Simcenter Nastran

Rotor Dynamics User’s Guide

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Introduction to Rotor Dynamics

1 Introduction to Rotor Dynamics

Simcenter Nastran includes a rotor dynamics capability that lets you predict the dynamic behavior of rotating systems. Rotating systems are subject to additional forces not present in non-rotating systems. These additional forces are a function of rotational speed and result in system modal frequencies that vary with the speed of rotation.

In a rotor dynamics analysis, the system's critical speed is particularly important. The critical speed corresponds to a rotation speed that is equal to the modal frequency. Because the critical speed is the speed at which the system can become unstable, engineers must be able to accurately predict those speeds as well as detect possible resonance problems in an analysis.

With frequency response analyses, the user can predict the steady-state response for different rotor speeds. Asynchronous analysis can be done by keeping the rotor speed constant and varying the excitation frequency. In the synchronous option, the excitation frequency is equal to, or a multiple of the rotor speed. Grid point displacement, velocity and acceleration, element forces and stresses can be recovered as function of rotor speed or excitation frequency.

Transient response in the time domain can be used in order to study the behavior of the rotor when passing a critical speed. Here, the user can define a sweep function of the excitation. In the transient analysis the grid point displacement, velocity and acceleration, element forces and stresses can be calculated as function of time.

Both modal and direct methods can be applied for complex eigenvalues, frequency response and transient response analyses.

Maneuver load analysis is a linear static structural analysis that accounts for inertial loads. The Simcenter Nastran rotor dynamics capabilities allow you to account for gyroscopic forces and forces due to damping of the rotor in a maneuver load analysis.

This guide describes the method of the rotor dynamic analysis in Simcenter Nastran, as well as the required input and modelling techniques. It also provides information about the different output data formats and describes ways that you can further post-process your data. Finally, this guide contains a number of example problems in which results from Simcenter Nastran are compared to theoretical results.

1.1 Overview of the Simcenter Nastran Rotor Dynamics Capabilities

In Simcenter Nastran, you can perform rotor dynamics analyses on structures with up to ten spinning rotors using either direct or modal solutions. You perform a rotor dynamics analysis in Simcenter Nastran using solution sequence 107 or 110 (Complex eigenvalue analysis). To compute the response of a rotating system in frequency domain, solution sequence 108 or 111 (Frequency response analysis) can be used. For transient analysis in the time domain solution sequence 109 or 112 (Transient response analysis) can be used. For maneuver load analysis, solution sequence 101 (Linear static analysis) can be used. The analysis types that are supported in rotor dynamic analysis are listed in Table 1.

Solution sequence	Analysis	Results
101	Linear static	Stress, bearing forces, damping forces
107	Direct complex eigenvalue	Campbell diagram, critical speeds, damping
108	Direct frequency response	Steady-state response in the frequency domain
109	Direct transient response	Transient response in the time domain
110	Modal complex eigenvalue	Campbell diagram, critical speeds, damping
111	Modal frequency response	Steady-state response in the frequency domain
112	Modal transient response	Transient response in the time domain

Table 1 Solution sequences supported in rotor dynamics

Simcenter Nastran commands and entries unique to rotor dynamics include:

- The RMETHOD case control command which is used to select the appropriate ROTORD bulk entry.
- The ROTORD bulk entry which is used to define rotor dynamic solution options.
- The ROTORG bulk entry which is used to define the portions of the model associated with a specific rotor.
- The ROTORB and CBEAR bulk entries which are used to model bearings. The PBEAR bulk entry is used to define bearing properties for bearings modeled using CBEAR entries.

For detailed information on creating Simcenter Nastran input files for rotor dynamic analysis, see the "Defining Simcenter Nastran Input for Rotor Dynamics" chapter.

In a rotor dynamics analysis, Simcenter Nastran takes into account all gyroscopic forces or Coriolis forces acting on the system. It also includes geometric (differential) stiffness and centrifugal softening (also referred to as spin softening).

1.1.1 Complex Eigenvalue Analysis

When you solve your model, Simcenter Nastran calculates the complex eigenvalues for each selected rotor speed, along with the damping, and the whirl direction. The software determines the whirl direction from the complex eigenvectors. The points in the rotor move on elliptical trajectories.

- If the motion is in the sense of rotation, the motion is called forward whirl.
- If the motion is against the sense of rotation, the motion is called backwards whirl.

In addition to this data, the software also calculates:

- Whirl modes (system modal frequencies that vary with rotational speed)
- Critical speeds
- Complex mode shapes (which you can view in a post-processor that supports the visualization of complex modes)

Simcenter Nastran writes the results of a rotor dynamics analysis to F06 or OP2 files for post-processing. You can also use parameters (ROTCSV, ROTGPF) in the input file to have the software generate additional types of ASCII output files (CSV and GPF files) which contain data that is specially formatted for post-processing your results with other tools. For example, you can generate a CSV file which contains data that are formatted to let you create a Campbell diagram of the eigenfrequencies using a program like Excel. You can also use the CSV file data to plot the damping as a function of rotor speed to help detect resonance points and regions of instability. More information is provided about these files later in this guide.

It is recommended to always do a complex eigenvalue analysis and to establish a Campbell and damping diagram. The results can be used as reference for response analysis and the physical behavior can be checked.

1.1.2 Frequency Response Analysis

In the frequency response analysis the complex nodal displacement, velocity and accelerations and also element forces and stresses can be plotted as function of frequency or rotor speed. The results can be output in the F06 file, OP2 file. The results can be plotted with standard Simcenter Nastran output commands in the post-processor. Also the Punch file can be used by defining XYPUNCH commands in the case control section.

For the definition of the dynamic loads, standard Simcenter Nastran commands are used. Rotating forces can be defined by applying forces in x- and y-directions with an appropriate phase lag between the components. If the force is of an unbalance type, the force can be multiplied by the square of the rotor speed as centrifugal force. The excitation order can be defined by the user.

1.1.3 Transient Response Analysis

In the transient response analysis the nodal displacement velocity or acceleration, as well as element forces and stress can be written to the F06 or OP2 file for post-processing. Also the Punch file can be used by defining XYPUNCH commands in the case control section like for the frequency response analysis.

For the definition of the dynamic loads, standard Simcenter Nastran commands are used. In this analysis type, the user must define a sweep function for the dynamic excitation if harmonic forces like mass unbalance are used. A phase lag can be defined in order to simulate rotating forces. Otherwise, general excitation types like force impulses can be applied. In addition, the force can be multiplied by the square of the rotor speed to simulate mass unbalance.

1.1.4 Maneuver Load Analysis

In a manuever load analysis the nodal displacement, as well as element forces and stress among others can be written to the F06 or OP2 file for post-processing. Also the Punch file can be used by defining XYPUNCH commands in the case control section.

1.2 Ability to Solve the Model in the Fixed or Rotating Reference System

In Simcenter Nastran, you can analyze a rotor in fixed and rotating reference systems (frames of reference). The criteria for determining which reference system to use are described later in this guide.

- In the fixed system, the nodal rotations and nodal inertia values are used.
- In the rotating system, the mass and the nodal displacements are used.

Solid models have no nodal rotations and must be analyzed in the rotating system. For special cases, there are options for analysis in the fixed system, but they must be used with care.

1.3 Support for General and Line Models

Many rotor dynamics programs require the use of a line model with concentrated mass and inertia. Line models can be used for shafts with rigid rotor disks, but not for elastic structures like propellers. These structures should be analyzed in the rotating reference system. In contrast, the rotor dynamic analysis capability in Simcenter Nastran supports the use of general models (A), unsymmetric models (B), and line models (C), as shown in the figure below.

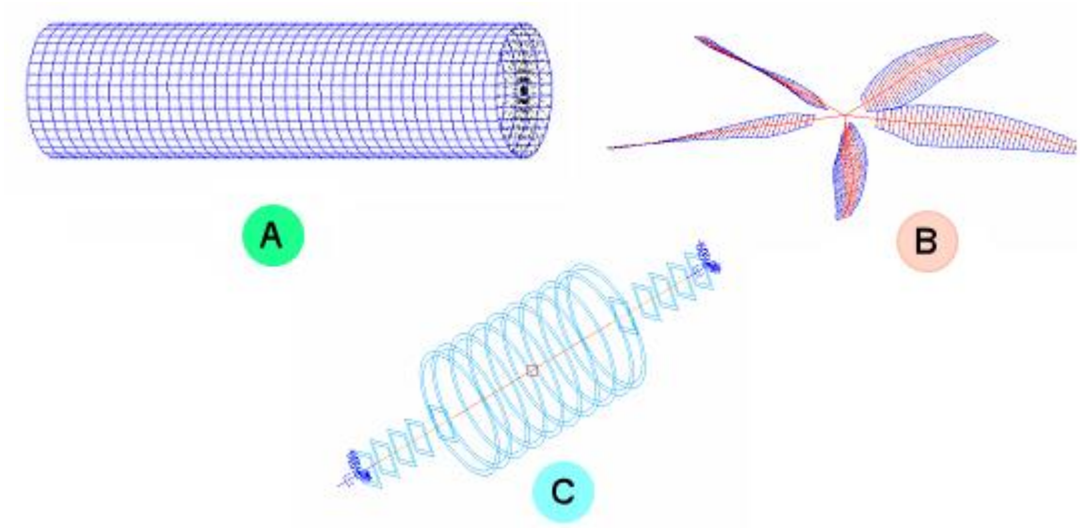


Fig. 1 Examples of Types of Models Supported by Simcenter Nastran Rotor Dynamics

You can use the full range of 1D (beam), 2D (shell) or 3D (solid) elements to mesh your rotor dynamics model. You can model bearing supports as either rigid or compliant. To model bearing supports as rigid, use fixed boundary conditions. To model bearing supports as compliant, use spring elements like CBUSH or CELASi, or use CBEAR elements. CDAMPi elements can be used in conjuction with CELASi elements to model damping in the bearing supports. CBEAR and CBUSH elements allow you to define both stiffness and damping in bearing supports. CBEAR elements allow you to model stiffness, damping, and mass.

1.4 Symmetric and Unsymmetric Rotors and Supports

In Simcenter Nastran, you can model rotor systems with symmetric and unsymmetric rotors and supports. However, some symmetry rules must still be followed as indicated in Table 2.

	Symmetric rotors	**Unsymmetric rotors**
Symmetric supports	Fixed and rotating reference systems	Rotating reference system
Unsymmetric supports	Fixed and rotating reference systems(1)	Rotating reference system(2)

(1) For symmetric rotors and unsymmetric supports, you can use a rotating reference system in SOL 107, 108, and 109 only. To do so, you must specify the ROTCOUP parameter.

(2) For unsymmetric rotors and unsymmetric supports, you can use a rotating reference system in SOL 107, 108, and 109 only. To do so, you must specify the ROTCOUP parameter.

Table 2 Symmetry Rules

1.5 Multiple Rotors

In Simcenter Nastran, up to ten rotors may be included in a single model. The rotors may run at different speeds and may rotate in different directions. Coaxial rotors can also be analyzed where one rotor supports the other. The speed of any rotor is a multiple of the reference rotor speed. The multiple can be fixed or can be a function of reference rotor speed.

1.6 Modal and Direct Method

For all rotor dynamic options, modal and direct methods can be used. The computing time with the direct methods can be significantly higher than with the modal methods. With the modal methods, truncation errors may occur. The user must decide which method is applicable. As a best practice, check the results from a modal solve with the results from a direct solve at fewer rotor speeds.

1.7 Synchronous and Asynchronous Analysis

In the modal frequency analysis, Campbell and damping diagrams are established. You can then interpret the results and judge if there are critical speeds inside the operating range. In addition, a synchronous analysis is done which calculates the critical speeds directly. This analysis is always performed, unless you switch it off. The results of the synchronous analysis can be used as verification of the asynchronous results with the Campbell diagram. It is not recommended to make only a synchronous analysis for the complex eigenvalue analysis.

In the frequency response analysis, the structure can be analyzed for a fixed rotor speed, but with frequency dependent excitation. This is the asynchronous case. The rotor can also be analyzed with varying rotor speed and excitation forces that depend on the rotor speed. This is the synchronous option. In this case, the system matrices are updated at each speed. You can select the excitation order of the applied force. For example, EORDER = 1.0 on the ROTORD bulk entry specifies unbalance excitation for rotors analyzed in the fixed system. For multiple rotors running at different speeds, the synchronous option cannot be used.

For transient analysis, both synchronous and asynchronous analysis are possible. In the synchronous case, the sweep functions which define the excitation as function of time must be compatible with the range of the rotor speed.

1.8 Mode Tracking

In the complex modal analysis, the solutions at each rotor speed are sorted by the value of the eigenfrequencies. For different modes, the frequency may increase or decrease with rotor speed and the eigenfrequency may couple with or cross those of other modes. Thus, it is important to be able to draw lines which connect the correct solutions. This process is called mode tracking. The results after mode tracking are listed in the F06 output file, stored in the OP2 file and eventually written to a CSV or GPF file. The Campbell and the damping diagrams can then be established by the post-processor.

CHAPTER

2

Theoretical Foundation of Rotor Dynamics

2 Theoretical Foundation of Rotor Dynamics

For a rotating structure, additional terms occur in the equations of motion depending on the chosen analysis system. In the software, the rotor is rotating about the positive z-axis. The equations of motion are described in this coordinate system. You can define a local coordinate system with the z-axis pointing in the rotor direction. You can then orient the rotor in other directions. To distinguish between the rotating and fixed parts of the structure, you can define a set of GRID points that belong to the rotor. If you do not define any sets of GRID points and do not select a local coordinate system with the RCORDi field on the ROTORD entry, the software assumes that all GRID points rotate about the basic z-axis.

For multiple rotors, the GRID points of the bearings must be defined in order to obtain the correct partition of the bearing damping to the specific rotor.

2.1 Additional Terms in the Equations of Motion

2.1.1 Coriolis Forces and Gyroscopic Moments

Rotor dynamics analyses in Simcenter Nastran include both Coriolis forces and gyroscopic moments.

- In the rotating analysis system, the Coriolis forces of the mass points are included.
- In the fixed analysis system, the gyroscopic moments due to nodal rotations are included.

2.1.2 Centrifugal Softening

This type of centrifugal softening occurs in analyses performed in the rotating reference system.

2.1.3 Centrifugal Stiffening Due to Centrifugal Forces

This type of stiffening occurs for blade structures such as propellers, helicopter rotors, and wind turbines. In Simcenter Nastran, you can perform a static analysis for unit rotor speed prior to the dynamic portion of the analysis to include the centrifugal stiffening effects.

This is necessary for rotor blades, hollow shafts and other models where the stretching of the structure due to the steady centrifugal force occurs.

2.1.4 Damping

The damping in a rotor system is divided into two parts:

- Internal damping acting on the rotating part of the structure.

- External damping acting on the fixed part of the structure and in the bearings.

Internal damping has a destabilizing effect. This can cause the rotor to become unstable at speeds above the critical speed. External damping has a stabilizing effect.

In the fixed system, the external damping of the bearing also acts as an antisymmetric stiffness term multiplied by the rotor speed.

In the rotating system, the damping from the rotor also acts as an antisymmetric stiffness term multiplied by the rotor speed.

Damping can be defined by the following inputs:

1. Viscous damping from CBEAR, CBUSH, and CDAMPi elements contribute to the viscous damping matrix.
2. Structural damping from PARAM G contributes to the complex stiffness matrix.
3. Structural damping from GE on MATi, PBUSH, and PELAS bulk entries contribute to the complex stiffness matrix.

In rotor dynamic analysis the complex stiffness matrix is not used. Thus, for modal solution sequences, the imaginary part of the stiffness matrix must be converted to viscous damping. This conversion is done by dividing by the eigenfrequency.

For direct solution sequences the imaginary part of the stiffness matrix is converted by dividing by:

- Parameter W3 for damping defined by PARAM G specification.
- Parameter W4 for damping defined by GE specification.

For the direct methods, the parameter W3 and W4 are required. In Simcenter Nastran the complex stiffness matrix is used. For transient analysis, there is no complex stiffness matrix and the W3 and W4 parameters are required for the transient modal solution.

Modal damping defined using a SDAMP case control command that points to a TABDMP bulk entry is not used in rotor dynamic solutions.

2.2 Equation of Motion for the Fixed Reference System

The physical equation of motion for a damped structure without external forces can be written as:

$$[M]\{\ddot{q}\} + \left(\Omega[C] + [D_I + D_A]\right)\{\dot{q}\} + \left([K] + \Omega[D_B]\right)\{q\} = \{0\} \tag{1}$$

In the modal solutions, the real eigenvalue problem is first solved and the modal vectors are collected into the modal matrix $[\Phi]$. Then the generalized equation of motion is as follows:

$$\left[\bar{M}\right]\{\ddot{q}\}+\left(\Omega\left[\bar{C}\right]+\left[\bar{D}_I+\bar{D}_A\right]\right)\{\dot{q}\}+\left(\left[\bar{K}\right]+\Omega\left[\bar{D}_B\right]\right)\{q\}=\{0\} \tag{2}$$

The generalized matrices are:

$$\left[\bar{M}\right]=[\Phi]^T[M][\Phi]=[I] \quad \text{generalized mass matrix} \tag{3}$$

$$\left[\bar{C}\right]=[\Phi]^T[C][\Phi] \quad \text{generalized antisymmetric gyroscopic matrix} \tag{4}$$

$$\left[\bar{D}_I\right]=[\Phi]^T[D_I][\Phi] \quad \text{generalized internal viscous damping matrix} \tag{5}$$

$$\left[\bar{D}_A\right]=[\Phi]^T[D_A][\Phi] \quad \text{generalized external viscous damping matrix} \tag{6}$$

$$\left[\bar{K}\right]=[\Phi]^T[K][\Phi]=\mathrm{diag}\left[\omega_0^2\right] \quad \text{generalized elastic stiffness matrix} \tag{7}$$

$$\left[\bar{D}_B\right]=[\Phi]^T[D_B][\Phi] \quad \text{generalized antisymmetric internal damping matrix} \tag{8}$$

For a rotating mass point the terms are as follows.

The mass matrix is the same as for non-rotating structures. Here, a lumped mass approach is used:

$$[M]=\begin{bmatrix} m & & & & & \\ & m & & & & \\ & & m & & & \\ & & & \Theta_x & & \\ & & & & \Theta_y & \\ & & & & & \Theta_z \end{bmatrix} \tag{9}$$

In the gyroscopic matrix, only the polar moment of inertia appears. A model without polar moments of inertia has no rotational effects in the fixed system. Only the rotational degrees of freedom are used.

$$[C]=\begin{bmatrix} 0 & 0 & 0 & & & \\ 0 & 0 & 0 & & & \\ 0 & 0 & 0 & & & \\ & & & 0 & -\Theta_z & 0 \\ & & & \Theta_z & 0 & 0 \\ & & & 0 & 0 & 0 \end{bmatrix} \tag{10}$$

Internal rotor damping matrix:

$$[D_I] = \begin{bmatrix} d_{I,x} & & & & & \\ & d_{I,y} & & & & \\ & & d_{I,z} & & & \\ & & & d_{I,Rx} & & \\ & & & & d_{I,Ry} & \\ & & & & & d_{I,Rz} \end{bmatrix} \tag{11}$$

External damping acting in the non-rotating bearings:

$$[D_A] = \begin{bmatrix} d_{A,x} & & & & & \\ & d_{A,y} & & & & \\ & & 0 & & & \\ & & & d_{A,Rx} & & \\ & & & & d_{A.Ry} & \\ & & & & & 0 \end{bmatrix} \tag{12}$$

Because the displacements in the rotating part act as a velocity in the fixed part, an antisymmetric matrix appears in the stiffness term:

$$[D_B] = \begin{bmatrix} 0 & d_{I,T} & 0 & & & \\ -d_{I,T} & 0 & 0 & & & \\ 0 & 0 & 0 & & & \\ & & & 0 & d_{I,R} & 0 \\ & & & -d_{I,R} & 0 & 0 \\ & & & 0 & 0 & 0 \end{bmatrix} \tag{13}$$

In the above equations, the following notation has been used:

m	mass
Θ_x	Inertia about x-axis
Θ_y	Inertia about y-axis
Θ_z	Inertia about z-axis
d_I	Viscous damping of rotor
d_A	Viscous damping of rotor
Subscript T	Translation
Subscript R	Rotation

In equation (13), damping in x- and y-directions are assumed equal.

2.2.1 Including Steiner’s Inertia Terms in the Analysis

The ZSTEIN option on the ROTORD entry lets you include the Steiner’s inertia terms in the analysis so you can analyze solid models in the fixed reference system. In this case, the polar moment of inertia is calculated as:

$$\Theta_z = \Theta_P = \sum m\left(dx^2 + dy^2\right) \tag{14}$$

You can only use the ZSTEIN option if the local rotations of the nodes are representative for that part. For example, you can analyze a solid shaft or a stiff rim of an electric generator in this way but not a propeller or an elastic structure. Additionally, to use the ZSTEIN option, the nodes must have rotations. In a model with solid elements, this can be obtained by adding a layer of thin shell elements around the solid elements. However, if you use this approach, you must ensure that the nodal rotations of the shell elements are not constrained by the AUTOSPC option.

In general, the ZSTEIN option must be used carefully. The preferred solution for general finite element models is to perform the rotor dynamics analysis in the rotating reference system.

2.3 Equation of Motion for the Rotating Reference System

The physical equation of motion for a damped structure without external forces can be written as:

$$[M]\{\ddot{q}\} + \left(2\Omega[C] + [D_I + D_A]\right)\{\dot{q}\} + \left([K] + \Omega[D_B] - \Omega^2[Z] + \Omega^2[K_G]\right)\{q\} = \{0\} \tag{15}$$

Using the modal matrix $[\Phi]$ found from the real eigenvalue analysis, the generalized equation of motion is as follows:

$$[\bar{M}]\{\ddot{q}\} + \left(2\Omega[\bar{C}] + [\bar{D}_I + \bar{D}_A]\right)\{\dot{q}\} + \left([\bar{K}] + \Omega[\bar{D}_B] - \Omega^2[\bar{Z}] + \Omega^2[\bar{K}_G]\right)\{q\} = \{0\} \tag{16}$$

The following additional terms occur:

$[\bar{Z}] = [\Phi]^T[Z][\Phi]$ generalized centrifugal softening matrix (17)

$[\bar{K}_G] = [\Phi]^T[K_G][\Phi]$ generalized geometric of differential stiffness matrix (18)

The gyroscopic matrix is different from the non-rotating formulation and Coriolis terms for the mass occur.

$$[C] = \begin{bmatrix} 0 & -m & 0 & 0 & 0 & 0 \\ m & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & \frac{1}{2}\left(\Theta_z - \left(\Theta_x + \Theta_y\right)\right) & 0 \\ 0 & 0 & 0 & -\frac{1}{2}\left(\Theta_z - \left(\Theta_x + \Theta_y\right)\right) & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 \end{bmatrix} \quad (19)$$

For a symmetric rotor, the inertia about the x- and y- axes are equal:

$$\Theta_x = \Theta_y = \Theta_A$$

For a thin disk with polar moment of inertia Θ_p, the following relation is valid:

$$\Theta_z = \Theta_P = 2\Theta_A$$

and the gyroscopic terms vanish:

$$\frac{1}{2}\left(\Theta_z - \left(\Theta_x + \Theta_y\right)\right) = \frac{1}{2}\left(\Theta_P - 2\Theta_A\right) = 0$$

The damping terms are similar to those of the fixed system:

$$[D_I] = \begin{bmatrix} d_{I,x} & & & & & \\ & d_{I,y} & & & & \\ & & d_{I,z} & & & \\ & & & d_{I,Rx} & & \\ & & & & d_{I,Ry} & \\ & & & & & d_{I,Rz} \end{bmatrix} \quad (20)$$

$$[D_A] = \begin{bmatrix} d_{A,x} & & & & & \\ & d_{A,y} & & & & \\ & & 0 & & & \\ & & & d_{A,Rx} & & \\ & & & & d_{A,Ry} & \\ & & & & & 0 \end{bmatrix} \quad (21)$$

Here, the external damping occurs as an antisymmetric stiffness term:

$$[D_B] = \begin{bmatrix} 0 & -d_A & 0 & & & \\ d_A & 0 & 0 & & & \\ 0 & 0 & 0 & & & \\ & & & 0 & -d_A & 0 \\ & & & d_A & 0 & 0 \\ & & & 0 & 0 & 0 \end{bmatrix} \tag{22}$$

The centrifugal softening matrix is:

$$[Z] = \begin{bmatrix} m & 0 & 0 & & & \\ 0 & m & 0 & & & \\ 0 & 0 & 0 & & & \\ & & & -(\Theta_z - \Theta_y) & 0 & 0 \\ & & & 0 & -(\Theta_z - \Theta_x) & 0 \\ & & & 0 & 0 & 0 \end{bmatrix} \tag{23}$$

For a symmetric rotor the inertia terms become:

$$\Theta_z - \Theta_y = \Theta_z - \Theta_x = \Theta_P - \Theta_A$$

For a disk, the terms are:

$$\frac{1}{2}\Theta_P$$

To obtain the geometric or differential stiffness matrix $[K_G]$, you must insert a static SUBCASE prior to the modal analysis. The load from the static subcase must be referenced in the modal subcase by a STATSUB command. A unit rotor speed of 1 rad/sec (hence, $1/2\pi$ Hz = 0.159155 Hz) must be specified on the RFORCE, RFORCE1, or RFORCE2 bulk entry in your Simcenter Nastran input file.

In Simcenter Nastran rotor dynamic analysis the lumped mass matrix is used. This is normally sufficient for reasonably fine models.

In Simcenter Nastran, rotor dynamic matrices are generated and reduced to the real modal problem of h-set size by transforming the complex physical system in g-set to a modal basis using the eigenfrequencies and mode shapes of the non-rotating full structure. With this method, even large structures can be analyzed efficiently. In the direct methods, the rigid elements, MPC, and SPC degrees of freedom are eliminated in the standard way and the equations are solved directly.

2.4 Real Eigenvalue Analysis for the Modal Solutions

As a first step in a modal rotor dynamics analysis, the software performs a real eigenvalue analysis:

$$\left(-\omega_0^2[M]+[K]\right)\{\varphi\}=\{0\} \tag{24}$$

The solution eigenvectors $\{\varphi\}$ are collected into the modal matrix $[\Phi]$. The displacement vectors can be described by a linear combination of the modes: $\{u\}=[\Phi]\{q\}$. This is mathematically true if all modes are used. It is a reasonable approximation when a sufficient number of modes are considered. The selected modes used for the subsequent rotor dynamic analysis must be chosen according to the following criteria:

- The frequency range of the real modes should be well above the frequency region of interest in the rotor dynamic analysis.
- The real modes must be able to describe the rotor motion. Modal displacement in x- and y-direction must be included.
- The modes must be able to represent the generalized forces.

After the establishment of the modal matrices, the software executes a rotor loop over the defined rotor speeds, and the results for each rotor speed are collected for post-processing. In addition to this, a synchronous analysis is performed to calculate possible critical speeds where the imaginary parts of the eigenvalues are equal to the rotor speed.

2.5 Reduction to the Analysis Set for the Direct Methods

For the direct methods, the matrices in the g-set are reduced to the n-set by eliminating the MPC and rigid elements. Then the SPC degrees of freedom are eliminated to obtain the f-set. In the reductions, the unsymmetry of the gyroscopic matrix is accounted for. The matrices can then be used in the direct solutions.

A-Set Dynamic Reduction
A dynamic reduction can optionally be performed to increase SOL 107 efficiency. You specify the exterior a-set DOF with ASET or ASET1 bulk entries. The DOF which are not included in the a-set become the o-set. The software reduces the o-set into modal coordinates with a real eigenvalue analysis. A complex eigenvalue analysis then occurs on the combined a-set and reduced o-set.

The reduction of the o-set is a symmetric condensation, although the a-set remains unsymmetric. For best accuracy and performance, the a-set must be selected carefully to preserve the unsymmetric rotor dynamic effects. For a solid rotor example, it is sufficient if you specify points on the center line as a-set grid points. For turbines, you must also specify the grid points on the blades.

You must include an EIGRL bulk entry in addition to the EIGC bulk entry. The EIGRL entry defines the options for the o-set real eigenvalue analysis. The METHOD case control command must be included to select the EIGRL entry. An appropriate EIGRL setting is problem dependent; more modes improves accuracy, but also increases run time.

You must also create at least as many spoints with the SPOINT bulk entry as there are modal coordinates, and their identification numbers should be included in a QSET bulk entry. The number of modes found in the real eigenvalue analysis is the number of modal coordinates. You can include more spoints than necessary if the number of modal coordinates is unknown.

The file r1b1_aset.dat can be found at *install_dir/nxnr/nast/tpl* to demonstrate the inputs.

Complex Modal Reduction
Complex modal reduction is also available to increase SOL 107 efficiency. When you use complex modal reduction, the software computes the complex modes and uses them to project the problem into modal space where, depending on the number of modes computed, the problem size is typically reduced. The software then performs an eigensolution on the reduced problem at each rotor speed. The software then projects the results of the eigensolution back into physical space for presentation and subsequent post-processing. Complex modal reduction is specified using the CMR describer on the RMETHOD case control command.

Complex modal reduction is applicable to problems containing unsymmetric stiffness and unsymmetric viscous damping. This makes a SOL 107 solve with complex modal reduction ideal for reducing the computational effort required to solve rotor dynamics problems that contain sources of unsymmetric stiffness and unsymmetric viscous damping like journal bearings.

For additional information on complex modal reduction, see the “Defining Simcenter Nastran Input for Rotor Dynamics” chapter.

2.6 Fixed System Eigenvalue Problem

For harmonic motion $\{q(t)\} = \{q\} e^{\lambda t}$, the following eigenvalue problem is solved in the fixed reference system:

$$\left(\lambda^2 \left[\bar{M}\right] + \lambda\left(\Omega\left[\bar{C}\right] + \left[\bar{D}_I + \bar{D}_A\right]\right) + \left(\left[\bar{K}\right] + \Omega\left[\bar{D}_B\right]\right)\right)\{q\} = \{0\} \tag{25}$$

A loop is made over the selected rotor speeds and the results are stored for post-processing.

2.6.1 Synchronous Analysis

For the points of intersection with the 1P-line, the eigenfrequency is equal to the rotor speed: $\omega = \Omega$. Neglecting the damping, $\lambda = j\omega$. To obtain the resonance points, the following eigenvalue equation is solved:

$$\left(-\Omega^2\left(\left[\bar{M}\right]+j\left[\bar{C}\right]\right)+\left[\bar{K}\right]\right)\{q\}=\{0\} \tag{26}$$

The imaginary parts of the solutions are the critical rotor speeds. For the modes that don't cross the 1P-line, the imaginary part is zero.

For models with multiple rotors, the eigenvalue problem is solved for each relative rotor speed. Not all solutions may be relevant and the user must verify the results of the synchronous analysis with the Campbell diagram.

2.7 Rotating System Eigenvalue Problem

The eigenvalue problem in the rotating system is:

$$\left(\lambda^2\left[\bar{M}\right]+\lambda\left(2\Omega\left[\bar{C}\right]+\left[\bar{D}_I+\bar{D}_A\right]\right)+\left(\left[\bar{K}\right]+\Omega\left[\bar{D}_B\right]-\Omega^2\left[\bar{Z}\right]+\Omega^2\left[\bar{K}_G\right]\right)\right)\{q\}=\{0\} \tag{27}$$

After the rotor loop, the post-processing is initiated. In the complex eigenvalue analysis, the software sorts the solutions in frequencies. With increasing rotor speed, the modes are changing, and there may be crossing of lines or coupling between different modes. To generate the data necessary for later creating Campbell diagrams, Simcenter Nastran sorts the solutions by automatically applying a mode tracking algorithm. This algorithm extrapolates the previous results and looks for the results of eigenfrequency and damping which best match the previous solution. To get smooth curves, you should use a sufficiently high number of rotor speeds. For large models, the computing time for the complex modes of the generalized system is generally lower than the real eigenvalue analysis required for the modal formulation prior to the rotor loop.

2.7.1 Synchronous Analysis

In the case of analysis in the rotating system, the forward whirl resonance points are found for the intersection of the 0P-line (abscissa). In this case the frequency is zero: $\omega = 0$ and the critical speed is found from the static part of the equation. Also here the damping is neglected.

$$\left(\left[\bar{K}\right]-\Omega^2\left(\left[\bar{Z}\right]-\left[\bar{K}_G\right]\right)\right)\{q\}=\{0\} \tag{28}$$

The backward whirl resonances can be found at the intersection with the 2P line.

$$\left(\left[\bar{K}\right]-\Omega^2\left(j4\left[\bar{C}\right]-4\left[\bar{M}\right]+\left[\bar{Z}\right]-\left[\bar{K}_G\right]\right)\right)\{q\}=\{0\} \tag{29}$$

In the program, both eigenvalue problems are solved and the results are concatenated into one output block. If the whirling direction could be calculated, the backward solutions are removed from the solution of the first and the forward solutions from the solution of the last eigenvalue problem.

Also for analysis in the fixed system, the eigenvalue problems are solved for each relative rotor speed. Not all solutions may be relevant and the user must verify the results of the synchronous analysis with the Campbell diagram.

2.8 Solution Interpretation

The solutions at each rotor speed are the complex conjugate pairs of eigenvalues $\lambda = \delta \pm j\omega$.

The real part is a measure of the amplitude amplification. Positive values lead to an increase in amplitude with time and the mode is unstable. The system is stable when the real part is negative. The damping is defined as:

$$\zeta = \frac{\delta}{\omega} \tag{30}$$

and is the fraction of critical viscous damping This damping is printed in the Campbell summary and written to the output files. In the complex modal analysis of Simcenter Nastran the g-damping is output: $g = 2\zeta$.

The imaginary part is the oscillatory part of the solution. The eigenfrequency is then $f = \frac{\omega}{2\pi}$ Hz. You can use the FUNIT option on the ROTORD entry to have the software output the solution eigenfrequencies in the units [rad/s], [Hz] or [RPM], or [CPS]. You can use the RUNIT option on the ROTORD entry to enter and output the rotor speed in the same units. The default is [Hz] for the frequencies and [RPM] for the rotor speed.

Fig. 2 shows a plot of the complete set of solutions for positive (rotation vector in positive z-axis) and negative (rotation vector in negative z-axis) rotor speeds. The plot is symmetric about both axes, and in the rotor dynamic analysis, only the first quadrant is plotted, as shown in Fig. 3. This is called a Campbell diagram. The analysis was performed in the fixed system. The model is a rotating disk on a cantilevered shaft.

The whirl direction of the modes can be calculated from the eigenvectors. The complex physical eigenvectors are $\{u\} = [\Phi]\{q\}$. The real and imaginary parts of the displacement in the x- and y-directions are collected into two vectors for each node i:

$$\{v_{1,i}\} = \begin{Bmatrix} u_{x,i}^{Im} \\ u_{y,i}^{Im} \\ 0 \end{Bmatrix} \tag{31}$$

$$\{v_{2,i}\} = \begin{Bmatrix} u_{x,i}^{Re} \\ u_{y,i}^{Re} \\ 0 \end{Bmatrix} \tag{32}$$

The whirl direction is found from the direction of the cross product

$$\{w\} = \{v_1\} \times \{v_2\} \tag{33}$$

If the vector $\{w\}$ is pointing in the positive z-direction, Simcenter Nastran marks the solution as forward whirl. If the vector is pointing in the negative z-direction, the whirl is backward. If the absolute value of the component in the z-direction is less than a prescribed value (defined with the ORBEPS option on the ROTORD entry), the motion is found to be linear.

In Fig. 3, the forward whirl modes (solutions 2 and 4) are plotted as red lines and the backward (solution 1 and 3) as blue lines. Linear modes are plotted as green lines (not shown in this example).

In most of the Campbell diagrams shown in this guide, the colors were selected based on the whirl direction color codes. These color codes are included in the CSV and GPF files. For analyses in the fixed system, the forward whirl modes are generally those with increasing frequency, and the backward whirl are those with decreasing frequency. The linear modes are straight horizontal lines.

The critical speeds are the resonance points with the 1P excitation lines. Mass imbalance will excite the forward whirl. There may be excitation also of the backwards whirl, for example due to excitation via the foundation. For helicopter rotors and wind turbines, there is also excitation of higher orders (2P, 3P etc.)

When the whirl direction is known, a conversion between the rotating and the fixed analysis system can be done by adding and subtracting the rotor speed with the backwards and forward whirl motions respectively. The same model was analyzed in the rotating system with results shown in Fig. 5. The curves are identical to those of Fig. 4 except for the color of the translation mode (solution 1 in Fig. 5 and solution 2 in Fig. 4) after the singular point, where the solution with negative frequency becomes positive and the solution with positive frequency becomes negative in the rotating system.

In the rotating system, the critical speed for the forward whirl motion is the crossing with the x-axis in the plots, i.e. the frequency becomes zero. The critical speeds for the backwards whirl motion are the crossings with the 2P line.

The real part of the solution is shown in Fig. 6. The real part of the solution is the same for both analysis systems. Because the damping is the real part of the solution divided by the frequency (see equation (30)), the damping curves are different for both systems as shown in Fig. 7 and Fig. 8.

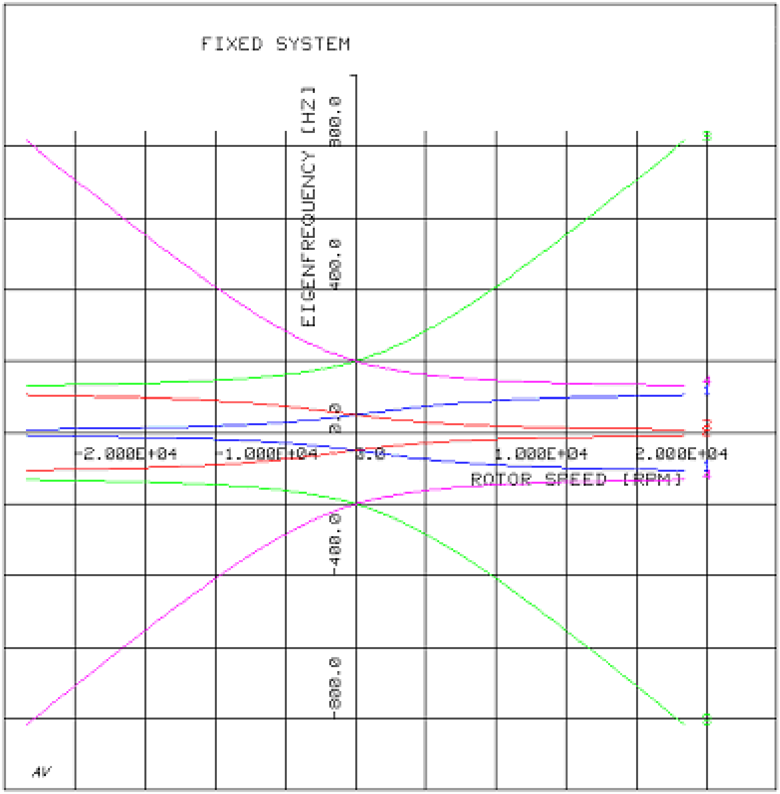


Fig. 2 Rotor Dynamic Analysis Example for Positive and Negative Rotor Speeds

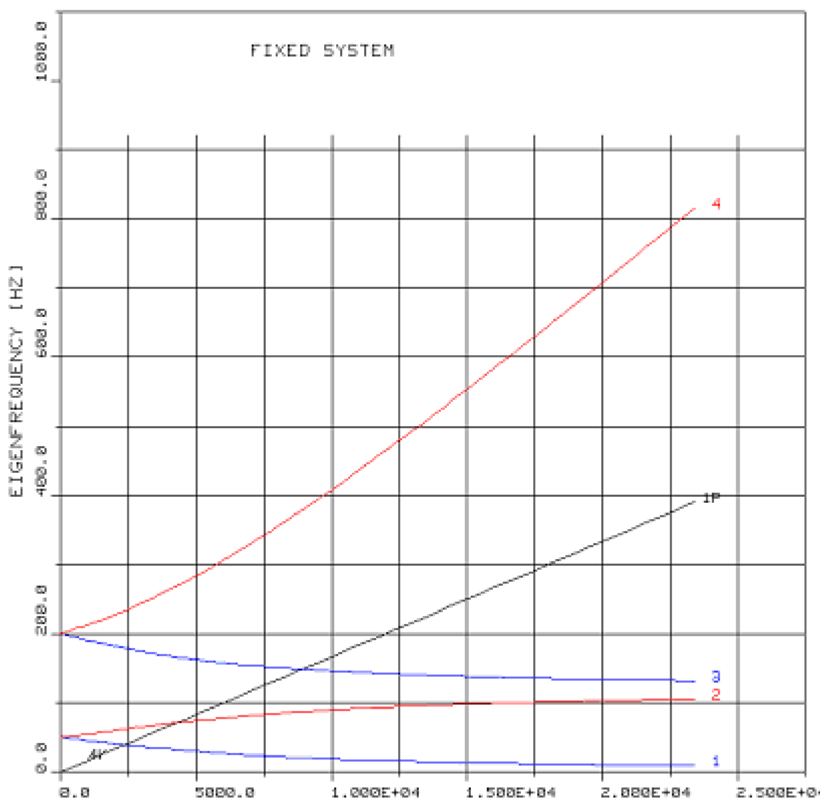


Fig. 3 Campbell Diagram of a Rotor Analyzed in the Fixed System

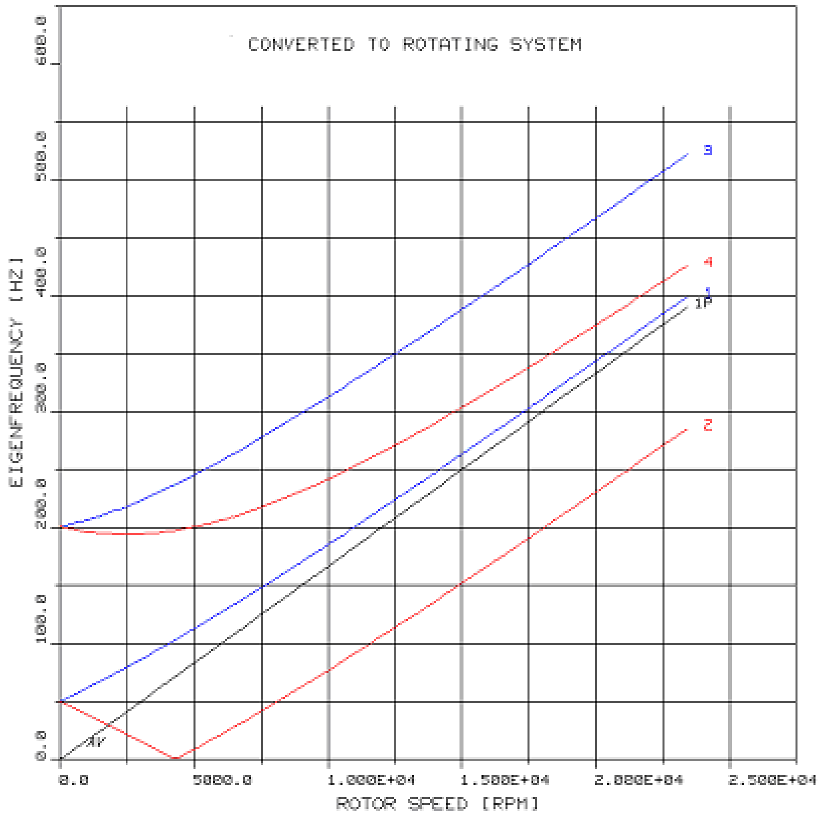


Fig. 4 Campbell Diagram Converted to the Rotating System

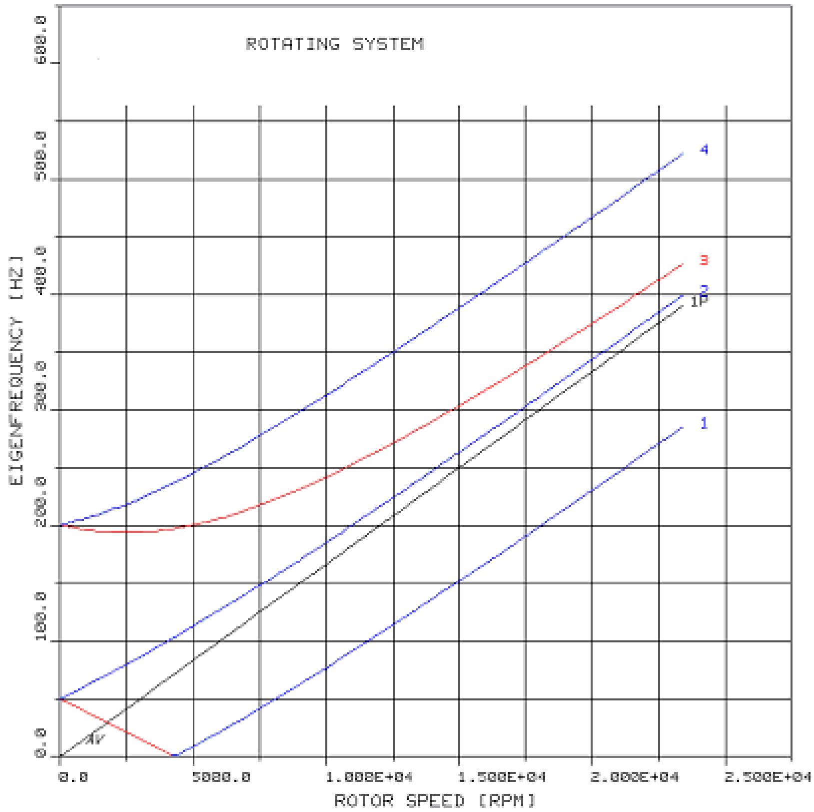


Fig. 5 Campbell Diagram of a Rotor Analyzed in the Rotating System

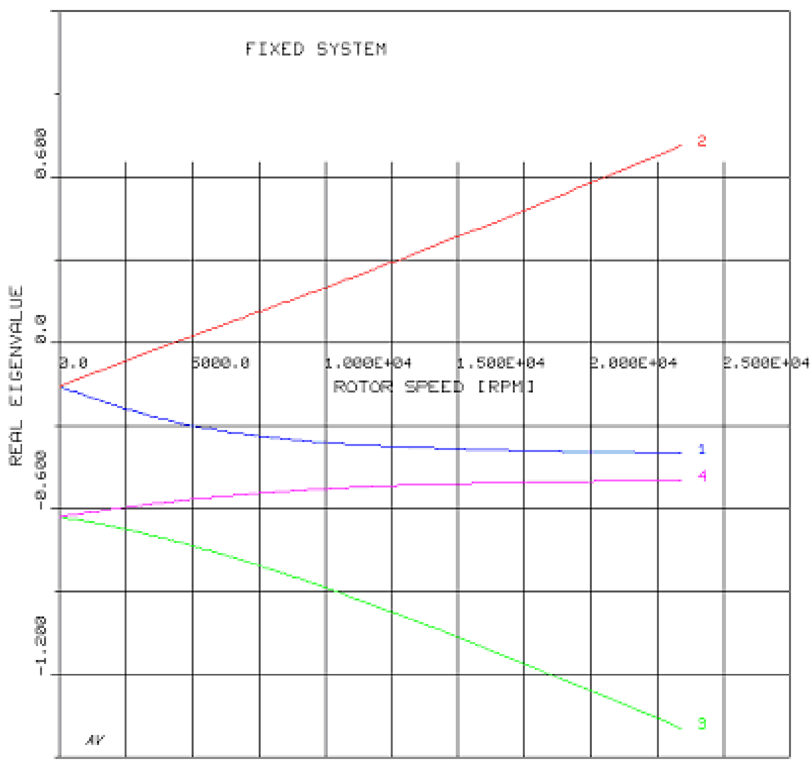


Fig. 6 Real Part of the Solution

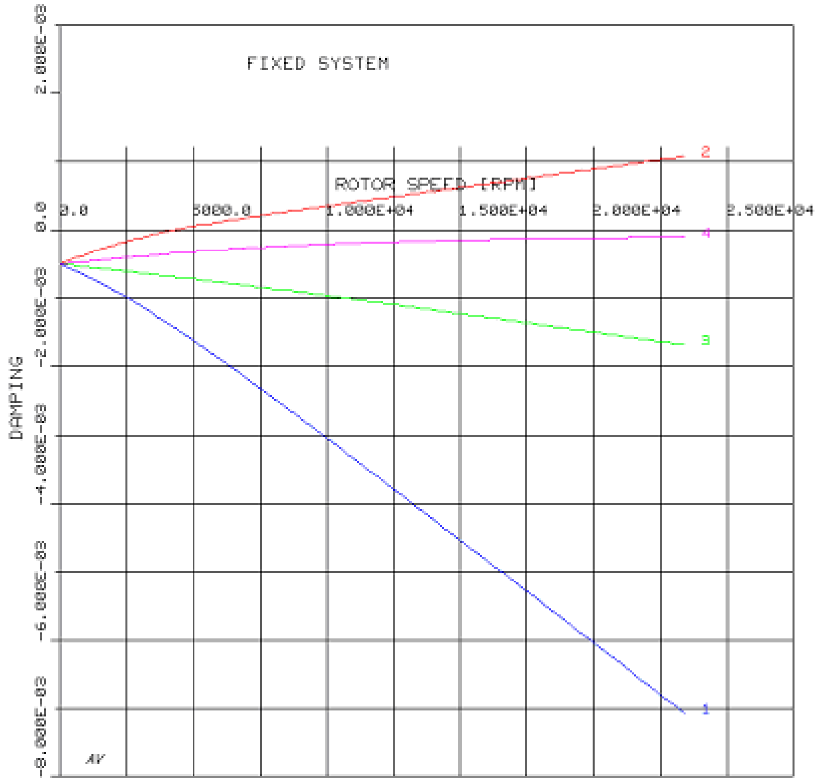


Fig. 7 Damping Diagram in the Fixed System

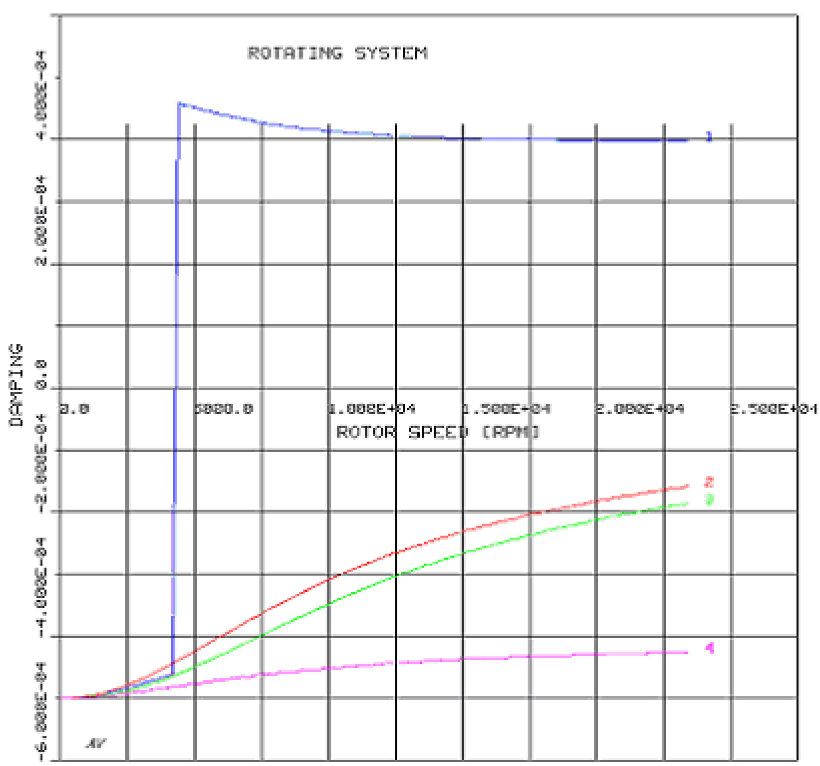


Fig. 8 Damping Diagram in the Rotating System

2.9 Equation of Motion for Frequency Response

In the following sections, only the modal equations are shown. For the direct solutions the bar above the symbols must be removed and the modal matrix is not used.

