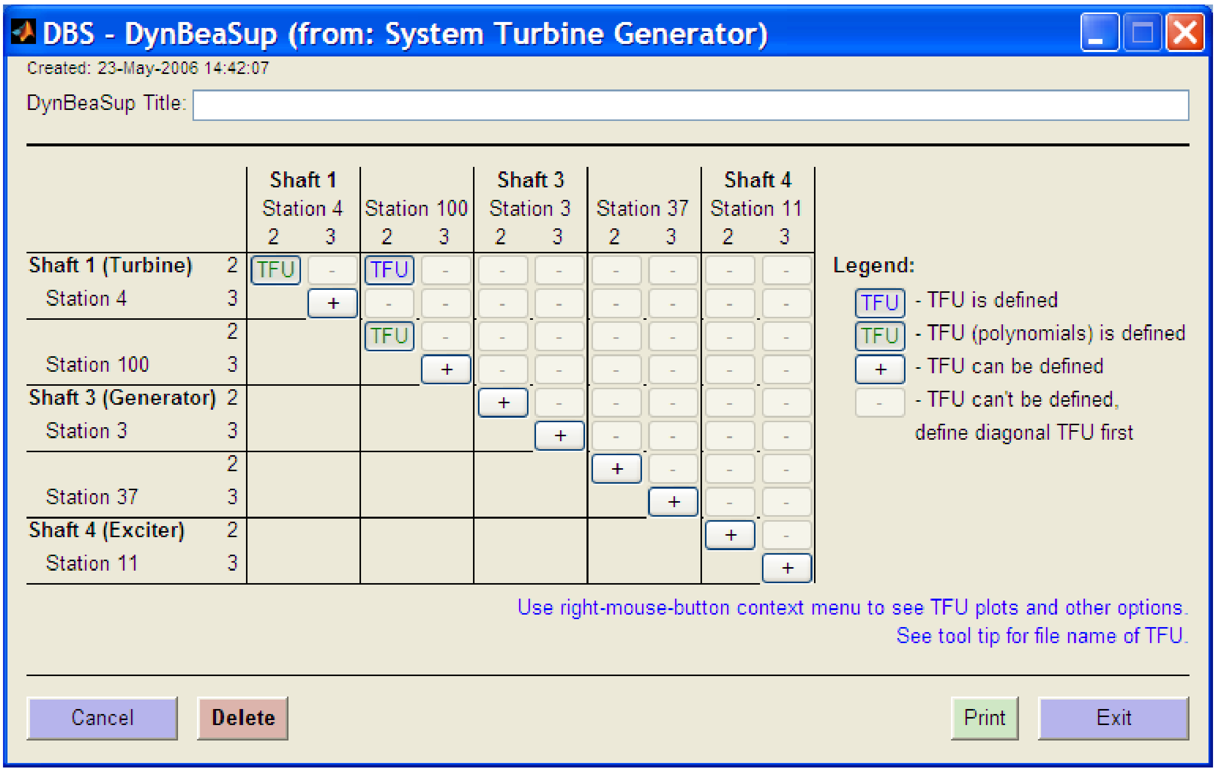


Figure 2.1.2: GUI to define DBS

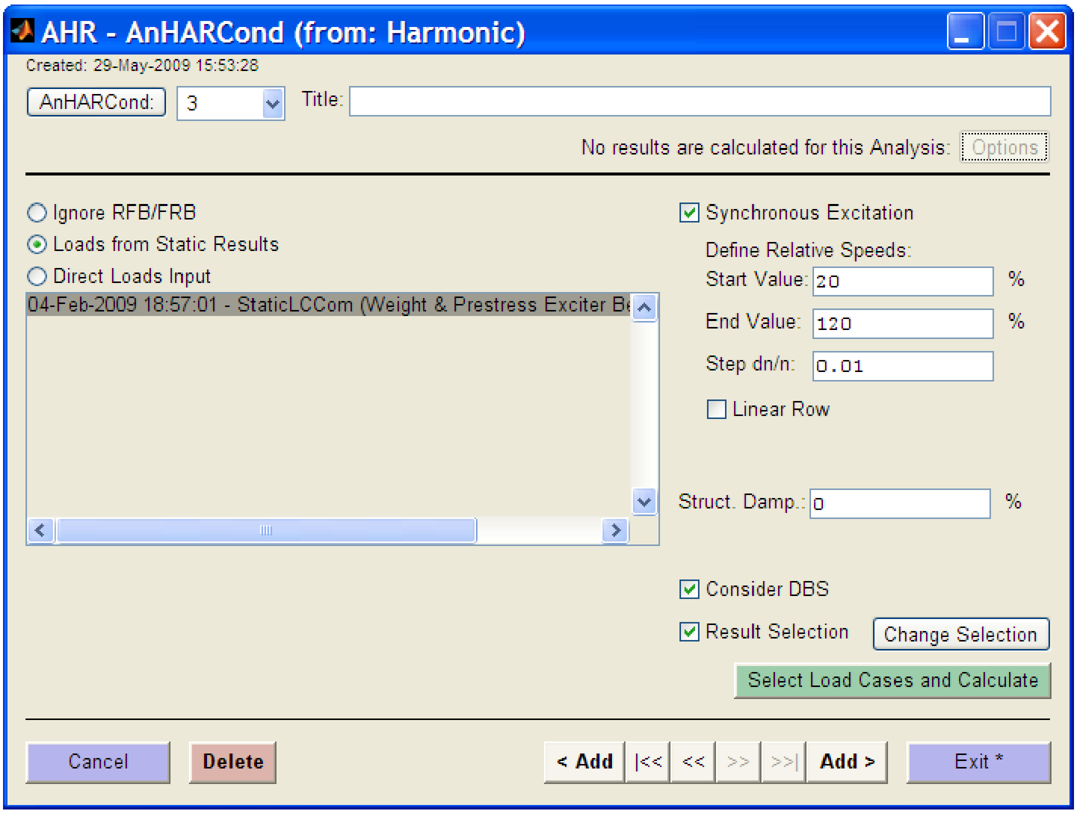


Figure 2.1.3: HAR GUI with check box to consider DBS

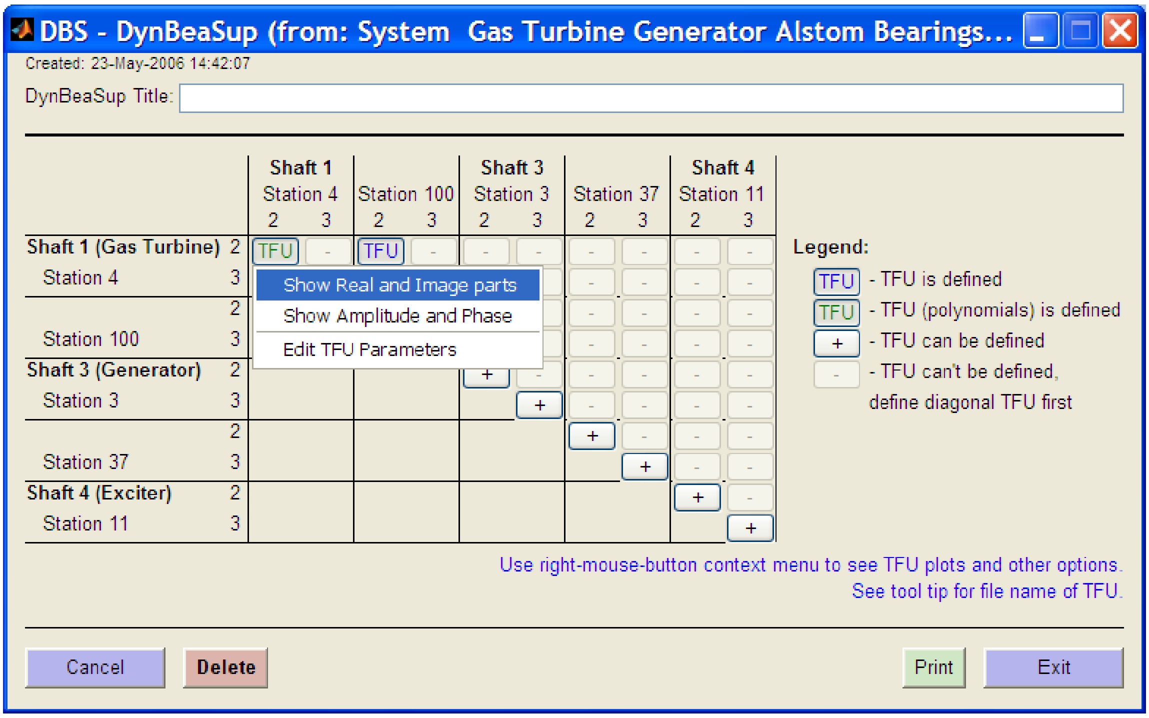


Figure 2.1.4: DBS GUI with the menu of TFU controls

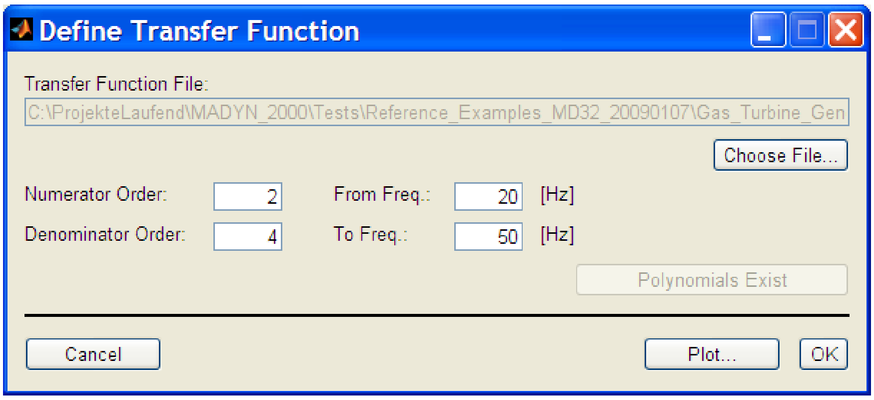


Figure 2.1.5: Menu to define numerator and denominator polynomials

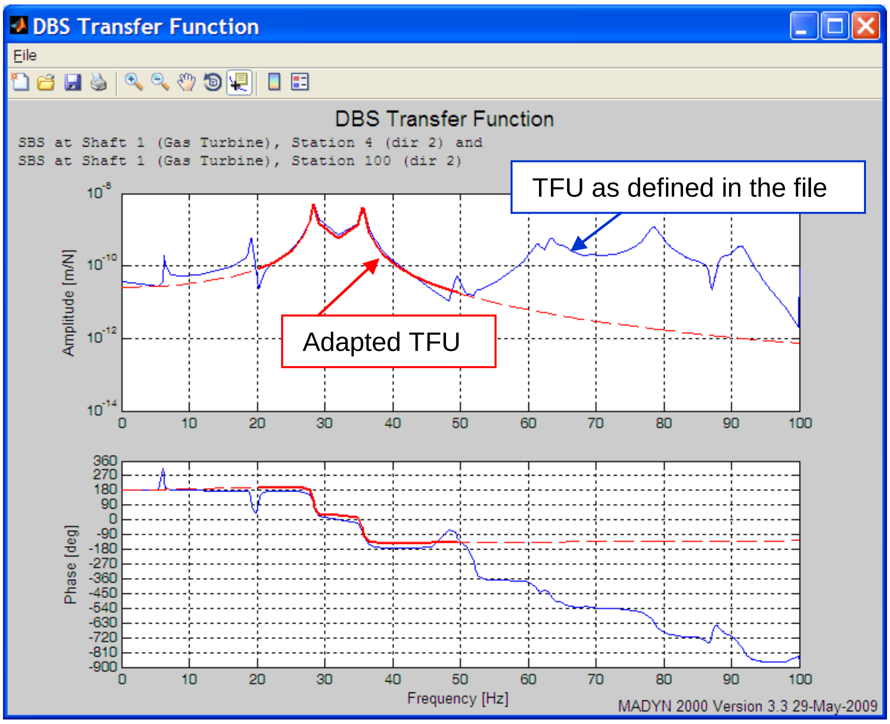


Figure 2.1.6: TFU plot with adapted polynomial transfer function, adaptation in the range 20 to 50 Hz

2.2 Behaviour of a Turbine Generator Train with DBS

In the following the influence of the supports on the behaviour of a turbine generator shaft train is shown. Results of eigenvalue analyses and harmonic response analyses for the following cases are shown:

1.) The turbine is supported on SBS with no resonances in the speed range of interest
2.) The turbine supports are DBS with several resonances

The system and the TFUs of the DBS are shown in figure 2.2.1. There are clear resonances at 28Hz and 64Hz (TFU station 4) and 36Hz (TFU station 100). The TFUs are adapted by the polynomial transfer functions in the frequency range of interest by an adequate order to model the mentioned resonances (station 4 numerator and denominator order 4, station 100 numerator order 0 and denominator order 2). Transfer functions are only considered for the vertical direction, since there are no pronounced resonances in the horizontal direction. The coupling between the supports is also neglected for the eigenvalue analysis, since it turned out to have no influence on the coupled rotor casing behaviour.

The natural modes of the system with and without DBS can be seen in figure 2.2.2 a to c. The appearance of two vertical turbine bending modes at 39Hz and 54Hz due to the dynamic bearing support can be clearly seen. Moreover the vertical tilting turbine mode drops from 34Hz to 27Hz.

The unbalance responses to two turbine load cases as shown in figure 2.2.3 were calculated. For the unbalance response the original list for the transfer function is used, not the polynomial form. The results can be seen in the figures 2.2.4 and 2.2.5 (velocity of the support vibrations) and figures 2.2.6 and 2.2.7 (relative displacement between support and shaft). It can be seen that the resonance of the support vibration slightly above 2’000rpm is shifted to about 1’600rpm when considering the dynamic bearing supports. This resonance is caused by the tilting vertical turbine mode. Moreover resonances appear at 2’400rpm and slightly above 3’000rpm due to the vertical turbine bending modes coupled with the casing vibration.

Turbine Generator

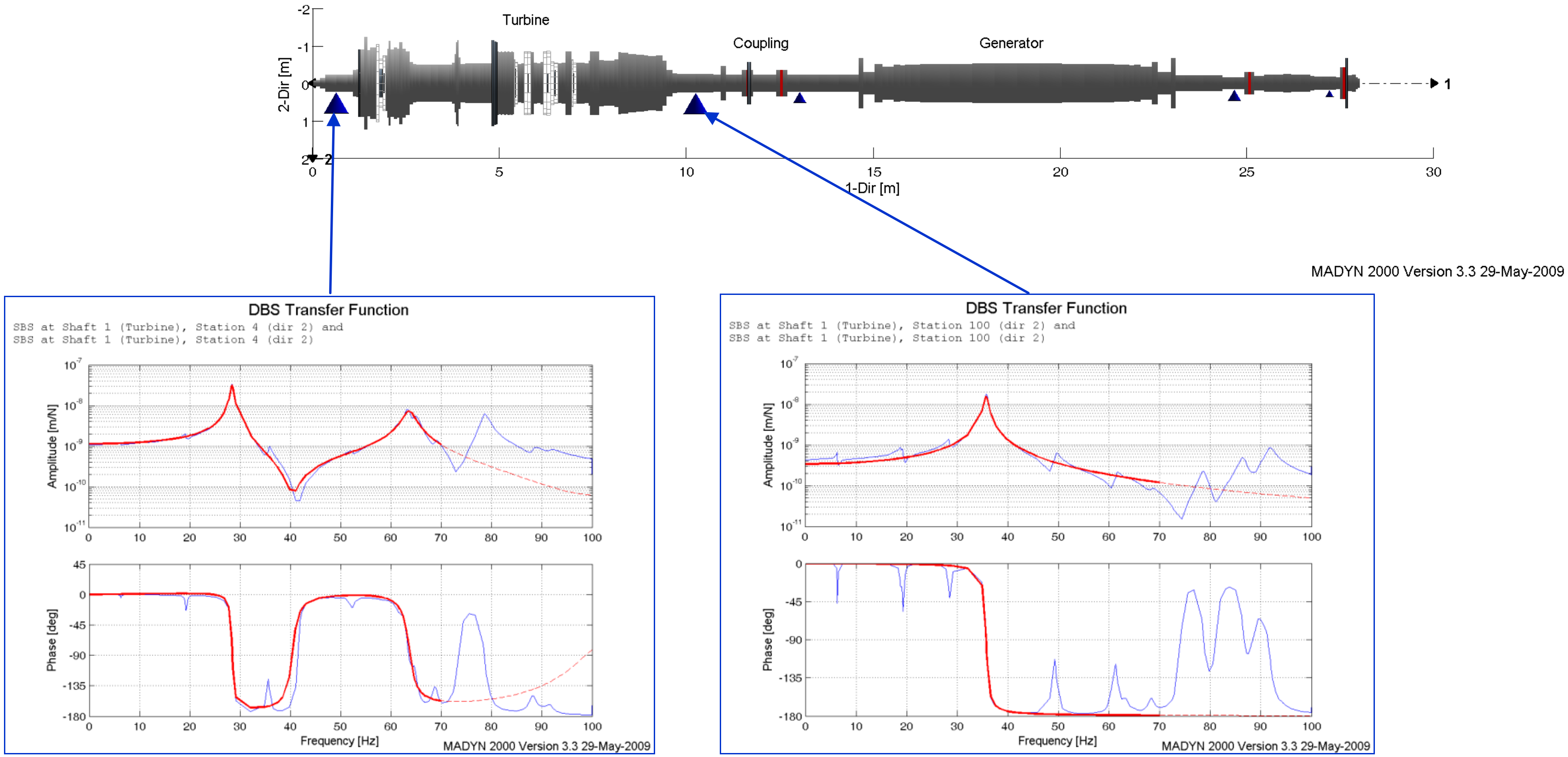


Figure 2.2.1: System consisting of a turbine and generator with TFUs at the turbine supports in vertical 2-direction