2.9.1 Fixed Reference System:

The governing equation of frequency response in modal space with rotor dynamics terms, in fixed reference system considering the load to be independent of the speed of the rotation, is:

$$\left(-\omega_k^2\left[\bar{M}\right]+j\omega_k\left(\Omega\left[\bar{C}\right]+\left[\bar{D}_I+\bar{D}_A\right]\right)+\left(\left[\bar{K}\right]+\Omega\left[\bar{D}_B\right]\right)\right)\left\{\bar{u}(\omega_k)\right\}=\bar{p}\left\{(\omega_k)\right\}\quad k=1,2,\ldots m \tag{34}$$

In the modal case, the force is the generalized force

$$\bar{p}\left\{(\omega_k)\right\}=\left[\Phi\right]^T p\left\{(\omega_k)\right\} \tag{35}$$

and the solution vector is the generalized displacement. The physical displacement is found from

$$u\left\{(\omega_k)\right\}=\left[\Phi\right]\bar{u}\left\{(\omega_k)\right\} \tag{36}$$

For the recovery of element forces and stresses, the standard Simcenter Nastran methods are used

Here m denotes the number of excitation frequencies of dynamic load. This is called an asynchronous solution and applicable to cases like gravity loads. The load in this case could still have frequency dependence as shown by the m discrete excitation frequencies defined on the FREQ entry. The rotor speed is constant and the asynchronous analysis is working along a vertical line in the Campbell diagram.

In the case the load is dependent on the speed of rotation (called synchronous analysis) is found by putting $\omega=\Omega$. The governing equation is as follows:

$$\left(-\Omega_k^2\left(\left[\bar{M}\right]-j\left[\bar{C}\right]\right)+j\Omega_k\left(\left[\bar{D}_I+\bar{D}_A\right]-j\left[\bar{D}_B\right]\right)+\left[\bar{K}\right]\right)\left\{u(\Omega_k)\right\}=p\left\{(\Omega_k)\right\}\quad k=1,2,\ldots n \tag{37}$$

Here n denotes the number of Ω_k rotation speeds at which the analysis is executed. Such is the case for example with centrifugal loads due to mass unbalance. In this case analysis is done along the 1P excitation line.

2.9.2 Rotating Reference System:

The governing equation of frequency response for asynchronous solution in rotating reference system is:

$$\left(-\omega_k^2\left[\bar{M}\right]+j\omega_k\left(2\Omega\left[\bar{C}\right]+\left[\bar{D}_I+\bar{D}_A\right]\right)+\left(\left[\bar{K}\right]+\left[\bar{D}_B\right]-\Omega^2\left[\bar{Z}\right]+\Omega^2\left[\bar{K}_G\right]\right)\right)\{u(\omega_k)\}=\{p(\omega_k)\} \quad k=1,2,\dots,m \tag{38}$$

The synchronous analysis in rotating reference system is found along the 0P line, by putting $\omega=0$. This is the response to the forwards whirl:

$$\left(\left(\left[\bar{K}\right]+\left[\bar{D}_B\right]-\Omega_k^2\left[\bar{Z}\right]+\Omega_k^2\left[\bar{K}_G\right]\right)\right)\{u(\Omega_k)\}=\{p(\Omega_k)\} \qquad k=1,2,\dots,n \tag{39}$$

The response to the backwards whirl is found along the 2P excitation line, by inserting $\omega=2\Omega$:

$$\left(\Omega_k^2\left(-4\left[\bar{M}\right]+4j\left[\bar{C}\right]-\left[\bar{Z}\right]+\left[\bar{K}_G\right]\right)+j2\Omega_k\left[\bar{D}_I+\bar{D}_A\right]+\left(\left[\bar{K}\right]+\left[\bar{D}_B\right]\right)\right)\{u(\Omega_k)\}=\{p(\Omega_k)\} \quad k=1,2,\dots,n \tag{40}$$

The user can select any order q of the excitation and generally calculate the following response:

$$\left(-(q\Omega_k)^2\left[\bar{M}\right]+j(q\Omega_k)\left(2\Omega_k\left[\bar{C}\right]+\left[\bar{D}_I+\bar{D}_A\right]\right)+\left(\left[\bar{K}\right]+\left[\bar{D}_B\right]-\Omega_k^2\left[\bar{Z}\right]+\Omega_k^2\left[\bar{K}_G\right]\right)\right)\{u(\Omega_k)\}=\{p(\Omega_k)\} \quad k=1,2,\dots,n \tag{41}$$

This is also the case for the non-rotating (fixed) system.

When studying the response to the forward whirl in the rotating system in a synchronous analysis, a forcing function of 0P must be used. This is simply a constant steady force. When the backward whirl resonance is to be analyzed, a forcing function of 2P must be applied.

2.9.3 Comparison of the Results with the Campbell Diagram

The results of a frequency response analysis can be compared to the Campbell diagram as shown in Fig. 9 for an asynchronous analysis at a rotor speed of 300 Hz. The peaks found with the frequency response analysis (right in the figure) are close to the predicted frequencies in the Campbell diagram (left in the figure). A similar example is shown for a synchronous analysis in Fig. 10. Here the resonance peaks occur around 400 Hz. Both direct and modal analyses were done. The red lines in the response diagrams were found with the direct method (SOL 108) and the blue lines with the modal method (SOL 111). The difference in the curves are due to truncation errors in the modal method. This is a fictive model and the order of magnitude of the results are arbitrarily chosen.

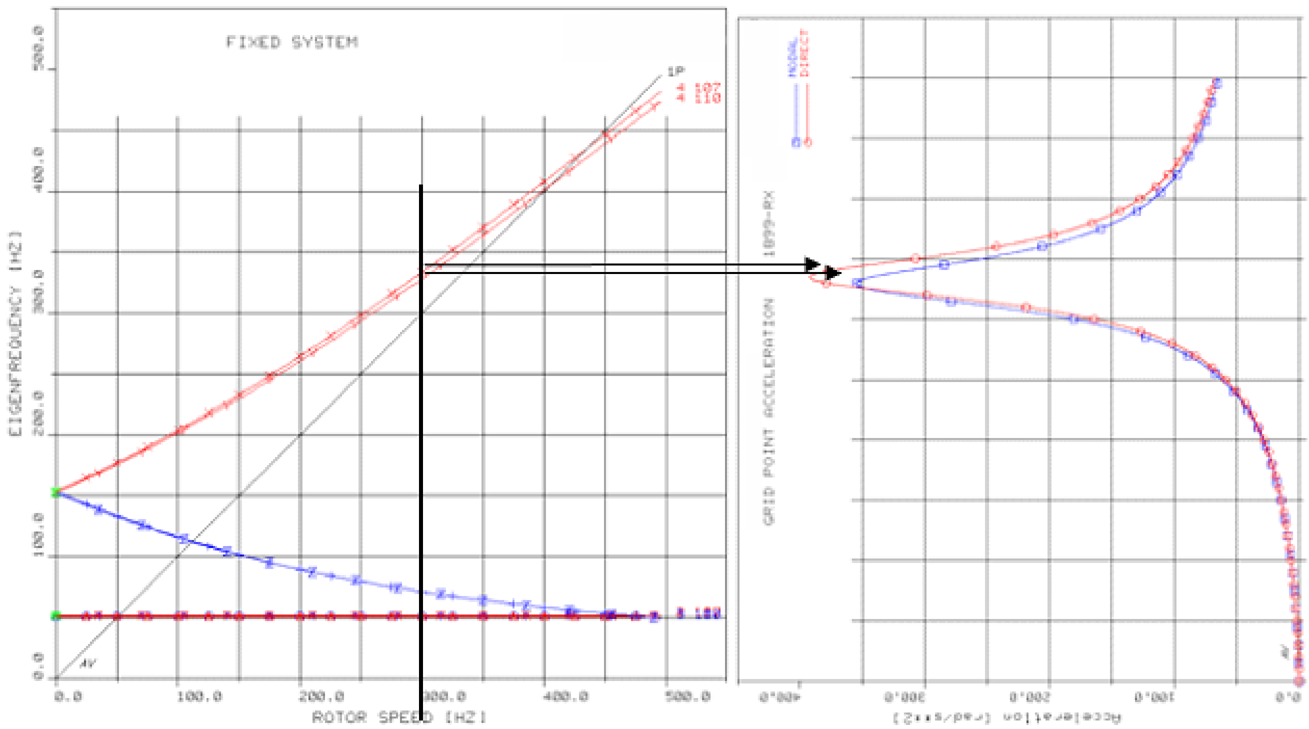


Fig. 9 Resonance peak of the tilting mode for an asynchronous analysis compared to the Campbell diagram

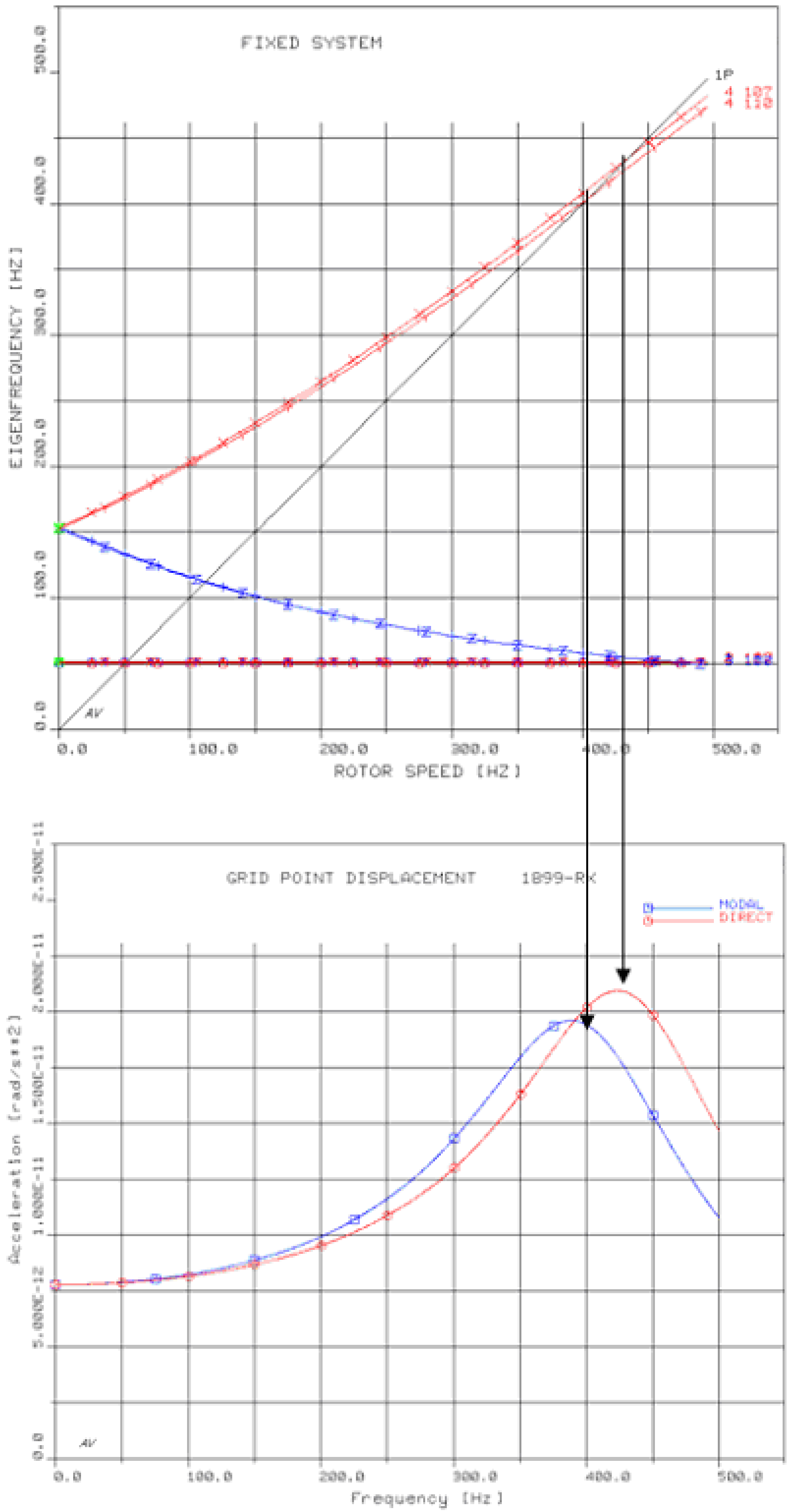


Fig. 10 Synchronous frequency response analysis compared to Campbell diagram

2.10 Equations of Motion for the Transient Response Analysis

The equations of motion are shown for the modal method. The equations for the direct formulation can be found by simply leaving out the bar and using the physical matrices after the MPC and SPC reduction.

2.10.1 Equations of motion for the Fixed System

In a modal transient analysis, the following equation is solved in the fixed reference system:

$$[\bar{M}]\{\ddot{\bar{u}}(t)\}+([\bar{D}_I+\bar{D}_A]+\Omega[\bar{C}])\{\dot{\bar{u}}(t)\}+([\bar{K}]+\Omega[\bar{D}_B])\{\bar{u}(t)\}=\{\bar{p}(t)\} \tag{42}$$

The generalized force is similar to the case of frequency response:

$$\bar{p}\{(t)\}=[\Phi]^T p\{(t)\} \tag{43}$$

The physical displacement method is found from:

$$u\{(t)\}=[\Phi]\bar{u}\{(t)\} \tag{44}$$

For the recovery of element forces and stresses, the standard Simcenter Nastran methods are used.

2.10.2 Equations of motion for the Rotating System

In the rotating reference system, following equation is solved:

$$[\bar{M}]\{\ddot{\bar{u}}(t)\}+([\bar{D}_I+\bar{D}_A]+2\Omega[\bar{C}])\{\dot{\bar{u}}(t)\}+([\bar{K}]+\Omega[\bar{D}_B]+\Omega^2([\bar{K}_G]-\bar{Z})\{\bar{u}(t)\}=\{\bar{p}(t)\} \tag{45}$$

2.10.3 Forcing Function and Initial Conditions

The equations are solved numerically with the standard Simcenter Nastran numerical methods. The initial conditions are equal to zero:

$$\{\bar{u}(0)\}=0, \quad \{\dot{\bar{u}}(0)\}=0 \tag{46}$$

The forcing function must be defined by the user with standard Simcenter Nastran entries.

For the transient analysis, the user must provide an excitation function which is compatible with the time step (TSTEP) used and with the rotor speed values on the ROTORD entry in Simcenter Nastran. For synchronous and asynchronous analysis, a time function with linearly varying frequency must be defined as shown in Fig. 11 .

$$p = p_0 \sin \omega(t)\, t \tag{47}$$

For a linearly varying frequency we have:

$$\omega(t) = 2\pi\, a\, t \tag{48}$$

The slope of the curve is a . The angle is:

$$\varphi(t) = \omega(t)\, t = 2\pi\, a\, t^2 \tag{49}$$

The instantaneous frequency is the time derivative of the angle:

$$\omega = \frac{d\varphi}{dt} = 4\pi\, a\, t \tag{50}$$

In order to obtain a frequency F at time T we have:

$$F = \frac{\omega_{end}}{2\pi} = 2\, a\, T \tag{51}$$

The slope of the curve is then given by:

$$a = \frac{F}{2T} \tag{52}$$

For example to simulate 0 to 500 Hz in 10 seconds, the constant is:

$$a = \frac{500}{2 \cdot 10} = 25$$

At the end of the simulation, the frequency is:

$$F = 2 \cdot 25 \cdot 10 = 500 \text{ Hz}$$

and the period is:

$$T_{end} = \frac{1}{500} = 0.002$$

The time step is now dependent on the desired number of integration points per cycle. A reasonable value is 10 points and hence, a time step of 0.0002 seconds can be used. This means 50,000 time steps for the simulation of 10 seconds. The excitation frequency as function of time is shown in Fig. 11.

Similar to the frequency domain where a rotating force was defined by real and imaginary parts (or 90 degree phase shift) in x- and y-direction, a rotating force can be defined in the time domain by taking sine and cosine functions for the x- and y- axis respectively. Sine and cosine functions are shown for the first second of simulation in Fig. 12. The last 0.01 second of simulation is shown in Fig. 13. Here, the integration points are shown by symbols. These are the points also defined on the TSTEP bulk entry and care must be taken to generate enough points for a period. In the figure shown, there are 10 time steps in the last period which is normally sufficient.

The data must be entered into Simcenter Nastran by means of TABLED1 entries. Pre-processors with function creation capabilities, like Simcenter, can be used to export the TABLED1 entries. Excel can also be used.

When studying the response to the forward whirl in the rotating system a synchronous analysis, a forcing function of 0P must be used. This is simply a constant steady force. When the backward whirl resonance is to be analyzed, a forcing function of 2P must be applied.

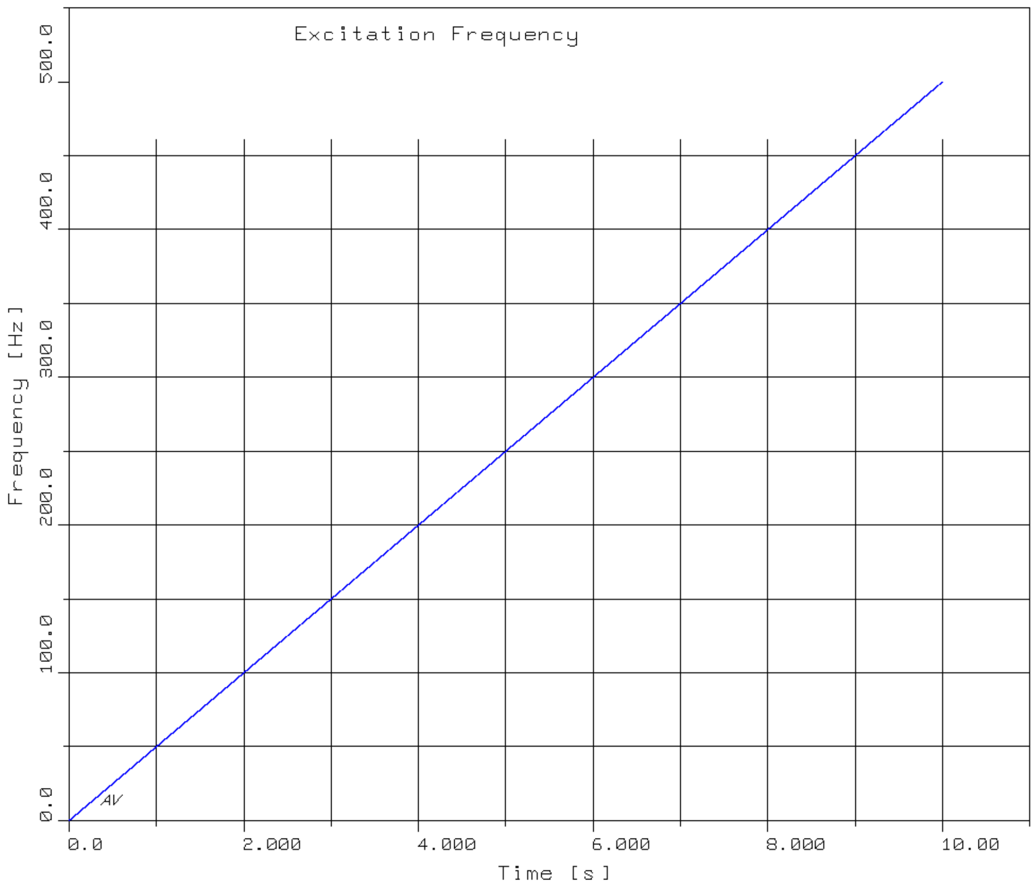


Fig. 11 Frequency as function of time. Sweep function

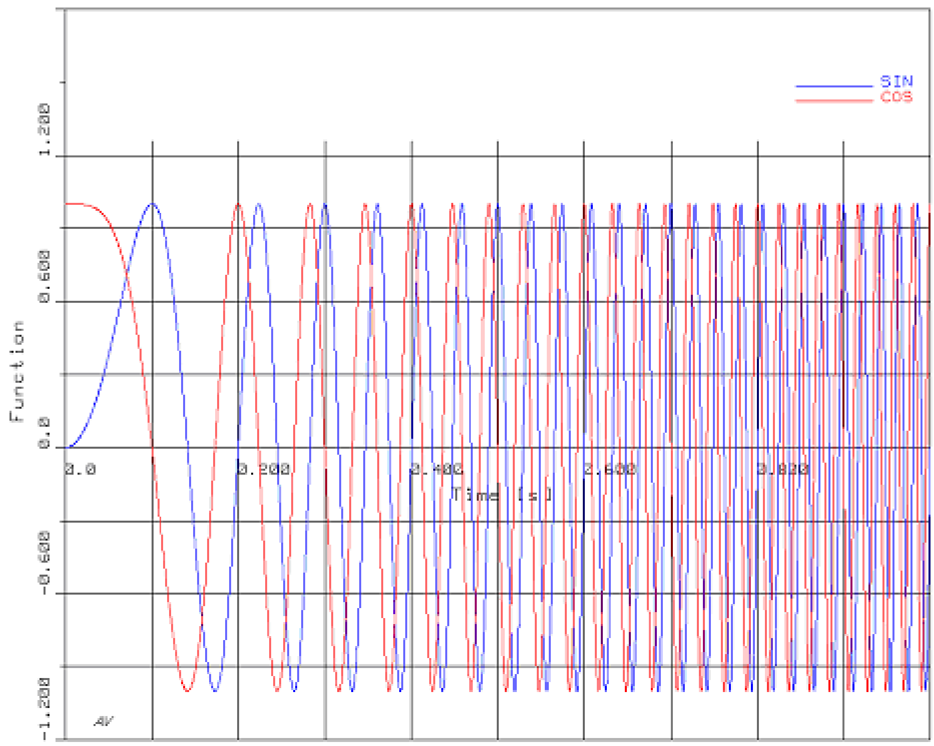


Fig. 12 Time functions during the first second of simulation

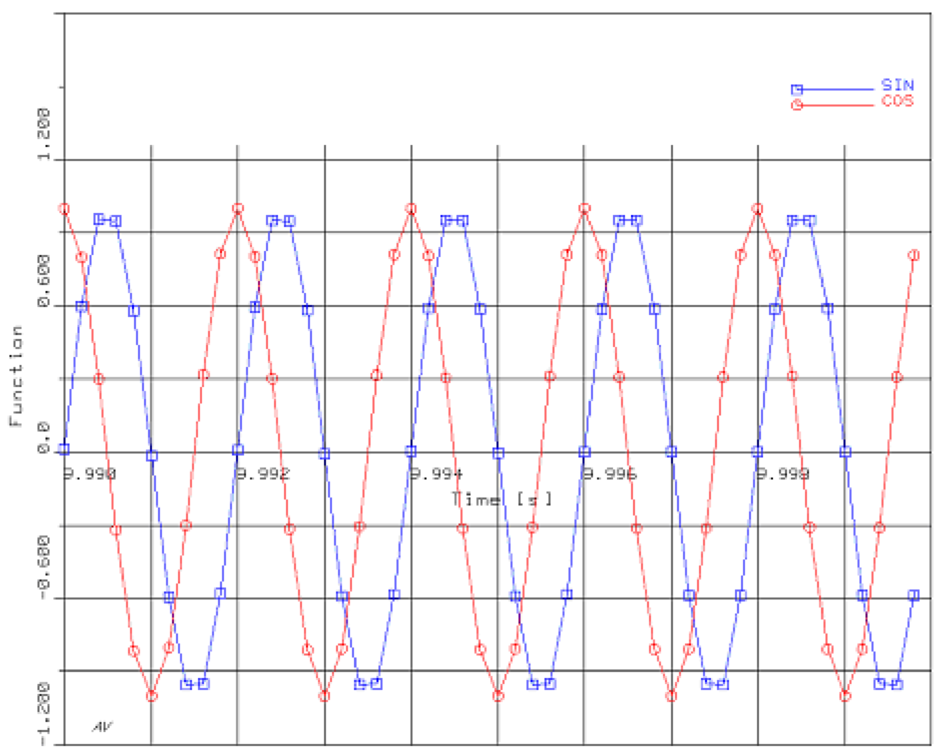


Fig. 13 Time functions during the last 0.01 second of simulation. The frequency is 500 Hz.

2.10.4 Asynchronous Analysis

As for the frequency response, the asynchronous analysis is done for a fixed rotor speed defined on the ROTORD entry. The excitation in a frequency sweep is along a vertical line in the Campbell diagram. The number of values in the sweep function must be equal to the number defined on the TSTEP entry.

2.10.5 Synchronous Analysis

In the synchronous analysis, excitation is along an excitation line in the Campbell diagram. For the fixed system, the critical speed due to an unbalance force is found by exciting the rotor along the 1P line. In the rotating system, the force is constant along the 0p line (abscissa). The backward whirl is excited along the 2P line and the sweep function must be defined with the double frequency. In the synchronous case, the number of rotor speeds on the ROTORD entry must be equal to the number of time step values defined on the TSTEP entry. The system matrices are updated for each rotor speed and a running restart technique is used in the time integration. Therefore, the F04 file can become very large.

2.10.6 Other types of Analysis

The transient response analysis is flexible and the excitation can be chosen independently on the rotor speed. For example, a constant excitation frequency can be applied to a synchronous analysis with variable rotor speed. This is equivalent to a horizontal line in the Campbell diagram. The EORDER field on the ROTORD entry has no effect on a transient analysis.

Also, a non-harmonic forcing function like an impulse can be applied.

2.10.7 Comparing the Results with the Campbell Diagram

For a transient response analysis, a comparison with the Campbell diagram should be made. A comparison is shown in Fig. 14. The time response curve is shown at the upper left side. A line at the time with maximum amplitudes is drawn to the figure below which shows the sweep frequency as function of time. Finally, a horizontal line is drawn from this frequency to the Campbell diagram in the lower right part of the figure. There a vertical line is drawn at the rotor speed of the asynchronous analysis at 300 Hz in this case. The crossing point with the forward whirl motion of the tilting mode corresponds to the frequency found in the transient analysis. A similar plot for a synchronous analysis is shown in Fig. 15. Here the excitation is along the 1P line and the time with maximum frequency corresponds to the 1P crossing of the translation mode in the Campbell diagram.

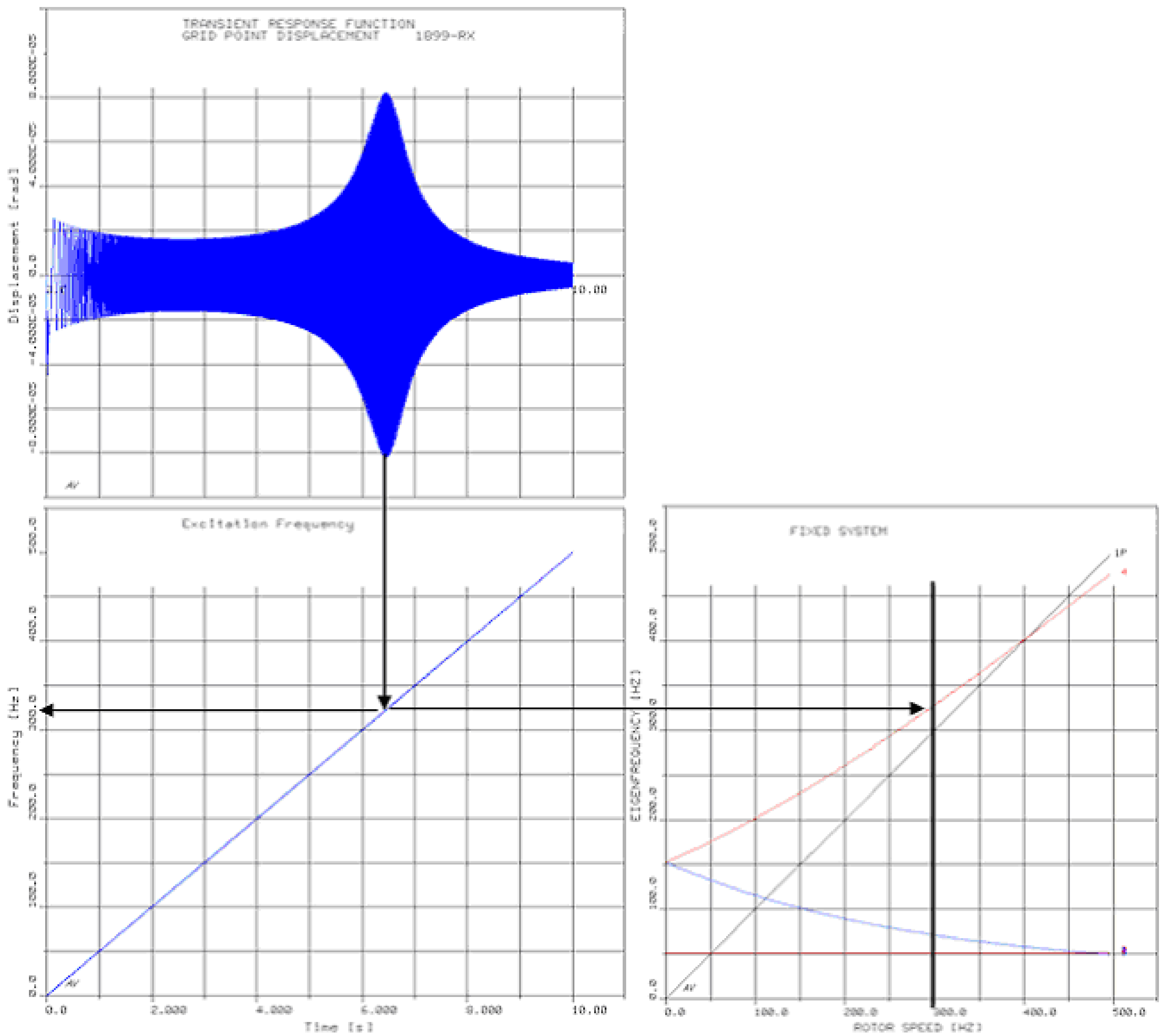


Fig. 14 Transient response of the tilting motion for an asynchronous analysis at 300 Hz rotor speed

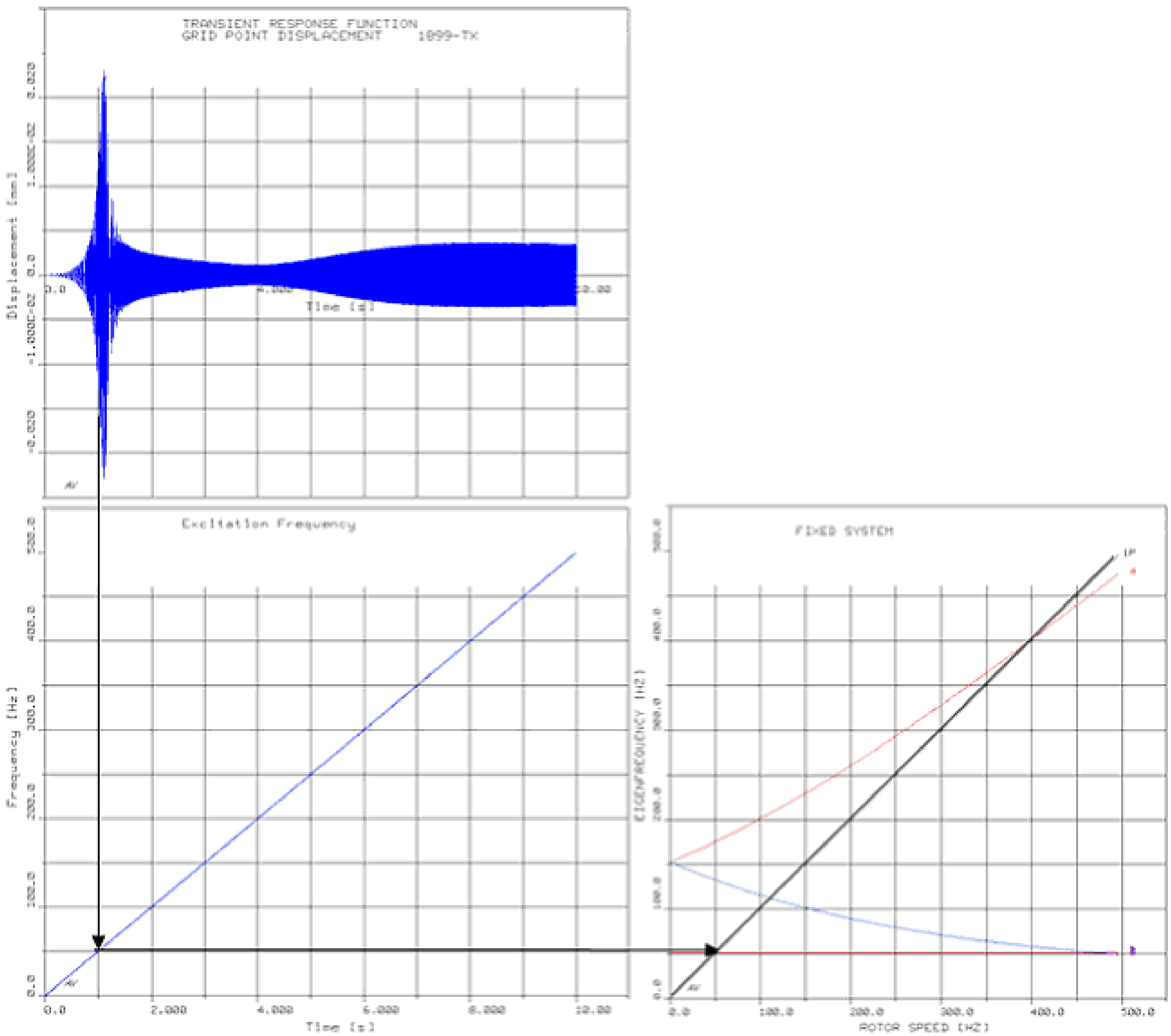


Fig. 15 Running through the translation peak at around 50 Hz

2.10.8 Influence of the Sweep Velocity

In the frequency response analysis the steady-state solution is calculated. In the transient analysis, the structure is excited by a sweep function. Steady-state amplitudes are only reached for a very slowly varying frequency. If the sweeping function is too fast, the steady-state amplitudes will not be reached.

Fig. 16 shows results with a fast sweep function. Amplitudes are lower for this case and the highest amplitudes occur at a later time because the response of the structure is delayed. In reality, a rotor passing a critical speed fast will usually not come into high amplitudes as a rotor passing slowly through the resonance point.

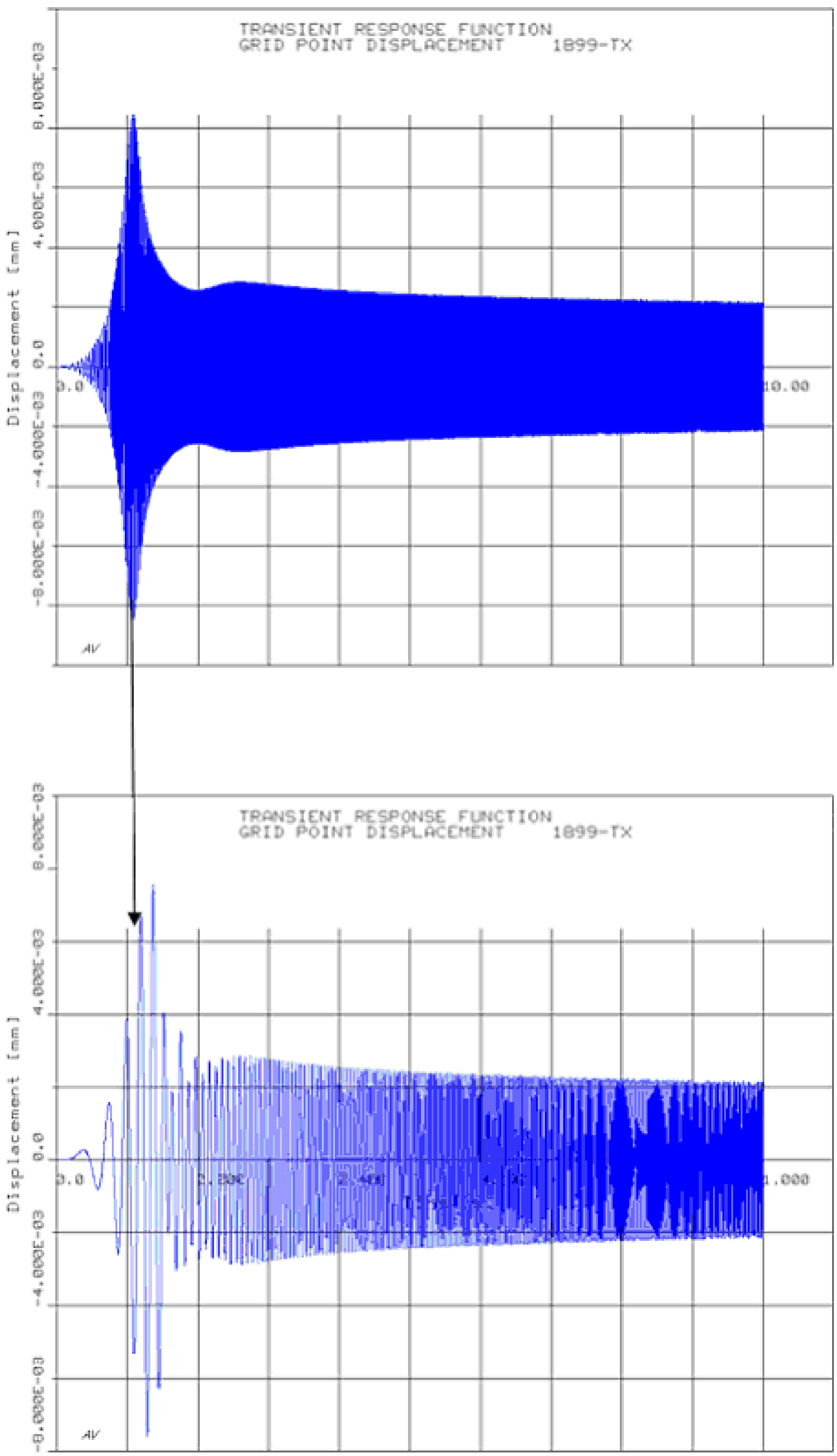


Fig. 16 Transient analysis with a fast sweep function

2.10.9 Instabilities

In frequency analysis, there are also solutions for the case of an unstable structure. The damping is then negative and there can be a peak. In transient analysis, the solution will diverge in the case of an unstable system. Fig. 17 shows a case where the rotor is first passing a resonance (critical speed) and then enters into an unstable condition.

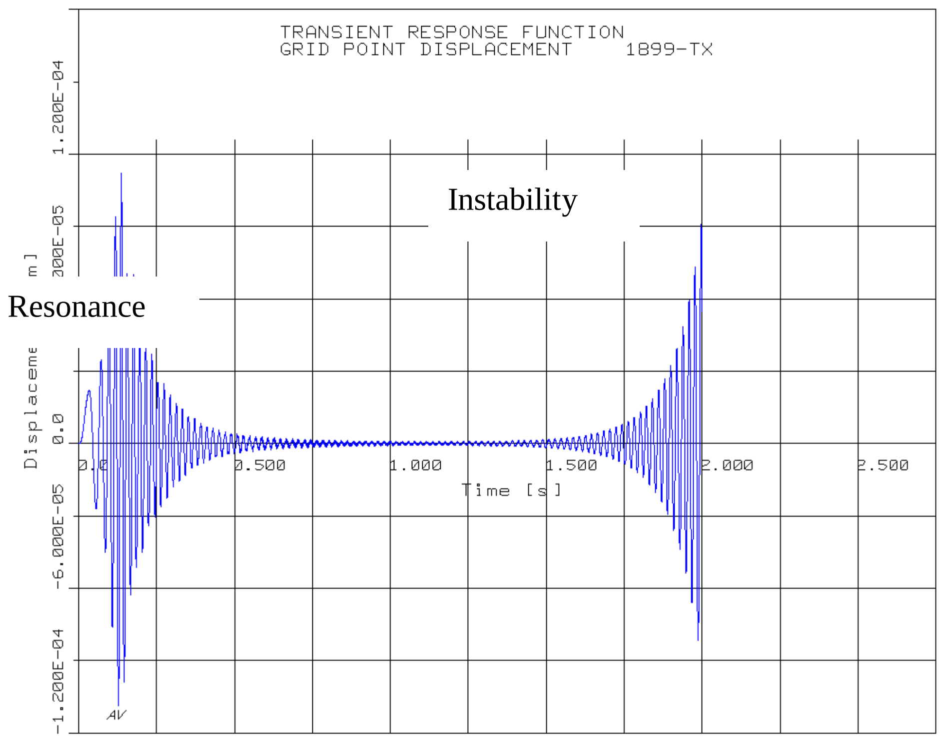


Fig. 17 Structure running through a resonance and entering into an instability

2.10.10 Initial Conditions

For all transient analysis, the initial conditions are zero (displacement and velocity). Thus, it will take some time until steady-state amplitudes are reached. In the example shown in Fig. 18, it takes 2 seconds of time until the rotor reaches the steady-state amplitudes. When the simulation starts at zero speed or zero frequency this is not important but when starting at a certain speed the effect can be significant.

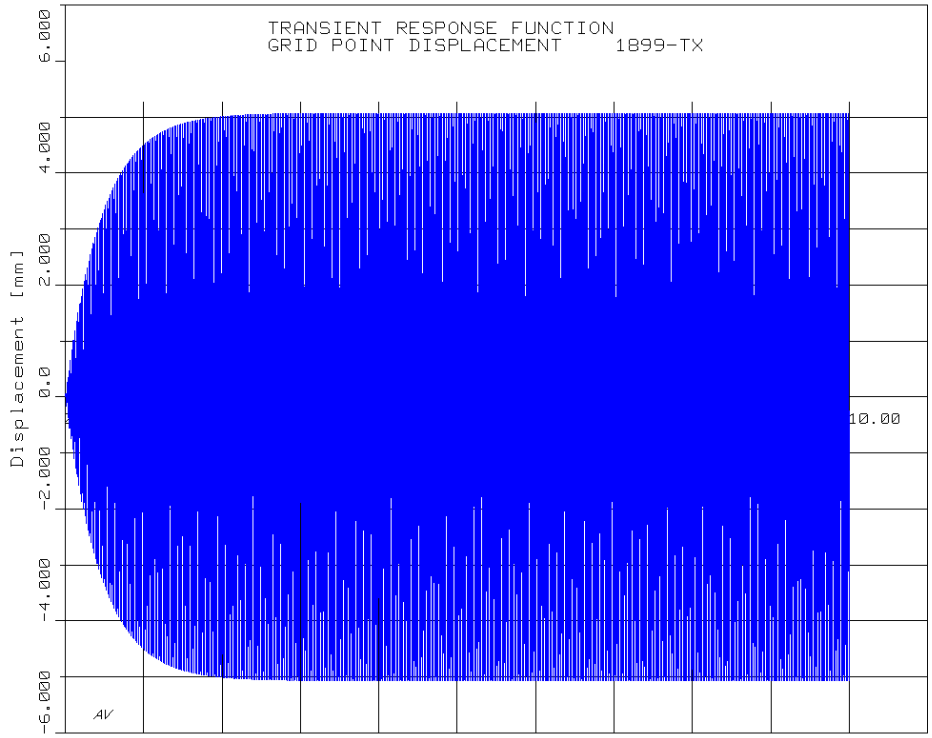


Fig. 18 Integration from zero initial conditions until steady-state response

2.11 Gyroscopic Moments in Maneuver Load Analysis

Maneuver load analyisis is a SOL 101 linear static structural analysis that accounts for inertial loads. Maneuver load analysis is commonly performed in the aircraft industry.

In maneuver load analysis, the inertial loads are calculated from grid point accelerations. The grid point accelerations are calculated from the rigid-body motion you specify for the model using the ACCELERATION, ACCELERATION1, RFORCE, RFORCE1, and RFORCE2 bulk entries.

If the model contains rotors and the rigid body motion causes the axis of rotation of the rotor to change, gyroscopic moments result. For example, gyroscopic moments arise when the axis of rotation of a gas turbine engine changes during certain aircraft maneuvers. The rotor dynamics capabilities in Simcenter Nastran allow you to account for gyroscopic moments in a maneuver load analysis.

To use the Simcenter Nastran rotor dynamics capability, include a ROTORD bulk entry and either a RFORCE, RFORCE1, or RFORCE2 bulk entry in the input file for your maneuver load analysis. On the ROTORD bulk entry, specify ‘FIX’ in the REFSYS field.

Gyroscopic moments are computed for the grid points that are included on ROTORG bulk entries. Simcenter Nastran computes the gyroscopic moment from:

$$\{F_g\} = -\Omega[C]\{\omega\}$$

where Ω is the angular velocity of the rotor, $[C]$ is the gyroscopic matrix, and $\{\omega\}$ is the rigid body angular velocity vector for the model.

The rigid body angular velocity vector is given by:

$$\{\omega\} = 2\pi \begin{Bmatrix} \dot{\alpha}_x \\ \dot{\alpha}_y \\ \dot{\alpha}_z \end{Bmatrix}$$

You specify the components of the rigid body angular velocity vector with an RFORCE, RFORCE1, or RFORCE2 bulk entry. Use the RFORCE bulk entry to apply the angular motion to the entire model. Use the RFORCE1 bulk entry to apply the angular motion to a subset of the model. Use the RFORCE2 bulk entry to apply the angular motion to the entire model, but exclude either the gyroscopic or centrifugal forces.

For the gyroscopic moment calculation, Simcenter Nastran uses the value specified in the RSTART field of the ROTORD bulk entry as the angular velocity of the rotor.

2.12 Coupled, Time-Dependent Solutions

The Simcenter Nastran implementation of a fully coupled equation of motion uses the following form:

$$\begin{aligned}
&\left(\begin{bmatrix} M_\sigma & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 \\ 0 & 0 & M_\rho & 0 \\ 0 & 0 & 0 & M_\alpha \end{bmatrix} + m_r \begin{bmatrix} 0 & B_0 & H & 0 \\ B_0^T & B_0^T B_0 & B_0^T H & 0 \\ H^T & H^T B_0 & 0 & 0 \\ 0 & 0 & 0 & 0 \end{bmatrix} \right) \begin{Bmatrix} \ddot{\sigma} \\ \ddot{\beta} \\ \ddot{\rho} \\ \ddot{\alpha} \end{Bmatrix} + \\
&+2\Omega \left(\begin{bmatrix} 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 \\ 0 & 0 & C_\rho & 0 \\ 0 & 0 & 0 & C_\alpha \end{bmatrix} + m_r \begin{bmatrix} 0 & \bar{B}_0 & \bar{H} & 0 \\ 0 & B_0^T \bar{B}_0 & B_0^T \bar{H} & 0 \\ 0 & H^T \bar{B}_0 & 0 & 0 \\ 0 & 0 & 0 & 0 \end{bmatrix} \right) \begin{Bmatrix} \dot{\sigma} \\ \dot{\beta} \\ \dot{\rho} \\ \dot{\alpha} \end{Bmatrix} - \\
&-\Omega^2 \left(\begin{bmatrix} 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 \\ 0 & 0 & Z_\rho & 0 \\ 0 & 0 & 0 & Z_\alpha \end{bmatrix} - m_r \begin{bmatrix} 0 & \bar{\bar{B}}_0 & \bar{\bar{H}} & 0 \\ 0 & B_0^T \bar{\bar{B}}_0 & B_0^T \bar{\bar{H}} & 0 \\ 0 & H^T \bar{\bar{B}}_0 & 0 & 0 \\ 0 & 0 & 0 & 0 \end{bmatrix} \right) \begin{Bmatrix} \sigma \\ \beta \\ \rho \\ \alpha \end{Bmatrix} = \Omega^2 \begin{Bmatrix} -m_r \bar{\bar{H}}\{r\} \\ -m_r B_0^T \bar{\bar{H}}\{r\} \\ f_{c\rho} \\ f_{c\alpha} \end{Bmatrix}
\end{aligned} \tag{53}$$

This equation represents only the effects of the rotational part and is written in terms of a single rotating grid point. Equations of this type are assembled for the complete structure and added to the mass, inertia, stiffness, and viscous damping matrices of the fixed part. To adhere to standard notational convention, a double negative is used to separate the constant and time-dependent terms of the centrifugal matrix.

The constant terms include:

$$M_\sigma = m_r I, \quad M_\rho = m_r I, \quad C_\rho = m_r P^T, \quad Z_\rho = m_r J^T$$

where

$$[I] = \begin{bmatrix} 1 & 0 & 0 \\ 0 & 1 & 0 \\ 0 & 0 & 1 \end{bmatrix}, \ [J] = \begin{bmatrix} 1 & 0 & 0 \\ 0 & 1 & 0 \\ 0 & 0 & 0 \end{bmatrix}, \ [P] = \begin{bmatrix} 0 & 1 & 0 \\ -1 & 0 & 0 \\ 0 & 0 & 0 \end{bmatrix}$$

The coupling mass, gyroscopic, and centrifugal matrices are:

$$M_\alpha = \begin{bmatrix} \Theta_x & & \\ & \Theta_y & \\ & & \Theta_z \end{bmatrix}, C_\alpha = \begin{bmatrix} 0 & -(\Theta_x + \Theta_y - \Theta_z)/2 & \\ (\Theta_x + \Theta_y - \Theta_z)/2 & 0 & \\ & & 0 \end{bmatrix}, Z_\alpha = \begin{bmatrix} \Theta_y - \Theta_z & & \\ & \Theta_x - \Theta_z & \\ & & 0 \end{bmatrix}$$

where the Θ terms are the inertias associated with the rotating point relative to the indicated directions.

The other submatrices in the above equations include:

$$[H] = \begin{bmatrix} cos\,\Omega t & -sin\,\Omega t & 0 \\ sin\,\Omega t & cos\,\Omega t & 0 \\ 0 & 0 & 1 \end{bmatrix}$$

and its time derivatives:

$$\dot{H} = \Omega \bar{H}, \ddot{H} = \Omega^2 \bar{\bar{H}}$$

and:

$$B_0 = \begin{bmatrix} & & -x \\ & & -y \\ x & y & 0 \end{bmatrix} \sin \Omega t + \begin{bmatrix} & & -y \\ & & x \\ y & -x & \end{bmatrix} \cos \Omega t$$

and its time derivatives:

$$\bar{B}_0 = \dot{B}_0, \bar{\bar{B}}_0 = \ddot{B}_0$$

In these matrices, *x* and *y* are the offset distances of the rotating grid point from the coupling point.

The $\begin{Bmatrix} \sigma \\ \beta \\ \rho \\ \alpha \end{Bmatrix}$ vector and its time derivatives represent the translational and rotational degrees of freedom of the stationary coupling point and the rotating point, respectively. Hence the vector has a total of 12 entries. You specify the stationary coupling point with the ROTCOUP parameter.

On the right-hand side of the equation, $f_{c\rho}, f_{c\alpha}$ are the centrifugal forces that act on the rotating point whose coordinates are are given by $\{ r \}$.

The coupled equations are solved at various values of Ωt. These angles are referred to as azimuth angles. You specify the range of azimuth angles with the PHIBGN, PHINUM, and PHIDEL parameters.

2.13 Gyroscopic and Circulation Term Scaling

For frequency response and complex eigenvalue rotor dynamic analysis, you can optionally include gyroscopic and circulation terms in the mass, damping, and stiffness matrices when the analysis is performed in the fixed reference system by specifying PARAM,RLOOPNEW,YES.

The combined rotor mass, damping, and stiffness contributions, including gyroscopic and circulation terms, are shown below for both frequency response and complex eigenvalue analysis.