With DBS | Without DBS

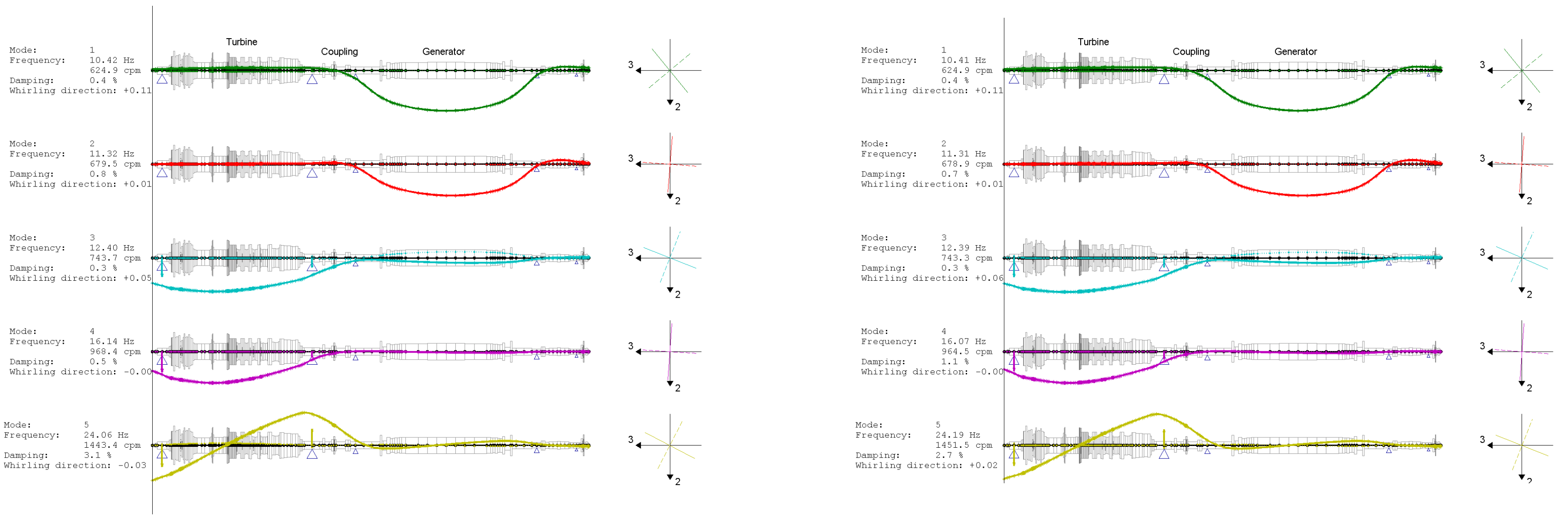
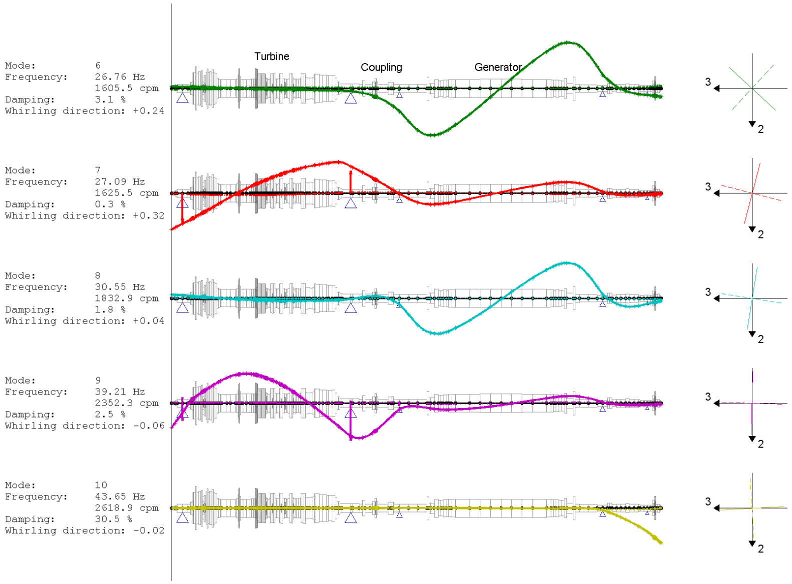


Figure 2.2.2 a: Natural mode with and without DBS

With DBS



Without DBS

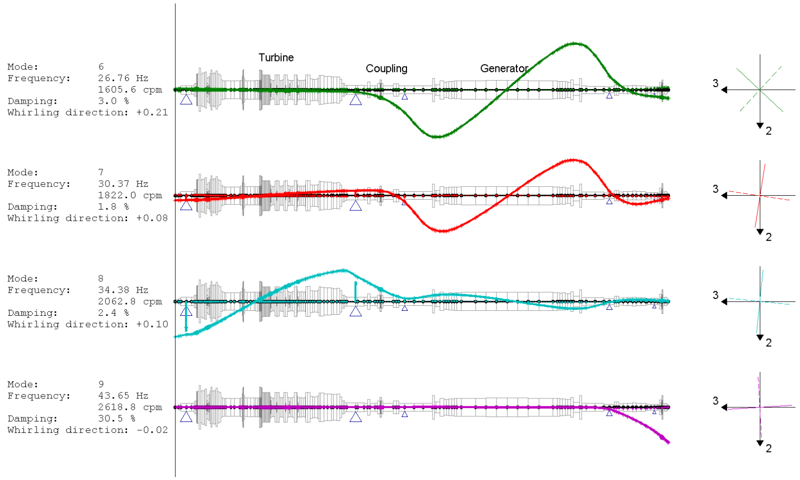
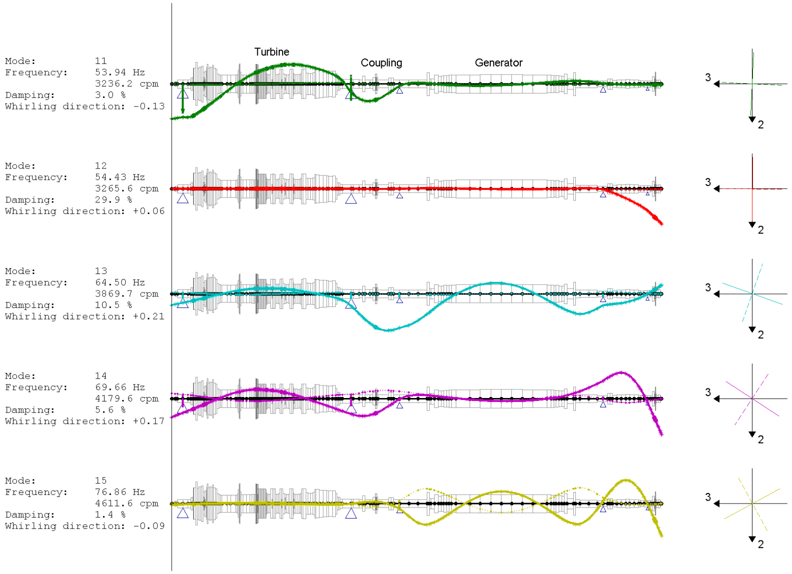


Figure 2.2.2 b: Natural mode with and without DBS

With DBS



Without DBS

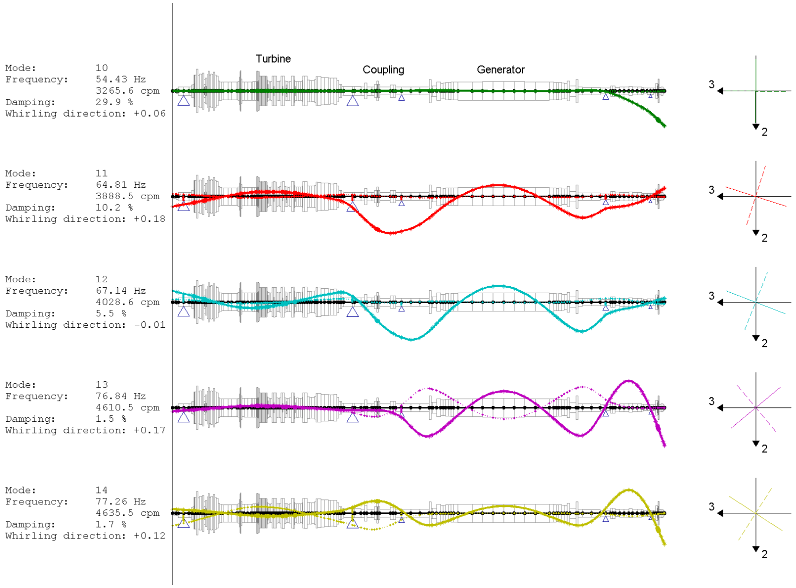
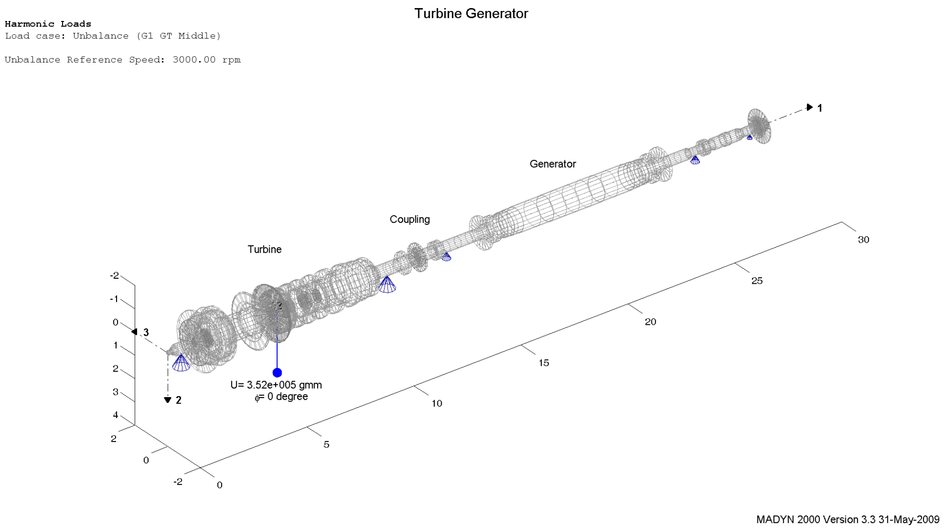


Figure 2.2.2 c: Natural mode with and without DBS

Load Case 1



Load Case 2

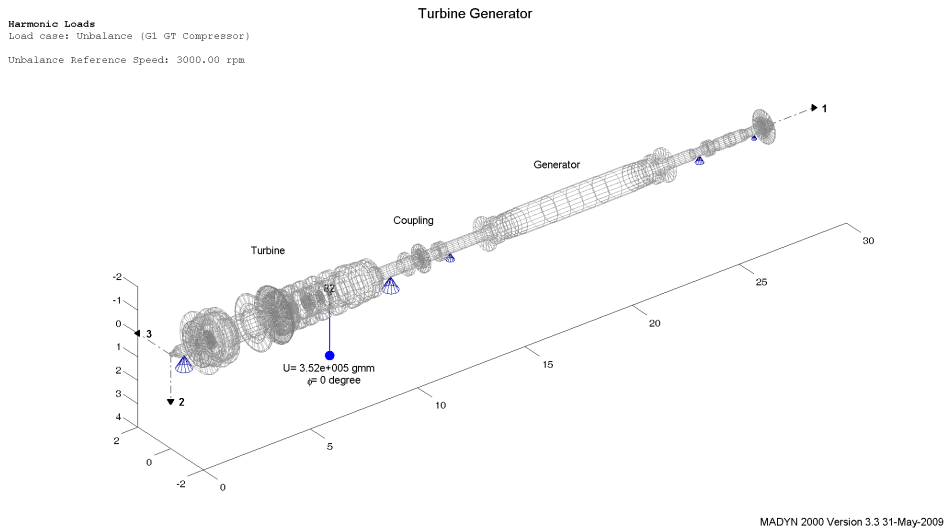
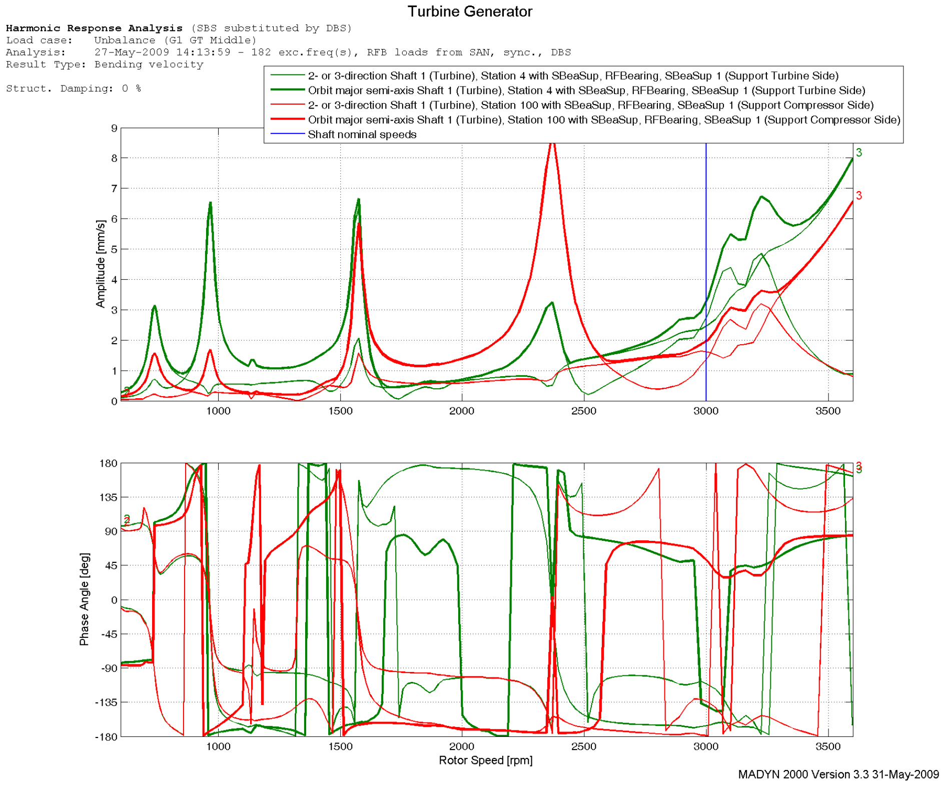