- Frequency response analysis:

$$M_{rotors} = \sum_{j=1}^{n} M_j^R$$

$$B_{rotors} = \sum_{j=1}^{n} B_j^R + \Omega_j(\Omega_{ref}) B_j^G$$

$$K_{rotors} = \sum_{j=1}^{n} (1 + \mathrm{i}G) K_j^R + \mathrm{i}K4_j^R + \Omega_j(\Omega_{ref}) \left(K_j^{CB} + \left(\frac{G}{\omega}\right) K_j^{CK} + \left(\frac{1}{\omega}\right) K_j^{CK4} \right)$$

- Asynchronous complex eigenvalue analysis (Ω_{ref} = constant):

$$M_{rotors} = \sum_{j=1}^{n} M_j^R$$

$$B_{rotors} = \sum_{j=1}^{n} B_j^R + \left(\frac{G}{W3\_j}\right) K_j^R + \left(\frac{1}{W4\_j}\right) K4_j^R + \Omega_j(\Omega_{ref}) B_j^G$$

$$K_{rotors} = \sum_{j=1}^{n} K_j^R + \Omega_j(\Omega_{ref})\left(K_j^{CB} + \left(\frac{G}{W3\_j}\right)K_j^{CK} + \left(\frac{1}{W4\_j}\right)K_j^{CK4}\right)$$

- Synchronous complex eigenvalue analysis ($\Omega_{ref} = \omega$):

$$M_{rotors} = \sum_{j=1}^{n} M_j^R - i\alpha_j B_j^G$$

$$B_{rotors} = \sum_{j=1}^{n} B_j^R + \left(\frac{G}{W3_j}\right)K_j^R + \left(\frac{1}{W4_j}\right)K4_j^R - i\alpha_j\left(K_j^{CB} + \left(\frac{G}{W3\_j}\right)K_j^{CK} + \left(\frac{1}{W4\_j}\right)K_j^{CK4}\right)$$

$$K_{rotors} = \sum_{j=1}^{n} K_j^R$$

In the above equations, the following nomenclature is used:

M^R	Rotor mass
B^R	Rotor viscous damping
K^R	Rotor stiffness
$K4^R$	Rotor hysteretic damping (GE on MATi entries)
B^G	Rotor gyroscopic terms
K^{CB}	Rotor circulation terms due to rotor viscous damping
K^{CK}	Rotor circulation terms due to structural damping (PARAM,G)
K^{CK4}	Rotor circulation terms due to hysteretic damping (GE on MATi entries)
G	Structural damping value (PARAM,G)
$W3\_j$	Rotor reference frequency for structural damping in rad/sec
$W4\_j$	Rotor reference frequency for material damping in rad/sec
ω	Excitation frequency in rad/sec
Ω	Rotor speed in rad/sec
Ω_{ref}	Rotor speed in rad/sec
α	Ratio of rotor speed to reference rotor speed (based on RSTART value on ROTORD entry)

CHAPTER

3

Defining Simcenter Nastran Input for Rotor Dynamics

3 Defining Simcenter Nastran Input for Rotor Dynamics

This chapter describes the inputs required to create a valid Simcenter Nastran input file for a rotor dynamic analysis.

3.1 File Management Section

In the File Management section of the input file, use assign statements to specify names for the .gpf and .csv output files the software generates. For example:

```
assign output4='filename.gpf',unit=22, form=formatted
```

```
assign output4='filename.csv',unit=25, form=formatted
```

More information regarding output files is provided in the “Interpretation of Rotor Dynamics Output” chapter. If the Campbell and damping diagrams are supported by the post-processor, these files are not needed.

3.2 Executive Control Section

In the Executive Control section of the input file, specify the appropriate solution sequence for the rotor dynamic analysis. SOLs 101 and 107-112 are valid solution sequences:

SOL 101 is the Linear Static solution sequence.

SOL 107 is the Direct Complex Eigenvalues solution sequence.

SOL 110 is the Modal Complex Eigenvalues solution sequence.

SOL 108 is the Direct Frequency Response solution sequence.

SOL 111 is the Modal Frequency Response solution sequence.

SOL 109 is the Direct Transient Response solution sequence.

SOL 112 is the Modal Transient Response solution sequence.

3.3 Case Control Section

In the Case Control section above any subcases, use the RMETHOD case control command to invoke the rotor dynamics capability. The format of the RMETHOD command is as follows:

```
RMETHOD(CMR=m)=n
```

where n references the ROTORD bulk entry to use for the rotor dynamic analysis.

The optional CMR describer is only valid when SOL 107 is used. Specifying the CMR describer invokes complex modal reduction. The value m references the EIGC bulk entry that specifies the complex eigenvalue method and related parameters for the software to use during the complex modal reduction.

When you use complex modal reduction, the software computes the complex modes and uses them to project the problem into modal space where, depending on the number of modes computed, the problem size is typically reduced. The complex modal reduction is performed at a reference rotor speed of zero unless you explicitly specify a different reference rotor speed. To specify a different reference rotor speed, use the ROTCMRF parameter.

The software then performs an eigensolution on the reduced problem at each reference rotor speed. You can expect the most accurate results to occur at reference rotor speeds near the reference rotor speed at which the complex modal reduction is performed. The software then projects the results of the eigensolution back into physical space for presentation and subsequent post-processing.

Complex modal reduction is applicable to problems containing unsymmetric stiffness and unsymmetric viscous damping. This makes a SOL 107 solve with complex modal reduction ideal for reducing the computational effort required to solve rotor dynamics problems that contain sources of unsymmetric stiffness and unsymmetric viscous damping like journal bearings.

In rotor dynamic analysis, you first perform a real eigenvalue analysis and examine the modes before proceeding to the next step in the analysis. As such, it is helpful to be able to select or deselect local modes or other modes, such as axial modes.

You can use the MODSEL case control command to optionally exclude specific modes. The format of the MODSEL case control command is as follows:

```
MODSEL=n
```

where n refers to a SET case control command that lists the mode numbers to retain. The default is to retain all modes. The mode numbers that are omitted from the list are removed from the modal space. Mode numbers larger than the number of eigenvalues computed are ignored.

This option is also very useful for studying the influence of specific modes.

3.4 Bulk Section

The ROTORD bulk entry is required in a rotor dynamic analysis. The ROTORD entry is used to define relevant rotor dynamics data. With the ROTORD entry, the first line and the first continuation line contain rotor dynamics data common to all the rotors in the system. Rotor-specific data is contained on additional continuation lines with one additional continuation line needed for each rotor. You can specify up to ten rotors in a system.

With the ROTORD entry, you can specify the following data that is common to all rotors in the model:

- The starting reference rotor speed, speed step, number of speed steps, and reference rotor speed units are specified using the RSTART, RSTEP, NUMSTEP, and RUNIT fields, respectively.

- Whether to perform the analysis in the fixed or rotating reference system is specified using the REFSYS field.
- Whether to perform a synchronous or an asynchronous analysis for frequency response is specified using the SYNC field.
- Whether or not to include Steiner's inertia terms is specified using the ZSTEIN field.
- The excitation type and excitation order are specified using the ETYPE and EORDER fields, respectively.
- The rotor speed for complex mode output, the units for frequency output, and .f06 file output options are specified using the CMOUT, FUNIT, and ROTPRT fields, respectively.
- The threshold value for the detection of whirl direction is specified using the ORBEPS field.
- When bearing stiffness, viscous damping, or mass is dependent on speed and displacement, or speed and force, the threshold value to evaluate solution convergence is specified using the THRSHOLD field. The maximum number of iterations permitted is specified using the MAXITER field.

With the ROTORD entry, you can specify the following rotor-specific data:

- The grids associated with each rotor are specified by referencing the RSETID field of ROTORG bulk entries from the RSETi fields.
- The stationary grids of bearings are also specified by referencing the RSETID field of ROTORB bulk entries from the RSETi fields. For each rotor, use the same identification number in the RSETID fields of the corresponding ROTORB and ROTORG bulk entries.
- The multiplier of the reference rotor speed that the software uses to determine the speed for each rotor is specified using the RSPEEDi fields. Use real input to specify fixed multipliers, or use integer input to reference TABLEDi bulk entries that contain speed-dependent multiplier data.

  For other options for specifying the relative speeds of rotors, see the Rotor Speed Specification Options section.
- The Cartesian coordinate system whose Z-axis defines the axis of rotation for each rotor is specified using the RCORDi fields.
- The reference frequencies for structural damping are specified using the W3_i and W4_i fields.
- The RFORCE, RFORCE1, or RFORCE2 bulk entry to use with each rotor is specified using the RFORCEi fields.
- The CBEAR elements that support each rotor are specified by referencing a GROUP bulk entry in the BRGSETi field. A distinct GROUP bulk entry is needed for each rotor that is supported by CBEAR elements.

For additional information on the ROTORD bulk entry, see the *Simcenter Nastran Quick Reference Guide*.

The ROTORG bulk entry is used to specify the rotating portions of the model. If the model contains multiple rotors, you must include a ROTORG entry for each rotor and the RSETi field on the ROTORD entry must reference the RSETID field on the corresponding ROTORG entry.

If the model contains only a single rotor and portions of the model are stationary, you must include a single ROTORG entry and the RSET1 field on the ROTORD entry must reference the RSETID field on the ROTORG entry. The software assumes any grids not specified on the ROTORG entry are stationary.

If the entire model is rotating as a single rotor, you can leave the RSET1 field blank on the ROTORD entry and omit including a ROTORG entry in the input file. In the absence of a ROTORG entry, the software assumes all grids are rotating.

3.4.1 Modeling Bearing Supports

You can use CBEAR elements to model the stiffness, damping, and inertial characteristics of bearings. You can use CBUSH, CELASi, and CDAMPi elements to model the stiffness and damping characteristics of bearings. As a best practice, use CBEAR elements to model bearings because unlike CBUSH, CELASi, and CDAMPi elements, CBEAR elements allow you to:

- Model unsymmetric bearing stiffness, damping, and mass (for example, that might result from journal bearings).
- Define not only constant and speed-dependent bearing stiffness, damping, or mass, but also speed and displacement-dependent, and speed and force-dependent bearing stiffness, damping, or mass.
- Include coupling terms for bearing stiffness, damping, or mass between the radial directions, between the radial directions and the axial direction, and between the rotational directions.
- Use composite radial relative displacement or force, composite axial relative displacement or force, or composite rotational relative displacement or force to look up values for bearing stiffness, damping, or mass.

In manuever load analysis, if you use CBEAR elements to model bearing supports, the bearing stiffness must be symmetric. If the bearing stiffness is not symmetric, the software automatically makes it symmetric by averaging the element stiffness and transpose of the element stiffness for each CBEAR element.

MODELING INDIVIDUAL ROTORS

The following procedure for modeling bearing supports is applicable to the following situations:

- Individual rotors that are connected to either a supporting structure or ground.

- Coaxial rotors where each rotor is connected to either a supporting structure or ground, and there is no interconnection between the rotors.

If the rotor is modeled with line elements, define three coincident grids along the axis of rotation of the rotor at the axial position of each bearing.

At each bearing location, include the grid that is used to define the line element connectivity on the ROTORG entry for the rotor. The software interprets this grid as the rotating grid. Include another coincident grid on the ROTORB entry for the rotor. The software interprets this grid as the stationary grid. Between these two grids, define an RBE2 element. The RBE2 element is necessary for the software to correctly partition the rotating and the stationary portions of the model. Use the grid that is listed on the ROTORB entry and the third coincident grid to define the connectivity of the CBEAR element, CBUSH element, or CELASi and CDAMPi elements.

If the rotor is modeled with shell or solid elements, mesh the rotor such that at each bearing location the element edges and element faces of the mesh form a cross section through the rotor. Define three coincident grids along the axis of rotation at the axial location of each bearing.

At each bearing location, include one of the coincident grids on the ROTORG entry for the rotor. The software interprets this grid as the rotating grid. Include another coincident grid on the ROTORB entry for the rotor. The software interprets this grid as the stationary grid. Between these two grids, define an RBE2 element. The RBE2 element is necessary for the software to correctly partition the rotating and the stationary portions of the model. Use the grid that is listed on the ROTORB entry and the third coincident grid to define the connectivity of the CBEAR element, CBUSH element, or CELASi and CDAMPi elements.

Connect the grids lying in the cross section of the mesh to the grid listed on the ROTORG entry with an RBE3 element. When you do so, define the grid listed on the ROTORG entry as the reference grid for the RBE3 element.

MODELING INTERCONNECTED COAXIAL ROTORS

The following procedure for modeling bearing supports is applicable to coaxial rotors where the rotors are interconnected. For example, if the outer rotor is connected to ground and the inner rotor is supported by the outer rotor, the rotors are interconnected.

When you have interconnected coaxial rotors, the procedure to model the rotor that is connected to either a supporting structure or ground is exactly the same as:

- An individual rotor that is connected to either a supporting structure or ground.
- A noninterconnected coaxial rotor that is connected to either a supporting structure or ground.

For the coaxial rotor that is supported by the other coaxial rotor, the procedure is slightly different. If the coaxial rotors are modeled with line elements, define three coincident grids along the axis of rotation of the supported coaxial rotor at the axial position of each bearing.

At each bearing location, include the grid that is used to define the line element connectivity on the ROTORG entry for the supported coaxial rotor. The software interprets this grid as rotating with the supported coaxial rotor. Then include the other two coincident grids on the ROTORB entry for both coaxial rotors. The software interprets these grids as rotating with the coaxial rotor that is connected to either a supporting structure or ground.

Between the coincident grid that is listed on the ROTORG entry for the supported coaxial rotor and one of the other coincident grids that are listed on the ROTORB entries, define an RBE2 element. Between the two grids that are listed on the ROTORB entries, define the connectivity of the CBEAR element, CBUSH element, or CELASi and CDAMPi elements.

Define a second RBE2 element between the grid that is part of the CBEAR, CBUSH, or CELASi and CDAMPi element connectivity, but not part of the connectivity of the RBE2 element that you already defined, and a forth coincident grid that is listed on the ROTORG entry for the rotor that is connected to either a supporting structure or ground. This grid may or may not be used in the definition of the bearing support that connects the other coaxial shaft to either a supporting structure or ground.

Both RBE2 elements are necessary for the software to correctly partition the model.

If the coaxial rotors are modeled with shell or solid elements, mesh the rotor such that at each bearing location the element edges and element faces of the mesh form a cross section through the rotor. Define three coincident grids along the axis of rotation of the supported coaxial rotor at the axial position of each bearing.

At each bearing location, include one of the coincident grids on the ROTORG entry for the supported coaxial rotor. The software interprets this grid as rotating with the supported coaxial rotor. Then include the other two coincident grids on the ROTORB entry for both coaxial rotors. The software interprets these grids as rotating with the coaxial rotor that is connected to either a supporting structure or ground.

Between the coincident grid that is listed on the ROTORG entry for the supported coaxial rotor and one of the other coincident grids that are listed on the ROTORB entries, define an RBE2 element. Between the two grids that are listed on the ROTORB entries, define the connectivity of the CBEAR element, CBUSH element, or CELASi and CDAMPi elements.

Define a second RBE2 element between the grid that is part of the CBEAR, CBUSH, or CELASi and CDAMPi element connectivity, but not part of the connectivity of the RBE2 element that you already defined, and another coincident grid that is listed on the ROTORG entry for the rotor that is connected to either a supporting structure or ground. This grid may or may not be used in the definition of the bearing support that connects the other coaxial shaft to either a supporting structure or ground.

Once again, both RBE2 elements are necessary for the software to correctly partition the model.

For the bearing supports between interconnected rotors, the software enters the stiffness and damping of CBUSH, CELASi, and CDAMPi elements twice when formulating the stiffness and damping matrices. Thus, when you use these elements to model a bearing support between interconnected rotors, halve the numerical value for the stiffness and damping on the property bulk entries that they reference. You do not need to do this when you use CBEAR elements to model bearing supports between interconnected rotors.

To associate CBEAR elements with a specific rotor, define a GROUP bulk entry for each rotor. On each GROUP entry, list all the CBEAR elements associated with the rotor. Then list the identification number of the GROUP entry in the corresponding BRGSETi field of the ROTORD entry. For interconnected coaxial rotors, CBEAR elements between the coaxial rotors must be included on the GROUP entries for both coaxial rotors.

If the model contains only one rotor, you do not need to associate the CBEAR elements with the rotor. By default, the software assumes that the CBEAR elements are associated with the rotor.

If you use CBUSH elements, or CELASi and CDAMPi elements to model the bearing supports for a rotor, you do not need to use a GROUP bulk entry to associate them with a specific rotor.

To associate the ROTORB entry for a rotor with the ROTORG entry for the same rotor, use the same identification number in the RSETID fields of both.

If the model contains only one rotor, you do not need to list the stationary grids on a ROTORB entry. By default, the software assumes that the coincident grids not listed on the ROTORG entry are stationary.

DEFINING PROPERTIES FOR CBEAR ELEMENTS

You define the stiffness, damping, and mass for each CBEAR element with a PBEAR bulk entry. With the PBEAR bulk entry, you can define the following stiffness, damping, and mass terms:

- Radial translation in the X- and Y-directions of the coordinate system that is referenced in the RCORDi field of the ROTORD bulk entry.
- Axial translation in the Z-direction of the coordinate system that is referenced in the RCORDi field of the ROTORD bulk entry.
- Coupling between the radial translations in the X- and Y-directions of the coordinate system that is referenced in the RCORDi field of the ROTORD bulk entry.
- Coupling between the radial translations in the X- and Y-directions of the coordinate system that is referenced in the RCORDi field of the ROTORD bulk entry and the axial translation in the Z-direction of the coordinate system that is referenced in the RCORDi field of the ROTORD bulk entry.
- Rotation about the X- and Y-directions of the coordinate system that is referenced in the RCORDi field of the ROTORD bulk entry.
- Coupling between the rotations about the X- and Y-directions of the coordinate system that is referenced in the RCORDi field of the ROTORD bulk entry.

With the PBEAR entry, you can define the stiffness, damping, and mass terms as dependent on speed and displacement or speed and force, or as dependent on speed only, or as a constant that is independent of speed, displacement, and force.

- To define a constant stiffness, damping, or mass, enter real values in the stiffness, damping, or mass fields.

- To define the stiffness, damping, or mass as speed-dependent, select ”K”, “KZ”, “KR”, “B”, “BZ”, “BR”, “M”, “MZ”, or “MR” as the type and enter integer values that reference TABLEDi bulk entries in the stiffness, damping, and mass fields. On the TABLEDi entries, enter the speed versus stiffness, speed versus damping, or speed versus mass data.
- To define the stiffness, damping, or mass as speed and displacement-dependent, or speed and force-dependent, select ”KD”, “KF”, “KDZ”, “KFZ”, “KDR”, “KFR”, “BD”, “BF”, “BDZ”, “BFZ”, “BDR”, “BFR”, “MD”, “MF”, “MDZ”, “MFZ”, “MDR”, or “MFR” as the type and enter integer values that reference TABLEST bulk entries in the stiffness, damping, and mass fields. The TABLEST bulk entries reference TABLEDi bulk entries that represent the stiffness, damping, or mass as a function of speed at various displacements or forces. The relative radial, axial, and rotational displacement, or relative radial, axial, and rotational force are used for the TABLEST look-up value.

If you specify bearing stiffness, damping, or mass as speed and displacement-dependent, you can optionally specify that the software use composite relative displacements for the stiffness, damping, or mass look up. Similarly, if you specify bearing stiffness, damping, or mass as speed and force-dependent, you can optionally specify that the software use composite relative forces for the stiffness, damping, and mass look up.

Composite relative displacements are linear combinations of radial and axial relative displacements. You specify the coefficients for the linear combinations. The coefficients weight the relative contributions of the radial and axial relative displacements. The composite relative displacements are then used by the software to look up the stiffness, damping, and mass values. You can also specify constants in the composite relative displacement equations. Use the constants to include preload displacements in the composite relative displacement equations.

Similarly, composite relative forces are linear combinations of radial and axial relative forces. You specify the coefficients for the linear combinations to weight the relative contributions of the radial and axial relative forces to the relative forces. The composite relative forces are then used by the software to look up the stiffness, damping, and mass values. You can also specify constants in the composite relative force equations. These constants represent preload forces in the composite relative force equations.

For additional information on using the CBEAR and PBEAR bulk entries, see the *Simcenter Nastran Quick Reference Guide*.

MODELING INDIVIDUAL ROTORS WITH THRUST BEARING SUPPORTS

Certain types of bearings and certain bearing/rotor configurations support thrust loads in one axial direction only. When you model the bearing supports for an individual rotor with CBEAR elements, you can model this type of behavior as follows.

Use a table to define the axial stiffness vs. relative axial displacement characteristic of the CBEAR elements such that:

$K_{zz} = 0$ for $\Delta \leq 0$

$K_{zz} > 0$ for $\Delta > 0$

where K_{zz} is the axial stiffness of the CBEAR element, and Δ is relative axial displacement between the grids that define the connectivity of the CBEAR element. The relative axial displacement is given by:

$$\Delta = \Delta_{GB} - \Delta_{GA}$$

where Δ_{GB} is axial displacement of the grid entered in the GB field of the CBEAR bulk entry, and Δ_{GA} is axial displacement of the grid entered in the GA field of the CBEAR bulk entry.

For an individual rotor, one of the grids that define the CBEAR element connectivity is also listed on the ROTORB bulk entry for that rotor. Generally, it does not matter whether you enter this grid in the GA field or the GB field. However, when you model a bearing that supports thrust loads in only one axial direction, it does matter.

- For a CBEAR element to support thrust loads in the +Z-direction only, enter the grid listed on the ROTORB bulk entry in the GB field of the CBEAR bulk entry.
- For a CBEAR element to support thrust loads in the –Z-direction only, enter the grid listed on the ROTORB bulk entry in the GA field of the bulk entry.

3.5 Coupled, Time-Dependent Solutions

When modeling symmetric or unsymmetric rotors on unsymmetric supports in the rotating reference system, time-dependent coupling terms arise in the equation of motion. Simcenter Nastran can include these terms in the equation of motion for SOL 107, 108, 109, 110, 111, and 112 rotor dynamic analyses.

To include the coupling terms in the equation of motion, include the ROTCOUP parameter in the bulk data section of the input file. On the ROTCOUP parameter, specify the coupling point for each rotor in the model. Coupling points are grid points that Simcenter Nastran uses to compute the coupling components. Only grid points that are listed on a ROTORB bulk entry are valid candidates to be coupling points.

- If the model contains a single rotor, specify the coupling point with PARAM,ROTCOUP,*gridid*, where *gridid* is the grid identification number of the coupling point.
- If the model contains multiple rotors, specify the coupling points with PARAM,ROTCOUP,*setid*, where *setid* is the identification number of a SET case control command that lists the grid identification number of the coupling point for each rotor.

If you include the ROTCOUP parameter in the input file, but also specify that the analysis be performed in the fixed reference system, Simcenter Nastran ignores the ROTCOUP parameter specification.

When the time-dependent terms are included, the equation of motion is solved at discrete azimuth angles. Simcenter Nastran can either solve the equation of motion over a range of azimuth angles at a single rotor speed, or solve the equation of motion at a single azimuth angle over a range of rotor speeds.

- To solve over a range of azimuth angles at a single rotor speed, use the PHIBGN, PHIDEL, and PHINUM parameters to specify the azimuth angle range, and use the RSTART field of the ROTORD bulk entry to specify the rotor speed.
- To solve over a range of rotor speeds at a single azimuth angle, use the PHIBGN parameter to specify the azimuth angle, omit the PHIDEL and PHINUM parameters, and use the RSTART, RSTEP, and NUMSTEP fields of the ROTORD bulk entry to specify the rotor speed range. If you also omit the PHIBGN parameter, the solve is at an azimuth angle of zero because the default value for the PHIBGN parameter is zero.

If you specify the PHIBGN, PHIDEL, and PHINUM parameters and the RSTART, RSTEP, and NUMSTEP fields of the ROTORD bulk entry, the solve is over the azimuth angle range specified by the PHIBGN, PHIDEL, and PHINUM parameters, and at the rotor speed specified by the RSTART field of the ROTORD bulk entry.

You can use the FCTCi, FCTMi, and FCTZi parameters to exclude time-dependent coupling terms from the equation of motion. For more information on these parameters, see the *Simcenter Nastran Quick Reference Guide*.

3.6 Rotor Speed Specification Options

As opposed to basing all the rotor speeds off of the speed of an imaginary reference rotor, for complex eigenvalue and frequency response rotor dynamic analysis you can optionally use the ROTPARM bulk entry to designate a rotor in the physical system as the reference rotor. To do so, list the rotor in the REFROT field of the ROTPARM bulk entry. The software then applies the values for the RSTART, RSTEP, and NUMSTEP fields to the rotor you specify in the REFROT field and it uses the RSPEEDi field specifications on the ROTORD bulk entry to calculate the corresponding speed ratios and speeds for the other rotors in the system.

The following examples demonstrate how the software calculates rotor speeds when you include a value in the REFROT field of the ROTPARM bulk entry.

- Fixed rotor speed ratios

  For this case, you specify real values in the RSPEEDi fields of the ROTORD bulk entry. The software determines the rotor speed ratios by setting the speed ratio for the rotor listed in the REFROT field of the ROTPARM bulk entry to 1.0 and calculating the speed ratios for the other rotors from the RSPEEDi values.

  For example, suppose that in a two rotor model, RSPEED1 = 2.0 and RSPEED2 = 3.0. If you specify rotor 1 in the REFROT field of the ROTPARM bulk entry, the software sets the speed ratio for rotor 1 to 1.0 and calculates the speed ratio for rotor 2 to be 1.5 = 3.0 / 2.0. The software then calculates the speed of rotor 2 to be the product of 1.5 and the speed of rotor 1.

  If you specify rotor 2 as the reference rotor, the software sets the speed ratio for rotor 2 to 1.0 and calculates the speed ratio for rotor 1 to be 0.6667 = 2.0 / 3.0. The software then calculates the speed of rotor 1 to be the product of 0.667 and the speed of rotor 2.

- Table of rotor speed ratios vs. reference rotor speed

  For this case, you specify integer values in the RSPEEDi fields of the ROTORD bulk entry that reference TABLEDi bulk entries. On the TABLEDi bulk entries, you define how the speed ratio between the rotor and the imaginary reference rotor varies as a function of the imaginary reference rotor speed.

  The procedure the software uses to calculate the rotor speed ratios is best demonstrated in an example.

  Suppose that in a two rotor model, the speed ratio as a function of the imaginary reference rotor speed for rotor 1 is given by

  ```
  TABLED1, 1,
  , 0.0, 0.0, 1000.0, 2.0, 2000.0, 2.5, ENDT
  ```

  The speed ratio as a function of the imaginary reference rotor speed for rotor 2 is given by

  ```
  TABLED1, 2,
  , 0.0, 0.0, 500.0, 2.0, 3000.0, 3.0, ENDT
  ```

  Also suppose that rotor 1 is specified in the REFROT field of the ROTPARM bulk entry. The software begins by creating a table of rotor speed for the rotor specified in the REFROT field vs. imaginary reference rotor speed. Using the TABLED1 bulk entry for rotor 1, the table is as follows:

REFROT rotor	Imaginary reference rotor
0.0	0.0
2000.0 = 2.0 x 1000.0	1000.0
5000.0 = 2.5 x 2000.0	2000.0

  Suppose the software needs the rotor speeds for a reference rotor speed of 1500.0. That is, the speed of the rotor specified in the REFROT field of the ROTPARM bulk entry is 1500.0. The software interpolates the REFROT rotor vs. imaginary reference rotor tabular data to obtain an imaginary reference rotor speed of 750.0.

  To determine the speed ratio for rotor 2 that corresponds to an imaginary reference rotor speed of 750.0, the software interpolates the TABLED1 data for rotor 2. Thus, the software calculates the speed ratio for rotor 2 to be:

  $$2.1 = 2.0 + (3.0 - 2.0) [(750.0- 500.0) / (3000.0 - 500.0)]$$

  Using the interpolated speed ratio for rotor 2 and the imaginary reference rotor speed, the software then calculates the speed of rotor 2 to be 1575.0 = 2.1 x 750.0. Thus, the speed ratio of rotor 2 relative to the rotor listed in the REFROT field is:

  $$1.05 = 1575.0 /1500.0.$$

- Direct specification of rotor speeds

  For this case you specify:

1. Integer values in the RSPEEDi fields of the ROTORD bulk entry that reference DDVAL bulk entries.
2. PARAM,RSPDTYPE,DDVAL
3. DDVAL bulk entries for each rotor in the system.

On each DDVAL bulk entry, you list the speed of the rotor for each spin state of the system. A spin state is the set of all rotor speeds that occur simultaneously.

When you use this approach, the software interpolates the data on the DDVAL bulk entries to determine the rotor speeds directly.

For example, suppose that in a three rotor model, the DDVAL bulk entry for rotor 1 is given by

```
DDVAL, 1, 1000.0, 4000.0
```

The DDVAL bulk entry for rotor 2 is given by

```
DDVAL, 2, 2000,0, 6000.0
```

The DDVAL bulk entry for rotor 3 is given by

```
DDVAL, 3, 3000,0,12000.0
```

If rotor 2 is the specified in the REFROT field of the ROTPARM bulk entry and the analysis is to be performed with the speed of rotor 2 at 4000.0, the corresponding speed for rotor 1 is:

2500.0 = 1000.0 + (4000.0 - 1000.0) / 2

and the corresponding speed for rotor 3 is:

7500.0 = 3000.0 + (12000.0 - 3000.0) / 2.

You can use fixed rotor speed ratios and tables of rotor speed ratios vs. reference rotor speed in combination. However, if you use the DDVAL approach, all rotors must reference DDVAL bulk entries.

As a best practice, use the DDVAL approach to define speed ratios that vary.

To have the software use the imaginary reference rotor approach to calculating rotor speeds, specify PARAM,RLOOPNEW,NO (the default) and leave the REFROT field blank on the ROTPARM bulk entry.

3.7 Parameters

When performing rotor dynamics analysis, you must turn off residual vectors. To do so, use the RESVEC parameter.

You can also optionally specify the following parameters.

Parameter	Default value	Description
BRSYMFAC	0.0	Controls whether the stiffness, damping, and inertia that the software uses for CBEAR elements is the stiffness, damping, and mass defined on PBEAR entries or the

		symmetrized versions of the stiffness, damping, and mass. By default, the software uses the stiffness, damping, and mass defined on the PBEAR bulk entries, except for SOL 101 rotordynamic analysis where it always uses the symmetrized stiffness.
FCTCi FCTMi FCTZi	1.0	Valid for SOLs 107-112 when a rotating reference system is used. By default, time-dependent coupling terms are included in the equation of motion. Use FCTCi, FCTMi, and FCTZi to selectively exclude time-dependent coupling terms from the equation of motion.
MODTRK	2	Selects the mode tracking method. See the “Mode Tracking Parameters” section for more information.
PHIBGN	0.0	Valid for SOLs 107-112 when a rotating reference system is used and the ROTCOUP parameter is specified. PHIBGN specifies the beginning of the range of azimuth angle in degrees. PHIDEL specifies the azimuth angle increment in degrees. PHINUM specifies the number of azimuth angle increments.
PHIDEL	0.0	See PHIBGN
PHINUM	1	See PHIBGN
RLOOPNEW	NO	For frequency response and complex eigenvalue rotor dynamic analysis, specify PARAM,RLOOPNEW,YES to include gyroscopic and circulation terms in the mass, damping, and stiffness matrices when the analysis is performed in the fixed reference system.
ROTCMRF	0.0	Specifies the reference rotor speed to perform complex modal reduction. Only valid for SOL 107 when complex modal reduction is used.
ROTCOUP	No default	Valid for SOLs 107-112 when a rotating reference system is used. Triggers the inclusion of time-dependent coupling terms in the equation of motion and specifies the coupling points for each rotor.
ROTCSV	No default	Defines the comma separated ASCII file for processing with Excel.
ROTGPF	No default	Defines the .gpf file used by the post-processor COLMAT. See ref. [3] for more information.
ROTSYNC	YES	Valid for SOLs 107 and 110. If PARAM,ROTSYNC,NO is specified, synchronous analysis is skipped when there are no solutions because there is no intersection with the 0-P line. If PARAM,ROTSYNC,YES (default) and the NUMSTEP field on the ROTORD bulk entry is set to zero, only synchronous analysis is performed. If PARAM,ROTSYNC,NO and the NUMSTEP field is set

		to zero, no analysis is performed.

For more information on these parameters, see the *Simcenter Nastran Quick Reference Guide*.

3.7.1 Mode Tracking Parameters

The quality of a Campbell diagram depends on the accuracy of the mode tracking. Three mode tracking methods are available in Simcenter Nastran. Use the MODTRK parameter to specify the mode tracking method:

- PARAM,MODTRK,1 (pre- NX Nastran 7 method). Outer loop over rotor speed, inner loop over degrees of freedom. This method does not work well for the direct method (SOL 107) because new solutions can enter and old solutions can leave the solution space. This is the default mode tracking method in versions of Simcenter Nastran prior to NX Nastran 7.
- PARAM,MODTRK,2 (method introduced in NX Nastran 7). Outer loop over degrees of freedom, inner loop over rotor speed. Process repeats until all solutions have been tracked. This is the default mode tracking method since NX Nastran 7.
- PARAM,MODTRK,3 (method introduced in NX Nastran 8.5). Eigenvectors and eigenvalues are used to track the modes. This method is applicable to models that have any combination of unsymmetric stiffness, viscous damping, mass, and structural damping.
- PARAM,MODTRK,4 (method introduced in NX Nastran 11). The initial reference rotor modes are used as the base vectors for complex eigenvalue calculations at subsequent reference rotor speeds. Mode tracking begins at the first nonzero reference rotor speed. Modes for the reference rotor speed of zero are excluded from the Campbell diagram results.

If you specify PARAM,MODTRK,2 or PARAM,MODTRK,3, you can can optionally specify a number of other parameters to tweak the mode tracking method.

If you specify PARAM,MODTRK,2, the following parameters are valid:

Parameter	Default value	Description
MTREPSI	0.01	Relative tolerance for the imaginary part (eigenfrequency). When a root has been found in the extrapolation, a check is made if the correct solution has been found. If the present value is outside of this tolerance, the solution is skipped.
MTREPSR	0.05	Relative tolerance for the real part (eigenfrequency). When a root has been found in the extrapolation, a check is made if the right solution has been found. If the present

		value is outside of this tolerance, the solution is skipped.
MTRFCTD	0.5	Threshold value for damping. This is in order to disregard the real part for solutions with low damping.
MTRFCTV	0.0	Reference value for converting aerodynamic speed to rotor speed for wind turbines. Used only for wind turbines in SOL 145.
MTRFMAX	0.0	Maximum frequency to consider. If zero, all frequencies are used. With this parameter, high frequency solutions can be filtered out.
MTRRMAX	0.0	Maximum absolute value of real part of solution to consider. If zero, all real part values are used. With this parameter, solutions with high real parts can be filtered out. It can be useful for disregarding numerical solutions which are not physically relevant.
MTRSKIP	5	If roots are not found for a specific speed, this speed is skipped and the next speed is analyzed. If there are more than MTRSKIP values missing, the curve is not considered. The solution will be marked as unused and will be considered in the next loop.

If MTREPSR and MTREPSI are chosen too small, the solution may be lost for some speed values. If they are too large, there may be lines crossing from one solution to another. Problems may occur for turbines with many elastic blades with equal frequencies. Then, a cluster of lines and crossings may occur.

If you specify PARAM,MODTRK,3, the following parameters are valid:

Parameter	Default value	Description
MTRFMAX	0.0	Maximum frequency to consider. If zero, all frequencies are used. With this parameter, high frequency solutions can be filtered out.
MTRRMAX	0.0	Maximum absolute value of real part of solution to consider. If zero, all real part values are used. With this parameter, solutions with high real parts can be filtered out. It can be useful for disregarding numerical solutions which are not physically relevant.

For more information on these parameters, see the *Simcenter Nastran Quick Reference Guide*.

3.8 Mode Filtering

You can use strain energy-based criteria and kinetic energy-based criteria to identify the modes that have minimal impact on the dynamic response in a SOL 107 or SOL 110 complex eigenvalue analysis in rotor dynamics. You can omit these modes from the list of modes that are tracked throughout the remainder of the rotor dynamics analysis. Because modes of little importance are eliminated, the analysis results produce a less cluttered and potentially more meaningful Campbell diagram.

To determine the importance of modes, you can use either or both of the following criteria:

- The ratio of the strain energy of a specific rotor to the total strain energy of the system for a given mode.
- The ratio of the kinetic energy of a specific rotor to the total kinetic energy of the system for a given mode.

To identify the modes which contribute little to the dynamic response, use the new ROTPARM bulk entry. With the ROTPARM bulk entry, you specify:

- The energy-based criterion for the software to use.
- The threshold value for the criterion below which modes are deemed of little importance to the dynamic response.
- The reference rotor. The reference rotor is the rotor whose strain or kinetic energy is used to calculate the ratio of energy of the rotor to the total energy of the system.

You specify the reference rotor in the REFROT field of the ROTPARM bulk entry.

If you specify the option that uses both criteria, only modes whose ratios fall below the threshold for both criteria are deemed of little importance.

For meaningful results, on the ROTORD bulk entry, you must set the RSPEEDi field of the reference rotor to 1.0. The software does not trap other values.

In the results file, the important modes are denoted by “***” in the summary line. The modes of little importance are not denoted in any way.

To omit the modes that are deemed of little importance, in the input file, include PARAM,MODTRK,4. The software does not omit modes if you specify any other mode tracking option or no mode tracking option. Modes are also not omitted if you specify PARAM,MODTRK,4, but do not specify one of the energy-based criteria on the ROTPARM bulk entry.

The MODTRK = 4 option uses the modes for the reference rotor at the initial non-zero speed as the base vectors for the complex eigenvalue calculations at subsequent rotor speeds. This approach improves the quality of the mode tracking and reduces the computational time for complex mode calculations.

3.9 Solution-Specific Data

In all modal solutions (SOL 110, 111, and 112), a METHOD case control command and an EIGRL bulk entry are required.

In complex eigenvalue solutions (SOL 107 and 110), a CMETHOD case control command and an EIGC bulk entry are required.

In frequency response solutions (SOL 108 and 111), DLOAD and FREQ case control commands, and RLOADi and FREQi bulk entries are required.

In transient response solutions (SOL 109 and 112), DLOAD and TSTEP case control commands, and TLOADi and TSTEP bulk entries are required.

In the direct complex eigenvalue solution (SOL 107), the EFLOAD case control command can be used to define an external force field. The typical application is to apply forces as a result of an electromagnetic field. Simcenter Nastran converts the electromagnetic field surface loads, which come from a third-party party electromagnetic simulation product like MAXWELL, to Simcenter Nastran structural loads. This is useful for analyzing structural components in motors or other electromechanical devices. See “External Force Fields” in the *Simcenter Nastran User’s Guide* for a description of the inputs and solution steps.

3.10 Superelement Reduction of Supporting Structures

In all rotor dynamic solutions, you can improve the computational efficiency of the rotor dynamic analysis by modeling the stationary portions of the rotor dynamics model as an external, internal, or partitioned superelement. You use the same procedure and Simcenter Nastran user inputs that you would if the model were not a rotor dynamics model.

For detailed information on how to create external, internal, and partitioned superelements, see the *Simcenter Nastran Superelement User’s Guide*.

3.11 Superelement-style Reduction of Rotors

In the direct complex eigenvalue (SOL 107), direct frequency response (SOL 108), and direct transient response (SOL 109) solutions, you can improve computational efficiency by applying superelement-style reduction to the rotors. However, the implementation of superelement-style reduction in rotor dynamic analysis is distinctly different from the implementation of superelements in other types of analysis. For example, none of the Simcenter Nastran user inputs for modeling superelements in other types of analysis are applicable to this capability.

The procedure to apply superelement-style reduction to a rotor is as follows:

- Include a ROTSE bulk entry for each rotor you want to reduce. The presence of a ROTSE bulk entry triggers the superelement-style reduction capability. Match the value in the RSETID field of each ROTSE bulk entry with the corresponding RSETi field for the rotor on the ROTORD bulk entry. For each rotor that you define a ROTSE bulk entry, the software will automatically assign the grids on the corresponding ROTORG bulk entry to a unique o-set.
- On each ROTSE bulk entry, specify any grids that are listed on the corresponding ROTORG bulk entry that need to be removed from the o-set and placed in the a-set. Typically, these are the grids that connect the rotor to the supporting structure, the grids where loads like mass imbalance are applied to the rotor, and the grids that provide a better representation of mass distribution for the rotor.
- On each ROTSE bulk entry, specify whether the software should use real or complex modal reduction. Generally, you will want to select complex modal reduction.

If a model contains ROTSE bulk enties and you use the model in a SOL 101 or 110-112 rotor dynamic analysis, Simcenter Nastran ignores the ROTSE bulk entries and does not reduce the rotors to superelements.

CHAPTER

4

Interpretation of Rotor Dynamics Output

4 Interpretation of Rotor Dynamics Output

When you perform a rotor dynamics analysis with Simcenter Nastran, the software writes the results to the F06 file, OP2 file, and to two ASCII files that you can use to post-process your results.

4.1 The F06 File

In the F06 file, the first part of the printed output is a list of rotor speeds, complex eigenvalues, frequency, damping, and whirl direction. The software prints one table for each solution. The solution numbers correspond to the real modes at low rotor speeds. At higher speeds, the complex solution modes are generally a combination of the original real modes. The eigenvalue routine simply sorts the solution according to the value of the imaginary part (eigenfrequency). Because there may be coupling or crossing of solution frequency and damping lines, the software automatically uses a mode tracking algorithm to sort the solutions in a reasonable way for creating the Campbell diagram summary. The printed output represents the results after the software has applied the mode tracking algorithm. See Table 5 in Section 6.1.1 for an example.

The second part of the printed output is a summary of the results from the Campbell diagram. This summary includes:

- Resonance of forward whirl.
- Resonance of backward whirl.
- Instabilities, which are the points of zero damping.
- Critical speeds from the synchronous analysis (only for analyses in the fixed system).

An example of a typical F06 file is shown in Table 6. Note that for an analysis in the fixed system, a synchronous analysis is always performed prior to the rotor loop.

4.2 The OP2 File

If you set PARAM, POST,-2 in your input file, Simcenter Nastran writes your results to the OP2 file. With this option, the software includes all the Campbell diagram summary data (contained in the CDDATA data block) in a format identical to the one used in the CSV and GPF files.

You can then view these complex modes in post-processing software that supports the visualization of complex mode shapes.

4.3 The CSV File for Creating Campbell Diagrams

In addition to writing your rotor dynamics results to the standard Simcenter Nastran F06 and OP2 files, you can also choose to have the software write specially formatted results to a CSV (.csv) file. The CSV file is a comma separated, ASCII formatted file that lets you easily import the Campbell diagram data into another program, such as Excel, for post-processing. Only the Campbell diagram data is written to the CSV file. You must add the necessary commands to actually create the Campbell diagrams in Excel.

Use the following parameter to create the CSV file:

PARAM ROTCSV *unit*

where *unit* defines the unit number of CSV files to which the software should write the results. Additionally, you must also designate the name for the CSV file in the File Management section of your input file. For example:

ASSIGN OUTPUT4='filename.csv',UNIT=25, FORM=FORMATTED

Here, unit 25 has been used. You must select a unit which is not already used to define other Simcenter Nastran files. For example, unit 12 is used for OP2 file for standard post-processing.

The software writes the data in the CSV file in sections because Simcenter Nastran has a limited record size. The CSV file format depends on the mode tracking method used.

If PARAM,MODTRK,1 is specified, the format is as follows:

Column Identifier	Column Contents
10001	Rotor speeds in the (FUNIT) specified on the ROTORD bulk entry.
20001, 20002, etc.	Up to 7 columns per section of eigenfrequencies in the analysis system (REFSYS) and the units (RUNIT) specified on the ROTORD bulk entry. If your model contains more than 7 solutions, the software adds additional sections.
30001, 30002, etc.	Up to 7 columns per section of damping values. If your model contains more than 7 solutions, the software adds additional sections.
40001, 40002, etc.	Real part of the eigensolution in the same format as the eigenfrequencies.

Column Identifier	Column Contents
50001, 50002	Whirl directions in the same format as the eigenfrequencies. • 2.0 indicates backward whirl • 3.0 indicates forward whirl • 4.0 indicates linear motion
50000	Used for plotting the 1P line.
60001, 60002	Eigenfrequencies that have been converted to the rotating or fixed reference system in the units (RUNIT) specified on the ROTORD bulk entry.
70001, 70002	Whirl directions in the converted system using the same format as in the section with the identifier 50001.
70000	Used for plotting the 1P line.

Table 13 shows an example of the contents of a typical CSV file when PARAM,MODTRK,1 is specified.

If PARAM,MODTRK,2 or PARAM,MODTRK,3 is specified, the format is as follows: (SOL = solution number)

Column Identifier	Column Contents
10000+(10*SOL)+1	Rotor speeds in the units (FUNIT) specified on the ROTORD bulk entry.
10000+(10*SOL)+2	Eigenfrequencies in the analysis system (REFSYS) and the units (RUNIT) specified on the ROTORD bulk entry.
10000+(10*SOL)+3	Damping values.
10000+(10*SOL)+4	Real part of the eigensolution in the same format as the eigenfrequencies.
10000+(10*SOL)+5	Imaginary part of the eigensolution in the same format as the eigenfrequencies.

Column Identifier	Column Contents
10000+(10*SOL)+6	Whirl direction: • 2.0 indicates backward whirl • 3.0 indicates forward whirl • 4.0 indicates linear motion
10000+(10*SOL)+7	Eigenfrequencies that have been converted to the rotating or fixed system in the units (RUNIT) specified on the ROTORD bulk entry.
10000+(10*SOL)+8	Whirl directions in the converted system.
10000+(10*SOL)+9	Imaginary part of the converted eigensolution.
10000+(10*SOL)+10	Real part of the converted eigensolution.
9000	Rotor speed, first and last in the requested speed unit.
9001	1P line in requested frequency
9002	2P line, used only for rotating system
9003	3P line
9004	4P line
9005	5P line
9006	Not used.

4.4 The GPF File for Additional Post-Processing

You can also have the software write your rotor dynamics results to a GPF (.gpf) file. Like the CSV file, the GPF file is a specially formatted ASCII file. The GPF file is designed to be used with the COLMAT post-processor [ref. 3]. In the case of the GPF file, the software includes all commands necessary to generate the plots in the file. Table 12 shows an example of a typical GPF file.

You use the following parameter to invoke the GPF file:

PARAM ROTGPF *unit*

where *unit* defines the unit number of GPF files to which the software should write the results. Additionally, you must also designate the name for the GPF file in the File Management section of your input file. For example:

ASSIGN OUTPUT4='filename.gpf',UNIT=22, FORM=FORMATTED

Here, unit 22 has been used. You must select a unit which is not already used to define other Simcenter Nastran files.

4.5 Output for Frequency Response

The frequency response output for rotor dynamics in F06 and OP2 files is similar to that of the standard modal frequency response analysis output. In addition the ROTORG bulk entry is stored on the DYNAMICS data block of .op2 file, containing the rotating subset of grids for post-processing purposes. In synchronous analysis, the frequencies are listed in RPM for response results.

Also, the punch file can be used for XYPUNCH commands in the case control deck.

4.6 Output for Transient Response

The transient output is similar to that for frequency response except that the response data is a function of time.

4.7 Complex Modes

Complex modes can be output at the speed specified in the CMOUT field on the ROTORD entry. The output can be written to the F06 or the OP2 files. The output files can be imported into post-processors like Simcenter, which can plot the complex modes as real and imaginary parts. The imaginary part is phase shifted by 90 degrees with respect to the real part. If the real and imaginary parts are assembled according to a rotating pointer in the complex plane, animation of the rotor modes can be established and the whirling motion studied.

In some post-processors, the animations of complex modes are only linear motions of the real or imaginary parts, which are less useful.

CHAPTER

5

Modeling Considerations and Selecting a Reference System

5 Modeling Considerations and Selecting a Reference System

Different problems require different solution strategies. The correct analysis type depends on the model you are analyzing.

5.1 Choosing Between the Fixed and Rotating Reference System

For all rotor dynamic analyses, you can analyze symmetric rotors on symmetric supports in both the fixed and rotating reference systems, you can analyze symmetric rotors on unsymmetric supports in the fixed reference system, and you can analyze unsymmetric rotors on symmetric supports in the rotating reference system.

For SOL 107, 108, and 109, you can analyze symmetric or unsymmetric rotors on unsymmetric supports in the rotating reference system.

- In the fixed reference system, the motion is observed relative to the stationary system.
- In the rotating reference system, the motion is observed relative to the rotor.

5.2 Translation and Tilt Modes

Translation modes have no rotational terms in the fixed system analysis, except for the damping. The typical behavior of tilting modes in the fixed system is demonstrated by the horizontal line at 10 Hz as shown in Fig. 20. The behavior in the rotating system is shown in Fig. 19. Only the positive frequencies are plotted. At the singular point where the eigenfrequency is zero, the centrifugal softening force is equal to the elastic stiffness, hence $k = m\omega^2 = m\Omega^2$

In the fixed system, the solution of the tilting modes of the backward whirl tends to zero. The forward whirl tends to 2P for a disk as shown in Fig. 33. In the rotating system, there is a double solution for both whirl modes approaching the 1P line asymptotically as shown in Fig. 35

5.3 Calculating Geometric Stiffness

You must define an RFORCE, RFORCE1, or RFORCE2 bulk entry and use the RFORCEi option on the ROTORD entry to have the software calculate the geometric (differential) stiffness matrix. If you are performing an analysis on a model comprised of elastic rotors, such as blades or rotating thin tubes, you should always calculate the geometric stiffness matrix. Generally, if you are performing an analysis on a model comprised of solid rotors, the geometric stiffness is not necessary.

Additionally, to obtain the geometric stiffness matrix, you must insert a static SUBCASE prior to the modal analysis. The load from the static subcase must be referenced in the modal subcase by a STATSUB command. You must define a unit rotor speed of 1 rad/sec (hence, $1/2\pi$ Hz = 0.159155 Hz) on the RFORCE, RFORCE1, or RFORCE2 bulk entry in the bulk section of your input file.

5.4 Steiner's Term in the Centrifugal Matrix

The ZSTEIN option on the ROTORD entry lets you include Steiner's inertia terms in your analysis. You should use the ZSTEIN option carefully. Importantly, you cannot use the ZSTEIN option if you are also having the software calculate the geometric stiffness matrix (RFORCE, RFORCE1, or RFORCE2 bulk entry and RFORCEi option on the ROTORD entry). However, you should use the ZSTEIN option if you are analyzing solid rotors, such as the one described in Section **Error! Reference source not found.**.

5.5 Whirl Motion

In Simcenter Nastran, you use the CMOUT = -1.0 option on the ROTORD entry to have the software calculate the whirl direction for all RPM using complex eigenvectors. The whirl direction is useful when you need to convert the results from one reference system to the other. In this case, the solutions are converted automatically from one system to the other by adding and subtracting the rotor speed to the backwards and forward whirl motion respectively.

5.6 Damping

Damping is applied as viscous damping and may be defined by the following methods:

- You can define physical damping with unit force/velocity using CDAMPi/PDAMP bulk entries. The equivalent viscous damping can be found for a simple system with:

$$\zeta = \frac{D}{2m\omega} \tag{54}$$

- You can define structural damping on the MATi entry. With rotor dynamics, this damping is converted to viscous damping using the calculated eigenfrequencies without rotation.

You can apply both damping types to the rotating and the non-rotating parts. The standard Simcenter Nastran damping output is in units of “g”. In the rotor dynamic analysis, the damping unit is the fraction of critical viscous damping (Lehr damping). The relation is

$$\zeta = \frac{g}{2} \tag{55}$$

If you use the ROTPRT = 1 or 3 option on the ROTORD entry, the software prints the equivalent damping factors and the generalized matrices to the F06 file.