Figure 2.2.3: Unbalance load cases

With DBS



Without DBS

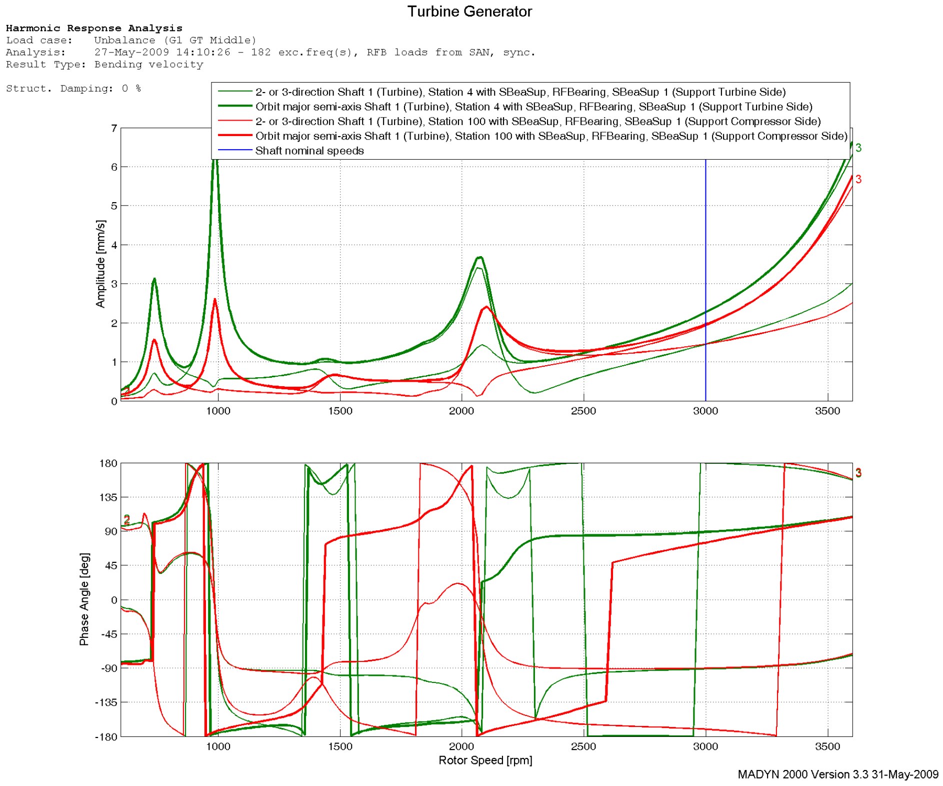
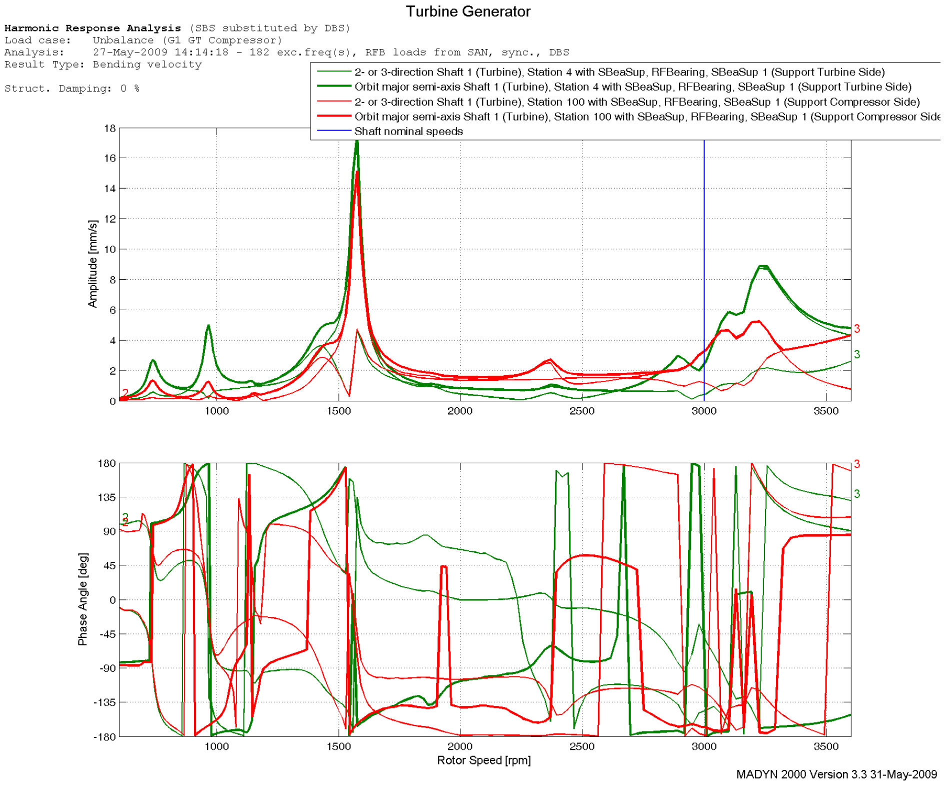


Figure 2.2.4: Response of supports to load case 1

With DBS



Without DBS

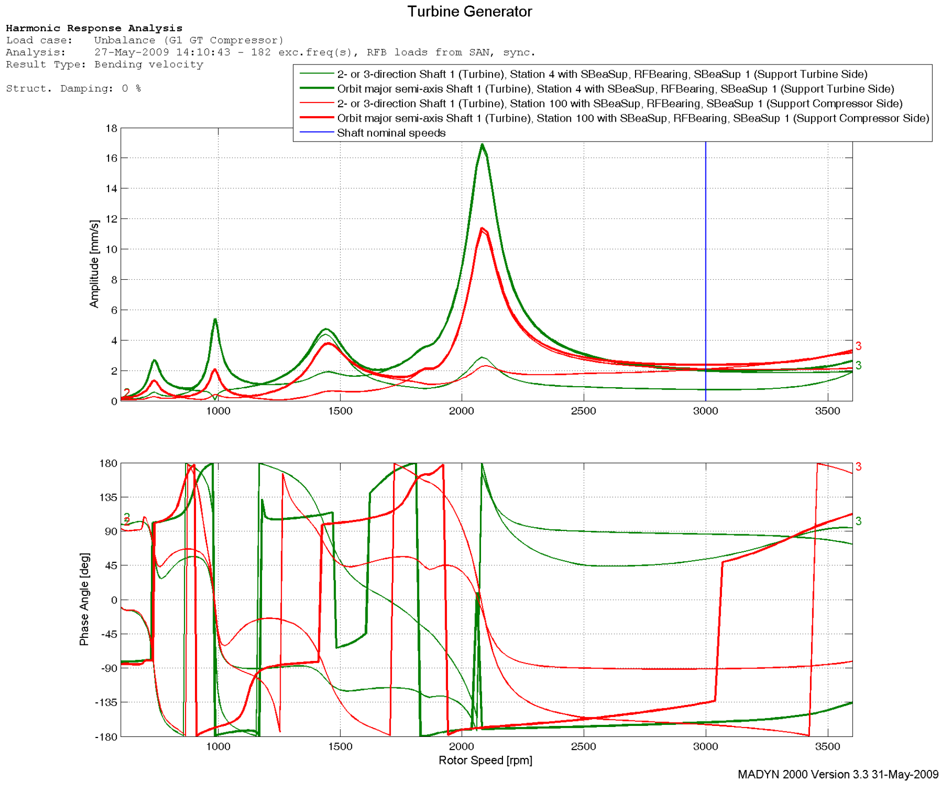
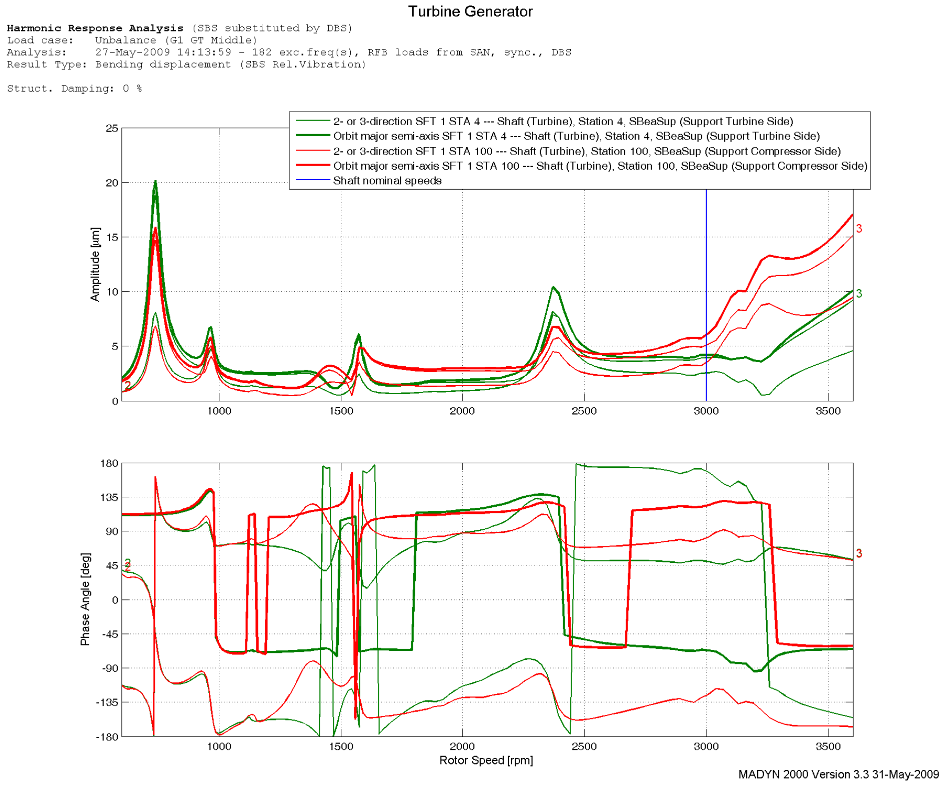