In Simcenter Nastran, the standard damping matrices are used. They are partitioned into the rotating and the non-rotating parts. The antisymmetric matrices are calculated in the following way:

$$\left[D_B\right] = \frac{1}{2}\left(\left[B\right]\left[D\right] + \left[D\right]\left[B\right]\right) \tag{56}$$

where $\left[D\right]$ is the damping matrix which is in general not a diagonal matrix and $\left[B\right]$ is a Boolean matrix defined as:

$$\left[B\right] = \begin{bmatrix} 0 & -1 & 0 & & & \\ 1 & 0 & 0 & & & \\ 0 & 0 & 0 & & & \\ & & & 0 & -1 & 0 \\ & & & 1 & 0 & 0 \\ & & & 0 & 0 & 0 \end{bmatrix} \tag{57}$$

5.7 Multiple Rotors

A structure with up to ten rotors can be analyzed. The rotors are defined using the ROTORD bulk entry and the grid points associated with each rotor are selected using ROTORG bulk entries.

In order to calculate the damping coupling between the rotors and the fixed part of the structure, the appropriate damping in the fixed part must be assigned to specific rotors by using ROTORB entries. The reference rotor speed is defined by the values RSTART, RSTEP, and NUMSTEP on the ROTORD entry. The relative rotor speeds for the different rotors are defined by the RSPEEDi fields on the ROTORD entry.

The Campbell and damping diagrams are established as function of the reference rotor speed. In order to find the possible critical speeds for the different rotors, the crossing with the 1P line multiplied by the relative speed must be used. In SOL 107 and 110 synchronous analysis, a loop over all rotors is done and all possible crossings are calculated. However, not all of the crossings may be relevant. This depends on the coupling between the rotors. An excitation of one rotor may lead to excitation of other rotors via the flexible supporting structure.

The response of the rotors can be analyzed with SOL 111 and SOL 108 in the frequency domain. In the synchronous case, only the reference rotor speed is used. The structure can be excited only with one frequency function. Therefore, separate analyses have to be made in order to study the response behavior if the rotors are running at different speeds. This means that the EORDER field on the ROTORD entry is common to all rotors and the relative speed is not considered in the forcing function. The RSPEED field cannot be applied in this case and it is not used for the forcing function. Also, when specifying

ETYPE=1, the reference rotor speed is used. Thus, synchronous analysis in the frequency domain with multiple rotors having different speeds must be performed with care.

For synchronous response analysis in the time domain, different sweep functions can be defined for different rotors and the restriction mentioned for the frequency response does not apply. However, ETYPE=1 cannot be used for multiple rotors having different speeds. The EORDER and the RSPEED values are not used in transient response analysis, but the sweep function is defined by the user.

For multiple rotors having different speeds, the conversion from fixed to rotating system or vice versa does not work because the software does not know which solution belongs to which rotor. The modes may be coupled via the fixed part. Also, for coaxial rotors, the modes are coupled.

5.8 Numerical Problems

Numerical problems may occur for the calculation of the synchronous critical speeds in SOL 107 and SOL 110 because the eigenvalue problem is numerically not well conditioned. This is due to the missing damping in the synchronous analysis and because there are less solutions (number of crossing points) then the order of the matrices. The synchronous analysis can be skipped by including PARAM,ROTSYNC,NO in the input file.

In SOL 107 there may be cases where the solution cannot be found because the problem is numerically ill conditioned. This can frequently be overcome by switching to the single vector complex Lanczos method by selecting the system cell 108 on the NASTRAN card:

NASTRAN SYSTEM(108)=2 $

This may not work for all machines. Synchronous analysis is only intended as a check for the Campbell diagram and not as a stand alone solution

Also, there may be difficulties with the calculation of the whirling directions because the eigenvector can be almost real instead of complex. This can also be overcome by using the single vector method as described above.

In SOL 109 and 112 there may be numerical problems in the time integration. This can happen for the analysis in the rotating system where the stiffness matrix becomes zero at the critical speed. In this case, the problem can frequently be solved by selecting a larger time step.

For the crossing with the 2P line, unstable time integration may occur. The problem is that the time step must be small in order to integrate over the period of the harmonic function. Therefore, the integration for high rotor speeds can fail. In this case, the integration time must be reduced. In the transient analysis, the integration can only be performed for a stable rotor. If the real part of the solution gets positive at a certain speed, the integration must end before this instability point.

When analyzing rotating shafts in the rotating system, the shaft torsion frequency may drop to zero. This is because the centrifugal softening term is linear and is therefore acting in the tangential direction. The torsional mode is normally not important in the rotor dynamic analysis and can be left out with the MODSEL case control command in the modal method. In the direct method the shaft torsion can be constrained.

5.9 Other Hints

In frequency and transient response analysis, the forcing function may be zero for zero speed. In this case, the program will stop. The problem can be solved by starting at a small speed.

In SOL 107 and SOL 110 the whirling direction is only calculated for CMOUT = -1. The complex modes are output for a certain speed defined by CMOUT.

When working with shell and solid models, it is recommended to check the results with a simple beam model.

CHAPTER

6

Rotor Dynamics Examples Complex Modes

6 Rotor Dynamics Examples

The following sections contain rotor dynamic analysis examples. Input files (.dat files) for all the examples described in this chapter are included in the Simcenter Nastran Test Problem Library, which is located in the *install_dir*/nxn*r*/nast/tpl directory.

6.1 Simple Mass Examples

This example shows the solutions of rotor translational modes using a simple model.

6.1.1 Symmetric Model without Damping (rotor086.dat)

The simple model is that of a rotating mass. As shown in Section 2.2, there are no rotational effects for a mass point in the fixed system. The input deck for such a simple model is shown in Table 3 for analysis in the rotating system.

With the field ROTPRT =3 on the ROTORD entry, the generalized matrices are printed out as shown in Table 4. This is a useful option for checking the model and the analysis. The Campbell diagram summary is shown in Table 5. The detection of forward and backward whirl resonances is shown in Table 6. The critical speed for the forward and backward whirl was found at 600 RPM.

Fig. 19 shows the Campbell diagram of the mass point calculated in the rotating system. In Fig. 19, 1P and 2P lines are plotted and the resonance points at 600 RPM can be seen. The conversion to the fixed system is shown in Fig. 20. Here there are two solutions with constant frequencies equal to 10 Hz. The resonance points are the crossings with the 1P-line at 600 RPM, which is identical with the theoretical solution.

```
NASTRAN $
$
assign output4='rotor086.gpf',unit=22, form=formatted
assign output4='rotor086.csv',unit=25, form=formatted
$
SOL 110
$
TIME 20000
DIAG 8
$
CEND
$
DISP = ALL
RMETHOD = 99
$
METHOD = 1
CMETHOD = 2
$
BEGIN BULK
$
$       define units for CSV and GPF-files
$
PARAM   ROTGPF  22
PARAM   ROTCSV  25
PARAM   GRDPNT  0
PARAM   MODTRK  1
$
```

```
$       standard selection or real and complex modal analysis
$
EIGRL   1                       2       1
EIGC    2       CLAN                                    2
$
$       Definition of rotor data
$
$       SID     RSTART  RSTEP   NUMSTEP REFSYS  CMOUT   RUNIT   FUNIT
ROTORD  99      0.0     50.0    20      ROT     -1.0    RPM     HZ      +ROT0
$       ZSTEIN  ORBEPS  ROTPRT
+ROT0   NO              3                                               +ROT1
$       RID1    RSET1   RSPEED1 RCORD1  W3-1    W4-1    RFORCE1
+ROT1   1               1.0
$
GRID    101             0.0     0.0     0.0             3456
$
CELAS1  101     101     101     1
CELAS1  102     102     101     2
$
PELAS   101     394.784
PELAS   102     394.784
$
CONM2   100     101             0.100
$
ENDDATA
```

Table 3 Input Deck for a Simple Rotating Mass Point

```
^^^
 ^^^ GENERALIZED CORIOLIS/GYROSCOPIC MATRIX

 MATRIX CHH

 (  1)          1           2

    1   0.0000E+00 -1.0000E+00
    2   1.0000E+00  0.0000E+00

 ^^^
 ^^^ GENERALIZED CENTRIFUGAL MATRIX

 MATRIX ZHH

 (  1)          1           2

    1   1.0000E+00  0.0000E+00
    2   0.0000E+00  1.0000E+00

 ^^^
 ^^^ GENERALIZED MASS MATRIX

 MATRIX MHH

 (  1)          1           2

    1   1.0000E+00  0.0000E+00
    2   0.0000E+00  1.0000E+00

 ^^^
 ^^^ GENERALIZED STIFFNESS MATRIX

 MATRIX KHHRE

 (  1)          1           2

    1   3.9478E+03  0.0000E+00
    2   0.0000E+00  3.9478E+03
```

Table 4 Output of Generalized Matrices

C A M P B E L L   D I A G R A M   S U M M A R Y

SOLUTIONS AFTER MODE TRACKING

SOLUTION NUMBER 1

STEP	ROTOR SPEED RPM	EIGENVALUE REAL	EIGENVALUE IMAG	FREQUENCY HZ	DAMPING [% CRIT]	WHIRL DIRECTION
1	0.00000E+00	0.00000E+00	6.28318E+01	1.00000E+01	0.00000E+00	LINEAR
2	5.00001E+01	-1.36530E-14	6.80678E+01	1.08333E+01	-2.00579E-16	BACKWARD
3	1.00000E+02	2.84839E-15	7.33038E+01	1.16667E+01	3.88573E-17	BACKWARD
4	1.50000E+02	2.71453E-15	7.85398E+01	1.25000E+01	3.45624E-17	BACKWARD
5	2.00000E+02	8.05262E-15	8.37758E+01	1.33333E+01	9.61210E-17	BACKWARD
6	2.50001E+02	9.74266E-15	8.90118E+01	1.41667E+01	1.09454E-16	BACKWARD
7	3.00001E+02	-1.26389E-14	9.42478E+01	1.50000E+01	-1.34103E-16	BACKWARD
8	3.50001E+02	1.40863E-14	9.94838E+01	1.58333E+01	1.41594E-16	BACKWARD
9	4.00001E+02	1.87628E-14	1.04720E+02	1.66667E+01	1.79171E-16	BACKWARD
10	4.50001E+02	6.50573E-15	1.09956E+02	1.75000E+01	5.91668E-17	BACKWARD
11	5.00001E+02	2.53821E-14	1.15192E+02	1.83334E+01	2.20347E-16	BACKWARD
12	5.50001E+02	1.43726E-14	1.20428E+02	1.91667E+01	1.19346E-16	BACKWARD
13	6.00001E+02	-1.41645E-13	1.25664E+02	2.00000E+01	-1.12718E-15	BACKWARD
14	6.50002E+02	-2.87140E-14	1.30900E+02	2.08334E+01	-2.19358E-16	BACKWARD
15	7.00002E+02	1.74764E-13	1.36136E+02	2.16667E+01	1.28375E-15	BACKWARD
16	7.50002E+02	-2.85480E-12	1.41372E+02	2.25000E+01	-2.01936E-14	BACKWARD
17	8.00002E+02	1.21181E-13	1.46608E+02	2.33334E+01	8.26564E-16	BACKWARD
18	8.50002E+02	-1.17065E-13	1.51844E+02	2.41667E+01	-7.70953E-16	BACKWARD
19	9.00002E+02	9.30786E-15	1.57080E+02	2.50000E+01	5.92556E-17	BACKWARD
20	9.50002E+02	8.71649E-14	1.62316E+02	2.58334E+01	5.37008E-16	BACKWARD

C A M P B E L L   D I A G R A M   S U M M A R Y

SOLUTIONS AFTER MODE TRACKING

SOLUTION NUMBER 2

STEP	ROTOR SPEED RPM	EIGENVALUE REAL	EIGENVALUE IMAG	FREQUENCY HZ	DAMPING [% CRIT]	WHIRL DIRECTION
1	0.00000E+00	0.00000E+00	6.28318E+01	1.00000E+01	0.00000E+00	LINEAR
2	5.00001E+01	-8.19654E-15	5.75958E+01	9.16666E+00	-1.42311E-16	FORWARD
3	1.00000E+02	-3.52221E-15	5.23598E+01	8.33333E+00	-6.72694E-17	FORWARD
4	1.50000E+02	4.21261E-15	4.71238E+01	7.49999E+00	8.93944E-17	FORWARD
5	2.00000E+02	2.89132E-29	4.18878E+01	6.66666E+00	6.90253E-31	FORWARD
6	2.50001E+02	5.22323E-15	3.66518E+01	5.83332E+00	1.42509E-16	FORWARD
7	3.00001E+02	-8.12545E-15	3.14158E+01	4.99999E+00	-2.58642E-16	FORWARD
8	3.50001E+02	-2.88400E-16	2.61798E+01	4.16665E+00	-1.10161E-17	FORWARD
9	4.00001E+02	1.68787E-15	2.09438E+01	3.33332E+00	8.05901E-17	FORWARD
10	4.50001E+02	4.36586E-15	1.57078E+01	2.49998E+00	2.77942E-16	FORWARD
11	5.00001E+02	1.23902E-15	1.04718E+01	1.66664E+00	1.18319E-16	FORWARD
12	5.50001E+02	1.74205E-15	5.23584E+00	8.33309E-01	3.32717E-16	FORWARD
13	6.00001E+02	-5.43173E-18	1.64315E-04	2.61515E-05	-3.30569E-14	BACKWARD
14	6.50002E+02	-3.19462E-15	5.23617E+00	8.33362E-01	-6.10107E-16	BACKWARD
15	7.00002E+02	2.19492E-15	1.04722E+01	1.66670E+00	2.09596E-16	BACKWARD
16	7.50002E+02	2.99485E-14	1.57082E+01	2.50003E+00	1.90655E-15	BACKWARD
17	8.00002E+02	-5.31201E-15	2.09442E+01	3.33337E+00	-2.53627E-16	BACKWARD
18	8.50002E+02	5.02973E-15	2.61802E+01	4.16670E+00	1.92120E-16	BACKWARD
19	9.00002E+02	-2.38171E-14	3.14162E+01	5.00004E+00	-7.58117E-16	BACKWARD
20	9.50002E+02	-8.66900E-16	3.66522E+01	5.83337E+00	-2.36521E-17	BACKWARD

Table 5 Campbell Diagram Summary

```
         D E T E C T I O N   O F   R E S O N A N C E S   A N D   I N S T A B I L I T I E S

                       A N A L Y S I S   I N   R O T A T I N G   S Y S T E M

 FORWARD WHIRL RESONANCE        SOLUTION      ROTOR SPEED     WHIRL
                                 NUMBER           RPM       DIRECTION

                                    2        6.00000E+02    FORWARD


 BACKWARD WHIRL RESONANCE       SOLUTION      ROTOR SPEED     WHIRL
                                 NUMBER           RPM       DIRECTION

                                     1        6.00000E+02    BACKWARD


 INSTABILITIES                  SOLUTION      ROTOR SPEED     WHIRL
                                 NUMBER           RPM       DIRECTION

                                              NONE   FOUND

CRITICAL SPEEDS FROM SYNCHRONOUS ANALYSIS
                                SOLUTION      ROTOR SPEED     WHIRL
                                 NUMBER           RPM       DIRECTION

                                     1        6.00000E+02    FORWARD
                                     2        6.00000E+02    BACKWARD
```

Table 6 The F06 File Results of Rotating System Analysis

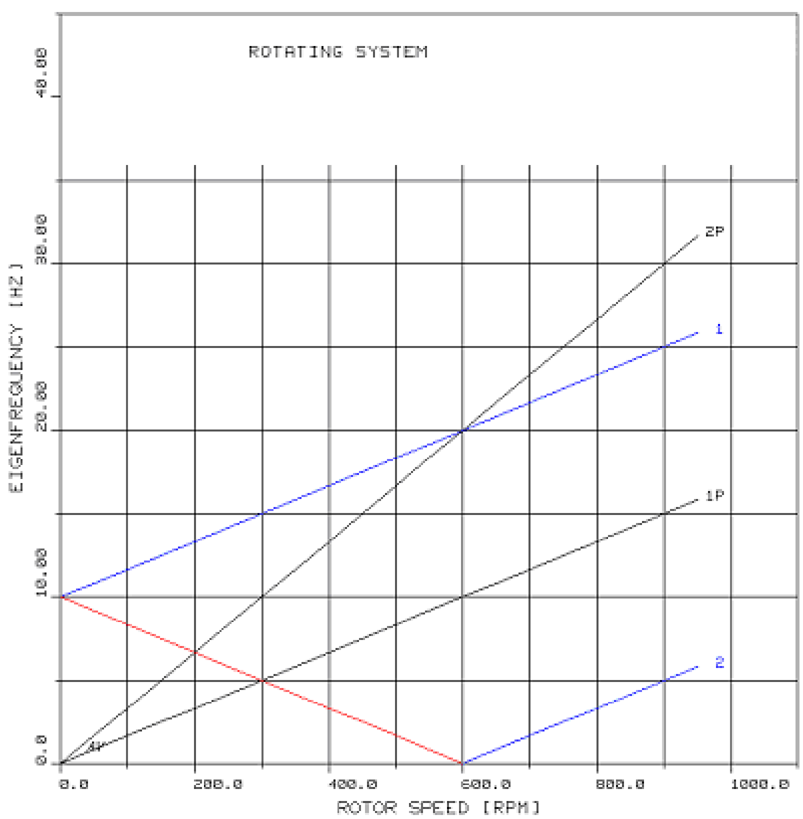


Fig. 19 Campbell Diagram for Rotating Mass Point Calculated in the Rotating System

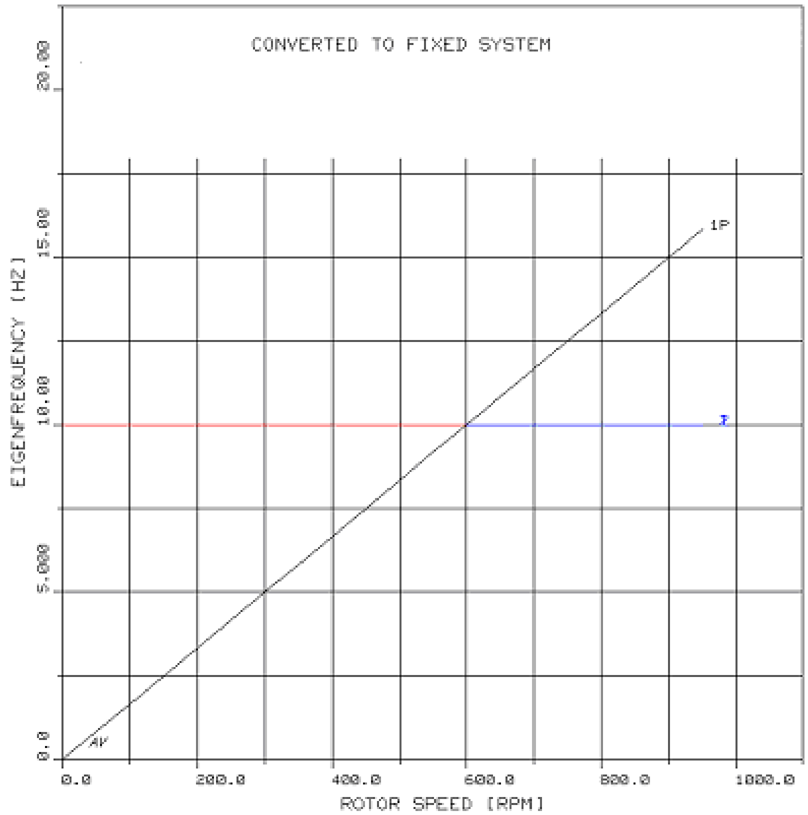


Fig. 20 Campbell Diagram for Rotating Mass Point Calculated in the Rotating System and Converted to the Fixed System

The same model can be calculated in the fixed system by changing the entry ROT to FIX on the ROTORD entry (rotor087.dat). Because there is no influence on rotation, the program cannot detect the whirl directions as shown in the Campbell diagram summary in Table 5. The results of the whirl resonances and the synchronous option are shown in Table 8. The Campbell diagram is shown in Fig. 21.

C A M P B E L L   D I A G R A M   S U M M A R Y

SOLUTIONS AFTER MODE TRACKING

SOLUTION NUMBER 1

STEP	ROTOR SPEED RPM	EIGENVALUE REAL	IMAG	FREQUENCY HZ	DAMPING [% CRIT]	WHIRL DIRECTI
1	0.00000E+00	0.00000E+00	6.28318E+01	1.00000E+01	0.00000E+00	LINEAR
2	5.00001E+01	0.00000E+00	6.28318E+01	1.00000E+01	0.00000E+00	LINEAR
3	1.00000E+02	0.00000E+00	6.28318E+01	1.00000E+01	0.00000E+00	LINEAR
4	1.50000E+02	0.00000E+00	6.28318E+01	1.00000E+01	0.00000E+00	LINEAR
5	2.00000E+02	0.00000E+00	6.28318E+01	1.00000E+01	0.00000E+00	LINEAR
6	2.50001E+02	0.00000E+00	6.28318E+01	1.00000E+01	0.00000E+00	LINEAR
7	3.00001E+02	0.00000E+00	6.28318E+01	1.00000E+01	0.00000E+00	LINEAR
8	3.50001E+02	0.00000E+00	6.28318E+01	1.00000E+01	0.00000E+00	LINEAR
9	4.00001E+02	0.00000E+00	6.28318E+01	1.00000E+01	0.00000E+00	LINEAR
10	4.50001E+02	0.00000E+00	6.28318E+01	1.00000E+01	0.00000E+00	LINEAR
11	5.00001E+02	0.00000E+00	6.28318E+01	1.00000E+01	0.00000E+00	LINEAR
12	5.50001E+02	0.00000E+00	6.28318E+01	1.00000E+01	0.00000E+00	LINEAR
13	6.00001E+02	0.00000E+00	6.28318E+01	1.00000E+01	0.00000E+00	LINEAR
14	6.50002E+02	0.00000E+00	6.28318E+01	1.00000E+01	0.00000E+00	LINEAR
15	7.00002E+02	0.00000E+00	6.28318E+01	1.00000E+01	0.00000E+00	LINEAR
16	7.50002E+02	0.00000E+00	6.28318E+01	1.00000E+01	0.00000E+00	LINEAR
17	8.00002E+02	0.00000E+00	6.28318E+01	1.00000E+01	0.00000E+00	LINEAR
18	8.50002E+02	0.00000E+00	6.28318E+01	1.00000E+01	0.00000E+00	LINEAR
19	9.00002E+02	0.00000E+00	6.28318E+01	1.00000E+01	0.00000E+00	LINEAR
20	9.50002E+02	0.00000E+00	6.28318E+01	1.00000E+01	0.00000E+00	LINEAR

C A M P B E L L   D I A G R A M   S U M M A R Y

SOLUTIONS AFTER MODE TRACKING

SOLUTION NUMBER 2

STEP	ROTOR SPEED RPM	EIGENVALUE REAL	IMAG	FREQUENCY HZ	DAMPING [% CRIT]	WHIRL DIRECTI
1	0.00000E+00	0.00000E+00	6.28318E+01	1.00000E+01	0.00000E+00	LINEAR
2	5.00001E+01	0.00000E+00	6.28318E+01	1.00000E+01	0.00000E+00	LINEAR
3	1.00000E+02	0.00000E+00	6.28318E+01	1.00000E+01	0.00000E+00	LINEAR
4	1.50000E+02	0.00000E+00	6.28318E+01	1.00000E+01	0.00000E+00	LINEAR
5	2.00000E+02	0.00000E+00	6.28318E+01	1.00000E+01	0.00000E+00	LINEAR
6	2.50001E+02	0.00000E+00	6.28318E+01	1.00000E+01	0.00000E+00	LINEAR
7	3.00001E+02	0.00000E+00	6.28318E+01	1.00000E+01	0.00000E+00	LINEAR
8	3.50001E+02	0.00000E+00	6.28318E+01	1.00000E+01	0.00000E+00	LINEAR
9	4.00001E+02	0.00000E+00	6.28318E+01	1.00000E+01	0.00000E+00	LINEAR
10	4.50001E+02	0.00000E+00	6.28318E+01	1.00000E+01	0.00000E+00	LINEAR
11	5.00001E+02	0.00000E+00	6.28318E+01	1.00000E+01	0.00000E+00	LINEAR
12	5.50001E+02	0.00000E+00	6.28318E+01	1.00000E+01	0.00000E+00	LINEAR
13	6.00001E+02	0.00000E+00	6.28318E+01	1.00000E+01	0.00000E+00	LINEAR
14	6.50002E+02	0.00000E+00	6.28318E+01	1.00000E+01	0.00000E+00	LINEAR
15	7.00002E+02	0.00000E+00	6.28318E+01	1.00000E+01	0.00000E+00	LINEAR
16	7.50002E+02	0.00000E+00	6.28318E+01	1.00000E+01	0.00000E+00	LINEAR
17	8.00002E+02	0.00000E+00	6.28318E+01	1.00000E+01	0.00000E+00	LINEAR
18	8.50002E+02	0.00000E+00	6.28318E+01	1.00000E+01	0.00000E+00	LINEAR
19	9.00002E+02	0.00000E+00	6.28318E+01	1.00000E+01	0.00000E+00	LINEAR
20	9.50002E+02	0.00000E+00	6.28318E+01	1.00000E+01	0.00000E+00	LINEAR

Table 7 Results of the Analysis in the Fixed System

```
          D E T E C T I O N   O F   R E S O N A N C E S   A N D   I N S T A B I L I T I E S

                          A N A L Y S I S   I N   F I X E D   S Y S T E M


WHIRL RESONANCE               SOLUTION      ROTOR SPEED     WHIRL
                               NUMBER           RPM       DIRECTION

                                    1       6.00000E+02    LINEAR
                                    2       6.00000E+02    LINEAR


INSTABILITIES                 SOLUTION      ROTOR SPEED     WHIRL
                               NUMBER           RPM       DIRECTION

                                            NONE  FOUND


CRITICAL SPEEDS FROM SYNCRONOUS ANALYSIS
                              SOLUTION      ROTOR SPEED     WHIRL
                               NUMBER           RPM       DIRECTION

                                    1       6.00000E+02    FORWARD
                                    2       6.00000E+02    FORWARD
```

Table 8 The F06 File Results in the Fixed System

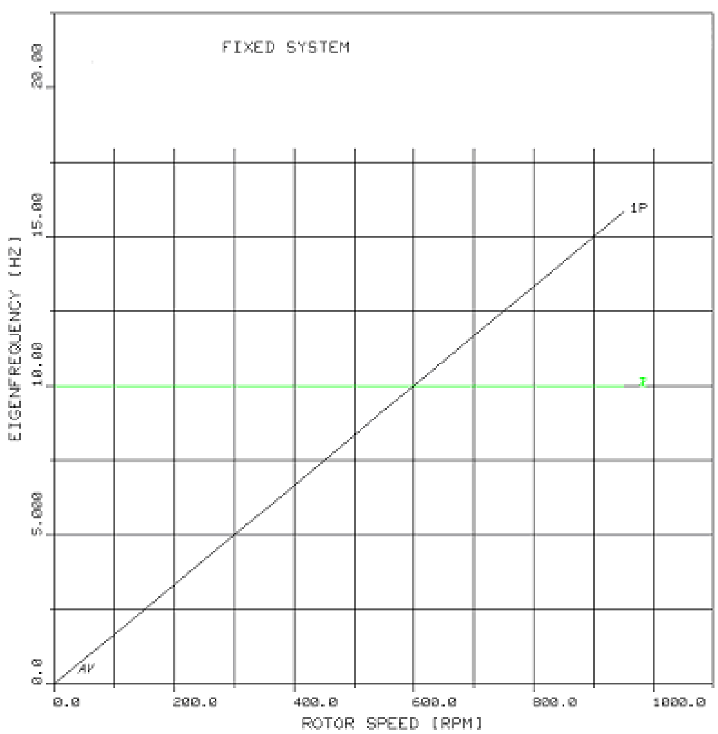


Fig. 21 Campbell Diagram of an Analysis in the Fixed System (No Whirl Directions Found)

6.1.2 Symmetric Model with Physical and Material Damping (rotor088.dat)

Replacing the CELAS elements with CROD elements and separating the non-rotating and the rotating parts with RBE2 elements, material damping can be defined in both systems. In this case, a SET must be defined which selects the rotating grid points. When damping is present, rotational effects are included and the program can calculate the whirl direction in both the rotating and the fixed analysis system. The new data deck is shown in Table 9. The Campbell diagram data is shown in Table 10. The Campbell diagrams for the rotating system and fixed system analyses are shown in Fig. 22 and Fig. 24, respectively.

The internal damping leads to instability above the critical speed. The external damping will stabilize the system. For a symmetric rotor, the instability point is at the rotor speed:

$$\Omega_{\text{Unstable}} = \Omega_0 \left(1 + \frac{\zeta_A}{\zeta_I}\right) \tag{58}$$

where Ω_0 is the critical speed (here 600 RPM). The internal (ζ_I) and external (ζ_A) damping values are:

$$\zeta_A = \zeta_I = \frac{g}{2} = 0.02$$

Hence, the instability point is at 1200 RPM. This is also calculated with Simcenter Nastran as shown in Table 11 and in the plot of the real part of the eigenvalues in Fig. 23. The real part of the eigenvalues are the same for analyses in both the rotating and fixed system. Because the damping is the real part of the eigenvalue divided by the imaginary part of the eigenvalue, the damping curves are different. The damping curves for the fixed system are shown in Fig. 25.

Table 12 shows the GPF file output, and Table 13 shows the output of the CSV file for the analysis in the rotating reference system. For information on the formating of the GPF and CSV files, see the “Interpretation of Rotor Dynamics Output” chapter.

```
NASTRAN $
$
assign output4='rotor088.gpf',unit=22, form=formatted
assign output4='rotor088.csv',unit=25, form=formatted
$
SOL 110
$
TIME 20000
DIAG 8
$
CEND
$
DISP = ALL
RMETHOD = 99
$
$
$
METHOD = 1
CMETHOD = 2
$
BEGIN BULK
$
```

```
PARAM   ROTGPF  22
PARAM   ROTCSV  25
PARAM   GRDPNT  0
PARAM   MODTRK  1
$       .       .       .       .       .       .       .
EIGRL   1                       2       1
EIGC    2       CLAN                                    2
$
ROTORG  11      101     THRU    103
$
$       SID     RSTART  RSTEP   NUMSTEP REFSYS  CMOUT   RUNIT   FUNIT
ROTORD  99      0.0     50.0    30      ROT     -1.0    RPM     HZ      +ROT0
$       ZSTEIN  ORBEPS  ROTPRT
+ROT0   NO      1.0E-5  3                                               +ROT1
$       RID1    RSET1   RSPEED1 RCORD1  W3-1    W4-1    RFORCE1
+ROT1   1       11      1.0     1
$
$
CORD2R  1                0.      0.      0.      0.      0.      1.     +XCRD001
+XCRD001 1.      0.      0.
$
$       Fixed
GRID    2                200.    0.      0.             123456
GRID    3                0.      200.    0.             123456
$
GRID    12               100.    0.      0.             3456
GRID    13               0.      100.    0.             3456
$       Rotating
GRID    101              0.      0.      0.             3456
$
GRID    102              100.    0.      0.             3456
GRID    103              0.      100.    0.             3456
$
RBE2    12      12       12     102
RBE2    13      13       12     103
$
$       connection 12 to 13 in the fixed system
$
CBAR    21      21      12      13      0.      0.      1.
PBAR    21      21      1000.   1.0+6   1.0+6   2.0+6
MAT1    21      200000.         0.3     0.
$
$
$       rotating part
$
CROD    101     101     101     102
CROD    102     102     101     103
$
PROD    101     101     0.394784
PROD    102     101     0.394784
MAT1    101     200000.         0.3     0.                      0.04
$
$       fixed part
$
CROD    1       1       2       12
CROD    2       2       3       13
$
PROD    1       1       0.394784
PROD    2       1       0.394784
MAT1    1       200000.         0.3     0.                      0.04
$
$
CONM2   100     101             0.100
$
ENDDATA
```

Table 9 Input File for a Simple Rotating Mass Point with Internal and External Material Damping

C A M P B E L L   D I A G R A M   S U M M A R Y

SOLUTIONS AFTER MODE TRACKING

SOLUTION NUMBER 1

STEP	ROTOR SPEED RPM	EIGENVALUE REAL	EIGENVALUE IMAG	FREQUENCY HZ	DAMPING [% CRIT]	WHIRL DIRECTION
1	0.00000E+00	-1.25664E+00	6.28193E+01	9.99800E+00	-2.00040E-02	LINEAR
2	5.00001E+01	-1.20427E+00	5.75833E+01	9.16467E+00	-2.09135E-02	FORWARD
3	1.00000E+02	-1.15190E+00	5.23474E+01	8.33134E+00	-2.20049E-02	FORWARD
4	1.50000E+02	-1.09953E+00	4.71115E+01	7.49802E+00	-2.33388E-02	FORWARD
5	2.00000E+02	-1.04716E+00	4.18756E+01	6.66471E+00	-2.50063E-02	FORWARD
6	2.50001E+02	-9.94787E-01	3.66398E+01	5.83141E+00	-2.71504E-02	FORWARD
7	3.00001E+02	-9.42418E-01	3.14041E+01	4.99811E+00	-3.00094E-02	FORWARD
8	3.50001E+02	-8.90050E-01	2.61683E+01	4.16482E+00	-3.40125E-02	FORWARD
9	4.00001E+02	-8.37682E-01	2.09327E+01	3.33154E+00	-4.00179E-02	FORWARD
10	4.50001E+02	-7.85316E-01	1.56970E+01	2.49826E+00	-5.00295E-02	FORWARD
11	5.00001E+02	-7.32950E-01	1.04615E+01	1.66499E+00	-7.00619E-02	FORWARD
12	5.50001E+02	-6.80586E-01	5.22592E+00	8.31731E-01	-1.30233E-01	FORWARD
13	6.00001E+02	-6.28222E-01	9.58121E-03	1.52490E-03	-6.55682E+01	BACKWARD
14	6.50002E+02	-5.75861E-01	5.24504E+00	8.34774E-01	-1.09792E-01	BACKWARD
15	7.00002E+02	-5.23500E-01	1.04804E+01	1.66801E+00	-4.99502E-02	BACKWARD
16	7.50002E+02	-4.71141E-01	1.57158E+01	2.50125E+00	-2.99788E-02	BACKWARD
17	8.00002E+02	-4.18784E-01	2.09511E+01	3.33448E+00	-1.99886E-02	BACKWARD
18	8.50002E+02	-3.66428E-01	2.61864E+01	4.16770E+00	-1.39931E-02	BACKWARD
19	9.00002E+02	-3.14074E-01	3.14217E+01	5.00091E+00	-9.99548E-03	BACKWARD
20	9.50002E+02	-2.61723E-01	3.66568E+01	5.83412E+00	-7.13980E-03	BACKWARD
21	1.00000E+03	-2.09373E-01	4.18920E+01	6.66732E+00	-4.99792E-03	BACKWARD
22	1.05000E+03	-1.57025E-01	4.71271E+01	7.50051E+00	-3.33195E-03	BACKWARD
23	1.10000E+03	-1.04680E-01	5.23622E+01	8.33370E+00	-1.99915E-03	BACKWARD
24	1.15000E+03	-5.23372E-02	5.75972E+01	9.16688E+00	-9.08677E-04	BACKWARD
25	1.20000E+03	3.14941E-06	6.28322E+01	1.00000E+01	5.01241E-08	BACKWARD
26	1.25000E+03	5.23408E-02	6.80671E+01	1.08332E+01	7.68959E-04	BACKWARD
27	1.30000E+03	1.04676E-01	7.33020E+01	1.16664E+01	1.42800E-03	BACKWARD
28	1.35000E+03	1.57008E-01	7.85368E+01	1.24995E+01	1.99916E-03	BACKWARD
29	1.40000E+03	2.09337E-01	8.37716E+01	1.33327E+01	2.49890E-03	BACKWARD
30	1.45000E+03	2.61663E-01	8.90063E+01	1.41658E+01	2.93983E-03	BACKWARD

C A M P B E L L   D I A G R A M   S U M M A R Y

SOLUTIONS AFTER MODE TRACKING

SOLUTION NUMBER 2

STEP	ROTOR SPEED RPM	EIGENVALUE REAL	EIGENVALUE IMAG	FREQUENCY HZ	DAMPING [% CRIT]	WHIRL DIRECTION
1	0.00000E+00	-1.25664E+00	6.28193E+01	9.99800E+00	-2.00040E-02	LINEAR
2	5.00001E+01	-1.30901E+00	6.80553E+01	1.08313E+01	-1.92345E-02	BACKWARD
3	1.00000E+02	-1.36138E+00	7.32914E+01	1.16647E+01	-1.85749E-02	BACKWARD
4	1.50000E+02	-1.41375E+00	7.85275E+01	1.24980E+01	-1.80032E-02	BACKWARD
5	2.00000E+02	-1.46612E+00	8.37636E+01	1.33314E+01	-1.75030E-02	BACKWARD
6	2.50001E+02	-1.51849E+00	8.89998E+01	1.41648E+01	-1.70617E-02	BACKWARD
7	3.00001E+02	-1.57086E+00	9.42361E+01	1.49981E+01	-1.66694E-02	BACKWARD
8	3.50001E+02	-1.62322E+00	9.94723E+01	1.58315E+01	-1.63183E-02	BACKWARD
9	4.00001E+02	-1.67559E+00	1.04709E+02	1.66649E+01	-1.60024E-02	BACKWARD
10	4.50001E+02	-1.72796E+00	1.09945E+02	1.74983E+01	-1.57166E-02	BACKWARD
11	5.00001E+02	-1.78032E+00	1.15181E+02	1.83317E+01	-1.54567E-02	BACKWARD
12	5.50001E+02	-1.83269E+00	1.20418E+02	1.91651E+01	-1.52194E-02	BACKWARD
13	6.00001E+02	-1.88505E+00	1.25654E+02	1.99985E+01	-1.50019E-02	BACKWARD
14	6.50002E+02	-1.93741E+00	1.30891E+02	2.08319E+01	-1.48017E-02	BACKWARD

15	7.00002E+02	-1.98977E+00	1.36128E+02	2.16654E+01	-1.46170E-02	BACKWARD
16	7.50002E+02	-2.04213E+00	1.41364E+02	2.24988E+01	-1.44459E-02	BACKWARD
17	8.00002E+02	-2.09449E+00	1.46601E+02	2.33323E+01	-1.42870E-02	BACKWARD
18	8.50002E+02	-2.14685E+00	1.51838E+02	2.41657E+01	-1.41391E-02	BACKWARD
19	9.00002E+02	-2.19920E+00	1.57074E+02	2.49992E+01	-1.40010E-02	BACKWARD
20	9.50002E+02	-2.25155E+00	1.62311E+02	2.58326E+01	-1.38718E-02	BACKWARD
21	1.00000E+03	-2.30390E+00	1.67548E+02	2.66661E+01	-1.37507E-02	BACKWARD
22	1.05000E+03	-2.35625E+00	1.72785E+02	2.74996E+01	-1.36369E-02	BACKWARD
23	1.10000E+03	-2.40859E+00	1.78022E+02	2.83331E+01	-1.35298E-02	BACKWARD
24	1.15000E+03	-2.46094E+00	1.83259E+02	2.91665E+01	-1.34287E-02	BACKWARD
25	1.20000E+03	-2.51328E+00	1.88496E+02	3.00000E+01	-1.33333E-02	BACKWARD
26	1.25000E+03	-2.56561E+00	1.93733E+02	3.08336E+01	-1.32430E-02	BACKWARD
27	1.30000E+03	-2.61795E+00	1.98970E+02	3.16671E+01	-1.31575E-02	BACKWARD
28	1.35000E+03	-2.67028E+00	2.04207E+02	3.25006E+01	-1.30763E-02	BACKWARD
29	1.40000E+03	-2.72261E+00	2.09444E+02	3.33341E+01	-1.29992E-02	BACKWARD
30	1.45000E+03	-2.77494E+00	2.14682E+02	3.41676E+01	-1.29258E-02	BACKWARD

Table 10 Campbell Summary for Model with Damping

```
               D E T E C T I O N   O F   R E S O N A N C E S   A N D   I N S T A B I L I T I E S

                                A N A L Y S I S   I N   R O T A T I N G   S Y S T E M



FORWARD WHIRL RESONANCE         SOLUTION      ROTOR SPEED     WHIRL
                                 NUMBER           RPM       DIRECTION

                                      1       5.99908E+02    FORWARD


BACKWARD WHIRL RESONANCE        SOLUTION      ROTOR SPEED     WHIRL
                                 NUMBER           RPM       DIRECTION

                                      2       5.99910E+02    BACKWARD


INSTABILITIES                   SOLUTION      ROTOR SPEED     WHIRL
                                 NUMBER           RPM       DIRECTION

START                                 1       1.20000E+03    FORWARD


CRITICAL SPEEDS FROM SYNCHRONOUS ANALYSIS
                                SOLUTION      ROTOR SPEED     WHIRL
                                 NUMBER           RPM       DIRECTION

                                       1       6.00000E+02    FORWARD
                                       2       6.00000E+02    BACKWARD
```

Table 11 Resonances for Model with Damping

```
data 10001
  0.00000E+00
  5.00001E+01
  1.00000E+02
  1.50000E+02
  ..........
  ..........
data 20001        20002
  9.99800E+00  9.99800E+00
  9.16467E+00  1.08313E+01
  8.33134E+00  1.16647E+01
  7.49802E+00  1.24980E+01
  ..........
  ..........
data 30001        30002
 -2.00040E-02 -2.00040E-02
 -2.09135E-02 -1.92345E-02
 -2.20049E-02 -1.85749E-02
 -2.33388E-02 -1.80032E-02
  ..........
  ..........
data 40001        40002
 -1.25664E+00 -1.25664E+00
 -1.20427E+00 -1.30901E+00
 -1.15190E+00 -1.36138E+00
 -1.09953E+00 -1.41375E+00
  ..........
  ..........
data 50001        50002
  4.00000E+00  4.00000E+00
  3.00000E+00  2.00000E+00
  3.00000E+00  2.00000E+00
  3.00000E+00  2.00000E+00
  ..........
  ..........
DATA 50000
   0.0000E+00
   0.0000E+00
   0.0000E+00
   0.0000E+00
  ..........
  ..........
$
heading " "
subtitl " ROTATING SYSTEM "
xaxis   "ROTOR SPEED [RPM] "
yaxis   "EIGENFREQUENCY [HZ] "
xgrid
ygrid
$
col  19991 = (  1.66667E-02 ) * col  10001
color  vector
funct    19991 10001     "1.00 P"    line   lcol   50000
col  19992 = (  3.33333E-02 ) * col  10001
color  vector
funct    19992 10001     "2.00 P"    line   lcol   50000
col  19993 = (  3.33333E-02 ) * col  10001
color  vector
funct    19993 10001     "2.00 P"    line   lcol   50000
funct    20001 10001     " 1                  "    line   lcol   50001
funct    20002 10001     " 2                  "    line   lcol   50002
plot window   1  15  1  15
$
heading " "
subtitl " ROTATING SYSTEM "
xaxis   "ROTOR SPEED [RPM] "
yaxis   "DAMPING "
xgrid
ygrid
$
color  1
```

```
funct    30001 10001     " 1                  "    line
funct    30002 10001     " 2                  "    line
plot window   1  15  1  15
$
heading " "
subtitl " ROTATING SYSTEM "
xaxis   "ROTOR SPEED [RPM] "
yaxis   "REAL EIGENVALUE "
xgrid
ygrid
$
color  1
funct    40001 10001     " 1                  "    line
funct    40002 10001     " 2                  "    line
plot window   1  15  1  15
data 60001        60002
  9.99800E+00  9.99800E+00
  9.99800E+00  9.99800E+00
  9.99801E+00  9.99801E+00
  9.99803E+00  9.99803E+00
  ...........
  ...........
data 70001        70002
  4.00000E+00  4.00000E+00
  3.00000E+00  2.00000E+00
  3.00000E+00  2.00000E+00
  3.00000E+00  2.00000E+00
  ...........
  ...........
DATA 70000
   0.0000E+00
   0.0000E+00
   0.0000E+00
   0.0000E+00
  ...........
  ...........
$
heading " "
subtitl " CONVERTED TO FIXED SYSTEM "
xaxis   "ROTOR SPEED [RPM] "
yaxis   "EIGENFREQUENCY [HZ] "
xgrid
ygrid
$
col  59991 = (  1.66667E-02 ) * col  10001
color  vector
funct    59991 10001     "1.00 P"    line   lcol   70000
funct    60001 10001     " 1                  "    line   lcol   70001
funct    60002 10001     " 2                  "    line   lcol   70002
plot window   1  15  1  15
```

Table 12 GPF Output File (Abbreviated Listing) for a Simple Rotating Mass Point

```
   10001
 0.00E+00
 5.00E+01
 1.00E+02
 1.50E+02
 ........
 ........
   20001     20002
 1.00E+01  1.00E+01
 9.16E+00  1.08E+01
 8.33E+00  1.17E+01
 7.50E+00  1.25E+01
 ........
 ........
   30001     30002
-2.00E-02 -2.00E-02
-2.09E-02 -1.92E-02
-2.20E-02 -1.86E-02
-2.33E-02 -1.80E-02
 ........
 ........
   40001     40002
-1.26E+00 -1.26E+00
-1.20E+00 -1.31E+00
-1.15E+00 -1.36E+00
-1.10E+00 -1.41E+00
 ........
 ........
   50001     50002
 4.00E+00  4.00E+00
 3.00E+00  2.00E+00
 3.00E+00  2.00E+00
 3.00E+00  2.00E+00
 ........
 ........
   50000
 0.00E+00
 0.00E+00
 0.00E+00
 0.00E+00
 ........
 ........
   60001     60002
 1.00E+01  1.00E+01
 1.00E+01  1.00E+01
 1.00E+01  1.00E+01
 1.00E+01  1.00E+01
 ........
 ........
   70001     70002
 4.00E+00  4.00E+00
 3.00E+00  2.00E+00
 3.00E+00  2.00E+00
 3.00E+00  2.00E+00
 ........
 ........
   70000
 0.00E+00
 0.00E+00
 0.00E+00
 0.00E+00
 ........
 ........
```

Table 13 CSV Output File (Abbreviated Listing) for a Simple Rotating Mass Point

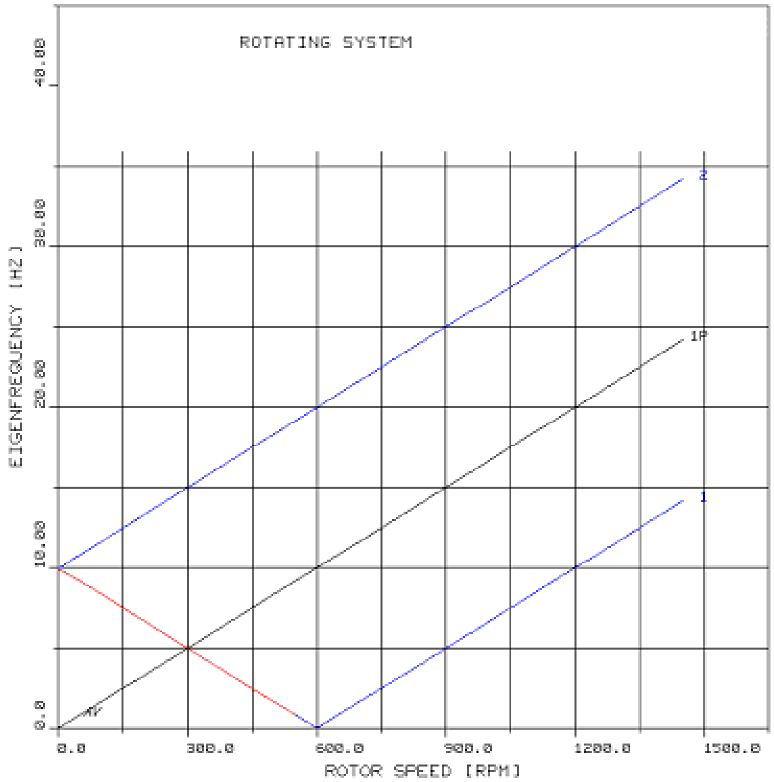


Fig. 22 Campbell Diagram in the Rotating Analysis System

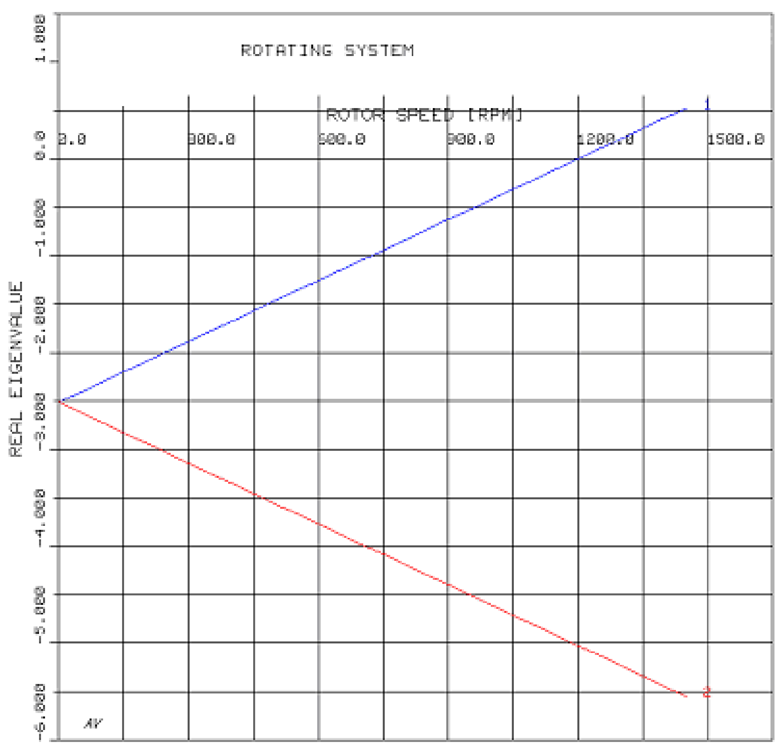


Fig. 23 Real Part of the Eigenvalues Calculated in the Rotating System

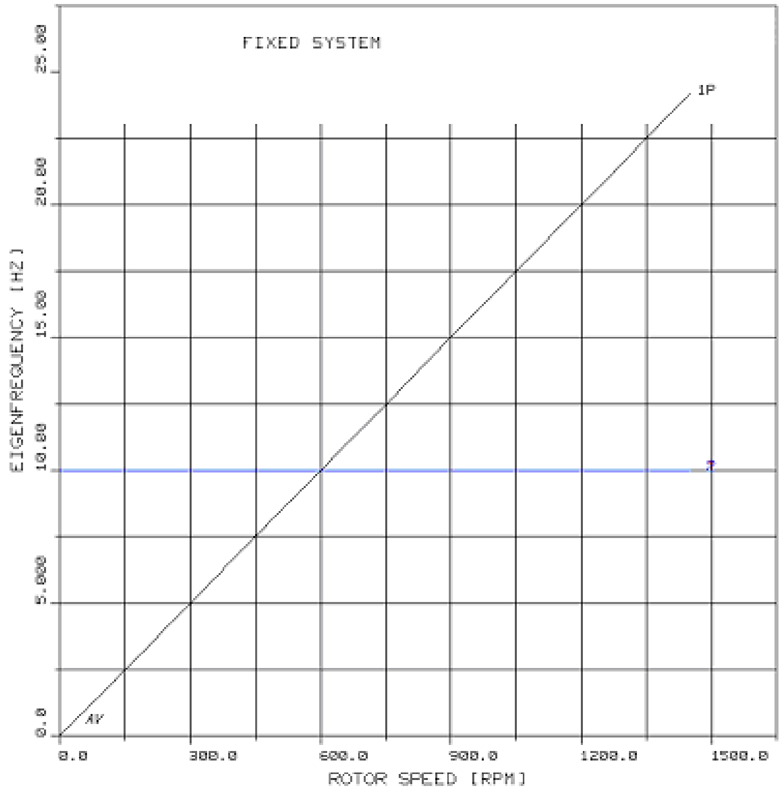


Fig. 24 Campbell Diagram in the Fixed System

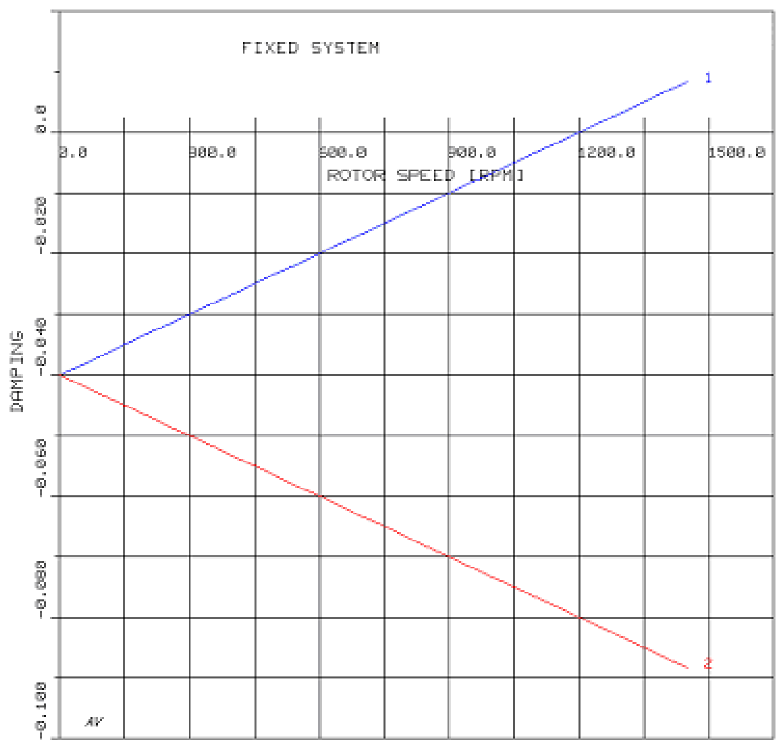


Fig. 25 Damping Diagram Calculated in the Fixed System

6.1.3 Unsymmetric Rotor with Damping (rotor089.dat)

When damping is introduced, the model must be separated into a rotating and a non-rotating part. The input deck for the modified one-mass model is shown in Table 14. The results are shown in Table 15. The Campbell diagram (Fig. 26) shows a region of zero frequency between 570 and 627 RPM. The real part of the eigenvalues are shown in Fig. 27. In the speed range of zero frequencies, this rotor becomes unstable. This is due to the centrifugal matrix. The instability can be called a centrifugal instability. This instability does not occur when calculating in the fixed system, which would not be correct for unsymmetric rotors.