Figure 2.2.5: Response of supports to load case 2

With DBS



Without DBS

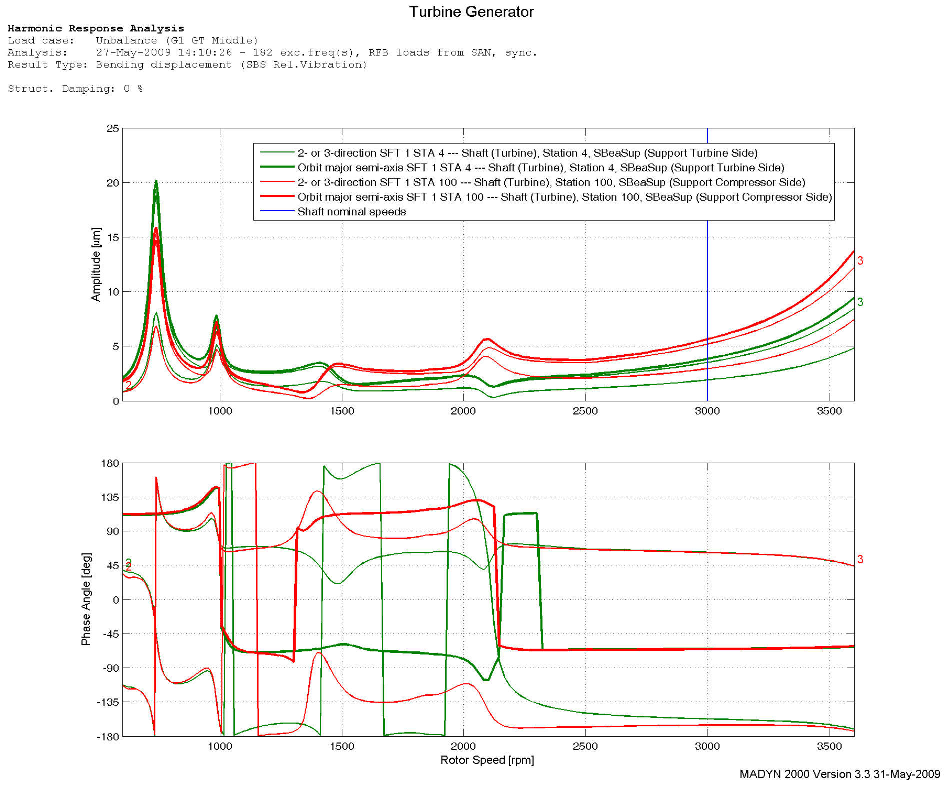
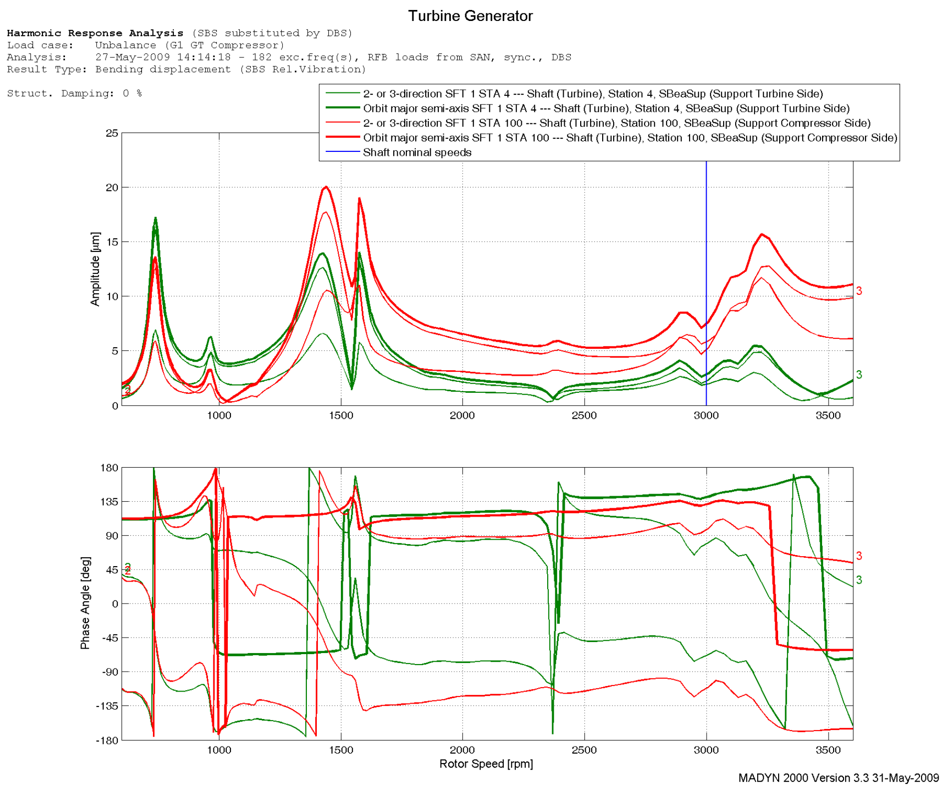


Figure 2.2.6: Response, relative shaft vibration to load case 1

With DBS



Without DBS

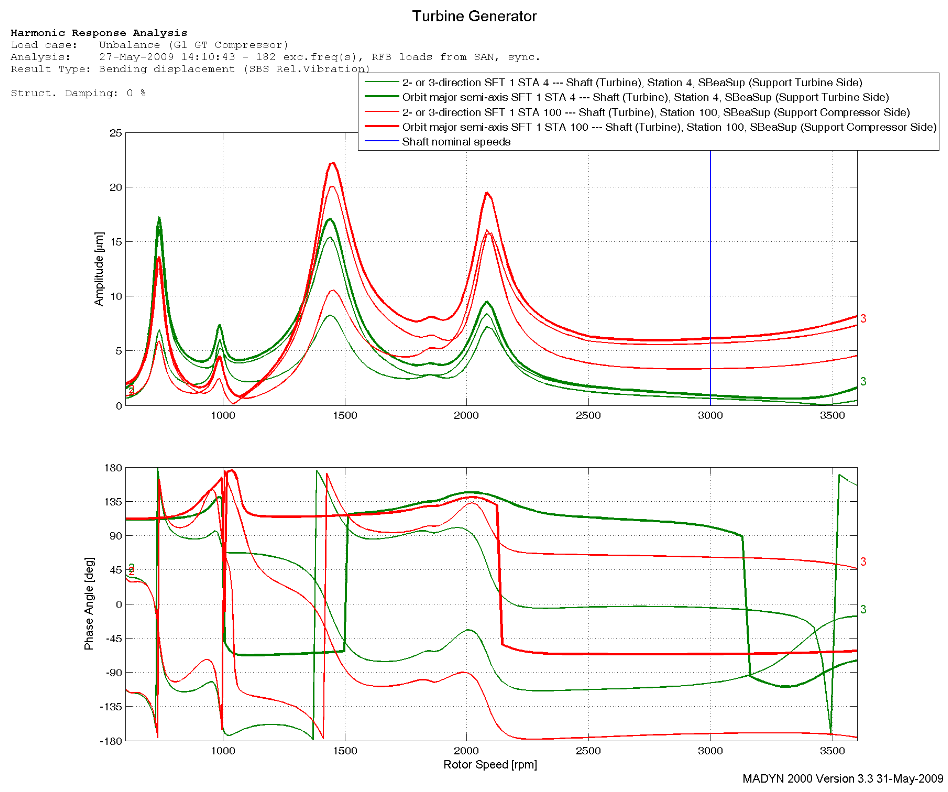


Figure 2.2.7: Response, relative shaft vibration to load case 2