In this example, approximately 2% internal and 1% external damping is used. One branch of the damping increases, and the other branch decreases with speed. At approximately 900 RPM, the real part gets positive, and an instability occurs due to the internal damping. If only internal damping is present, the instability occurs at the critical speed. Adding external damping shifts the damping instability point up to higher rotor speeds. Fig. 28 shows the results of the eigenfrequencies converted to the fixed system. The region of zero frequency in the rotating system is represented by the green line between the critical speeds.

The degree of unsymmetry is defined as:

$$\mu = \frac{k_x - k_y}{k_x - k_y} \tag{59}$$

Here, the value is:

$$\mu = \frac{434.2624 - 355.3056}{434.2624 + 355.3056} = \frac{78.9568}{789.568} = 0.1$$

The rotor is only slightly unsymmetric and the external damping is half of the internal damping. Therefore, the theoretical instability point is close to 900 RPM. Simcenter Nastran calculated 897.5 RPM.

```
NASTRAN $
$
assign output4='rotor089.gpf',unit=22, form=formatted
assign output4='rotor089.csv',unit=25, form=formatted
$
SOL 110
$
TIME 20000
DIAG 8
$
CEND
$
DISP = ALL
RMETHOD = 99
$
METHOD = 1
CMETHOD = 2
$
BEGIN BULK
$
PARAM   ROTGPF  22
PARAM   ROTCSV  25
PARAM   GRDPNT  0
PARAM   MODTRK  1
$       .       .       .       .       .       .       .
EIGRL   1                       2       1
```

```
EIGC    2       CLAN                                    2
$
ROTORG  11      101     THRU    102
$
$       SID     RSTART  RSTEP   NUMSTEP REFSYS  CMOUT   RUNIT   FUNIT
ROTORD  99      0.0     5.0     200     ROT     -1.0    RPM     HZ      +ROT0
$       ZSTEIN  ORBEPS  ROTPRT
+ROT0   NO      1.0E-5  3                                               +ROT1
$       RID1    RSET1   RSPEED1 RCORD1  W3-1    W4-1    RFORCE1
+ROT1   1       11      1.0     1
$
CORD2R  1               0.      0.      0.      0.      0.      1.      +XCRD0
+XCRD0  1.      0.      0.
$
$       Fixed
$
GRID    1               0.      0.      0.              3456
$
$       Rotor point
$
GRID    101             0.      0.      0.              3456
$
$      Constrained grid point for the internal damping
$
GRID    102             0.      0.      0.              123456
$
RBE2    1       1       12      101
$
$      Stiffness in the rotating part
$
CELAS1  101     101     101     1
CELAS1  102     102     101     2
$
PELAS   101     434.2624
PELAS   102     355.3056
$
$       Internal damping of the rotating part
$
CDAMP1  111     111     101     1       102     1
CDAMP1  112     112     101     2       102     2
$
PDAMP   111     434.26-3
PDAMP   112     355.30-3
$
$       External damping of the bearings
$
CDAMP1  121     121     1       1
CDAMP1  122     122     1       2
$
PDAMP   121     217.13-3
PDAMP   122     177.65-3
$
CONM2   100     101             0.100
$
ENDDATA
```

Table 14 Input File for an Unsymmetric Rotor with External and Internal Damping

```
           D E T E C T I O N   O F   R E S O N A N C E S   A N D   I N S T A B I L I T I E S

                        A N A L Y S I S   I N   R O T A T I N G   S Y S T E M


 FORWARD WHIRL RESONANCE         SOLUTION       ROTOR SPEED     WHIRL
                                  NUMBER           RPM        DIRECTION

 START                                 1        5.70002E+02     FORWARD
 END                                   1        6.25002E+02     FORWARD

 BACKWARD WHIRL RESONANCE        SOLUTION       ROTOR SPEED     WHIRL
                                  NUMBER           RPM        DIRECTION

                                       2        5.99994E+02     BACKWARD

 INSTABILITIES                   SOLUTION       ROTOR SPEED     WHIRL
                                  NUMBER           RPM        DIRECTION

 START                                 1        5.71005E+02     LINEAR
 START                                 1        8.97491E+02     FORWARD

 CRITICAL SPEEDS FROM SYNCHRONOUS ANALYSIS
                                 SOLUTION       ROTOR SPEED     WHIRL
                                  NUMBER           RPM        DIRECTION

                                       1        5.69210E+02     FORWARD
                                       2        6.29286E+02     FORWARD
                                       3        6.00375E+02     BACKWARD
```

Table 15 Unsymmetric Rotor Results with Internal and External Damping

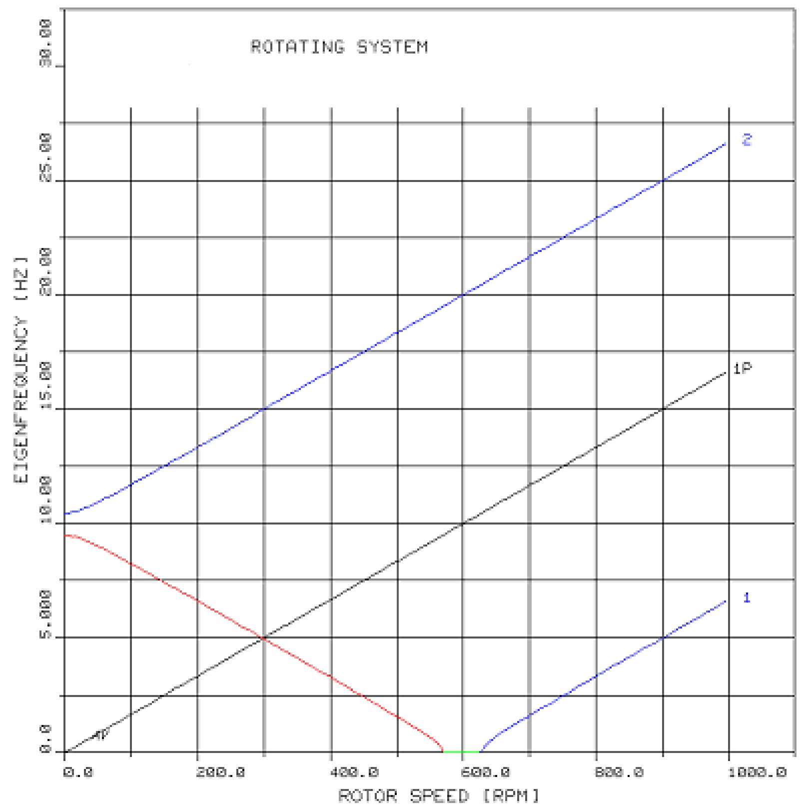


Fig. 26 Campbell Diagram of Rotating Mass Point with Unsymmetric Stiffness and Damping

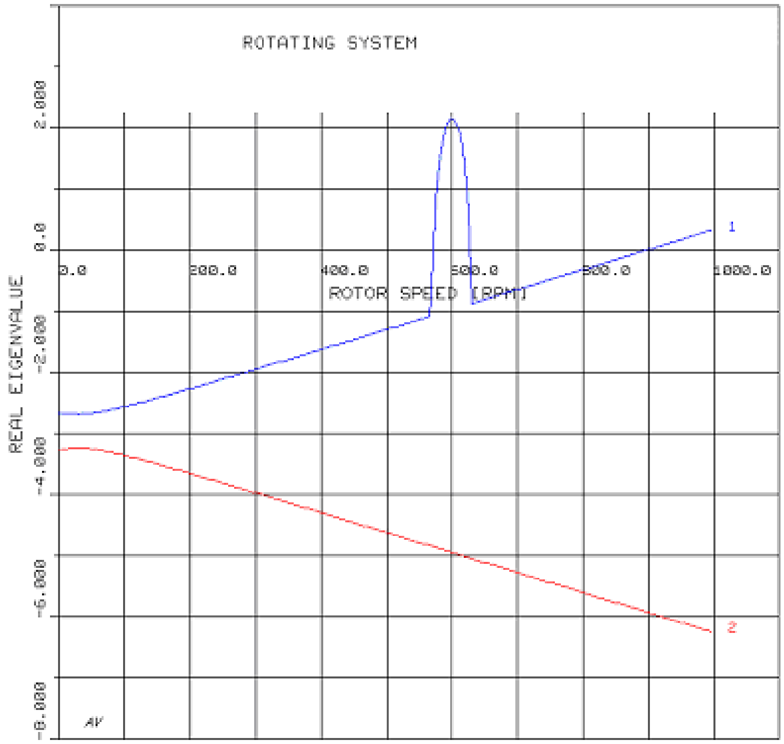


Fig. 27 Real Part of Solution with Centrifugal and Damping Instabilities

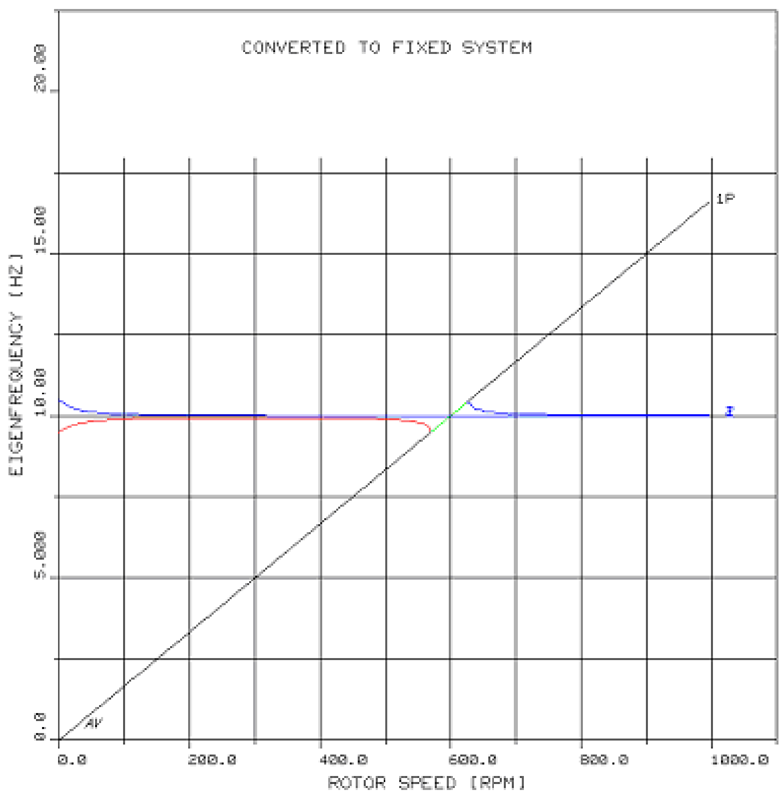


Fig. 28 Results of the Analysis in the Rotating System Converted to the Fixed System

6.1.4 Symmetric Rotor in Unsymmetric Bearings (rotor090.dat)

This type of application must be analyzed in the fixed reference system.

The instability point is given by:

$$\Omega_{\text{Unstable}} = \omega_0 \sqrt{\left(1+\frac{\zeta_a}{\zeta_i}\right)^2 + \left(\frac{\mu}{2\zeta_i}\right)^2} \tag{60}$$

$$\Omega_{\text{Unstable}} = \omega_0 \sqrt{\left(1+\frac{0.015}{0.030}\right)^2 + \left(\frac{0.1}{0.060}\right)^2} = \Omega_0\sqrt{1.5^2 + 1.6667^2} = \Omega_0 \cdot 2.24227$$

$$\omega_0 = \sqrt{\frac{k_x + k_y}{2m}} = \sqrt{\frac{434.2624 + 355.3056}{2 \bullet 0.1}} = 62.83184 \text{ rad/s}$$

With a critical speed of 600 RPM, the theoretical instability starts at 1345.36 RPM. Simcenter Nastran finds the instability at 1345.35 RPM.

```
NASTRAN $
$
assign output4='rotor090.gpf',unit=22, form=formatted
assign output4='rotor090.csv',unit=25, form=formatted
$
SOL 110
$
TIME 20000
DIAG 8
$
CEND
$
DISP = ALL
RMETHOD = 99
$
$
METHOD = 1
CMETHOD = 2
$
$
BEGIN BULK
$
PARAM   ROTGPF  22
PARAM   ROTCSV  25
PARAM   GRDPNT  0
PARAM   MODTRK  1
$       .       .       .       .       .       .       .
EIGRL   1                       2       1
EIGC    2       CLAN                                    2
$
ROTORG  11      101     THRU    102
$
$       SID     RSTART  RSTEP   NUMSTEP REFSYS  CMOUT   RUNIT   FUNIT
ROTORD  99      0.0     5.0     400     FIX     -1.0    RPM     HZ      +ROT0
$       ZSTEIN  ORBEPS  ROTPRT
+ROT0   NO      1.0E-9  3                                               +ROT1
$       RID1    RSET1   RSPEED1 RCORD1  W3-1    W4-1    RFORCE1
+ROT1   1       11      1.0     1
$
$
CORD2R  1               0.      0.      0.      0.      0.      1.      +XCRD0
+XCRD0  1.      0.      0.
$
$       Fixed
$
```

```
GRID    1               0.      0.      0.              3456
$
$       Rotor
$
GRID    101             0.      0.      0.              3456
GRID    102             0.      0.      0.              123456
$
RBE2    1       1       12      101
$
CELAS1  101     101     1       1
CELAS1  102     102     1       2
$
PELAS   101     434.2624
PELAS   102     355.3056
$
$       Internal damping
$
CDAMP1  111     111     101     1       102     1
CDAMP1  112     112     101     2       102     2
$
PDAMP   111     0.3770
PDAMP   112     0.3770
$
$       External damping
$
CDAMP1  121     121     1       1
CDAMP1  122     122     1       2
$
PDAMP   121     0.1885
PDAMP   122     0.1885
$
CONM2   100     101             0.100
$
ENDDATA
```

Table 16 Input File for Rotor with Unsymmetric Bearings

```
              D E T E C T I O N   O F   R E S O N A N C E S   A N D   I N S T A B I L I T I E S

                              A N A L Y S I S   I N   F I X E D   S Y S T E M



 WHIRL RESONANCE                SOLUTION      ROTOR SPEED     WHIRL
                                 NUMBER           RPM       DIRECTION

                                     1       5.74282E+02    LINEAR
                                     2       6.22451E+02    LINEAR


 INSTABILITIES                  SOLUTION      ROTOR SPEED     WHIRL
                                 NUMBER           RPM       DIRECTION

 START                               2       1.34535E+03    FORWARD


 CRITICAL SPEEDS FROM SYNCHRONOUS ANALYSIS
                                SOLUTION      ROTOR SPEED     WHIRL
                                 NUMBER           RPM       DIRECTION

                                     1       5.69210E+02    LINEAR
                                     2       6.29286E+02    LINEAR
```

Table 17 Whirl Resonance and Instability

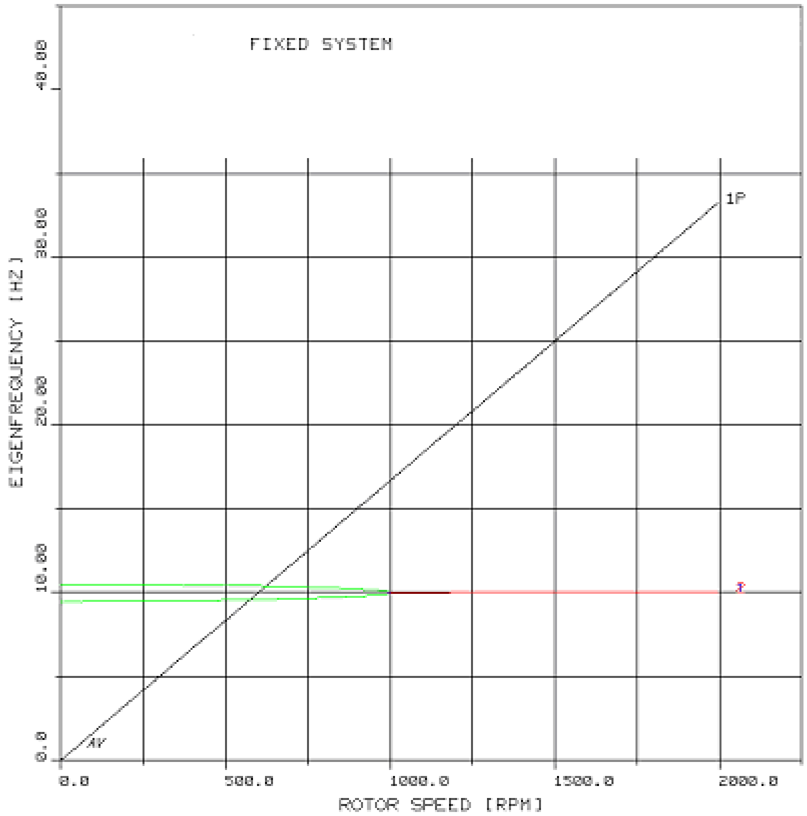


Fig. 29 Campbell Diagram of a Rotor with Unsymmetric Bearings in the Fixed System

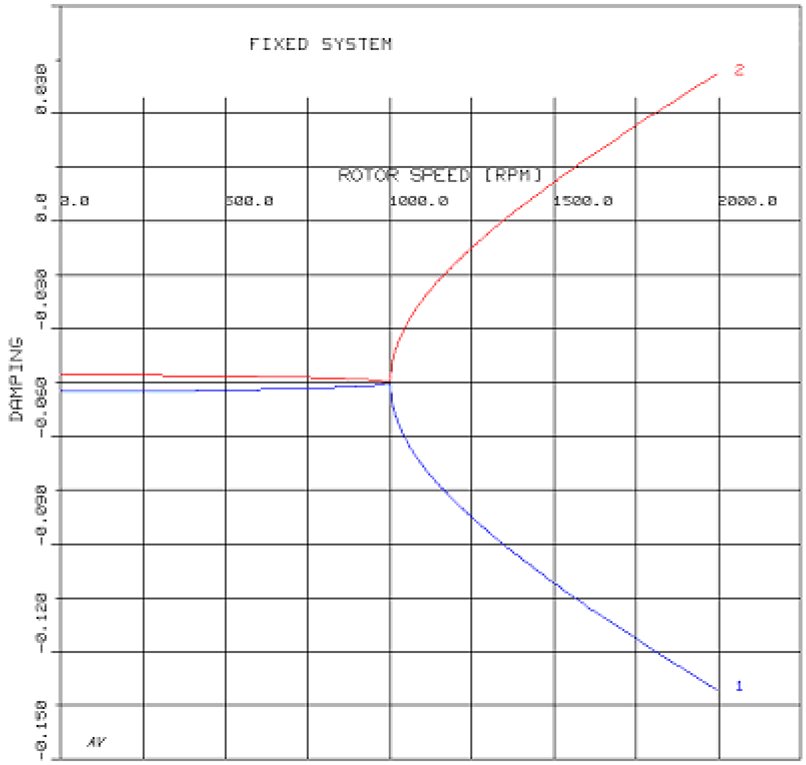


Fig. 30 Damping Diagram of a Rotor with Unsymmetric Bearings in the Fixed System

6.2 Laval Rotor Examples

A Laval rotor is a very simple example of a rotor. It consists of a rotating disk mounted on an elastic shaft supported by stiff or elastic bearings. The rotor is based on a simple steam turbine first patented by Carl G.P. de Laval in 1883. The Laval rotor is similar to an analytical rotor model published by Henry H. Jeffcott in 1919.

The theoretical solution for the Laval rotor can be derived from the equations of motion as published in ref. [1]. In this example, different variations are derived from the original model and compared with the theoretical solutions.

6.2.1 The Theoretical Model for the Laval Rotor

The length of the shaft is 1000, the polar moment of inertia is: $\Theta_P = 5000$, the moment of inertia for bending about x and y-axis is: $\Theta_A = 2500$. This corresponds to a thin disk. The mass is 0.040. The stiffness of the bearings is $k_x = k_y = 1974$ at each end both in x- and y-direction. The length of the shaft is 1000 ant the disk is mounted in the middle point. Hence, the distance between the bearing and the disk is a=500. The stiffness for the tilt motion is:

$$k_R = 2 k_X\, a^2 = 9.87E+8$$

For a cylinder with radius R and height H, the moments of inertia are given by:

$$\Theta_P = \frac{m}{2} R^2 \tag{61}$$

$$\Theta_A = \frac{m}{12}(3R^2 + H^2) = \frac{\Theta_P}{2} + \frac{m\,H^2}{12} \tag{62}$$

The mass of the disk is:

$$m = \rho\,\pi R^2\, H \tag{63}$$

In this example, the following values are used:

R=500

H=6.49

$\rho = 7.85E-9$

The following values are obtained:

$m = 0.40$

$\Theta_P = 5000$

$\Theta_A = 2500$

The bending eigenfrequency is:

$$f_B = \frac{1}{2\pi}\sqrt{\frac{k_x}{m}} = \sqrt{\frac{2 \cdot 1974}{0.04}} = \frac{314.17}{2\pi} = 50.0\,[\mathrm{Hz}]$$

The tilting eigenfrequency is then:

$$f_R = \frac{1}{2\pi}\sqrt{\frac{k_R}{\Theta_A}} = \sqrt{\frac{9.87E+8}{2500}} = \frac{628.33}{2\pi} = 100.0\,[\mathrm{Hz}]$$

The analytical solution as a function of rotor speed is given in [1]:

$$\omega_{1,2} = \frac{\Theta_P}{2\Theta_A}\Omega \pm \sqrt{\left(\frac{\Theta_P}{2\Theta_A}\Omega\right)^2 + \frac{k_R}{\Theta_A}} \tag{64}$$

The results are shown in Fig. 31. The two solutions are the red and green lines in the figure. Solution of the complex eigenvalue problem of the equations of motion yields complex conjugate pairs where the imaginary part represents the eigenfrequency. In Simcenter Nastran, all complex solutions are calculated, but only the solutions with positive eigenfrequency are used for post-processing and establishment of Campbell diagrams. Campbell diagrams are plots of eigenfrequencies as a function of rotor speed. The blue line is the solution with positive eigenfrequency (mirror of the green line) and represents the backward whirl motion. The red line is the forward whirl motion.

The asymptotic behavior is:

- The eigenfrequency of the backward whirl tends to zero for increasing rotor speed.
- The eigenfrequency of the forward whirl approaches the 2P (2 per rev) line for increasing rotor speeds. Since the forward whirl does not cross the 1P line, there is no critical speed.
- The crossing point between the 1P and the backward whirl mode is at 3465 RPM.

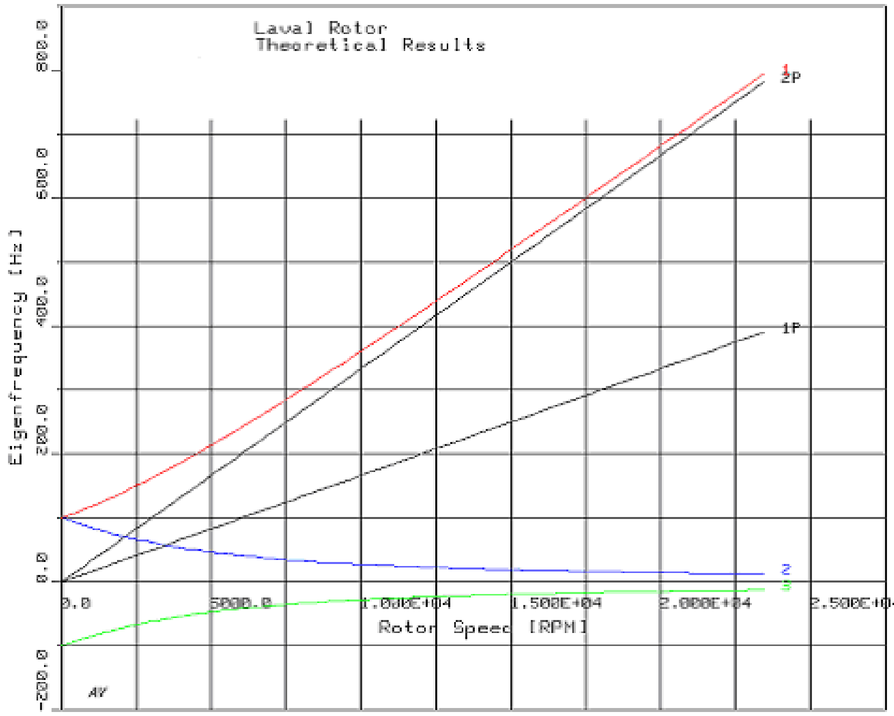


Fig. 31 Theoretical Results for the Laval Rotor

6.2.2 Analysis of the Laval Rotor (rotor091.dat, rotor092.dat)

Fig. 32 depicts a Laval rotor. A depiction of the FE representation (rotor091.dat) for the Laval rotor is also provided below.

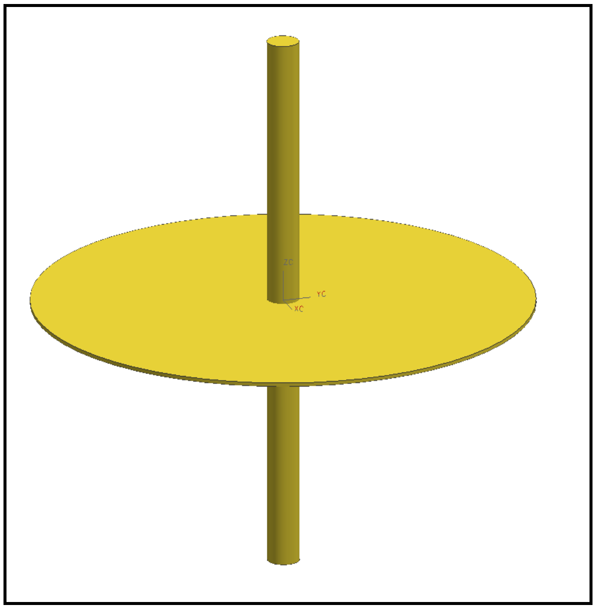
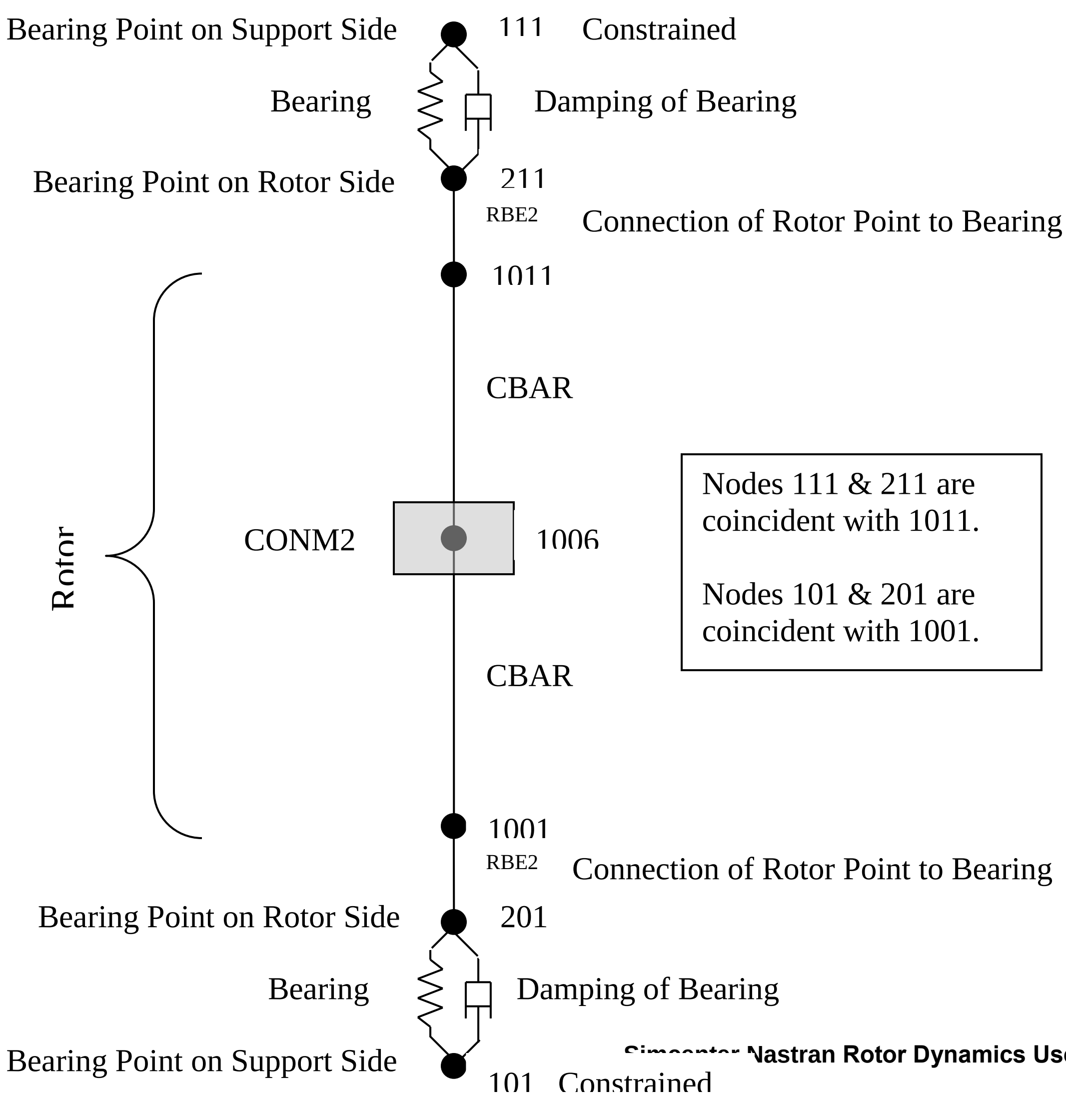


Fig. 32 Laval Rotor



Sketch of FE Model (rotor091.dat)

You can derive other models from this model (rotor091.dat) by changing the inertia parameters of the rotor disk. The Simcenter Nastran input file is shown in Table 18.

The results of the Laval rotor calculated in the fixed system are shown in Table 19. There is no critical speed for the forward whirl mode. The backward resonance is found at 3465.47 RPM for both the rotating and fixed system analysis. The synchronous analysis is only possible for the non-rotating formulation and the backward critical speed was found at 3464.16 RPM. In the synchronous analysis, the damping is actually neglected.

The intersection points with the 1P and the 2P curves are found by linear interpolation between the calculated values for the rotor speeds. Therefore, sufficient rotor speed values should be used. In the examples, 118 values for rotor speed and a step size of 200 RPM were used. With 24 values and a step size of 1000 RPM, the curves are reasonably smooth, but the intersection point with 1P is found at 3473.10 instead of 3464.47 RPM, which means an error of 0.25%.

The results from Simcenter Nastran are shown in Fig. 33. The symbols represent the theoretical solution. The results of Simcenter Nastran are identical to those of the theoretical solution. The conversion from the fixed system to the rotating system is shown in Fig. 34. The curves are found by subtracting the rotor speed from the forward whirl and adding the rotor speed to the backwards whirl. In this case two identical solutions are found. Both curves tend asymptotically to the 1P line.

The same rotor was analyzed in the rotating system by simply changing FIX to ROT on the ROTORD entry (rotor092.dat). The results of this same model calculated in the rotating system are shown in Table 20. The results in Fig. 35 are identical to those in Fig. 34. A conversion of these results to the fixed system is shown in Fig. 36. These curves are identical to those of Fig. 33.

Analysis in both systems leads to identical results, which are in agreement with theory.

```
NASTRAN $
$
assign output4='OUTDIR:rotor091.gpf',unit=22, form=formatted
assign output4='OUTDIR:rotor091.csv',unit=25, form=formatted
$
SOL 110
$
TIME 20000
DIAG 8
$
CEND
$
SPC = 1
$
SET 1 = 1006
$
DISP = 1
$
RMETHOD = 99
```

```
$
METHOD = 1
CMETHOD = 2
$
BEGIN BULK
$
PARAM,ROTGPF,22
PARAM,ROTCSV,25
PARAM,MODTRK,1
$
ROTORG  11     1001    THRU   1024
$
$       SID     RSTART  RSTEP   NUMSTEP REFSYS CMOUT   RUNIT   FUNIT
ROTORD  99      0.0     200.0   118     FIX     -1.0    RPM     HZ      +ROT0
$       ZSTEIN  ORBEPS  ROTPRT
+ROT0           1.0E-6                                                  +ROT1
$       RID1    RSET1   RSPEED1 RCORD1  W3-1    W4-1    RFORCE1
+ROT1           11      1.0     1
$
EIGRL   1                        4      1
EIGC    2       CLAN                                     4
$
$       Coordinate system for definition of rotor axis of rotation
$
CORD2R  1                 0.      0.      0.      0.      0.      1.     +XCRD001
+XCRD001 1.      0.      0.
$
$       Shaft stiffness. here a stiff massless shaft is used
$
PBAR    1000    1000     7853.98 4.909+6 4.909+6 9.817+6                 +P00V8AA
+P00V8AA 0.      50.     50.     0.      0.     -50.    -50.     0.     +P00V8AB
+P00V8AB 0.9     0.9
MAT1    1000     2.000+9         0.3     0.0
$
$      Nodes
$
GRID    1001              0.      0.     -500.
GRID    1006              0.      0.      0.
GRID    1011              0.      0.      500.
$
$      Elements
$
CBAR    1001    1000    1001    1006     1.      0.      0.
CBAR    1010    1000    1006    1011     1.      0.      0.
$
$       Mass and inertia of the rotor disk
$
CONM2   1106    1006            0.040                                  +C1106
+C1106  2500.0          2500.0                  5000.0
$
$       Bearing points on the support side (non rotating)
$
GRID    101             0.      0.     -500.
GRID    111             0.      0.      500.
$
$       Bearing points on the rotor side (coincident with the rotor nodes)
$
GRID    201             0.      0.     -500.
GRID    211             0.      0.      500.
$
$       Bearings
$
CELAS1  101     101     101     1       201     1
CELAS1  102     102     101     2       201     2
CELAS1  111     111     111     1       211     1
CELAS1  112     112     111     2       211     2
$
PELAS   101     1974.
PELAS   102     1974.
PELAS   111     1974.
PELAS   112     1974.
$
```

```
$       Damping of the bearings
$
CDAMP1  301     301     101     1       201     1
CDAMP1  302     302     101     2       201     2
CDAMP1  311     311     111     1       211     1
CDAMP1  312     312     111     2       211     2
$
$       Here the damping is not considered. Small values are used in order to avoid
$       numerical problems
$
PDAMP   301     1.0-6
PDAMP   302     1.0-6
PDAMP   311     1.0-6
PDAMP   312     1.0-6
$
$       Connection of rotor points to bearing
$
RBE2    201     1001    123456  201
RBE2    211     1011    123456  211
$
$       Constraints
$
SPC1    1       36      1001
SPC1    1       123456  101     111
$
ENDDATA
```

Table 18 Simcenter Nastran Input File for the Laval Rotor

```
              D E T E C T I O N   O F   R E S O N A N C E S   A N D   I N S T A B I L I T I E S

                              A N A L Y S I S   I N   F I X E D   S Y S T E M


 WHIRL RESONANCE                SOLUTION      ROTOR SPEED     WHIRL
                                 NUMBER           RPM       DIRECTION

                                      1       3.00005E+03    LINEAR
                                      2       3.00005E+03    LINEAR
                                      3       3.46447E+03    BACKWARD


 INSTABILITIES                  SOLUTION      ROTOR SPEED     WHIRL
                                 NUMBER           RPM       DIRECTION

                                              NONE  FOUND


 CRITICAL SPEEDS FROM SYNCHRONOUS ANALYSIS
                                SOLUTION      ROTOR SPEED     WHIRL
                                 NUMBER           RPM       DIRECTION

                                      1       3.00005E+03    LINEAR
                                      2       3.00005E+03    LINEAR
                                      3       3.46416E+03    BACKWARD
```

Table 19 Laval Rotor Results in the Fixed Reference System

```
              D E T E C T I O N   O F   R E S O N A N C E S   A N D   I N S T A B I L I T I E S

                            A N A L Y S I S   I N   R O T A T I N G   S Y S T E M


 FORWARD WHIRL RESONANCE        SOLUTION      ROTOR SPEED     WHIRL
                                 NUMBER           RPM       DIRECTION

                                      1       3.00005E+03    FORWARD
```

```
BACKWARD WHIRL RESONANCE          SOLUTION       ROTOR SPEED      WHIRL
                                   NUMBER            RPM         DIRECTION

                                        2       3.00005E+03     BACKWARD
                                        4       3.46447E+03     BACKWARD


INSTABILITIES                     SOLUTION       ROTOR SPEED      WHIRL
                                   NUMBER            RPM         DIRECTION

                                                NONE   FOUND


CRITICAL SPEEDS FROM SYNCHRONOUS ANALYSIS
                                  SOLUTION       ROTOR SPEED      WHIRL
                                   NUMBER            RPM         DIRECTION

                                        1       3.00005E+03     FORWARD
                                        2       3.00005E+03     BACKWARD
                                        3       3.46416E+03     LINEAR
                                        4       3.46416E+03     LINEAR
```

Table 20 Laval Rotor Results in the Rotating Reference System

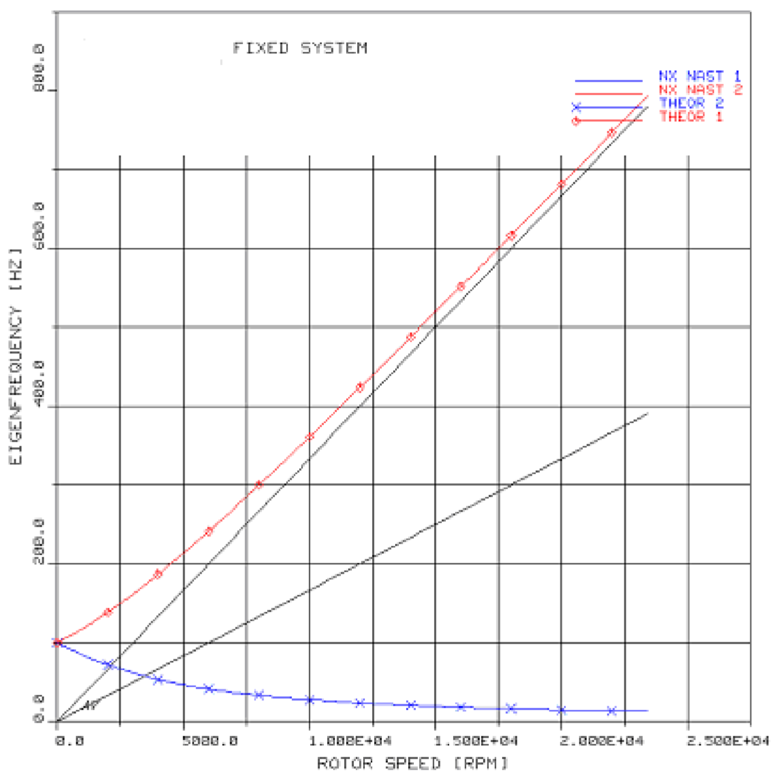


Fig. 33 Campbell Diagram of Laval Rotor Calculated in the Fixed System Compared to the Analytical Solution (Symbols)

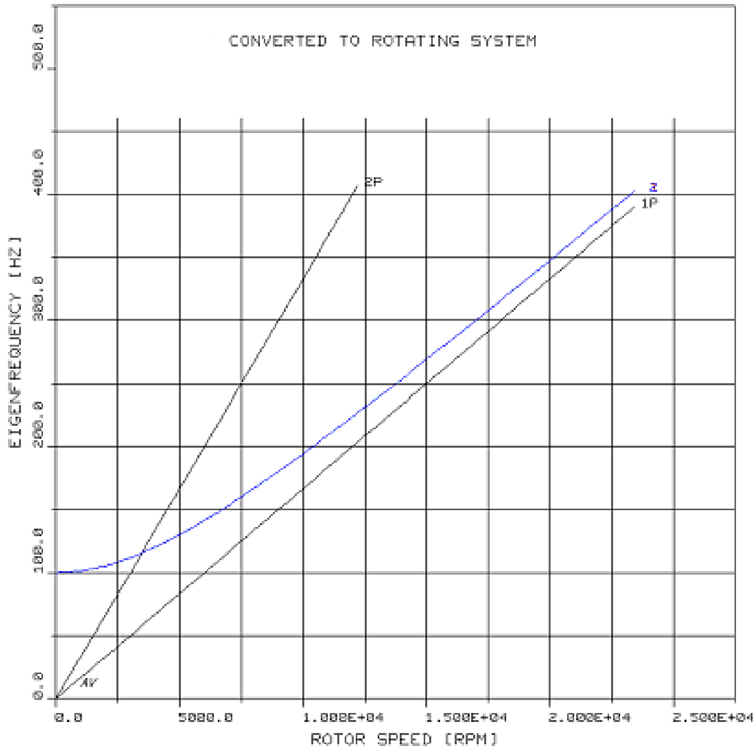


Fig. 34 Campbell Diagram of Laval Rotor Calculated in the Fixed System and Converted to the Rotating System

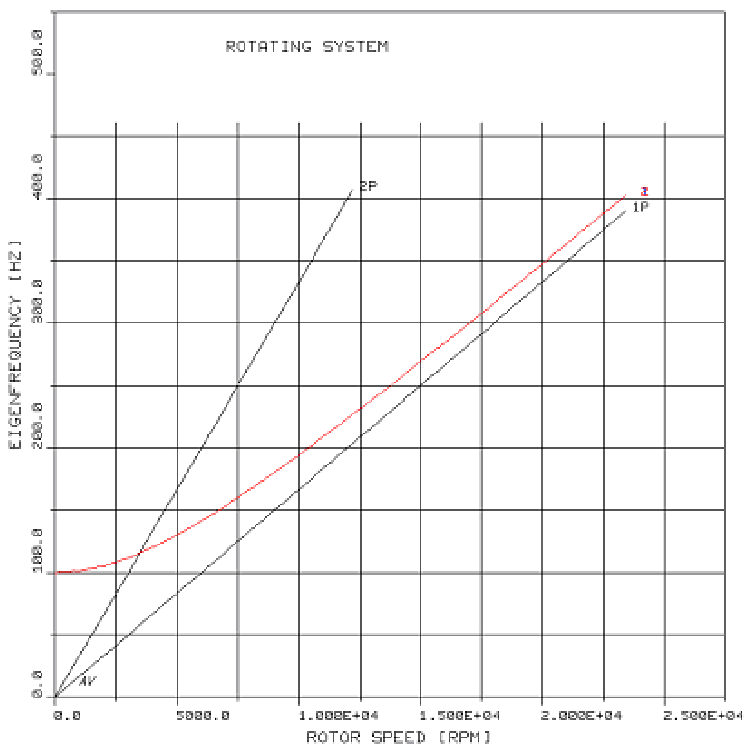


Fig. 35 Campbell Diagram of Laval Rotor Calculated in the Rotating System (Two Identical Solutions of Forward and Backward Whirl)

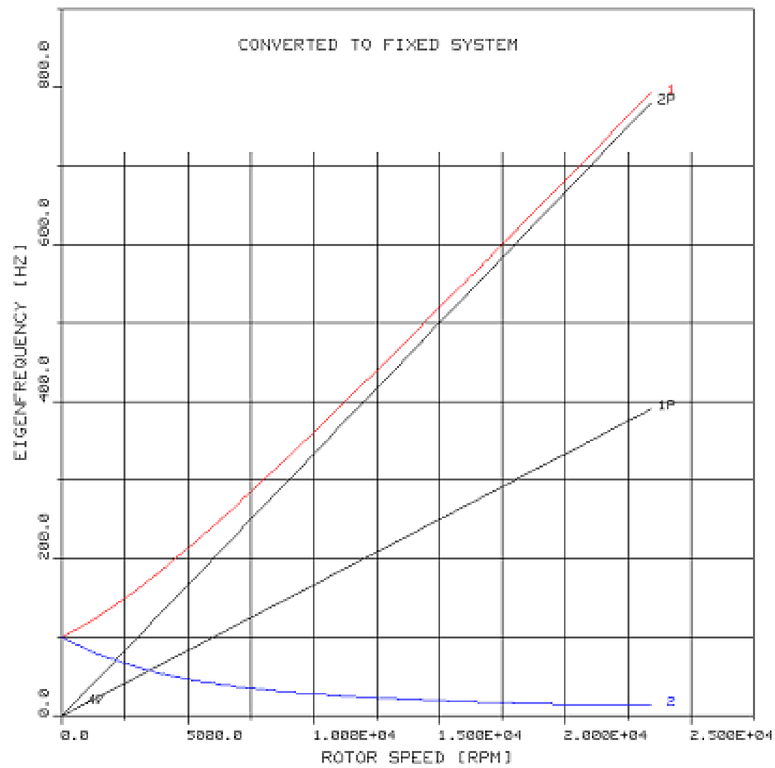


Fig. 36 Campbell Diagram of the Laval Rotor Calculated in the Rotating System and Converted to the Fixed System

6.3 Rotating Shaft Examples

A thin walled rotating shaft was studied analytically by Pedersen (ref. [2]). In this example, a tube with length of 1.0 meter, 0.30 m diameter, and a 0.005 m wall thickness is supported at each end by rigid or elastic bearings. The modulus of elasticity is 2.1E+11 N/m^2, Poisson's ratio is 0.3, and the density is 7850 kg/m^3. A shear factor of 0.5 was used in the analysis.

There are two different mode shapes:

- bending modes shapes.
- shear deformation mode shapes.

In the following cases, rigid bearings, and isotropic and anisotropic bearings are studied and compared to the analytical results. It must be noted that for the analytical results, trigonometric functions were used to describe the deformation. Hence, truncation errors may occur, and the results of the high frequency solutions may be imprecise for both methods.

6.3.1 Rotating Shaft with Rigid Bearings (rotor098.dat)

A Simcenter Nastran input deck is shown in Table 21. The rotor line is along the x-axis and the CORD2R entry defines the rotor coordinate system with rotation about the z-axis. The mass and inertia data must be given as discrete CONM2 entries. In the case of BEAM elements, the polar moment of inertia is calculated from the density on the MAT1 entry and the torsional moment of inertia J on the PBEAM entry. The CONM2 entries must be removed.

The eigenfrequencies of the bending modes, which are denoted by 1,2 for the first bending, 3,4 for the second bending, etc., behave like a rotating mass. The solutions, which are denoted by 7,8 for the first shear mode, 11,12 for the second shear mode, etc., behave much like a rotating disk.

The eigenfrequencies of the higher modes are slightly different, which may be due to the difference in discretization. The agreement of the Simcenter Nastran results (solid lines) with the analytical solution (dashed lines with symbols) is good, as shown in Fig. 37. The results of Simcenter Nastran for instabilities and critical speeds are shown in Table 22.

```
NASTRAN $
assign output4='rotor098.gpf',unit=22, form=formatted
assign output4='rotor098.csv',unit=25, form=formatted
$
sol 110
$
time 20000
CEND
$
SPC = 1
$
SET 2 = 1 THRU 18 EXCEPT 3,10
MODSEL = 2
$
RMETHOD = 99
$
METHOD = 1
CMETHOD = 2
$
BEGIN BULK
$
PARAM,ROTGPF,22
PARAM,ROTCSV,25
PARAM,GRDPNT,0
PARAM,MODTRK,1
$
ROTORG  11      1001    THRU    1041
$
$       Input for rotor dynamics
$
$       SID     RSTART  RSTEP   NUMSTEP REFSYS  CMOUT   RUNIT   FUNIT
ROTORD  99      0.0     2500.0  116     FIX     -1.0    RPM     HZ      +ROT0
$       ZSTEIN  ORBEPS  ROTPRT
+ROT0   NO      1.0E-8  3                                               +ROT1
$       RID1    RSET1   RSPEED1 RCORD1  W3-1    W4-1    RFORCE1
+ROT1   1       11      1.0     1       0.      0.      0
$
eigrl   1                       18      1
eigc    2       clan                                    16
$
$       Coordinates system defining rotor axis
$
```

```
cord2r  1               0.      0.      0.      1.      0.      0.      +xcrd001
+xcrd001 0.     1.      0.
$
$       Constraints
$
spc1    1       4       1001    thru    1041
spc     1       1001    1
$
$       Fixed bearings
$
spc1    1       23      1001    1041
$
$       Constraints of springs
$
spc1    1       123456  9101    9102
$
$
grid    9101             0.5    0.       2.186-8
grid    9102            -0.5    0.      -2.186-8
$
$
celas1  9001    9001    5001    2       9101    2
celas1  9002    9002    5001    3       9101    3
$
celas1  9003    9001    5041    2       9102    2
celas1  9004    9002    5041    3       9102    3
$
pelas   9001     5.000+9
pelas   9002     5.000+9
$
pdamp   9101     50000.
pdamp   9102     50000.
$
grid    5001             0.5    0.      0.
grid    5041            -0.5    0.      0.
$
$       Connection between fixed and rotating part
$
rbe2    5001    1001    123     5001
rbe2    5041    1041    123     5041
$
$       Structural model
$
pbar    1000    1000     4.634-3 5.042-5 5.042-5 1.008-4                +p00v8aa
+p00v8aa 0.      0.15    0.15    0.      0.     -0.15   -0.15    0.     +p00v8ab
+p00v8ab 0.5     0.5
$
$       Material with small damping
$
mat1    1000     2.10+11         0.3                            1.000-3
$
$       Nodes
$
grid    1001             0.5    0.      0.
grid    1002             0.475  0.      0.
grid    1003             0.45   0.      0.
grid    1004             0.425  0.      0.
grid    1005             0.4    0.      0.
grid    1006             0.375  0.      0.
grid    1007             0.35   0.      0.
grid    1008             0.325  0.      0.
grid    1009             0.3    0.      0.
grid    1010             0.275  0.      0.
grid    1011             0.25   0.      0.
grid    1012             0.225  0.      0.
grid    1013             0.2    0.      0.
grid    1014             0.175  0.      0.
grid    1015             0.15   0.      0.
grid    1016             0.125  0.      0.
grid    1017             0.1    0.      0.
grid    1018             0.075  0.      0.
grid    1019             0.05   0.      0.
```

```
grid    1020            0.025   0.      0.
grid    1021            -7.823-8 0.     0.
grid    1022            -0.025  0.      0.
grid    1023            -0.05   0.      0.
grid    1024            -0.075  0.      0.
grid    1025            -0.1    0.      0.
grid    1026            -0.125  0.      0.
grid    1027            -0.15   0.      0.
grid    1028            -0.175  0.      0.
grid    1029            -0.2    0.      0.
grid    1030            -0.225  0.      0.
grid    1031            -0.25   0.      0.
grid    1032            -0.275  0.      0.
grid    1033            -0.3    0.      0.
grid    1034            -0.325  0.      0.
grid    1035            -0.35   0.      0.
grid    1036            -0.375  0.      0.
grid    1037            -0.4    0.      0.
grid    1038            -0.425  0.      0.
grid    1039            -0.45   0.      0.
grid    1040            -0.475  0.      0.
grid    1041            -0.5    0.      0.
$
$       Elements
$
cbar    1001    1000    1001    1002    0.      0.      1.
cbar    1002    1000    1002    1003    0.      0.      1.
cbar    1003    1000    1003    1004    0.      0.      1.
cbar    1004    1000    1004    1005    0.      0.      1.
cbar    1005    1000    1005    1006    0.      0.      1.
cbar    1006    1000    1006    1007    0.      0.      1.
cbar    1007    1000    1007    1008    0.      0.      1.
cbar    1008    1000    1008    1009    0.      0.      1.
cbar    1009    1000    1009    1010    0.      0.      1.
cbar    1010    1000    1010    1011    0.      0.      1.
cbar    1011    1000    1011    1012    0.      0.      1.
cbar    1012    1000    1012    1013    0.      0.      1.
cbar    1013    1000    1013    1014    0.      0.      1.
cbar    1014    1000    1014    1015    0.      0.      1.
cbar    1015    1000    1015    1016    0.      0.      1.
cbar    1016    1000    1016    1017    0.      0.      1.
cbar    1017    1000    1017    1018    0.      0.      1.
cbar    1018    1000    1018    1019    0.      0.      1.
cbar    1019    1000    1019    1020    0.      0.      1.
cbar    1020    1000    1020    1021    0.      0.      1.
cbar    1021    1000    1021    1022    0.      0.      1.
cbar    1022    1000    1022    1023    0.      0.      1.
cbar    1023    1000    1023    1024    0.      0.      1.
cbar    1024    1000    1024    1025    0.      0.      1.
cbar    1025    1000    1025    1026    0.      0.      1.
cbar    1026    1000    1026    1027    0.      0.      1.
cbar    1027    1000    1027    1028    0.      0.      1.
cbar    1028    1000    1028    1029    0.      0.      1.
cbar    1029    1000    1029    1030    0.      0.      1.
cbar    1030    1000    1030    1031    0.      0.      1.
cbar    1031    1000    1031    1032    0.      0.      1.
cbar    1032    1000    1032    1033    0.      0.      1.
cbar    1033    1000    1033    1034    0.      0.      1.
cbar    1034    1000    1034    1035    0.      0.      1.
cbar    1035    1000    1035    1036    0.      0.      1.
cbar    1036    1000    1036    1037    0.      0.      1.
cbar    1037    1000    1037    1038    0.      0.      1.
cbar    1038    1000    1038    1039    0.      0.      1.
cbar    1039    1000    1039    1040    0.      0.      1.
cbar    1040    1000    1040    1041    0.      0.      1.
$
$       Mass data
$
conm2   2001    1001            0.4547                                  +con2001
+con2001 9.895-3         4.971-3                 4.971-3
conm2   2002    1002            0.90939                                 +con2002
+con2002 1.979-2         9.943-3                 9.943-3
```

```
conm2   2003    1003            0.90939                                 +con2003
+con2003 1.979-2        9.943-3                 9.943-3
conm2   2004    1004            0.90939                                 +con2004
+con2004 1.979-2        9.943-3                 9.943-3
conm2   2005    1005            0.90939                                 +con2005
+con2005 1.979-2        9.943-3                 9.943-3
conm2   2006    1006            0.90939                                 +con2006
+con2006 1.979-2        9.943-3                 9.943-3
conm2   2007    1007            0.90939                                 +con2007
+con2007 1.979-2        9.943-3                 9.943-3
conm2   2008    1008            0.90939                                 +con2008
+con2008 1.979-2        9.943-3                 9.943-3
conm2   2009    1009            0.90939                                 +con2009
+con2009 1.979-2        9.943-3                 9.943-3
conm2   2010    1010            0.90939                                 +con2010
+con2010 1.979-2        9.943-3                 9.943-3
conm2   2011    1011            0.90939                                 +con2011
+con2011 1.979-2        9.943-3                 9.943-3
conm2   2012    1012            0.90939                                 +con2012
+con2012 1.979-2        9.943-3                 9.943-3
conm2   2013    1013            0.90939                                 +con2013
+con2013 1.979-2        9.943-3                 9.943-3
conm2   2014    1014            0.90939                                 +con2014
+con2014 1.979-2        9.943-3                 9.943-3
conm2   2015    1015            0.90939                                 +con2015
+con2015 1.979-2        9.943-3                 9.943-3
conm2   2016    1016            0.90939                                 +con2016
+con2016 1.979-2        9.943-3                 9.943-3
conm2   2017    1017            0.90939                                 +con2017
+con2017 1.979-2        9.943-3                 9.943-3
conm2   2018    1018            0.90939                                 +con2018
+con2018 1.979-2        9.943-3                 9.943-3
conm2   2019    1019            0.90939                                 +con2019
+con2019 1.979-2        9.943-3                 9.943-3
conm2   2020    1020            0.90939                                 +con2020
+con2020 1.979-2        9.943-3                 9.943-3
conm2   2021    1021            0.90939                                 +con2021
+con2021 1.979-2        9.943-3                 9.943-3
conm2   2022    1022            0.90939                                 +con2022
+con2022 1.979-2        9.943-3                 9.943-3
conm2   2023    1023            0.90939                                 +con2023
+con2023 1.979-2        9.943-3                 9.943-3
conm2   2024    1024            0.90939                                 +con2024
+con2024 1.979-2        9.943-3                 9.943-3
conm2   2025    1025            0.90939                                 +con2025
+con2025 1.979-2        9.943-3                 9.943-3
conm2   2026    1026            0.90939                                 +con2026
+con2026 1.979-2        9.943-3                 9.943-3
conm2   2027    1027            0.90939                                 +con2027
+con2027 1.979-2        9.943-3                 9.943-3
conm2   2028    1028            0.90939                                 +con2028
+con2028 1.979-2        9.943-3                 9.943-3
conm2   2029    1029            0.90939                                 +con2029
+con2029 1.979-2        9.943-3                 9.943-3
conm2   2030    1030            0.90939                                 +con2030
+con2030 1.979-2        9.943-3                 9.943-3
conm2   2031    1031            0.90939                                 +con2031
+con2031 1.979-2        9.943-3                 9.943-3
conm2   2032    1032            0.90939                                 +con2032
+con2032 1.979-2        9.943-3                 9.943-3
conm2   2033    1033            0.90939                                 +con2033
+con2033 1.979-2        9.943-3                 9.943-3
conm2   2034    1034            0.90939                                 +con2034
+con2034 1.979-2        9.943-3                 9.943-3
conm2   2035    1035            0.90939                                 +con2035
+con2035 1.979-2        9.943-3                 9.943-3
conm2   2036    1036            0.90939                                 +con2036
+con2036 1.979-2        9.943-3                 9.943-3
conm2   2037    1037            0.90939                                 +con2037
+con2037 1.979-2        9.943-3                 9.943-3
conm2   2038    1038            0.90939                                 +con2038
+con2038 1.979-2        9.943-3                 9.943-3
```

```
conm2   2039    1039            0.90939                                 +con2039
+con2039 1.979-2        9.943-3                 9.943-3
conm2   2040    1040            0.90939                                 +con2040
+con2040 1.979-2        9.943-3                 9.943-3
conm2   2041    1041            0.4547                                  +con2041
+con2041 9.895-3        4.971-3                 4.971-3
Enddata
```

Table 21 Input File for the Rotating Shaft

```
               D E T E C T I O N   O F   R E S O N A N C E S   A N D   I N S T A B I L I T I E S

                              A N A L Y S I S   I N   F I X E D   S Y S T E M


 WHIRL RESONANCE                SOLUTION      ROTOR SPEED     WHIRL
                                 NUMBER           RPM       DIRECTION

                                      1       4.16308E+04    FORWARD
                                      2       3.81111E+04    BACKWARD
                                      3       1.15192E+05    FORWARD
                                      4       1.04875E+05    BACKWARD
                                      5       1.77400E+05    BACKWARD
                                      6       1.89816E+05    FORWARD
                                      7       1.19732E+05    BACKWARD
                                      9       2.48837E+05    BACKWARD
                                     10       2.60758E+05    FORWARD
                                     11       1.59666E+05    BACKWARD
                                     15       2.31731E+05    BACKWARD


 INSTABILITIES                  SOLUTION      ROTOR SPEED     WHIRL
                                 NUMBER           RPM       DIRECTION

 START                                1       4.16309E+04    FORWARD
 START                                3       1.15192E+05    FORWARD
 START                                6       1.89816E+05    FORWARD
 START                               10       2.60758E+05    FORWARD


 CRITICAL SPEEDS FROM SYNCHRONOUS ANALYSIS
                                SOLUTION      ROTOR SPEED     WHIRL
                                 NUMBER           RPM       DIRECTION

                                      1       3.81111E+04    BACKWARD
                                      2       4.16308E+04    FORWARD
                                      3       1.04875E+05    BACKWARD
                                      4       1.15192E+05    FORWARD
                                      5       1.19732E+05    BACKWARD
                                      6       1.59666E+05    BACKWARD
                                      7       1.77400E+05    BACKWARD
                                      8       1.89817E+05    FORWARD
                                     10       2.31730E+05    BACKWARD
                                     11       2.48837E+05    BACKWARD
                                     13       2.60758E+05    FORWARD
                                     14       3.19167E+05    BACKWARD
                                     15       3.29925E+05    FORWARD
```

Table 22 F06 File Results for the Rotating Shaft

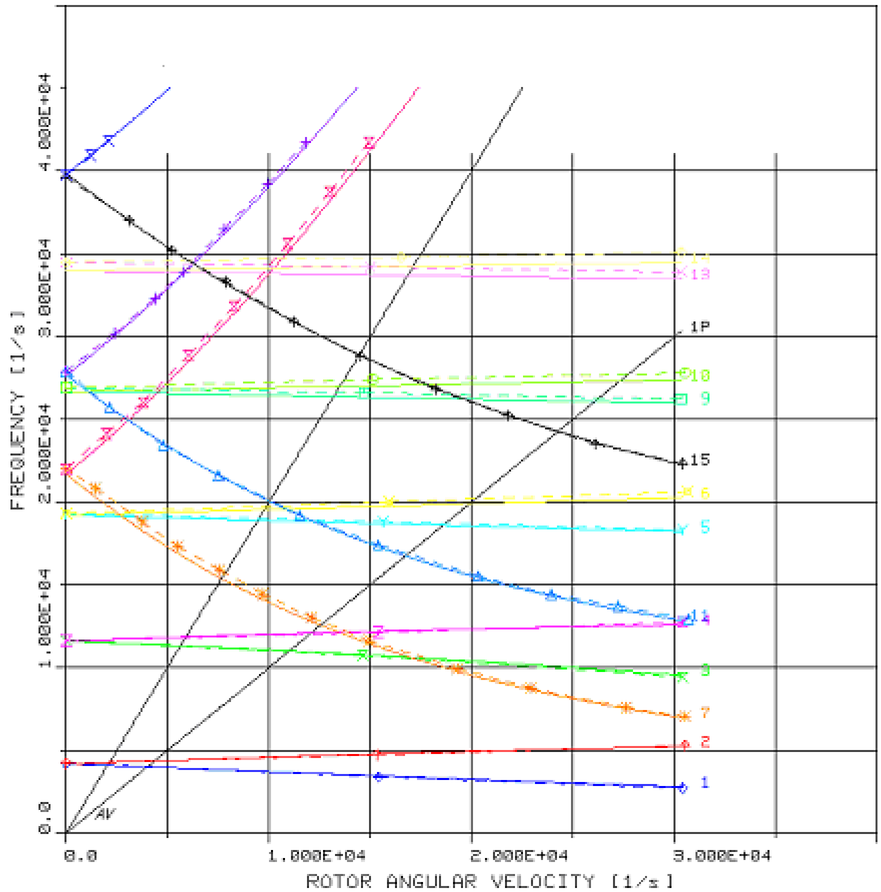


Fig. 37 Comparison of Simcenter Nastran Results (Solid Lines) with Analytical Solution (Symbols) for Rigid Bearing Example

6.3.2 Rotating Shaft with Elastic Isotropic Bearings (rotor095.dat)

The same model was analyzed with isotropic bearings (same stiffness in x- and y-directions) of 5.0E+9 N/m stiffness by removing the following entry from the input file (shown in Table 21):

```
spc1    1    12    1001    1041
```

In this example, the agreement between Simcenter Nastran and the analytical solution is good. The solutions of the higher modes are dependent on the number of modes accounted for in the complex modal analysis. This is also true for the analytic analysis, where the number of theoretical solutions was also limited. The corresponding Campbell diagram is shown in Fig. 38.

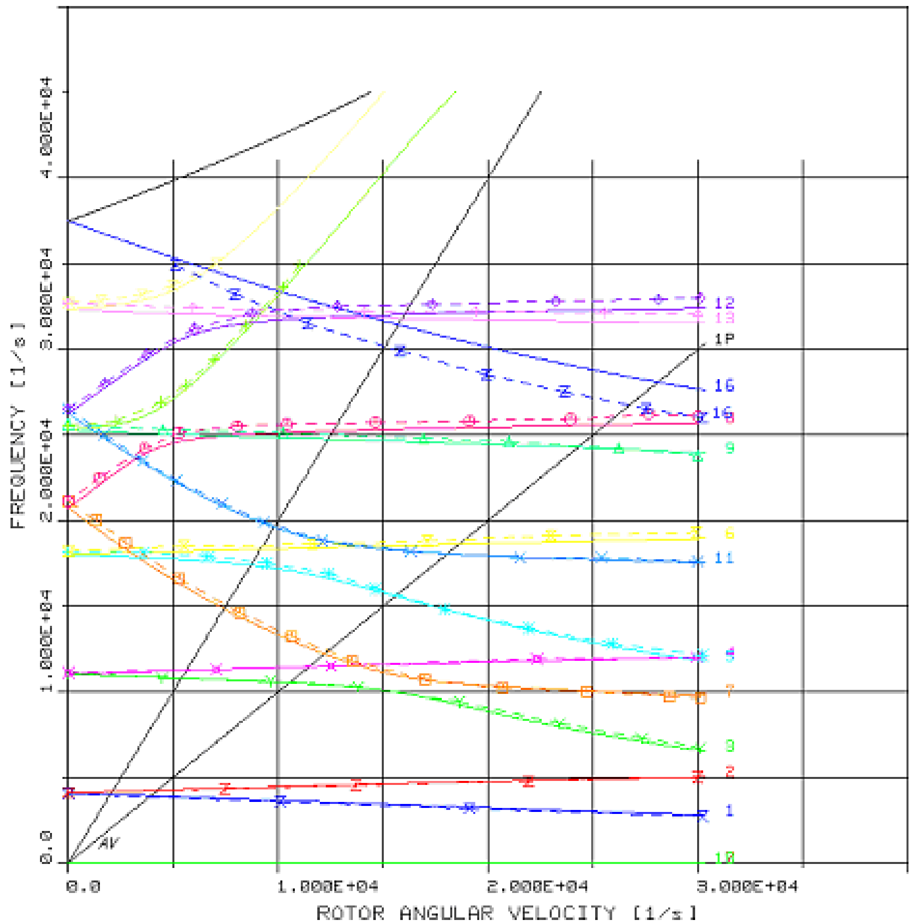


Fig. 38 Elastic Isotropic Bearing Results (Solid Lines) Compared with Analytical Solution (Symbols)

6.3.3 Rotating Shaft with Elastic Anisotropic Bearings

The case of anisotropic bearings was analyzed with Simcenter Nastran with good agreement with the analytical solution. The results of ref [2] were scanned in and may not be exact.

The model is obtained by changing

pelas 9002 5.000+9

to

pelas 9002 2.000+9

in the Simcenter Nastran input file.

The comparison is shown in Fig. 39, where good agreement was found between the Simcenter Nastran results and the theoretical results.

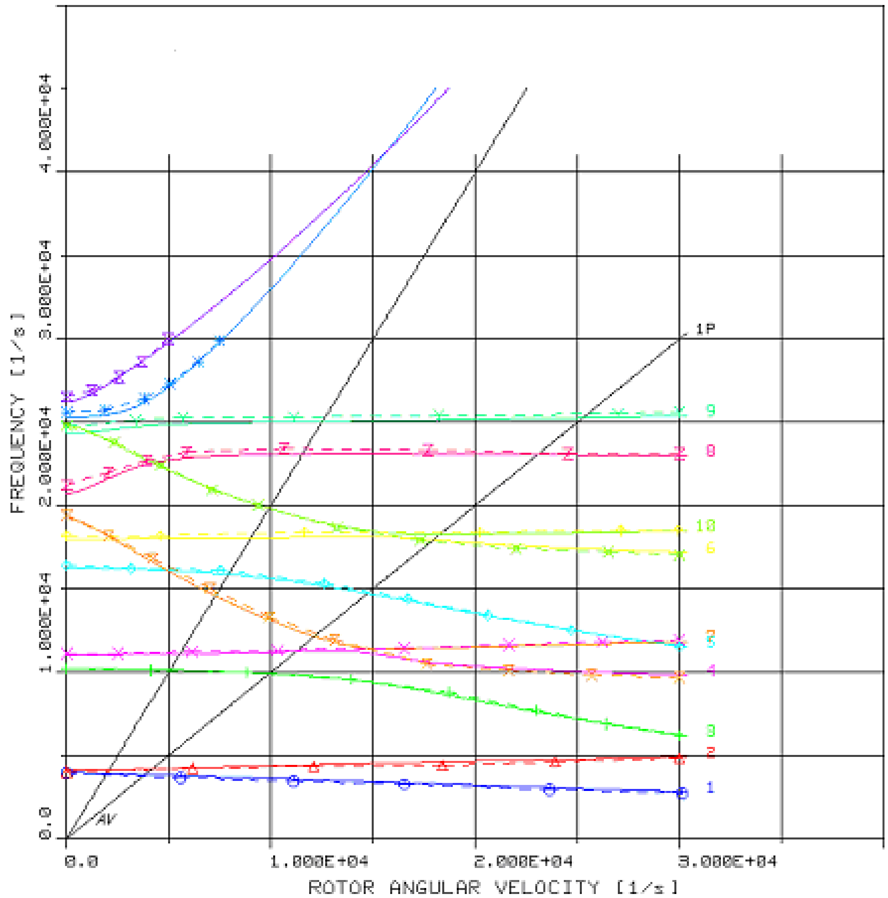


Fig. 39 Elastic Anisotropic Bearings (Solid Lines) Compared with Analytical Solution (Symbols)

6.3.4 Model with Two Rotors (rotor096.dat)

A model with two rotors can be obtained by duplicating the rotor structure of the example in Section 6.3.2. The input deck (both rotors are defined along the z-axis) is shown in Table 23.

The solutions with Simcenter Nastran are given in Table 24. All solutions appear twice. The Campbell diagram is shown in Fig. 40. The case of different rotor speeds can be analyzed by simply modifying the constant RSPEED2 for the second rotor. Fig. 41 shows the Campbell diagram for the same model, but with the second rotor rotating at twice the speed.

```
nastran $
$
assign output4='rotor096.gpf',unit=22, form=formatted
assign output4='rotor096.csv',unit=25, form=formatted
$
sol 110
$
time 20000
diag 8
$
CEND
$
SPC = 1
$
```

```
SET 2 = 1 THRU 36 EXCEPT 5,6,19,20
MODSEL = 2
$
RMETHOD = 99
$
METHOD = 1
CMETHOD = 2
$
BEGIN BULK
$
PARAM,ROTGPF,22
PARAM,ROTCSV,25
PARAM,GRDPNT,0
PARAM,MODTRK,1
$
$       Rotor 1
ROTORG  11      1001    THRU    1041
$       Rotor 2
ROTORG  12      2001    THRU    2041
$
ROTORB  11      5001    5041    9101    9102
$
ROTORB  12      5201    5241    9301    9302
$
$       SID     RSTART  RSTEP   NUMSTEP REFSYS  CMOUT   RUNIT   FUNIT
ROTORD  99      0.0     5000.0  58      FIX     -1.0    RPM     HZ      +ROT0
$       ZSTEIN  ORBEPS  ROTPRT
+ROT0   NO      1.0E-5  3                                               +ROT1
$       RID1    RSET1   RSPEED1 RCORD1  W3-1    W4-1    RFORCE1
+ROT1   1       11      1.0     1       0.      0.                      +ROT2
$       RID2    RSET2   RSPEED2 RCORD2  W3-2    W4-2    RFORCE1
+ROT2   2       12      1.0     2       0.      0.
$
$       Disregard axial displacement and rotation
$
eigrl   1                       36      1
eigc    2       clan                                    32
$
$
cord2r  1                0.      0.      0.      0.      0.      1.     +xcrd001
+xcrd001 1.      0.      0.
$
cord2r  2                0.      0.      0.      0.      0.      1.     +xcrd002
+xcrd002 1.      0.      0.
$
$       Rotation constrained
$
spc1    1       6       1001    thru    1041
spc1    1       6       2001    thru    2041
$
spc1    1       3       5001
spc1    1       3       5201
$
spc1    1       123456  9101    9102
spc1    1       123456  9301    9302
$
$       1. Rotor
$
grid    9101             0.      0.      -0.5
grid    9102             0.      0.       0.5
$
celas1  9001    9001    5001    1       9101    1
celas1  9002    9002    5001    2       9101    2
$
celas1  9003    9001    5041    1       9102    1
celas1  9004    9002    5041    2       9102    2
$
pelas   9001     5.00+9
pelas   9002     5.00+9
$
cdamp1  9101    9101    5001    1       9101    1
cdamp1  9102    9102    5001    2       9101    2
```

```
$
cdamp1  9103    9101    5041    1       9102    1
cdamp1  9104    9102    5041    2       9102    2
$
pdamp   9101     5.0+4
pdamp   9102     5.0+4
$
grid    5001             0.      0.     -0.5
grid    5041             0.      0.      0.5
$
rbe2    5001    5001    123     1001
rbe2    5041    5041    123     1041
$
$       2. Rotor
$
grid    9301             0.      0.      -0.5
grid    9302             0.      0.       0.5
$
celas1  9201    9201    5201    1       9301    1
celas1  9202    9202    5201    2       9301    2
$
celas1  9203    9201    5241    1       9302    1
celas1  9204    9202    5241    2       9302    2
$
pelas   9201     5.00+9
pelas   9202     5.00+9
$
cdamp1  9301    9301    5201    1       9301    1
cdamp1  9302    9302    5201    2       9301    2
$
cdamp1  9303    9301    5241    1       9302    1
cdamp1  9304    9302    5241    2       9302    2
$
pdamp   9301     5.0+4
pdamp   9302     5.0+4
$
grid    5201             0.      0.     -0.5
grid    5241             0.      0.      0.5
$
rbe2    5201    5201    123     2001
rbe2    5241    5241    123     2041
$
$       1. Rotor
$
pbar    1000    1000     4.634-3 5.042-5 5.042-5 1.008-4                +p00v8aa
+p00v8aa 0.      0.15    0.15    0.      0.     -0.15   -0.15    0.     +p00v8ab
+p00v8ab 0.5     0.5
$
mat1    1000     2.10+11         0.3     0.0                     0.02
$
$       Nodes
$
grid    1001             0.      0.     -0.5
grid    1002             0.      0.     -0.475
grid    1003             0.      0.     -0.45
grid    1004             0.      0.     -0.425
grid    1005             0.      0.     -0.4
grid    1006             0.      0.     -0.375
grid    1007             0.      0.     -0.35
grid    1008             0.      0.     -0.325
grid    1009             0.      0.     -0.3
grid    1010             0.      0.     -0.275
grid    1011             0.      0.     -0.25
grid    1012             0.      0.     -0.225
grid    1013             0.      0.     -0.2
grid    1014             0.      0.     -0.175
grid    1015             0.      0.     -0.15
grid    1016             0.      0.     -0.125
grid    1017             0.      0.     -0.1
grid    1018             0.      0.     -0.075
grid    1019             0.      0.     -0.05
grid    1020             0.      0.     -0.025
```

```
grid    1021            0.      0.      7.823-8
grid    1022            0.      0.      0.025
grid    1023            0.      0.      0.05
grid    1024            0.      0.      0.075
grid    1025            0.      0.      0.1
grid    1026            0.      0.      0.125
grid    1027            0.      0.      0.15
grid    1028            0.      0.      0.175
grid    1029            0.      0.      0.2
grid    1030            0.      0.      0.225
grid    1031            0.      0.      0.25
grid    1032            0.      0.      0.275
grid    1033            0.      0.      0.3
grid    1034            0.      0.      0.325
grid    1035            0.      0.      0.35
grid    1036            0.      0.      0.375
grid    1037            0.      0.      0.4
grid    1038            0.      0.      0.425
grid    1039            0.      0.      0.45
grid    1040            0.      0.      0.475
grid    1041            0.      0.      0.5
$
$       Elements
$
cbar    1001    1000    1001    1002    1.      0.      0.
cbar    1002    1000    1002    1003    1.      0.      0.
cbar    1003    1000    1003    1004    1.      0.      0.
cbar    1004    1000    1004    1005    1.      0.      0.
cbar    1005    1000    1005    1006    1.      0.      0.
cbar    1006    1000    1006    1007    1.      0.      0.
cbar    1007    1000    1007    1008    1.      0.      0.
cbar    1008    1000    1008    1009    1.      0.      0.
cbar    1009    1000    1009    1010    1.      0.      0.
cbar    1010    1000    1010    1011    1.      0.      0.
cbar    1011    1000    1011    1012    1.      0.      0.
cbar    1012    1000    1012    1013    1.      0.      0.
cbar    1013    1000    1013    1014    1.      0.      0.
cbar    1014    1000    1014    1015    1.      0.      0.
cbar    1015    1000    1015    1016    1.      0.      0.
cbar    1016    1000    1016    1017    1.      0.      0.
cbar    1017    1000    1017    1018    1.      0.      0.
cbar    1018    1000    1018    1019    1.      0.      0.
cbar    1019    1000    1019    1020    1.      0.      0.
cbar    1020    1000    1020    1021    1.      0.      0.
cbar    1021    1000    1021    1022    1.      0.      0.
cbar    1022    1000    1022    1023    1.      0.      0.
cbar    1023    1000    1023    1024    1.      0.      0.
cbar    1024    1000    1024    1025    1.      0.      0.
cbar    1025    1000    1025    1026    1.      0.      0.
cbar    1026    1000    1026    1027    1.      0.      0.
cbar    1027    1000    1027    1028    1.      0.      0.
cbar    1028    1000    1028    1029    1.      0.      0.
cbar    1029    1000    1029    1030    1.      0.      0.
cbar    1030    1000    1030    1031    1.      0.      0.
cbar    1031    1000    1031    1032    1.      0.      0.
cbar    1032    1000    1032    1033    1.      0.      0.
cbar    1033    1000    1033    1034    1.      0.      0.
cbar    1034    1000    1034    1035    1.      0.      0.
cbar    1035    1000    1035    1036    1.      0.      0.
cbar    1036    1000    1036    1037    1.      0.      0.
cbar    1037    1000    1037    1038    1.      0.      0.
cbar    1038    1000    1038    1039    1.      0.      0.
cbar    1039    1000    1039    1040    1.      0.      0.
cbar    1040    1000    1040    1041    1.      0.      0.
$
$       Mass data
$
conm2   1201    1001            0.4547                                  +con1201
+con1201 4.971-3        4.971-3                 9.895-3
conm2   1202    1002            0.90939                                 +con1202
+con1202 9.943-3        9.943-3                 1.979-2
conm2   1203    1003            0.90939                                 +con1203
```

```
+con1203 9.943-3        9.943-3             1.979-2
conm2   1204    1004            0.90939                             +con1204
+con1204 9.943-3        9.943-3             1.979-2
conm2   1205    1005            0.90939                             +con1205
+con1205 9.943-3        9.943-3             1.979-2
conm2   1206    1006            0.90939                             +con1206
+con1206 9.943-3        9.943-3             1.979-2
conm2   1207    1007            0.90939                             +con1207
+con1207 9.943-3        9.943-3             1.979-2
conm2   1208    1008            0.90939                             +con1208
+con1208 9.943-3        9.943-3             1.979-2
conm2   1209    1009            0.90939                             +con1209
+con1209 9.943-3        9.943-3             1.979-2
conm2   1210    1010            0.90939                             +con1210
+con1210 9.943-3        9.943-3             1.979-2
conm2   1211    1011            0.90939                             +con1211
+con1211 9.943-3        9.943-3             1.979-2
conm2   1212    1012            0.90939                             +con1212
+con1212 9.943-3        9.943-3             1.979-2
conm2   1213    1013            0.90939                             +con1213
+con1213 9.943-3        9.943-3             1.979-2
conm2   1214    1014            0.90939                             +con1214
+con1214 9.943-3        9.943-3             1.979-2
conm2   1215    1015            0.90939                             +con1215
+con1215 9.943-3        9.943-3             1.979-2
conm2   1216    1016            0.90939                             +con1216
+con1216 9.943-3        9.943-3             1.979-2
conm2   1217    1017            0.90939                             +con1217
+con1217 9.943-3        9.943-3             1.979-2
conm2   1218    1018            0.90939                             +con1218
+con1218 9.943-3        9.943-3             1.979-2
conm2   1219    1019            0.90939                             +con1219
+con1219 9.943-3        9.943-3             1.979-2
conm2   1220    1020            0.90939                             +con1220
+con1220 9.943-3        9.943-3             1.979-2
conm2   1221    1021            0.90939                             +con1221
+con1221 9.943-3        9.943-3             1.979-2
conm2   1222    1022            0.90939                             +con1222
+con1222 9.943-3        9.943-3             1.979-2
conm2   1223    1023            0.90939                             +con1223
+con1223 9.943-3        9.943-3             1.979-2
conm2   1224    1024            0.90939                             +con1224
+con1224 9.943-3        9.943-3             1.979-2
conm2   1225    1025            0.90939                             +con1225
+con1225 9.943-3        9.943-3             1.979-2
conm2   1226    1026            0.90939                             +con1226
+con1226 9.943-3        9.943-3             1.979-2
conm2   1227    1027            0.90939                             +con1227
+con1227 9.943-3        9.943-3             1.979-2
conm2   1228    1028            0.90939                             +con1228
+con1228 9.943-3        9.943-3             1.979-2
conm2   1229    1029            0.90939                             +con1229
+con1229 9.943-3        9.943-3             1.979-2
conm2   1230    1030            0.90939                             +con1230
+con1230 9.943-3        9.943-3             1.979-2
conm2   1231    1031            0.90939                             +con1231
+con1231 9.943-3        9.943-3             1.979-2
conm2   1232    1032            0.90939                             +con1232
+con1232 9.943-3        9.943-3             1.979-2
conm2   1233    1033            0.90939                             +con1233
+con1233 9.943-3        9.943-3             1.979-2
conm2   1234    1034            0.90939                             +con1234
+con1234 9.943-3        9.943-3             1.979-2
conm2   1235    1035            0.90939                             +con1235
+con1235 9.943-3        9.943-3             1.979-2
conm2   1236    1036            0.90939                             +con1236
+con1236 9.943-3        9.943-3             1.979-2
conm2   1237    1037            0.90939                             +con1237
+con1237 9.943-3        9.943-3             1.979-2
conm2   1238    1038            0.90939                             +con1238
+con1238 9.943-3        9.943-3             1.979-2
conm2   1239    1039            0.90939                             +con1239
```

```
+con1239 9.943-3         9.943-3                 1.979-2
conm2   1240    1040            0.90939                                 +con1240
+con1240 9.943-3         9.943-3                 1.979-2
conm2   1241    1041            0.4547                                  +con1241
+con1241 4.971-3         4.971-3                 9.895-3
$
pbar    2000    2000     4.634-3 5.042-5 5.042-5 1.008-4                +p20v8aa
+p20v8aa 0.      0.15    0.15    0.      0.     -0.15   -0.15    0.     +p20v8ab
+p20v8ab 0.5     0.5
$
mat1    2000     2.10+11        0.3     0.0                     0.02
$
$
$       Nodes
$
grid    2001             0.      0.     -0.5
grid    2002             0.      0.     -0.475
grid    2003             0.      0.     -0.45
grid    2004             0.      0.     -0.425
grid    2005             0.      0.     -0.4
grid    2006             0.      0.     -0.375
grid    2007             0.      0.     -0.35
grid    2008             0.      0.     -0.325
grid    2009             0.      0.     -0.3
grid    2010             0.      0.     -0.275
grid    2011             0.      0.     -0.25
grid    2012             0.      0.     -0.225
grid    2013             0.      0.     -0.2
grid    2014             0.      0.     -0.175
grid    2015             0.      0.     -0.15
grid    2016             0.      0.     -0.125
grid    2017             0.      0.     -0.1
grid    2018             0.      0.     -0.075
grid    2019             0.      0.     -0.05
grid    2020             0.      0.     -0.025
grid    2021             0.      0.      7.823-8
grid    2022             0.      0.      0.025
grid    2023             0.      0.      0.05
grid    2024             0.      0.      0.075
grid    2025             0.      0.      0.1
grid    2026             0.      0.      0.125
grid    2027             0.      0.      0.15
grid    2028             0.      0.      0.175
grid    2029             0.      0.      0.2
grid    2030             0.      0.      0.225
grid    2031             0.      0.      0.25
grid    2032             0.      0.      0.275
grid    2033             0.      0.      0.3
grid    2034             0.      0.      0.325
grid    2035             0.      0.      0.35
grid    2036             0.      0.      0.375
grid    2037             0.      0.      0.4
grid    2038             0.      0.      0.425
grid    2039             0.      0.      0.45
grid    2040             0.      0.      0.475
grid    2041             0.      0.      0.5
$
$       Elements
$
cbar    2001    2000    2001    2002     1.      0.      0.
cbar    2002    2000    2002    2003     1.      0.      0.
cbar    2003    2000    2003    2004     1.      0.      0.
cbar    2004    2000    2004    2005     1.      0.      0.
cbar    2005    2000    2005    2006     1.      0.      0.
cbar    2006    2000    2006    2007     1.      0.      0.
cbar    2007    2000    2007    2008     1.      0.      0.
cbar    2008    2000    2008    2009     1.      0.      0.
cbar    2009    2000    2009    2010     1.      0.      0.
cbar    2010    2000    2010    2011     1.      0.      0.
cbar    2011    2000    2011    2012     1.      0.      0.
cbar    2012    2000    2012    2013     1.      0.      0.
cbar    2013    2000    2013    2014     1.      0.      0.
```

```
cbar    2014    2000    2014    2015     1.      0.      0.
cbar    2015    2000    2015    2016     1.      0.      0.
cbar    2016    2000    2016    2017     1.      0.      0.
cbar    2017    2000    2017    2018     1.      0.      0.
cbar    2018    2000    2018    2019     1.      0.      0.
cbar    2019    2000    2019    2020     1.      0.      0.
cbar    2020    2000    2020    2021     1.      0.      0.
cbar    2021    2000    2021    2022     1.      0.      0.
cbar    2022    2000    2022    2023     1.      0.      0.
cbar    2023    2000    2023    2024     1.      0.      0.
cbar    2024    2000    2024    2025     1.      0.      0.
cbar    2025    2000    2025    2026     1.      0.      0.
cbar    2026    2000    2026    2027     1.      0.      0.
cbar    2027    2000    2027    2028     1.      0.      0.
cbar    2028    2000    2028    2029     1.      0.      0.
cbar    2029    2000    2029    2030     1.      0.      0.
cbar    2030    2000    2030    2031     1.      0.      0.
cbar    2031    2000    2031    2032     1.      0.      0.
cbar    2032    2000    2032    2033     1.      0.      0.
cbar    2033    2000    2033    2034     1.      0.      0.
cbar    2034    2000    2034    2035     1.      0.      0.
cbar    2035    2000    2035    2036     1.      0.      0.
cbar    2036    2000    2036    2037     1.      0.      0.
cbar    2037    2000    2037    2038     1.      0.      0.
cbar    2038    2000    2038    2039     1.      0.      0.
cbar    2039    2000    2039    2040     1.      0.      0.
cbar    2040    2000    2040    2041     1.      0.      0.
$
$       Mass data
$
conm2   2201    2001            0.4547                          +con2201
+con2201 4.971-3        4.971-3         9.895-3
conm2   2202    2002            0.90939                         +con2202
+con2202 9.943-3        9.943-3         1.979-2
conm2   2203    2003            0.90939                         +con2203
+con2203 9.943-3        9.943-3         1.979-2
conm2   2204    2004            0.90939                         +con2204
+con2204 9.943-3        9.943-3         1.979-2
conm2   2205    2005            0.90939                         +con2205
+con2205 9.943-3        9.943-3         1.979-2
conm2   2206    2006            0.90939                         +con2206
+con2206 9.943-3        9.943-3         1.979-2
conm2   2207    2007            0.90939                         +con2207
+con2207 9.943-3        9.943-3         1.979-2
conm2   2208    2008            0.90939                         +con2208
+con2208 9.943-3        9.943-3         1.979-2
conm2   2209    2009            0.90939                         +con2209
+con2209 9.943-3        9.943-3         1.979-2
conm2   2210    2010            0.90939                         +con2210
+con2210 9.943-3        9.943-3         1.979-2
conm2   2211    2011            0.90939                         +con2211
+con2211 9.943-3        9.943-3         1.979-2
conm2   2212    2012            0.90939                         +con2212
+con2212 9.943-3        9.943-3         1.979-2
conm2   2213    2013            0.90939                         +con2213
+con2213 9.943-3        9.943-3         1.979-2
conm2   2214    2014            0.90939                         +con2214
+con2214 9.943-3        9.943-3         1.979-2
conm2   2215    2015            0.90939                         +con2215
+con2215 9.943-3        9.943-3         1.979-2
conm2   2216    2016            0.90939                         +con2216
+con2216 9.943-3        9.943-3         1.979-2
conm2   2217    2017            0.90939                         +con2217
+con2217 9.943-3        9.943-3         1.979-2
conm2   2218    2018            0.90939                         +con2218
+con2218 9.943-3        9.943-3         1.979-2
conm2   2219    2019            0.90939                         +con2219
+con2219 9.943-3        9.943-3         1.979-2
conm2   2220    2020            0.90939                         +con2220
+con2220 9.943-3        9.943-3         1.979-2
conm2   2221    2021            0.90939                         +con2221
+con2221 9.943-3        9.943-3         1.979-2
```

```
conm2   2222    2022            0.90939                                 +con2222
+con2222 9.943-3        9.943-3                 1.979-2
conm2   2223    2023            0.90939                                 +con2223
+con2223 9.943-3        9.943-3                 1.979-2
conm2   2224    2024            0.90939                                 +con2224
+con2224 9.943-3        9.943-3                 1.979-2
conm2   2225    2025            0.90939                                 +con2225
+con2225 9.943-3        9.943-3                 1.979-2
conm2   2226    2026            0.90939                                 +con2226
+con2226 9.943-3        9.943-3                 1.979-2
conm2   2227    2027            0.90939                                 +con2227
+con2227 9.943-3        9.943-3                 1.979-2
conm2   2228    2028            0.90939                                 +con2228
+con2228 9.943-3        9.943-3                 1.979-2
conm2   2229    2029            0.90939                                 +con2229
+con2229 9.943-3        9.943-3                 1.979-2
conm2   2230    2030            0.90939                                 +con2230
+con2230 9.943-3        9.943-3                 1.979-2
conm2   2231    2031            0.90939                                 +con2231
+con2231 9.943-3        9.943-3                 1.979-2
conm2   2232    2032            0.90939                                 +con2232
+con2232 9.943-3        9.943-3                 1.979-2
conm2   2233    2033            0.90939                                 +con2233
+con2233 9.943-3        9.943-3                 1.979-2
conm2   2234    2034            0.90939                                 +con2234
+con2234 9.943-3        9.943-3                 1.979-2
conm2   2235    2035            0.90939                                 +con2235
+con2235 9.943-3        9.943-3                 1.979-2
conm2   2236    2036            0.90939                                 +con2236
+con2236 9.943-3        9.943-3                 1.979-2
conm2   2237    2037            0.90939                                 +con2237
+con2237 9.943-3        9.943-3                 1.979-2
conm2   2238    2038            0.90939                                 +con2238
+con2238 9.943-3        9.943-3                 1.979-2
conm2   2239    2039            0.90939                                 +con2239
+con2239 9.943-3        9.943-3                 1.979-2
conm2   2240    2040            0.90939                                 +con2240
+con2240 9.943-3        9.943-3                 1.979-2
conm2   2241    2041            0.4547                                  +con2241
+con2241 4.971-3        4.971-3                 9.895-3
$
enddata
```

Table 23 Simcenter Nastran Input File for Model with Two Coincident Rotors

```
              D E T E C T I O N   O F   R E S O N A N C E S   A N D   I N S T A B I L I T I E S

                              A N A L Y S I S   I N   F I X E D   S Y S T E M

WHIRL RESONANCE                SOLUTION      ROTOR SPEED     WHIRL
                                 NUMBER           RPM       DIRECTION


                                ROTOR NUMBER       1     RELATIVE SPEED  1.00000E+00

                                       1       4.04540E+04    FORWARD
                                       2       4.04540E+04    FORWARD
                                       3       3.73449E+04    BACKWARD
                                       4       3.73449E+04    BACKWARD
                                       5       1.09334E+05    FORWARD
                                       6       1.09334E+05    FORWARD
                                       7       1.00406E+05    BACKWARD
                                       8       1.00406E+05    BACKWARD
                                       9       1.77687E+05    FORWARD
                                      10       1.77687E+05    FORWARD
                                      11       1.48712E+05    BACKWARD
                                      12       1.48712E+05    BACKWARD
                                      13       1.16499E+05    BACKWARD
                                      14       1.16499E+05    BACKWARD
                                      15       2.44293E+05    FORWARD
                                      16       2.44293E+05    FORWARD
                                      17       2.31815E+05    BACKWARD
                                      18       2.31815E+05    BACKWARD
```

```
                                    21       1.72225E+05    BACKWARD
                                    22       1.72225E+05    BACKWARD
                                    29       2.67606E+05    BACKWARD
                                    30       2.67606E+05    BACKWARD

                               ROTOR NUMBER       2     RELATIVE SPEED  1.00000E+00

                                     1       4.04540E+04    FORWARD
                                     2       4.04540E+04    FORWARD
                                     3       3.73449E+04    BACKWARD
                                     4       3.73449E+04    BACKWARD
                                     5       1.09334E+05    FORWARD
                                     6       1.09334E+05    FORWARD
                                     7       1.00406E+05    BACKWARD
                                     8       1.00406E+05    BACKWARD
                                     9       1.77687E+05    FORWARD
                                    10       1.77687E+05    FORWARD
                                    11       1.48712E+05    BACKWARD
                                    12       1.48712E+05    BACKWARD
                                    13       1.16499E+05    BACKWARD
                                    14       1.16499E+05    BACKWARD
                                    15       2.44293E+05    FORWARD
                                    16       2.44293E+05    FORWARD
                                    17       2.31815E+05    BACKWARD
                                    18       2.31815E+05    BACKWARD
                                    21       1.72225E+05    BACKWARD
                                    22       1.72225E+05    BACKWARD
                                    29       2.67606E+05    BACKWARD
                                    30       2.67606E+05    BACKWARD


 INSTABILITIES                 SOLUTION      ROTOR SPEED     WHIRL
                                NUMBER           RPM       DIRECTION

 START                               1       4.57857E+04    FORWARD
 START                               2       4.57857E+04    FORWARD
 START                               5       2.07983E+05    FORWARD
 START                               6       2.07983E+05    FORWARD


 CRITICAL SPEEDS FROM SYNCHRONOUS ANALYSIS
                               SOLUTION      ROTOR SPEED     WHIRL
                                NUMBER           RPM       DIRECTION

                                     1       3.73455E+04    BACKWARD
                                     2       3.73455E+04    BACKWARD
                                     3       4.04540E+04    FORWARD
                                     4       4.04540E+04    FORWARD
                                     5       1.00399E+05    BACKWARD
                                     6       1.00399E+05    BACKWARD
                                     7       1.09317E+05    FORWARD
                                     8       1.09317E+05    FORWARD
                                     9       1.16498E+05    BACKWARD
                                    10       1.16498E+05    BACKWARD
                                    11       1.48717E+05    BACKWARD
                                    12       1.48717E+05    BACKWARD
                                    13       1.72253E+05    BACKWARD
                                    14       1.72253E+05    BACKWARD
                                    15       1.77641E+05    FORWARD
                                    16       1.77641E+05    FORWARD
                                    19       2.31636E+05    BACKWARD
                                    20       2.31636E+05    BACKWARD
                                    21       2.44248E+05    FORWARD
                                    22       2.44248E+05    FORWARD
                                    25       2.68027E+05    BACKWARD
                                    26       2.68027E+05    BACKWARD
                                    27       3.00735E+05    BACKWARD
                                    28       3.00735E+05    BACKWARD
                                    29       3.09666E+05    FORWARD
                                    30       3.09666E+05    FORWARD
                                    31       2.39797E+06    FORWARD
                                    32       2.39797E+06    FORWARD
```

33	3.73455E+04	BACKWARD
34	3.73455E+04	BACKWARD
35	4.04540E+04	FORWARD
36	4.04540E+04	FORWARD
37	1.00399E+05	BACKWARD
38	1.00399E+05	BACKWARD
39	1.09317E+05	FORWARD
40	1.09317E+05	FORWARD
41	1.16498E+05	BACKWARD
42	1.16498E+05	BACKWARD
43	1.48717E+05	BACKWARD
44	1.48717E+05	BACKWARD
45	1.72253E+05	BACKWARD
46	1.72253E+05	BACKWARD
47	1.77641E+05	FORWARD
48	1.77641E+05	FORWARD
51	2.31636E+05	BACKWARD
52	2.31636E+05	BACKWARD
53	2.44248E+05	FORWARD
54	2.44248E+05	FORWARD
57	2.68027E+05	BACKWARD
58	2.68027E+05	BACKWARD
59	3.00735E+05	BACKWARD
60	3.00735E+05	BACKWARD
61	3.09666E+05	FORWARD
62	3.09666E+05	FORWARD
63	2.39797E+06	FORWARD
64	2.39797E+06	FORWARD

Table 24 Simcenter Nastran Results for Two Rotors Turning at the Same Speed

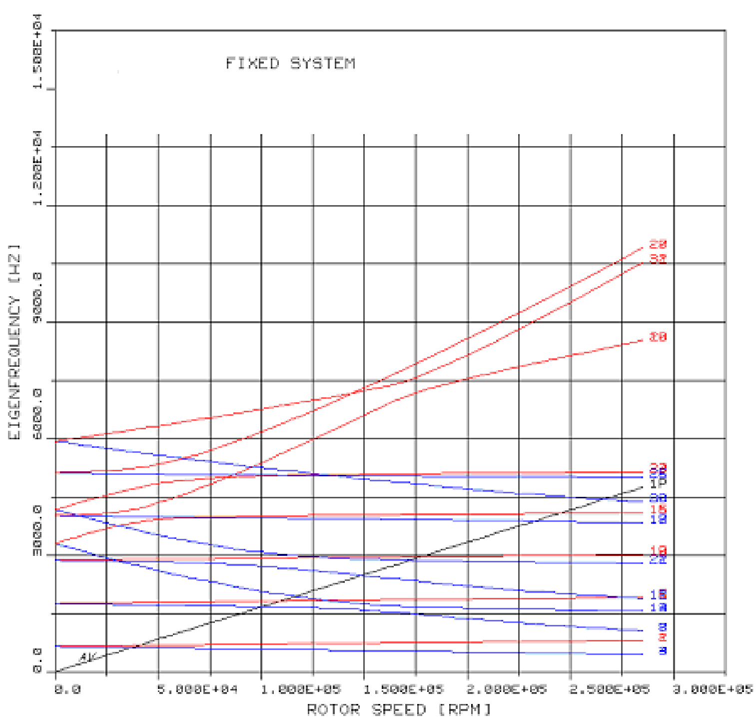


Fig. 40 Campbell Diagram for Two Rotors Turning at the Same Speed

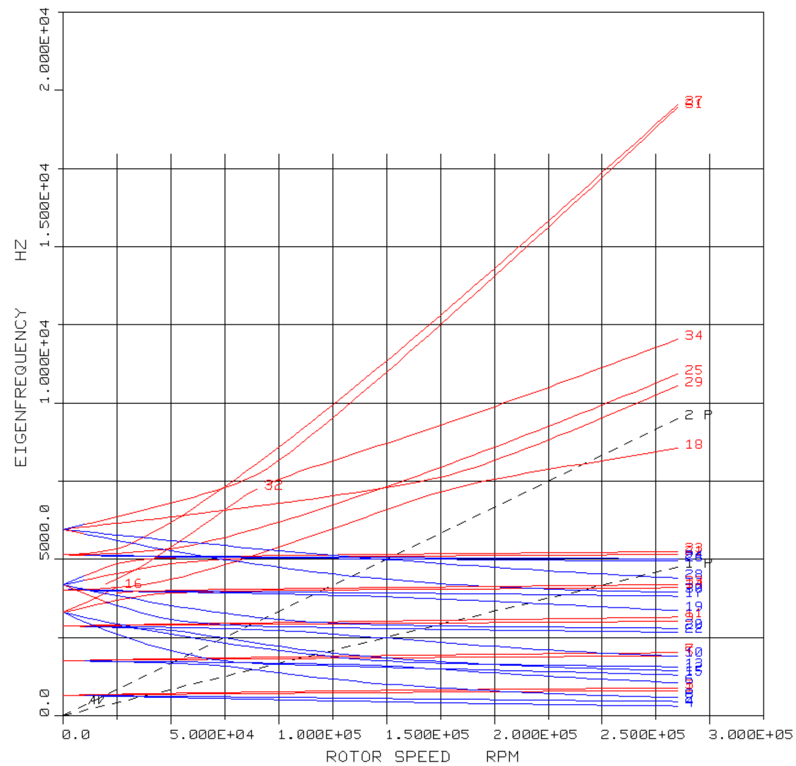


Fig. 41 Campbell diagram for two rotors. Second rotor turning at the double speed

6.3.5 Symmetric Shaft Modeled with Shell Elements (rotor097.dat)

A rotating shaft model with shell elements is shown in Fig. 38. A depiction of the upper end of the FE model is also provided below. This model is not a simple line model, and some attention must be given to the analysis method. The model is similar to that described in Section 6.3.2 except that the thickness has been increased to 20 mm to avoid an excessive number of local modes.

An abbreviated version of the input file (rotor097.dat) is shown in Table 25. Due to the length of the file, only some of the meshing data is included here. A complete version of the rotor097.dat file is available in the Test Problem Library. There are 36 elements in the tangential direction and 40 elements in the axial direction. Grid point and element labels start with 1002.

Because the shell elements have local rotations that are not directly related to the overall rotor rotations, the model must be analyzed in the rotating system, and the geometric stiffness matrix, due to the centrifugal force, must be included. Because many local modes occur for thin walled tubes, a mode selection was applied in order to select only the relevant bending and shear modes. If this model were analyzed in the fixed reference system, the nodal rotations of the elements in the plane of deformation would be constrained by AUTOSPC, and the gyroscopic matrix would be zero. Application of the parameter K6ROT would lead to unrealistic rotations.

An example of a typical bending mode is shown in Fig. 43 for the shell model and Fig. 44 for the beam model. The first shear mode is shown in Fig. 45 and Fig. 46. Using the post-processing capability available in FEMAP, this mode looks like the fourth bending mode. When plotting the rotation with FEMUTIL [ref. 3] in Fig. 47, the shear deformation is seen. The nodal rotations are practically equal for all nodes. An example of the many local modes is shown in Fig. 48.

The results from Simcenter Nastran for the critical speeds and instabilities are shown in Table 26. The Campbell diagram for the analysis in the rotating system is shown in Fig. 49. and the conversion to the fixed system in Fig. 50. Internal and external damping has been used. The real part of the eigenvalues are shown in Fig. 51. Here, the two instability points can be seen where the real parts of solutions 2 and 4 become positive. The result of an analysis in the rotating system with the first 50 modes (including the local modes) is shown in Fig. 52. There are 5 critical speeds as can be seen from the crossing points with the x-axis.

A comparison of the results with a beam model is shown in Fig. 53. There is a slight difference in the forward whirl solution of the shear mode. This is probably due to the fact that the shear mode is very sensitive to the shear factor (K1 and K2 on the PBAR entry), as shown in Fig. 54. In Fig. 54, the eigenvalues are normalized to the same eigenfrequencies without rotation.

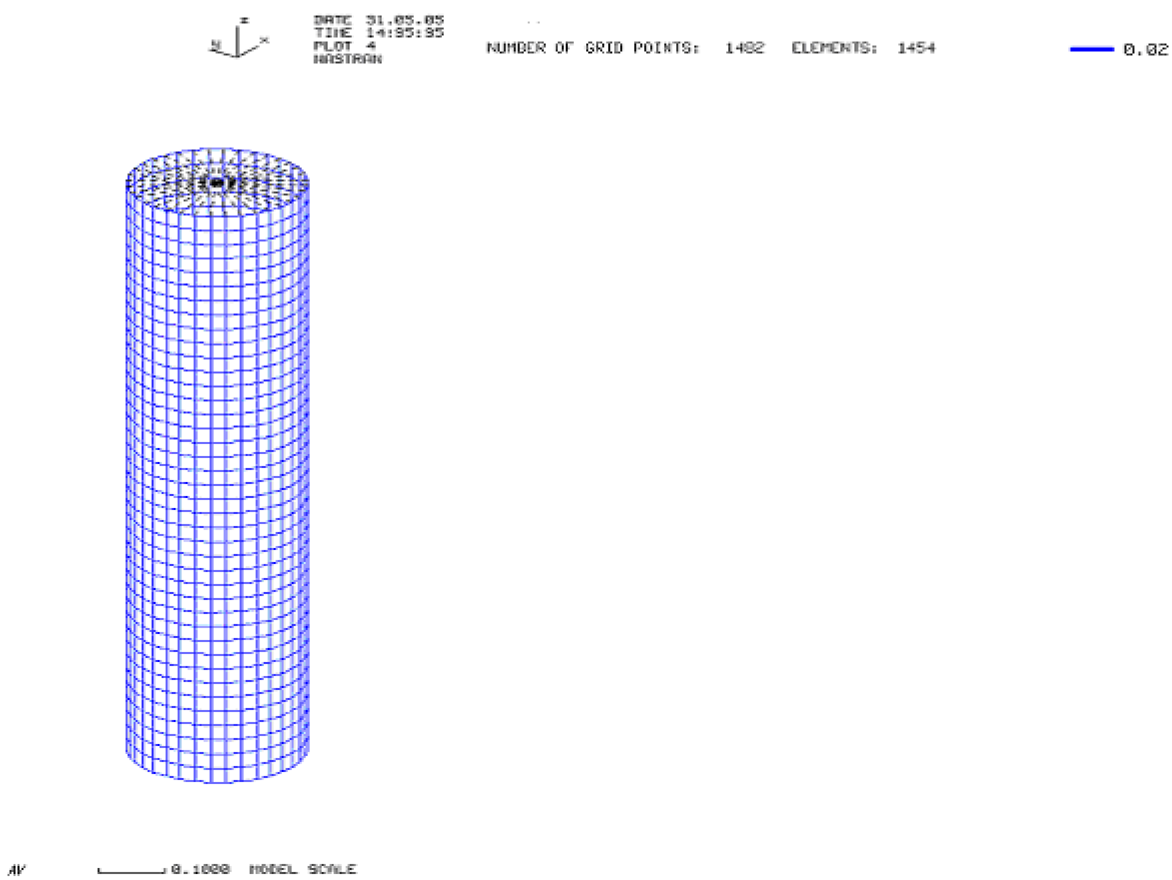


Fig. 42 Shell Model of the Rotating Shaft

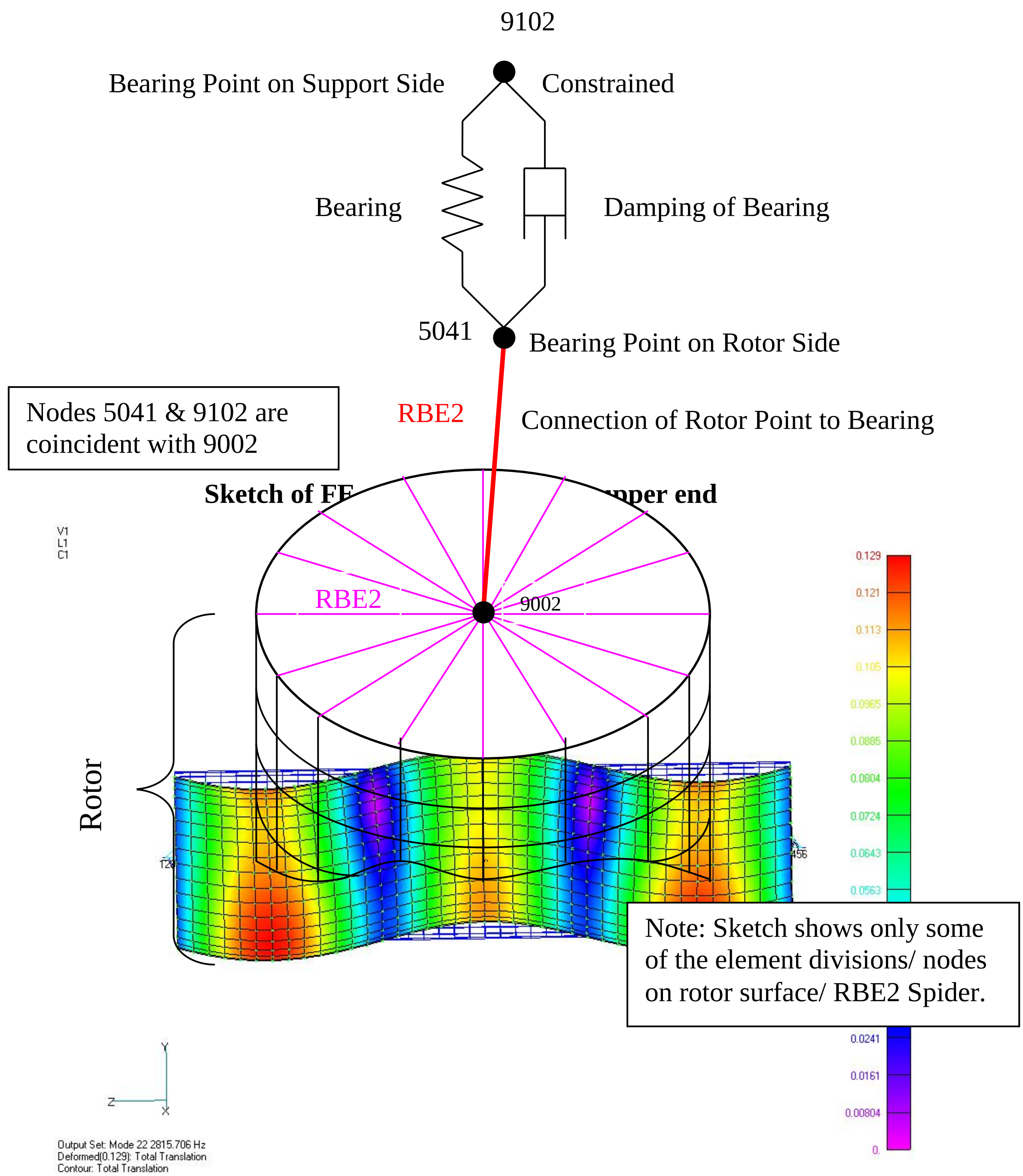


Fig. 43 Third Bending Mode of the Shell Model

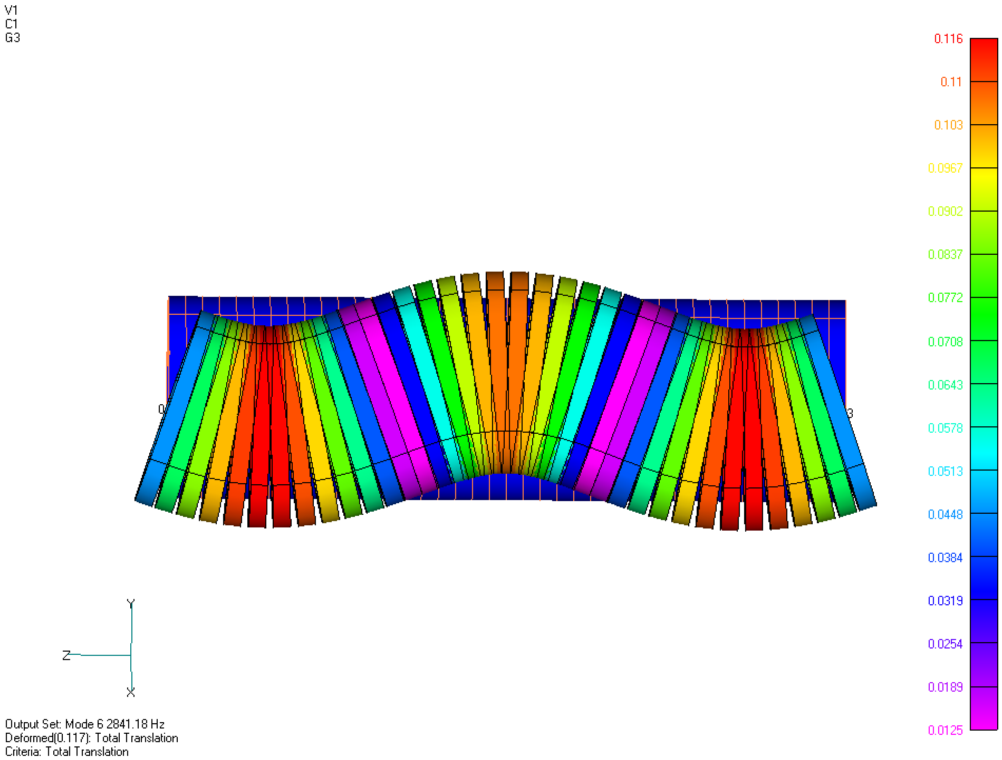


Fig. 44 Third Bending Mode of the Beam Model

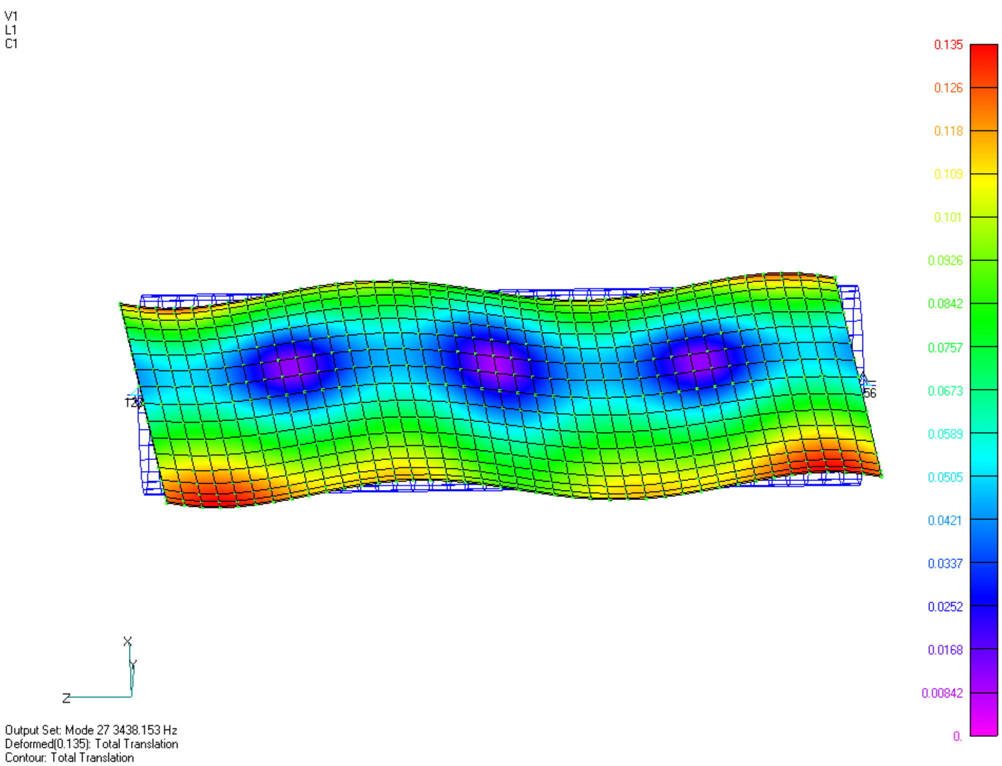


Fig. 45 First Shear Mode of the Shell Model

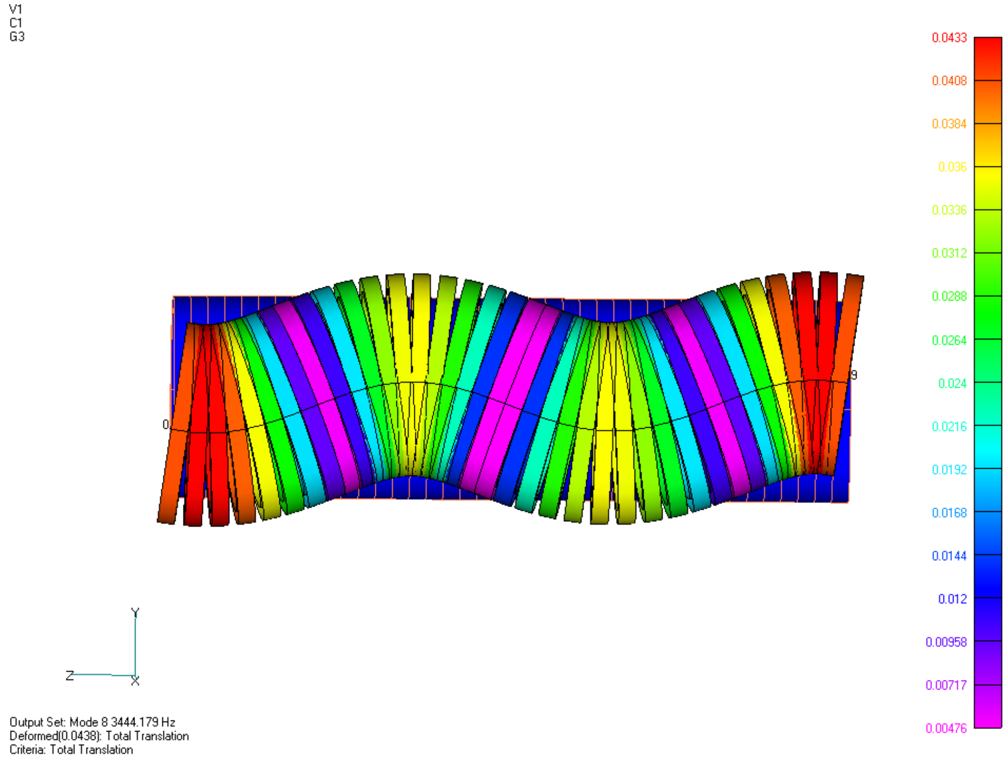


Fig. 46 First Shear Mode of the Beam Model

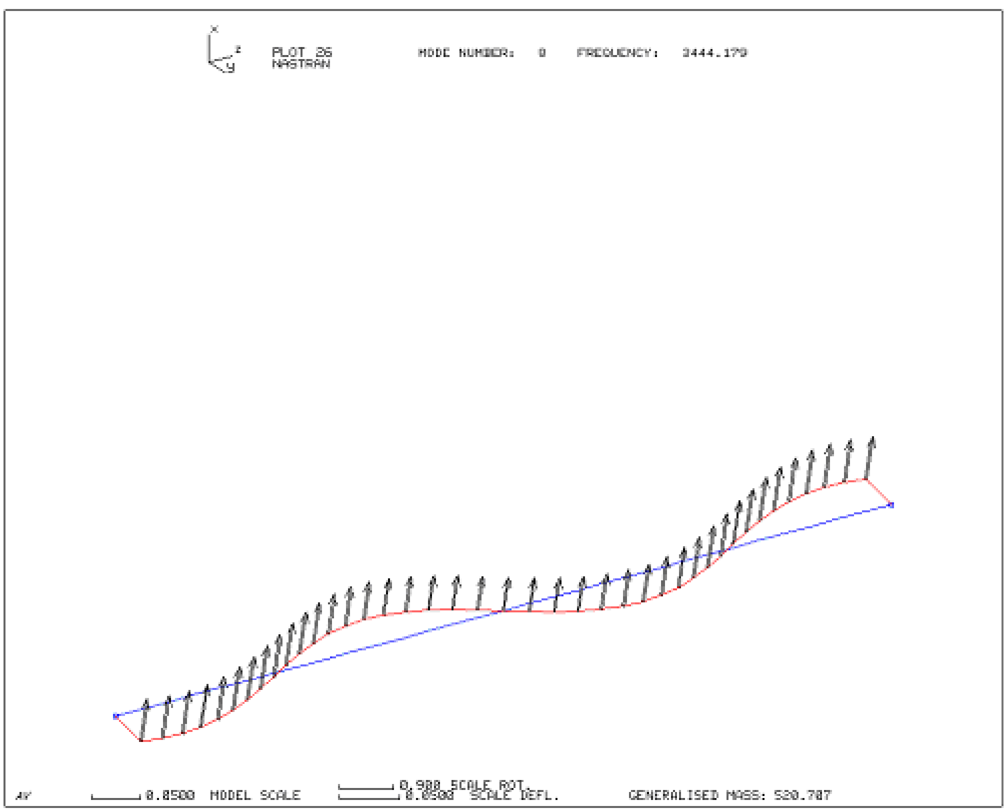


Fig. 47 First Shear Mode of the Beam Model

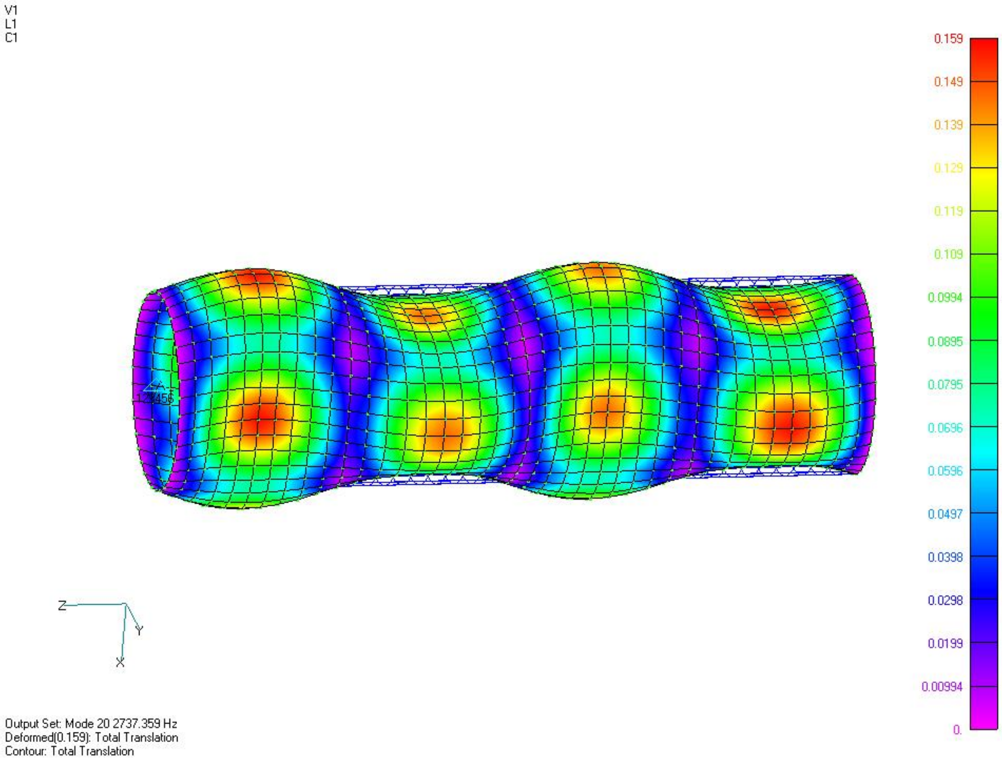


Fig. 48 Example of a Local Mode of the Shell Model

```
NASTRAN $
assign output4='rotor097.gpf',unit=22, form=formatted
assign output4='rotor097.csv',unit=25, form=formatted
$
sol 110
$
time 20000
CEND
$
ECHO = NONE
SPC = 1
$
SET 1 = 1011
$
DISP = 1
$
SET 2 = 1,2,9,10,22,23,27,28
MODSEL = 2
$
RMETHOD = 99
$
SUBCASE 1
LOAD = 1
$
SUBCASE 2
$
STATS = 1
METHOD = 1
CMETHOD = 2
$
BEGIN BULK
$
PARAM,ROTGPF,22
PARAM,ROTCSV,25
```

```
PARAM,COUPMASS,-1
PARAM,G,0.00
PARAM,K6ROT,10.0
PARAM,OGEOM,NO
PARAM,AUTOSPC,NO
PARAM,GRDPNT,0
PARAM,MODTRK,1
$
ROTORG  11      1001    THRU    2500
$
$       input for rotor dynamics
$
$       SID     RSTART  RSTEP   NUMSTEP REFSYS CMOUT   RUNIT   FUNIT
ROTORD  99      0.0     5000.0  58      ROT     -1.0    RPM     HZ          +ROT0
$       ZSTEIN  ORBEPS  ROTPRT
+ROT0   NO      1.0E-8  3                                                   +ROT1
$       RID1    RSET1   RSPEED1 RCORD1  W3-1    W4-1    RFORCE1
+ROT1   1       11      1.0     1       0.      0.      1
$
RFORCE  1       9001            0.159155 0.0    0.0     1.0    1
$
EIGRL   1                         100   1
EIGC    2       CLAN                                    32
EIGC    3       HESS                                    32
$
cord2r  1                0.      0.      0.      0.      0.      1.     +xcrd001
+xcrd001 1.      0.      0.
$
$       constrained
$
spc1    1       123456  9101    9102
$
grid    9101             0.      0.      -0.5
grid    9102             0.      0.       0.5
$
celas1  9001    9001    5001    1       9101    1
celas1  9002    9002    5001    2       9101    2
$
celas1  9003    9001    5041    1       9102    1
celas1  9004    9002    5041    2       9102    2
$
pelas   9001    18.983+9
pelas   9002    18.983+9
$
celas1  9013    9013    5001    3       9101    3
celas1  9016    9016    5001    6       9101    6
$
pelas   9013     5.000+9
pelas   9016     5.000+9
$
cdamp1  9101    9101    5001    1       9101    1
cdamp1  9102    9102    5001    2       9101    2
$
cdamp1  9103    9101    5041    1       9102    1
cdamp1  9104    9102    5041    2       9102    2
$
pdamp   9101    18.983+4
pdamp   9102    18.983+4
$
grid    5001             0.      0.     -0.5
grid    5041             0.      0.      0.5
$
rbe2    5001    5001    123456  9001
rbe2    5041    5041    123456  9002
$
$-------------------------------------------------------------------------------
$
grid    9001             0.      0.     -0.5
rbe2    9001    9001    123     1002    1003    1004    1005    1006    +rb00000
+rb000001007    1008    1009    1010    1011    1012    1013    1014    +rb00001
+rb000011015    1016    1017    1018    1019    1020    1021    1022    +rb00002
+rb000021023    1024    1025    1026    1027    1028    1029    1030    +rb00003
```

```
+rb000031031    1032    1033    1034    1035    1036    1037
$
grid    9002             0.      0.      0.5
rbe2    9002    9002    123     2442    2443    2444    2445    2446    +rb00004
+rb000042447    2448    2449    2450    2451    2452    2453    2454    +rb00005
+rb000052455    2456    2457    2458    2459    2460    2461    2462    +rb00006
+rb000062463    2464    2465    2466    2467    2468    2469    2470    +rb00007
+rb000072471    2472    2473    2474    2475    2476    2477
$
pshell  1001    1001     2.000-21001            1001
$
mat1    1001     2.10+11         0.3     7850.                   0.02
$
grid    1002             0.14    0.      -0.5
grid    1003             0.13787 0.02431-0.5
$
$Note: the remainder of the meshing data is not included here due to its length.
$See the rotor097.dat file in the Test Problem Library for the complete input file.
$
cquad4  2441    1001    2441    2406    2442    2477
$
enddata
```

Table 25 Input File for the Rotating Shaft Shell Model

```
           D E T E C T I O N   O F   R E S O N A N C E S   A N D   I N S T A B I L I T I E S

                           A N A L Y S I S   I N   R O T A T I N G   S Y S T E M



 FORWARD WHIRL RESONANCE        SOLUTION      ROTOR SPEED     WHIRL
                                 NUMBER           RPM       DIRECTION

                                      2       3.97736E+04    FORWARD
                                      4       1.09338E+05    FORWARD
                                      6       1.75318E+05    FORWARD


 BACKWARD WHIRL RESONANCE       SOLUTION      ROTOR SPEED     WHIRL
                                 NUMBER           RPM       DIRECTION

                                      1       3.67326E+04    BACKWARD
                                      3       1.01677E+05    BACKWARD
                                      5       1.61819E+05    BACKWARD
                                      8       1.28382E+05    BACKWARD


 INSTABILITIES                  SOLUTION      ROTOR SPEED     WHIRL
                                 NUMBER           RPM       DIRECTION

                                              NONE  FOUND


 CRITICAL SPEEDS FROM SYNCHRONOUS ANALYSIS
                                SOLUTION      ROTOR SPEED     WHIRL
                                 NUMBER           RPM       DIRECTION

                                      1       3.97666E+04    FORWARD
                                      2       1.09329E+05    FORWARD
```

3	1.75327E+05	FORWARD
4	3.67326E+04	BACKWARD
5	1.01656E+05	BACKWARD
6	1.28420E+05	BACKWARD
7	1.61831E+05	BACKWARD

Table 26 Results for the Rotating Shaft Shell Model with MODTRK = 1 (Rotating System)

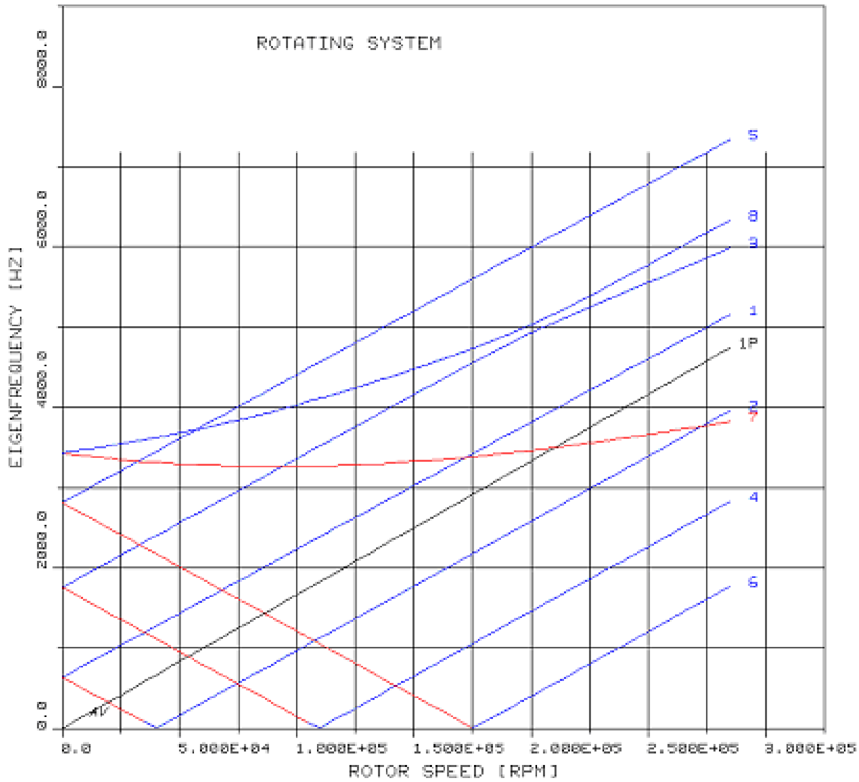


Fig. 49 Campbell Diagram for the Rotating Shell Model (Rotating System)

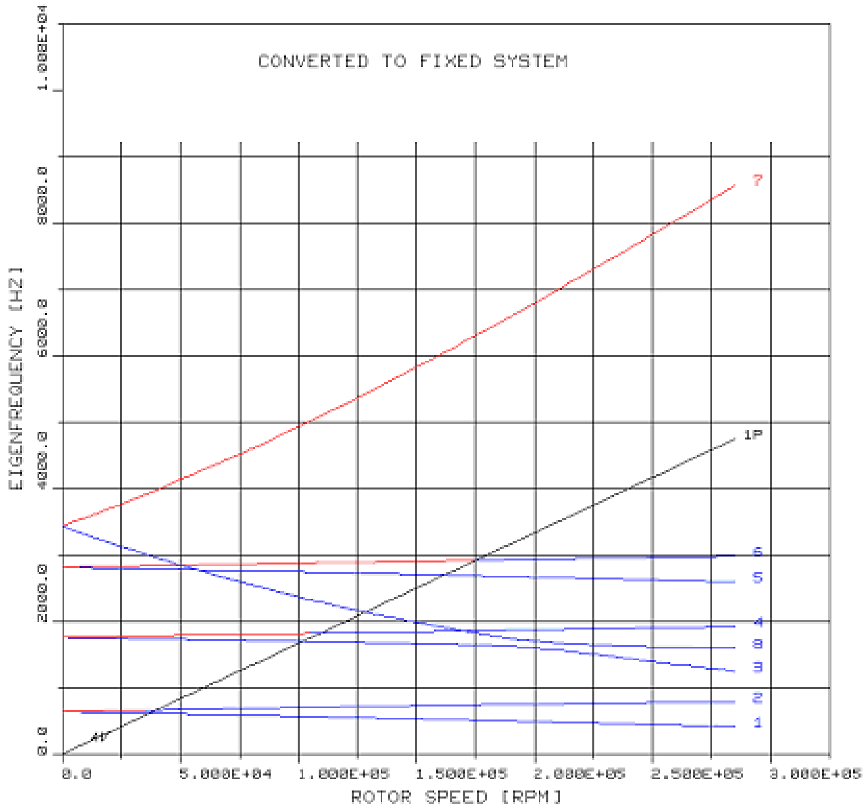


Fig. 50 Campbell Diagram for the Rotating Shell Model (Rotating System Converted to the Fixed System)

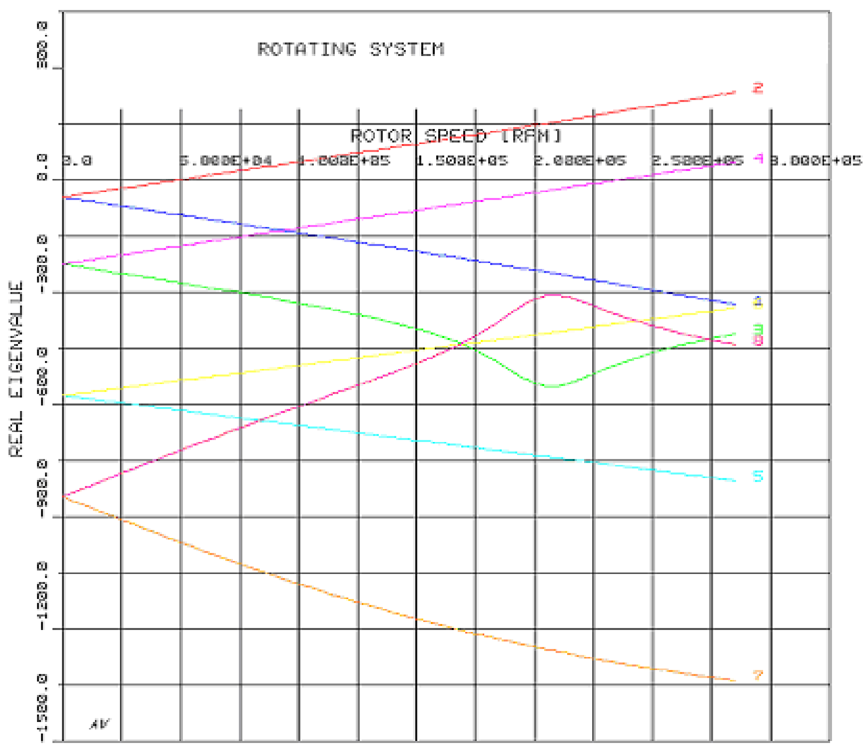


Fig. 51 Real Part of the Eigenvalues for the Rotating Shell Model

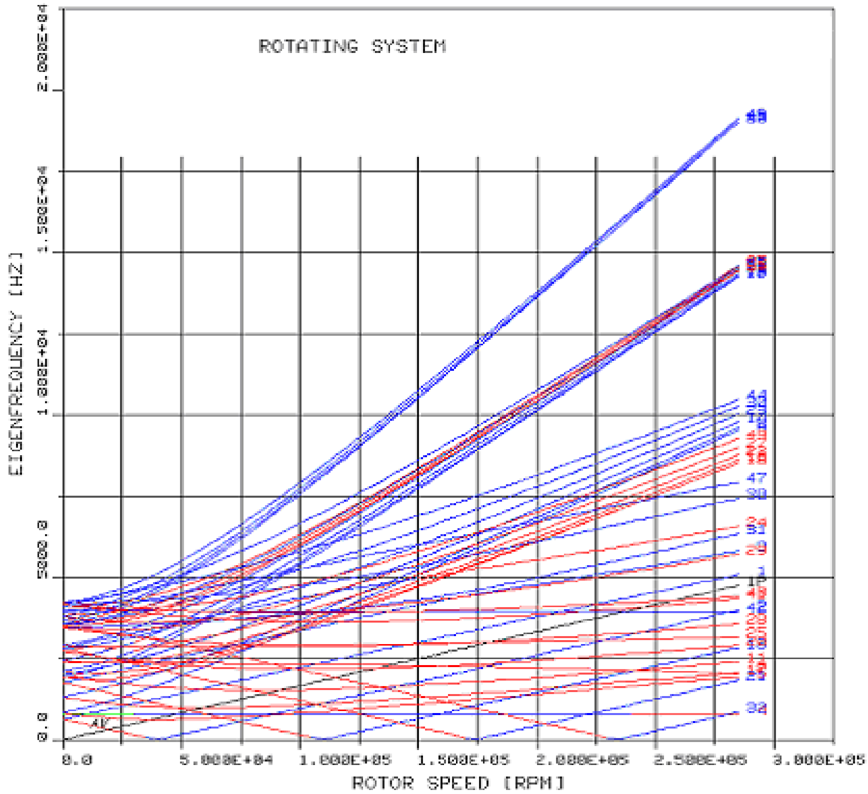


Fig. 52 Campbell Diagram for the Rotating Shell Model Including Local Modes

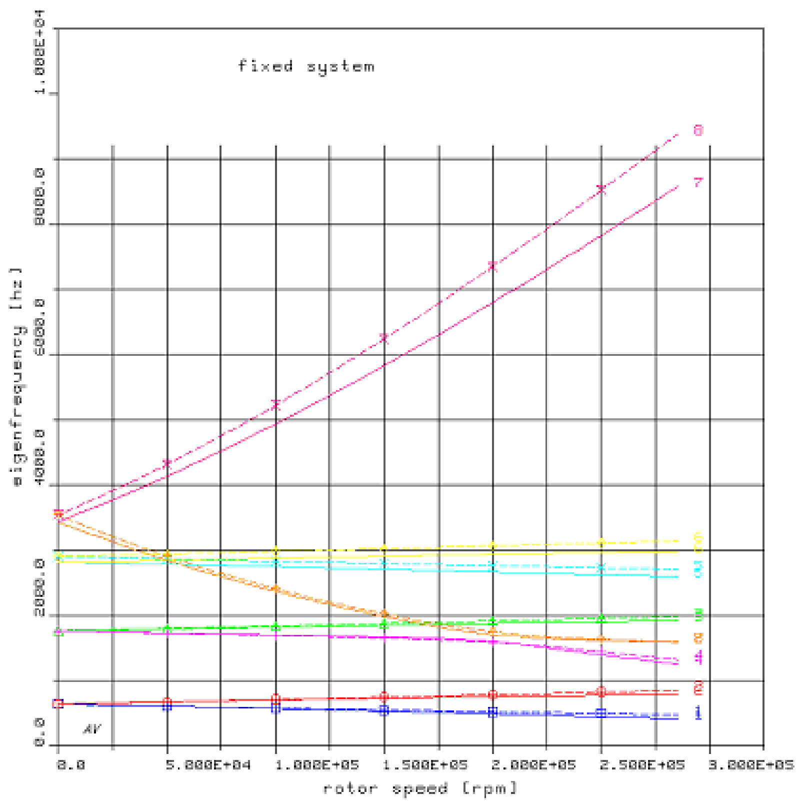


Fig. 53 Comparing Beam Model Results (Symbols) with Rotating Shell Model Results Calculated in the Rotating System and Converted to the Fixed System

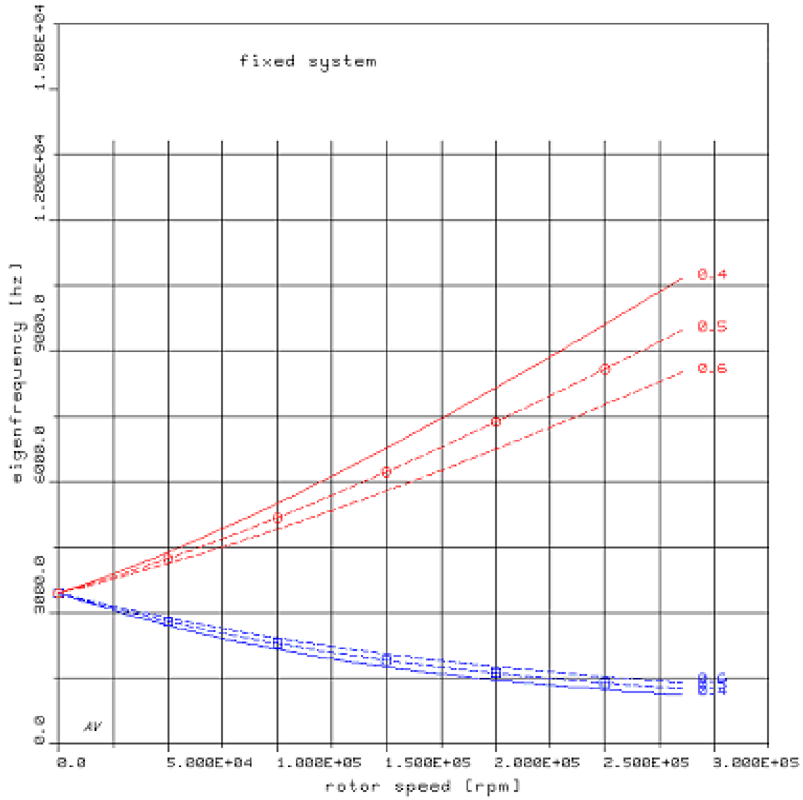


Fig. 54 Shear Factor Influence on the Shear Whirl Modes for the Beam Model

CHAPTER

7

Rotor Dynamics Examples Frequency Response

7 Frequency Response Examples

The following sections contain rotor dynamic modal frequency response analysis examples. Input files (.dat files) for the examples described in this chapter are included in the Simcenter Nastran Test Problem Library, which is located in the *install_dir*/nxnr/nast/tpl directory. To duplicate the results presented in this guide, you should add PARAM,MODTRK,1 in the bulk section of the .dat files.

7.1 Rotating Cylinder with Beam Elements

A steel cylinder with radius of 218.22 mm and a length of 436.44 mm is mounted on a shaft of length 1000 mm. The diameter of the shaft is 78.621 mm. The density of cylinder is 7.6578E-9 ton/mm^3.

The cylinder and shaft are modeled using CBAR elements. The mass of the cylinder is modeled with CONM2 elements. The stiffness and damping of bearings are modeled using CELAS1 and CDAMP1 elements. The model is analyzed for Campbell diagrams, critical speeds and modal frequency response.

The modal frequency response is calculated using various solution methods as given below in Table 30 and 31. The input for SYNC, REFSYS, ETYPE, EORDER, DLOAD, and DPHASE for each solution method is specified in these tables.

Analysis	Synchronous		Asynchronous	
Whirl	Backward	Forward	Backward	Forward
SYNC	1	1	0	0
REFSYS	FIX	FIX	FIX	FIX
ETYPE	1	1	1 (or 0)	1 (or 0)
EORDER	1.0	1.0	NA	NA
Force on DLOAD	mr	mr	mr (if ETYPE=1); mrΩ^2 (if ETYPE=0)	mr (if ETYPE=1); mrΩ^2 (if ETYPE=0)
Phase angle on DPHASE	Positive	Negative	Positive	Negative
Test Deck	rtr_mfreq21.dat	rtr_mfreq22.dat	rtr_mfreq23.dat	rtr_mfreq24.dat

Note: The value of EORDER does not matter in asynchronous analysis.

Table 27 Modal Frequency Response Solutions in the Fixed Reference System

Analysis	Synchronous		Asynchronous	
Whirl	Backward	Forward	Backward	Forward
SYNC	1	1	0	0
REFSYS	ROT	ROT	ROT	ROT
ETYPE	1	1	1 (or 0)	1 (or 0)
EORDER	0.0	2.0	NA	NA
Force on DLOAD	mr	mr	mr (if ETYPE=1); mrΩ^2 (if ETYPE=0)	mr (if ETYPE=1); mrΩ^2 (if ETYPE=0)
Phase angle on DPHASE	Positive	Negative	Positive	Negative
Test Deck	rtr_mfreq25.dat	rtr_mfreq26.dat	rtr_mfreq27.dat	rtr_mfreq28.dat

Note: The value of EORDER does not matter in asynchronous analysis.

Table 28 Modal Frequency Response Solutions in the Rotating Reference System

The Campbell diagrams, results for critical speeds and modal frequency response of various solutions are discussed below.

7.1.1 Campbell Diagrams

It is useful to analyze the structure with SOL 110 before proceeding to frequency response analyses with SOL 111 in order to check the expected resonance frequencies. An inspection of the damping values gives a hint as to the expected shape of the response curve: A low damping value leads to a strong resonance and narrow peak, whereas a large damping value leads to a broad resonance peak with less magnification.

The Campbell diagram for the non-rotating analysis system is shown in Fig. 55. The translation modes of forward and backward whirl (curves 1 and 2) are constant with rotor speed. The tilting mode (curve 3) is the backward whirl and curve 4 the forward whirl. In this figure, the crossing with the 1P line is 50 Hz for the forward and backward translation modes and 400 Hz for the forward whirl of the tilting mode and 110 Hz for the backward tilting mode.

The mass unbalance will excite the forward whirl at the 1P excitation (equal to the rotor speed). There may, however be excitation also for the backward whirl and there may also be other orders of excitation.

For these reasons, the user can define the excitation order. The whirl direction can be defined with the standard Simcenter Nastran entries for forces in the complex plane.

The Campbell diagram for analysis in the rotating system is shown in Fig. 56.

Subtracting the rotor speed from the forward whirl in Fig. 55 yields the red line in Fig. 56. This is, however, now the backwards whirl in the rotating system. At 400 Hz the backwards whirl frequency equals the rotor speed and the total motion is zero. At speeds above the zero frequency, the whirl direction changes.

The damping for the fixed system is shown in Fig. 57. The translation modes are lightly damped (2-3%) and the tilting modes have lager damping of around 6%. Hence, the peaks for the translation modes will be strong and narrow and for the tilting modes, the resonance peaks will be broad and less pronounced.

The summary of the results for analysis in the fixed and the rotating reference system are shown in Table 29 and Table 30, respectively. The critical speeds are the same in both systems and are summarized in Table 31.

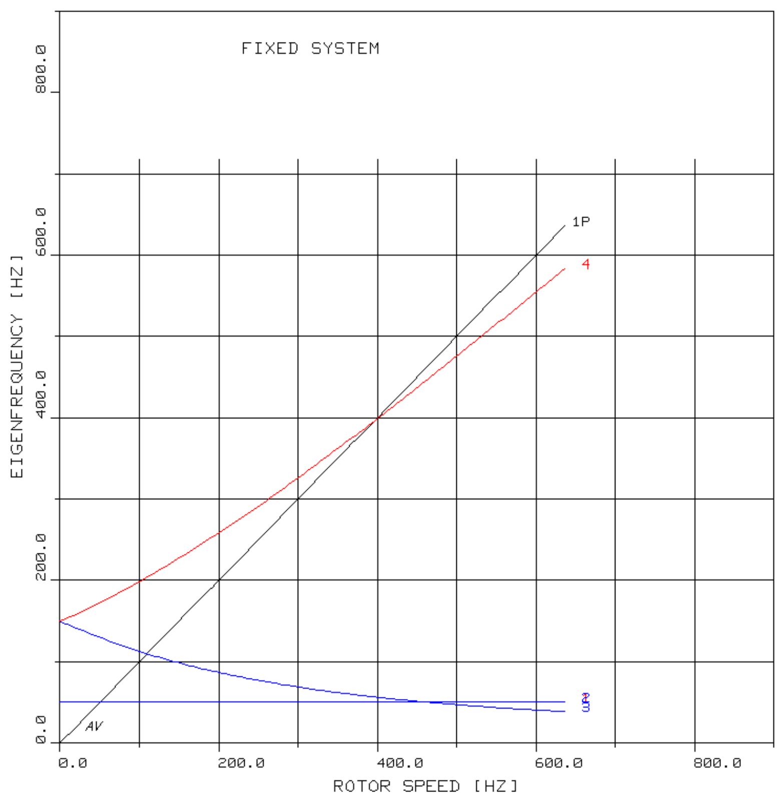


Fig. 55 Campbell diagram in the fixed system

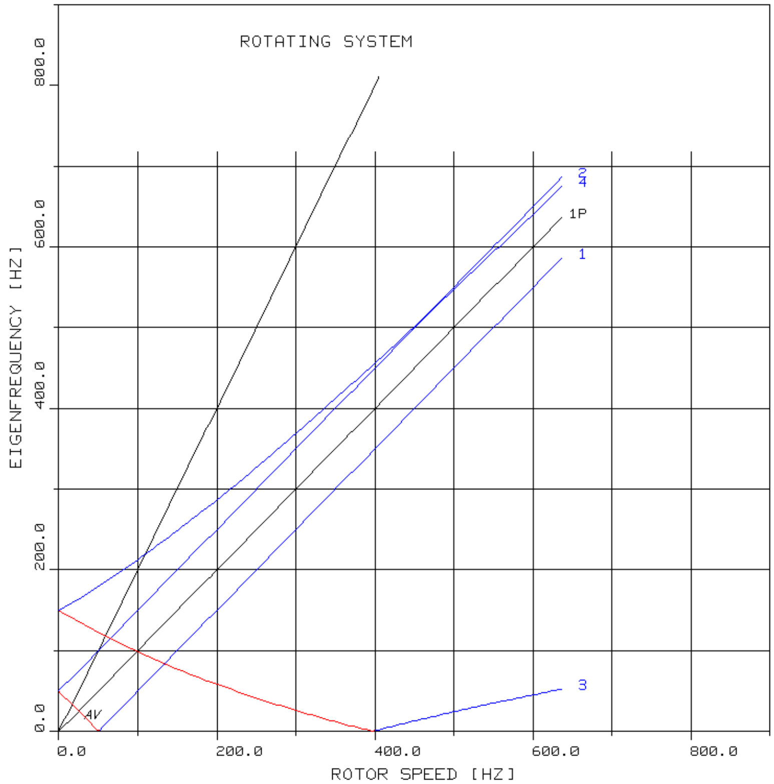


Fig. 56 Campbell diagram for analysis in the rotating system

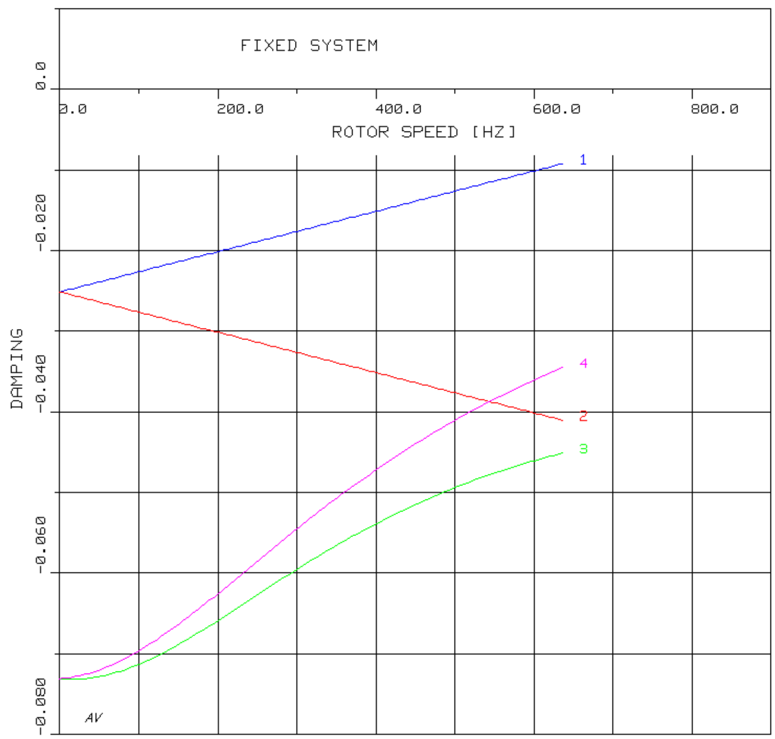


Fig. 57 Damping in the fixed system

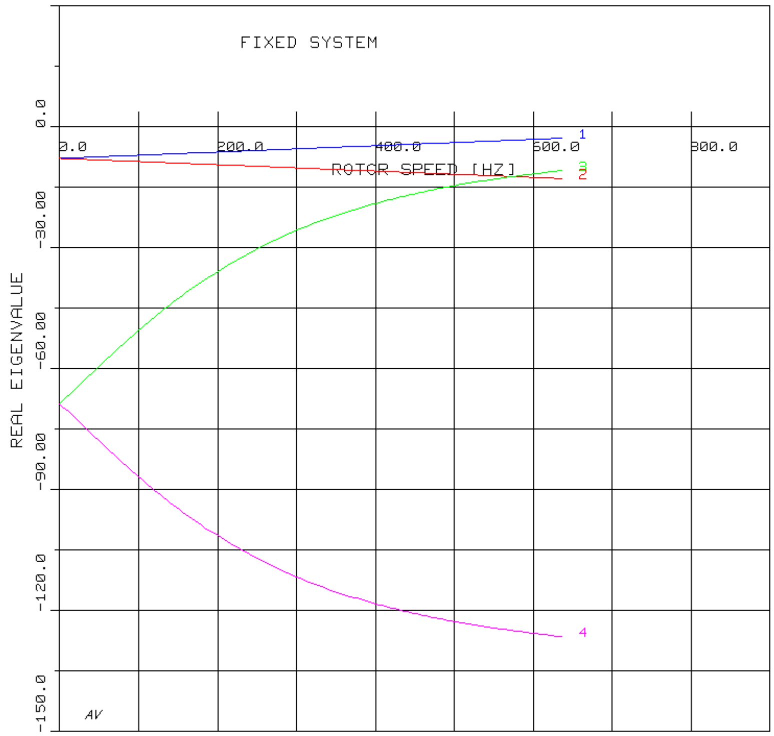


Fig. 58 Real eigenvalues in the fixed and the rotating system

```
D E T E C T I O N   O F   R E S O N A N C E S   A N D   I N S T A B I L I T I E S

               A N A L Y S I S   I N   F I X E D   S Y S T E M


WHIRL RESONANCE                SOLUTION      ROTOR SPEED     WHIRL
                                NUMBER           HZ        DIRECTION

                                     1       4.99846E+01     FORWARD
                                     2       4.99846E+01     BACKWARD
                                     3       1.09803E+02     BACKWARD
                                     4       3.96371E+02     FORWARD

INSTABILITIES                  SOLUTION      ROTOR SPEED     WHIRL
                                NUMBER           HZ        DIRECTION

                                             NONE   FOUND

CRITICAL SPEEDS FROM SYNCRONOUS ANALYSIS
                               SOLUTION      ROTOR SPEED     WHIRL
                                NUMBER           HZ        DIRECTION

                                     1       5.00004E+01     BACKWARD
                                     2       5.00004E+01     FORWARD
                                     3       1.10070E+02     BACKWARD
                                     4       3.96878E+02     FORWARD
```

Table 29 Critical speeds calculated in the fixed system

```
D E T E C T I O N   O F   R E S O N A N C E S   A N D   I N S T A B I L I T I E S

               A N A L Y S I S   I N   R O T A T I N G   S Y S T E M


FORWARD WHIRL RESONANCE        SOLUTION      ROTOR SPEED     WHIRL
                                NUMBER           HZ        DIRECTION

                                     1       4.99846E+01    FORWARD
                                     3       3.96371E+02    FORWARD


BACKWARD WHIRL RESONANCE       SOLUTION      ROTOR SPEED     WHIRL
                                NUMBER           HZ        DIRECTION

                                     2       4.99846E+01    BACKWARD
                                     4       1.09803E+02    BACKWARD


INSTABILITIES                  SOLUTION      ROTOR SPEED     WHIRL
                                NUMBER           HZ        DIRECTION

                                             NONE  FOUND
```

Table 30 Critical speeds calculated in the rotating system

	Forward	Backward
Translation	50.0	50.0
Tilt	396.4	109.8

Table 31 Critical speeds for the translation and tilt modes

7.1.2 Frequency Response Analysis in the Fixed System

The resonance peaks at the critical speeds can be found by analyzing the model for frequency response analysis.

The force is acting at a point outside of the axial middle point of the rotor in order to excite the tilt modes.

The unbalance force is defined as, $F_U = \Delta m\, R\, \Omega^2$

Assuming a radius of R=218.22 mm and a force of 1000 N at a rotor speed of 50 Hz (= 314.16 rad/s) leads to a mass of 4.643E-5 tons (because the model is in millimeters) or a mass unbalance of 0.01013 ton-mm or 10.13 kg-mm.

In order to simulate the centrifugal force for synchronous analysis, use the ETYPE field on the ROTORD entry.

ETYPE = 0 The user must define the force $F_U = \Delta m\, R\, \Omega^2$

ETYPE = 1 Define the unbalance $\Delta m\, R$ and the program will multiply by Ω^2.

For asynchronous analysis the ETYPE is also effective.

```
$       sid     s       s1      rload           rload
DLOAD   100     1.013-2 1.0     101     1.0     102
$
$               darea   delay   dphase  tabled          type
RLOAD1  101     131             141     111             0
RLOAD1  102     132             142     111             0
$
$       input  function
TABLED1 111                                                     +TBL111A
+TBL111A 0.0    1.0     100000. 1.0     ENDT
$
$       sid     grid    dof     force
DAREA   131     1008    1       1.0
DAREA   132     1008    2       1.0
$
DPHASE  141     1008    1       0.0
$       forwards
DPHASE  142     1008    2       90.0
```

Table 32 Definition of excitation force for the forward excitation

```
$       backwards
DPHASE  142     1008    2       -90.0
```

Table 33 Modification of the phase angle for backward whirl excitation

7.1.3 Synchronous Analysis

The synchronous analysis calculates the response along an excitation line in the Campbell diagram. A mass unbalance force is exciting the rotor in the 1P line.

The range of rotor speed must be defined similar to the critical speed analysis. The field EORDER must be equal to 1.0 (the default) which specifies excitation along the 1P line.

For mass unbalance, the force is:

$$F = \Omega^2\, \Delta m\, r$$

If ETYPE is set to 1.0, the user must input only the mass unbalance $\Delta m\, r$ and the program will multiply the excitation force at each speed by Ω^2 .

The FREQ entry must contain only one dummy value. The software calculates the rotor speed from the frequency.

$	SID	**RSTART**	**RSTEP**	**NUMSTEP**	REFSYS	CMOUT	RUNIT	FUNIT	
ROTORD	99	**1.0**	**1.**	**500**	FIX	-1.0	HZ	HZ	+ROT0
$	ZSTEIN	ORBEPS	ROTPRT	**SYNC**	**ETYPE**	**EORDER**			
+ROT0	NO	1.0E-5	3	**1**	**1**	**1.0**			+ROT1
$	RID1	RSET1	RSPEED1	RCORD1	W3-1	W4-1	RFORCE1		
+ROT1	1	11	1.0	1					
$									
FREQ	200	1.0							
$									

Table 34 ROTORD and FREQ entries for synchronous analysis

7.1.3.1 Forward Whirl

The results for the translation motion are shown in Fig. 59 with a narrow resonance peak at 50 Hz. The results for the tilting motion with a broad peak around 400 Hz is shown in Fig. 60. This agrees well with the expected results from the Campbell diagram in Fig. 55 and the damping values in Fig. 57. Here, EORDER = 1.0 has been used. This means, that the excitation force is zero for zero speed and the force increases with speed. For a constant force with EORDER = 0.0, the response tends to zero when frequency is increased above the resonance point as shown in Fig. 61.

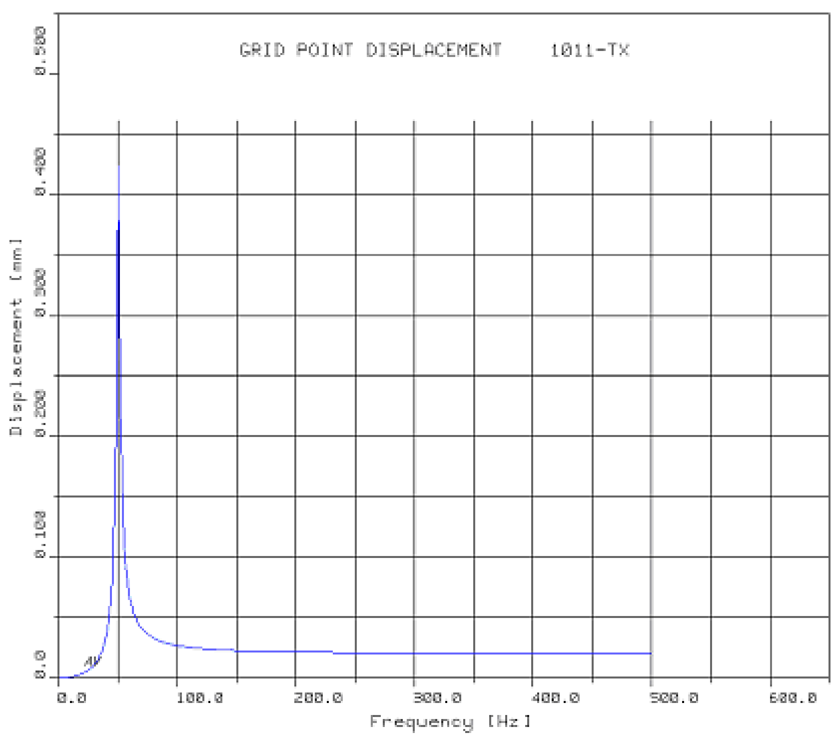


Fig. 59 Displacement of forward whirl of translation mode with resonance at 50 Hz

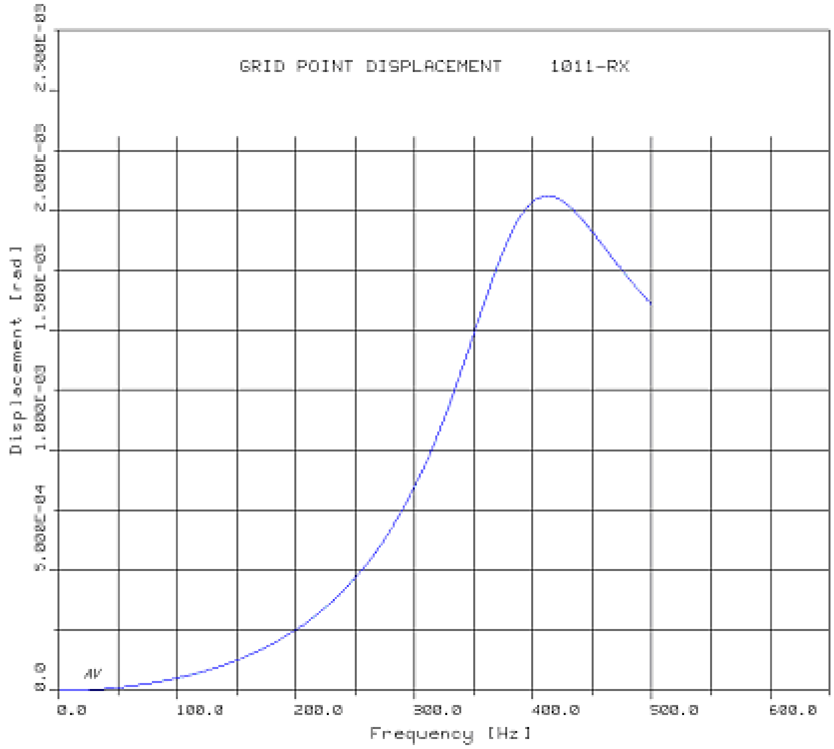


Fig. 60 Displacement of forward whirl of tilt mode with resonance around 400 Hz

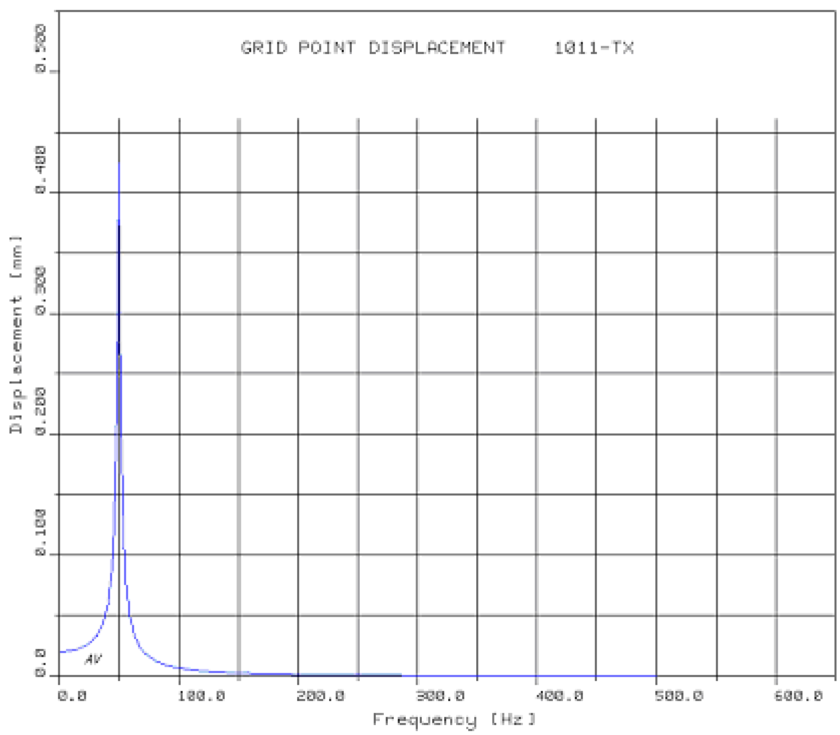


Fig. 61 Displacement of forward whirl of translation mode with resonance at 50 Hz, ETYPE=0

7.1.3.2 Backward whirl

The results for the backward whirl excitation are shown in Fig. 62 for the translation with a resonance peak at 50 Hz and in Fig. 63 for the tilting motion with resonance around 110 Hz, as expected from the Campbell plot in Fig. 55.

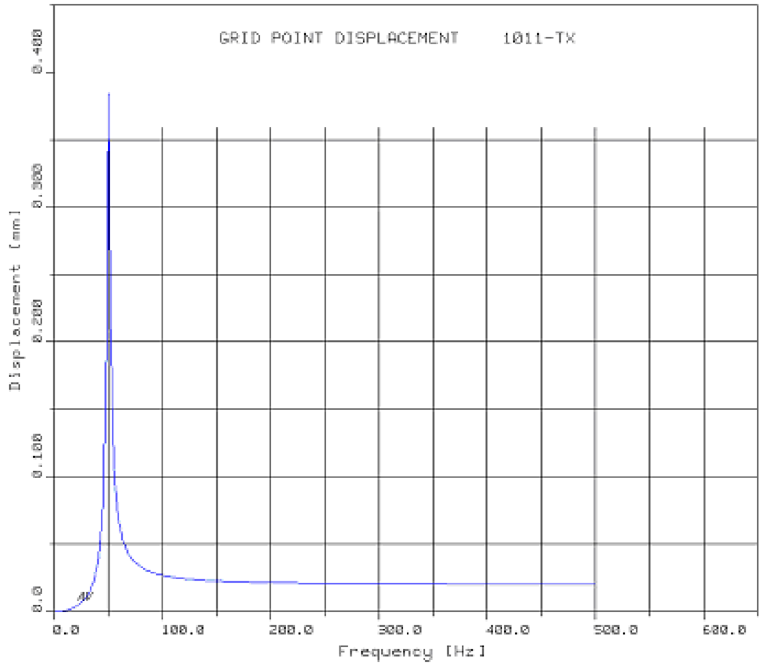


Fig. 62 Displacement of backward whirl of translation mode with resonance at 50 Hz

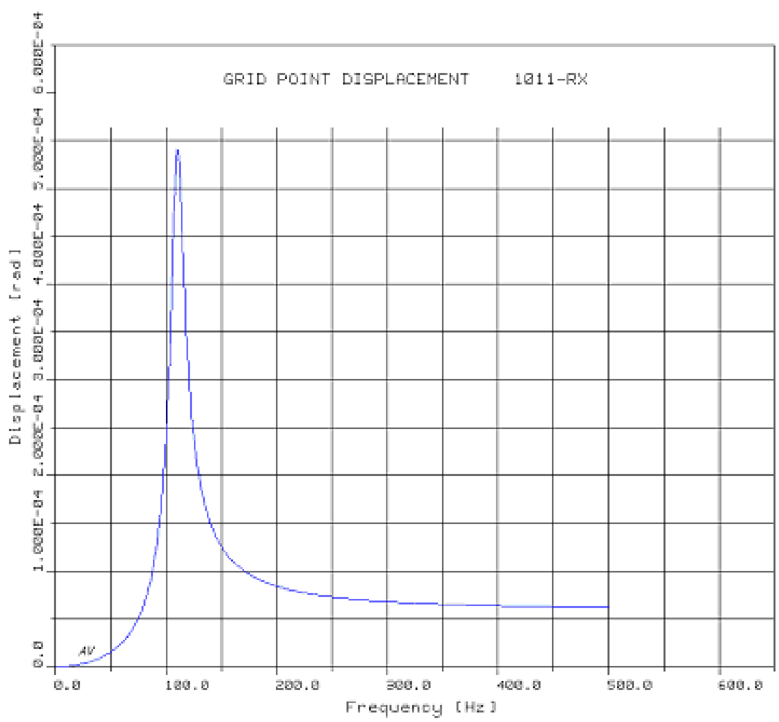


Fig. 63 Displacement of backward whirl of tilt mode with resonance around 110 Hz

7.1.4 Asynchronous Analysis

For an asynchronous analysis, the response is calculated along a vertical line in the Campbell diagram. The rotor speed is constant in this analysis and is defined by RSTART (=200.0Hz) on the ROTORD entry. The FREQ or FREQ1 entry defines the frequency range for the response calculations. The EORDER field value has no effect in this analysis. The value of ETYPE must be selected according to the nature of excitation. Either a mass unbalance (ETYPE = 1) or a force (ETYPE = 0).

```
$       SID     RSTART  RSTEP   NUMSTEP REFSYS  CMOUT   RUNIT   FUNIT
ROTORD  99      200.0   1.      1       FIX     -1.0    HZ      HZ      +ROT0
$       ZSTEIN  ORBEPS  ROTPRT  SYNC    ETYPE   EORDER
+ROT0   NO      1.0E-5  3       0       1       1.0                     +ROT1
$       RID1    RSET1   RSPEED1 RCORD1  W3-1    W4-1    RFORCE1
+ROT1   1       11      1.0     1
$
FREQ1   201     0.0     1.0     500
$
```

Table 35 ROTORD entry for Asynchronous Analysis

```
      51       2.00000E+02       -6.32365E+00  3.14066E+02        4.99852E+01 -2.01348E-02     FORWARD
      51       2.00000E+02       -9.46574E+00  3.14066E+02        4.99852E+01 -3.01393E-02     BACKWARD
      51       2.00000E+02       -3.59259E+01  5.45260E+02        8.67809E+01 -6.58875E-02     BACKWARD
      51       2.00000E+02       -1.01466E+02  1.62239E+03        2.58211E+02 -6.25410E-02     FORWARD
```

Table 36 Extracted values from the Campbell diagram of the 4 solutions at 200 Hz rotor speed

7.1.4.1 Forward Whirl

According to the Campbell diagram in Fig. 55, reading along the vertical line at 200 Hz, both forward and backward translation modes are at 50 Hz. The backward tilting mode is around 90 Hz and the forward tilting mode around 260 Hz. The exact values found with SOL 110 are shown in Table 36. The forward translation at 50 Hz is shown in Fig. 64 and the tilting motion in Fig. 65. Plotting the imaginary part versus the real part of the response peak, the Nyquist circle is obtained as shown in Fig. 66 and Fig. 67 for the two peaks respectively. The eigenfrequency and damping can be determined from the Nyquist plot and are compared to the values found from SOL 110 in Table 37. The agreement is good. The Nyquist analysis is not included in Simcenter Nastran and is used here only for verification. Programs from **Error! Reference source not found.**

	Frequency		Damping %	
	SOL 110	SOL 111 Nyquist	SOL 110	SOL 111 Nyquist
Translation	49.99	49.99	2.01	2.03
Tilt	258.21	258.31	6.25	6.28

Table 37 Comparison of frequencies and damping for SOL 110 and 111.

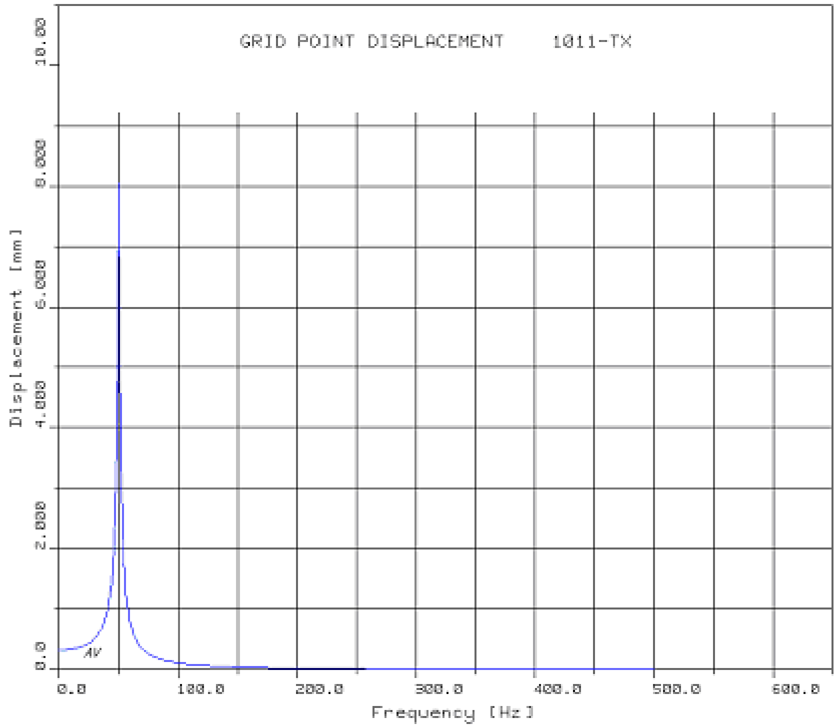


Fig. 64 Displacement for 200 Hz rotor speed, forward whirl

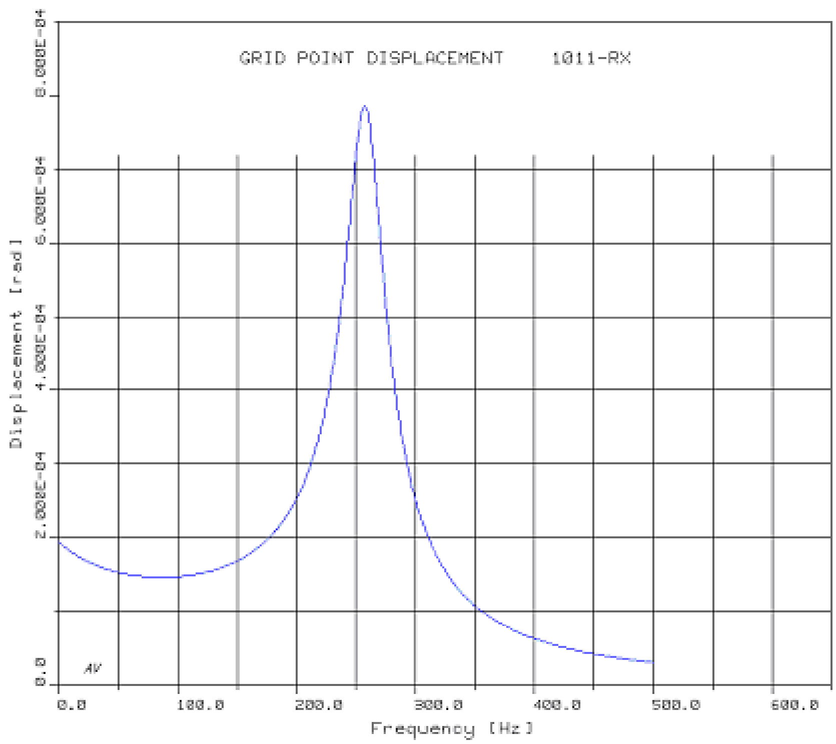


Fig. 65 Rotor speed 200 Hz, forward whirl, tilting motion displacement

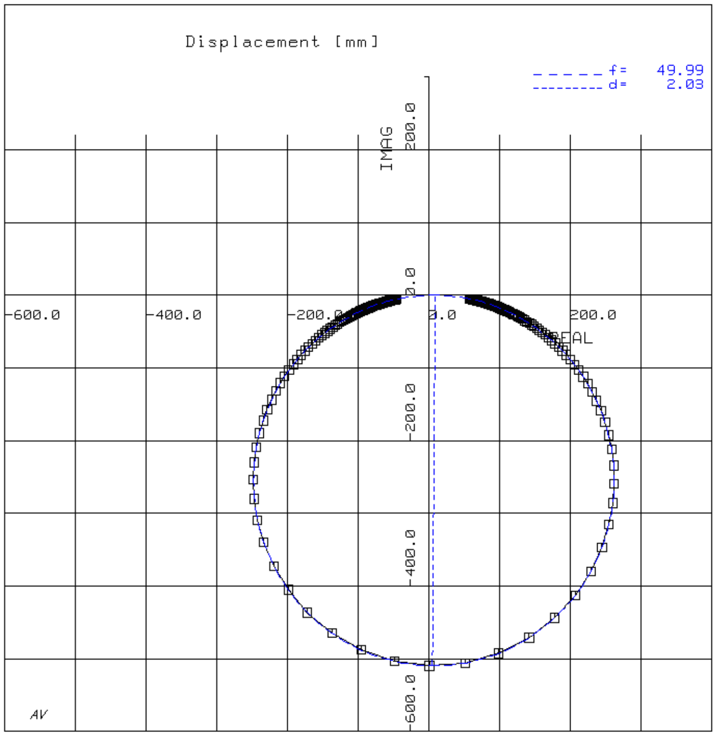


Fig. 66 Nyquist plot of translation resonance peak

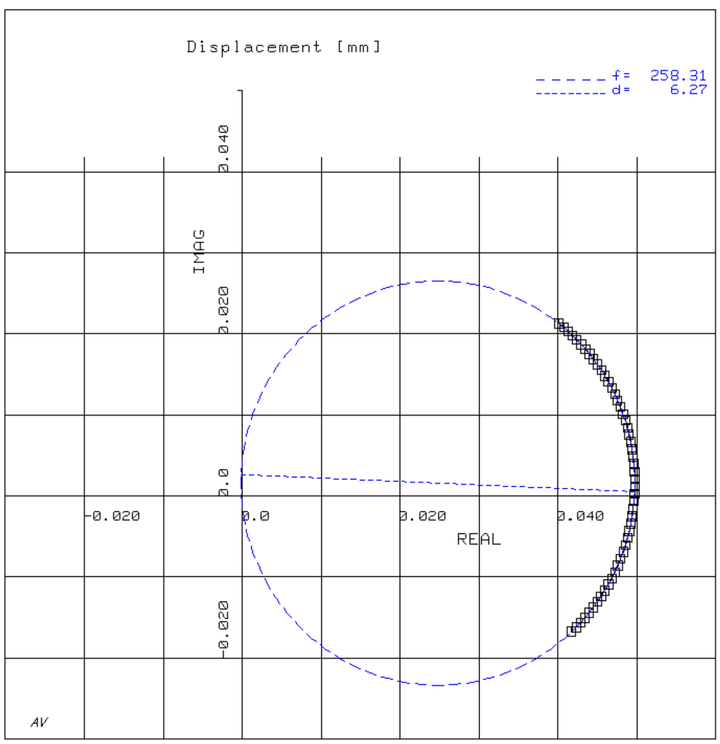


Fig. 67 Nyquist plot of tilting resonance peak

7.1.4.2 Backward Whirl

The results for the backward whirl excitation are shown for the translation and the tilting motion in Fig. 68 and Fig. 69 respectively. The resonance frequencies are close to the values shown in Table 36.

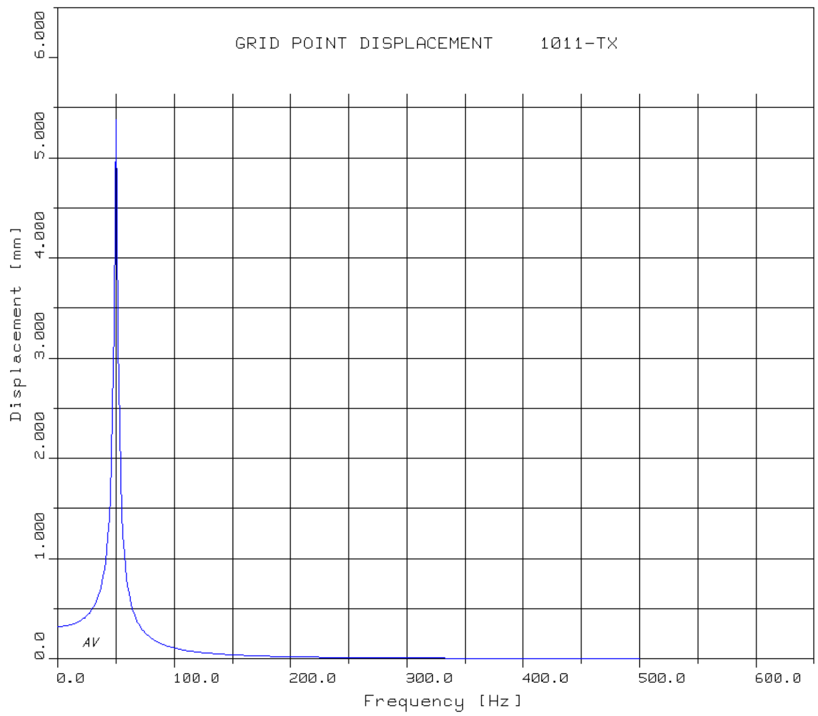


Fig. 68 Displacement response of backward whirl of translation motion

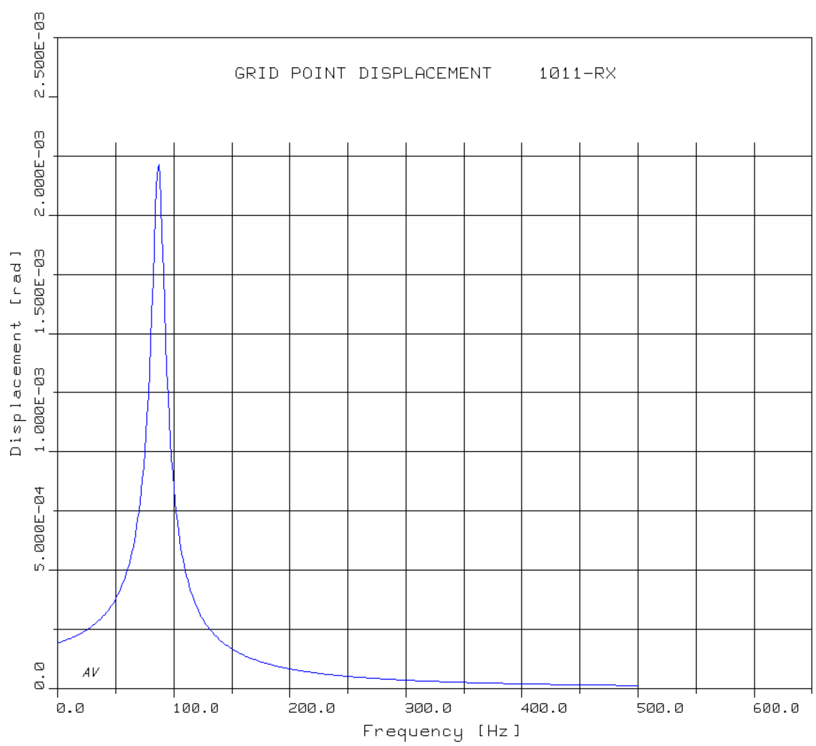


Fig. 69 Displacement response of backward whirl of tilt motion

7.1.5 Analysis in the rotating system

Forward whirl resonance is the intersection with the rotor speed axis, which means the 0P line. In this case EORDER must be equal to 0.0.

7.1.6 Synchronous Analysis in the Rotating System

7.1.6.1 Forward Whirl

The input is shown in Table 38. It must be noted, that the excitation direction must be changed when analyzing in the rotating system.

In the rotating system the forward 1P resonance is found for the 0P line. Because at the speeds where zero frequency is found, the whirl direction changes. For this reason, the results with forward and backward excitation are identical. The results shown in Fig. 70 and Fig. 71 are identical to those obtained for the fixed system in Fig. 59 and Fig. 60.

```
$
$       SID     RSTART  RSTEP   NUMSTEP REFSYS  CMOUT   RUNIT   FUNIT
ROTORD  99      1.0     1.      500     ROT     -1.0    HZ      HZ      +ROT0
$       ZSTEIN  ORBEPS  ROTPRT  SYNC    ETYPE   EORDER
+ROT0   NO      1.0E-5  3       1       1       0.0                     +ROT1
$       RID1    RSET1   RSPEED1 RCORD1  W3-1    W4-1    RFORCE1
+ROT1   1       11      1.0     1
$
$       BACKWARDS
DPHASE  142     1008    2       -90.0
$
```

Table 38 Input for synchronous analysis in the rotating system

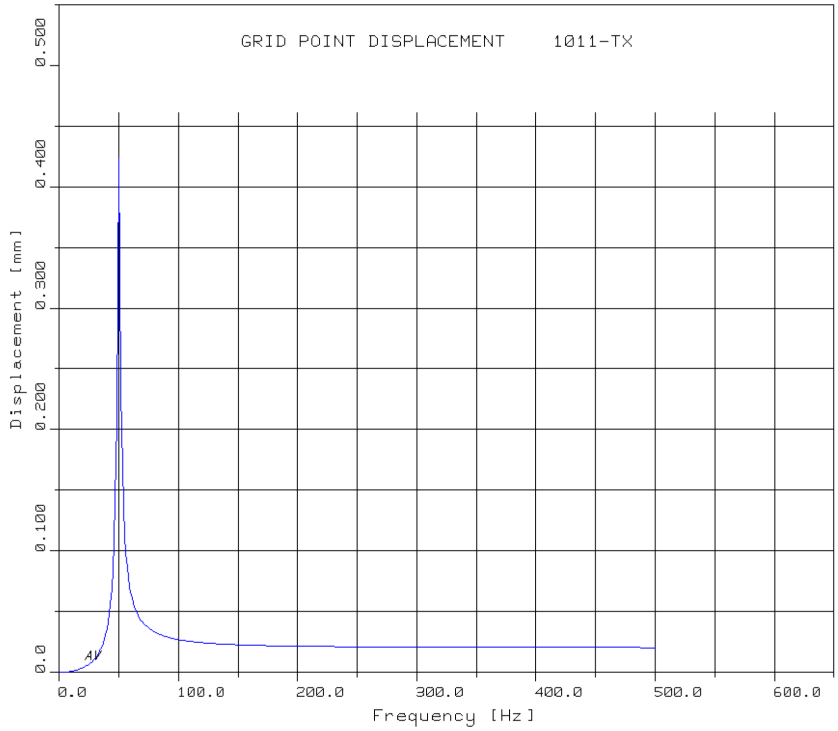


Fig. 70 Displacement response of translation motion to forward whirl excitation

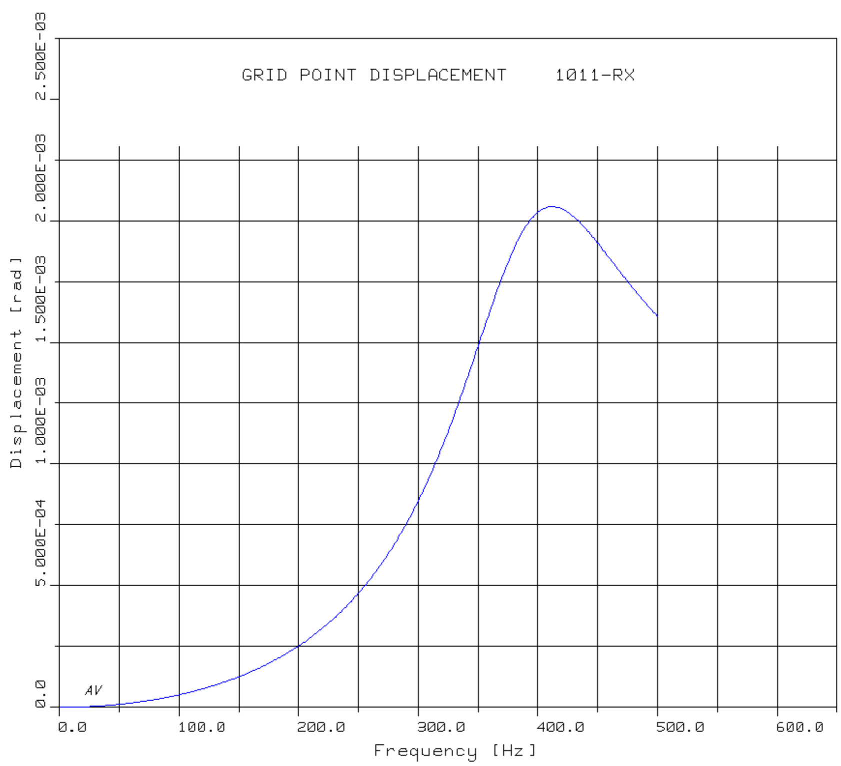


Fig. 71 Displacement response of tilting motion to forward whirl excitation

7.1.6.2 Backward Whirl

The resonances for the backward whirl modes are found for the 2P excitation line. Hence, EORDER=2.0 as shown in Table 39. The results shown in Fig. 72 and Fig. 73 are identical to the results found for the fixed system in Fig. 62 and Fig. 63.

$	SID	RSTART	RSTEP	NUMSTEP	**REFSYS**	CMOUT	RUNIT	FUNIT	
ROTORD	99	1.0	1.	500	ROT	-1.0	HZ	HZ	+ROT0
$	ZSTEIN	ORBEPS	ROTPRT	SYNC	ETYPE	**EORDER**			
+ROT0	NO	1.0E-5	3	1	1	**2.0**			+ROT1
$	RID1	RSET1	RSPEED1	RCORD1	W3-1	W4-1	RFORCE1		
+ROT1	1	11	1.0	1					
$									
$	FORWARDS								
DPHASE	**142**	**1008**	**2**	**90.0**					

Table 39 Input for backward whirl analysis in the rotating system

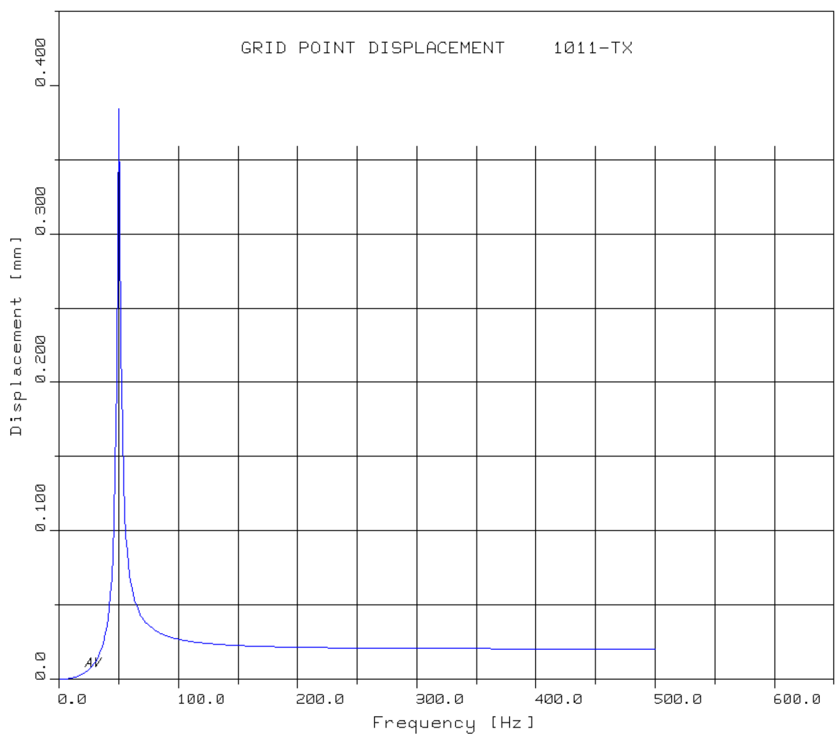


Fig. 72 Displacement response of translation motion to backwards whirl excitation

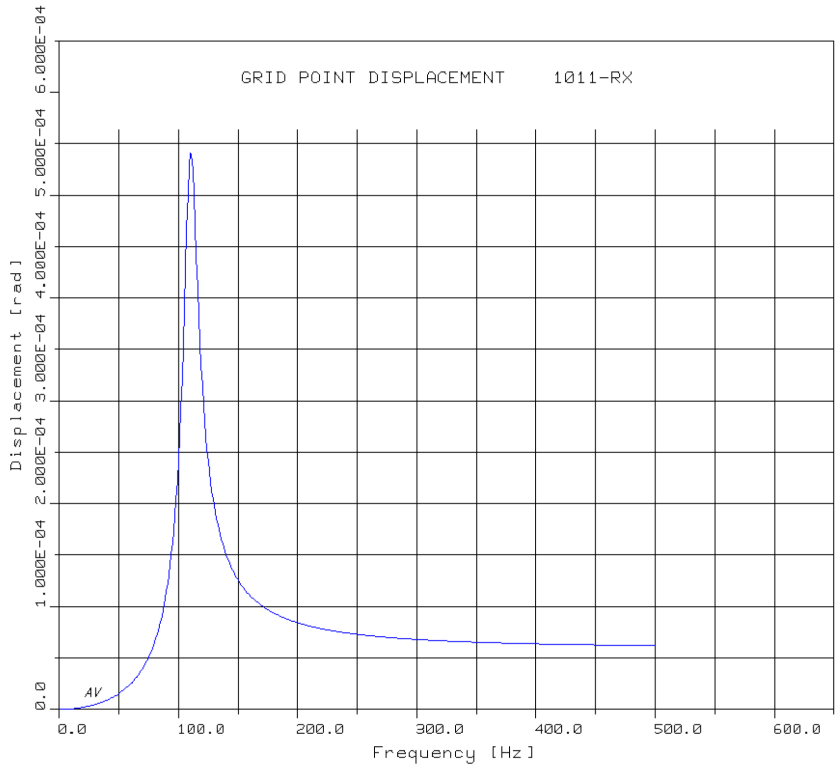


Fig. 73 Displacement response of tilt motion to backwards whirl excitation

7.1.7 Asynchronous Analysis

The asynchronous analyses for the rotor speed of 200 Hz were performed for the forward and backward whirl motion.

7.1.7.1 Forward Whirl

According to the Campbell diagram for the rotating system in Fig. 56, there are two forward resonances of the translation modes at 150 and 250 Hz respectively. The resonance of the tilt mode is around 290 Hz.

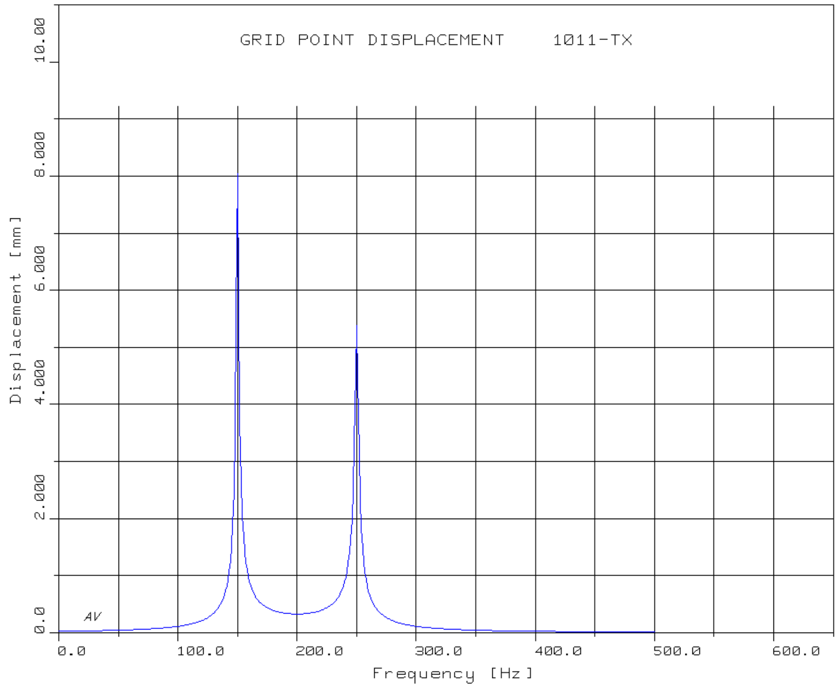


Fig. 74 Translation response for forward asynchronous analysis at 200 Hz rotor speed

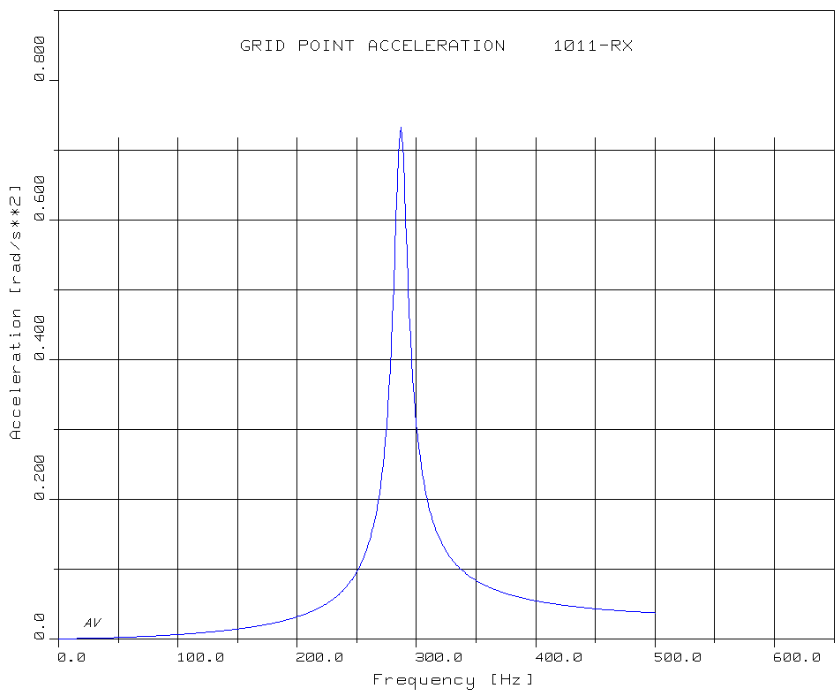


Fig. 75 Tilt response for forward asynchronous analysis at 200 Hz rotor speed

7.1.7.2 Backward Whirl

According to the Campbell diagram in Fig. 56 there is no backward whirl resonance at 200 Hz rotor speed. This is confirmed in the response analysis for the translation motion shown in Fig. 76. The response of the tilt motion is found around 60 Hz as shown in Fig. 76.

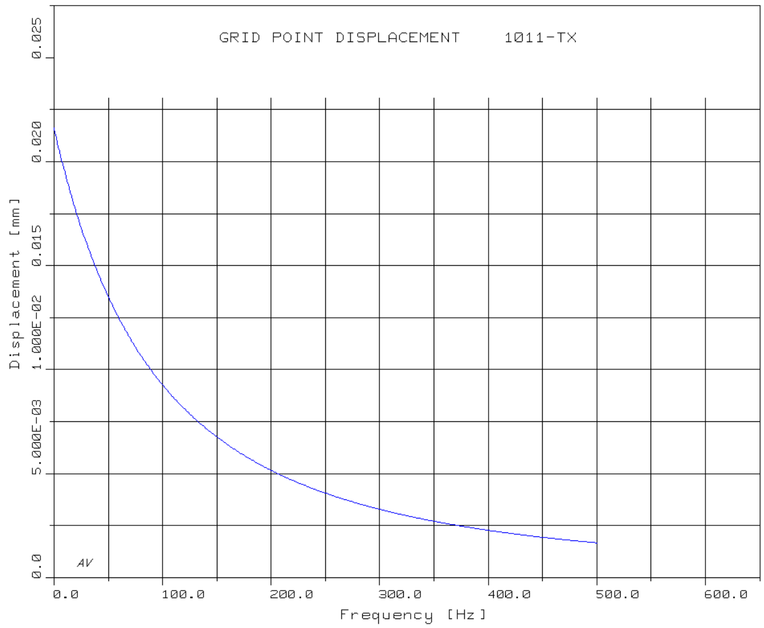


Fig. 76 Translation response for backward asynchronous analysis at 200 Hz rotor speed

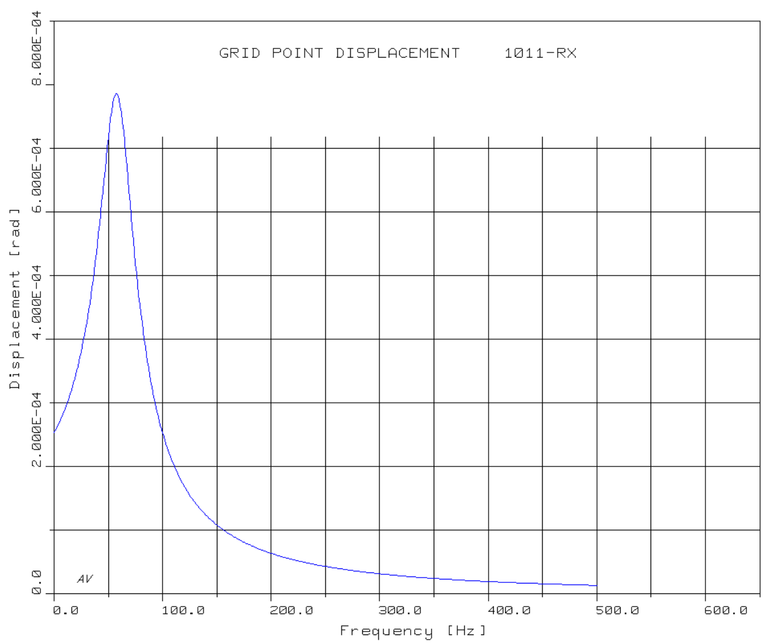


Fig. 77 Tilt response for backward asynchronous analysis at 200 Hz rotor speed

7.2 Rotating Shaft with Shell Elements

The symmetric shaft example (rotor097.dat) is modified for modal frequency response analysis. The model is analyzed with synchronous and asynchronous excitation in a rotating reference system. The shell model cannot be analyzed in the fixed system. Elastic rotors that cannot be analyzed with a line model must be analyzed in the rotating reference system. In the rotating system, the geometric stiffness matrix must be accounted for. The force may be of type mass unbalance, which will excite the rotor in the forward whirl mode. The mass unbalance force is:

$f = m\ r\ \Omega^2$

You can define the dynamic excitation force with data as given in Table 40. Because the rotor speed is contained in the equation, you can define the ETYPE field of the ROTORD entry as 1. In this case, the force is multiplied by Ω^2 and you must define a dynamic force equal to m r.

```
$
$
$       SID     RSTART  RSTEP   NUMSTEP REFSYS  CMOUT   RUNIT   FUNIT
ROTORD  99      500.0   500.0   400     ROT     -1.0    RPM     HZ      +ROT0
$       ZSTEIN  ORBEPS  ROTPRT  SYNC    ETYPE   EORDER
+ROT0   NO      1.0E-8  3       1       1       0.0                     +ROT1
$+ROT0   NO      1.0E-8  3       1                                       +ROT1
$       RID1    RSET1   RSPEED1 RCORD1  W3-1    W4-1    RFORCE1
+ROT1   1       11      1.0     1       0.      0.      15
$
$
$       0.1 kg UNBALANCE AT R=0.14 M
$
$       SID     S       S1      RLOAD           RLOAD
DLOAD   100     0.014   1.0     101     1.0     102
$
$               DAREA   DELAY   DPHASE  TABLED          TYPE
RLOAD1  101     131             141     111             0
RLOAD1  102     132             142     111             0
$
$       INPUT FUNCTION FORCE
TABLED1 111                                                     +TBL111A
+TBL111A 0.0    1.0     100.    1.0     ENDT
$
$       SID     GRID    DOF
DAREA   131     1899    1       1.0
DAREA   132     1899    2       1.0
$
DPHASE  141     1899    1       0.0
$       FORWARDS
$DPHASE  142     1899    2       90.0
$       BACKWARDS
DPHASE  142     1899    2       -90.0
$
FREQ     201     0.0
$FREQ1    200     0.0    5000.0   58
```

Table 40 Input of dynamic excitation force.

7.2.1 Synchronous Analysis

The input deck for this example is rtr_mfreq03.dat. This example is analyzed using a rotating reference system and hence the forces must be inverted. The Campbell diagram in the rotating system is shown in Fig. 78. The forward whirl resonance is shown in Fig. 79 where EORDER = 0.0 has been used. The backwards whirl resonance is shown in Fig. 80 where EORDER = 2.0 has been used. The peaks are in accordance with the critical speeds found in SOL 110 as shown in Table 41 . The critical speeds are also shown in Fig. 81 for the fixed reference system. The critical speeds are the same in both systems but the frequencies differ by ±Ω.

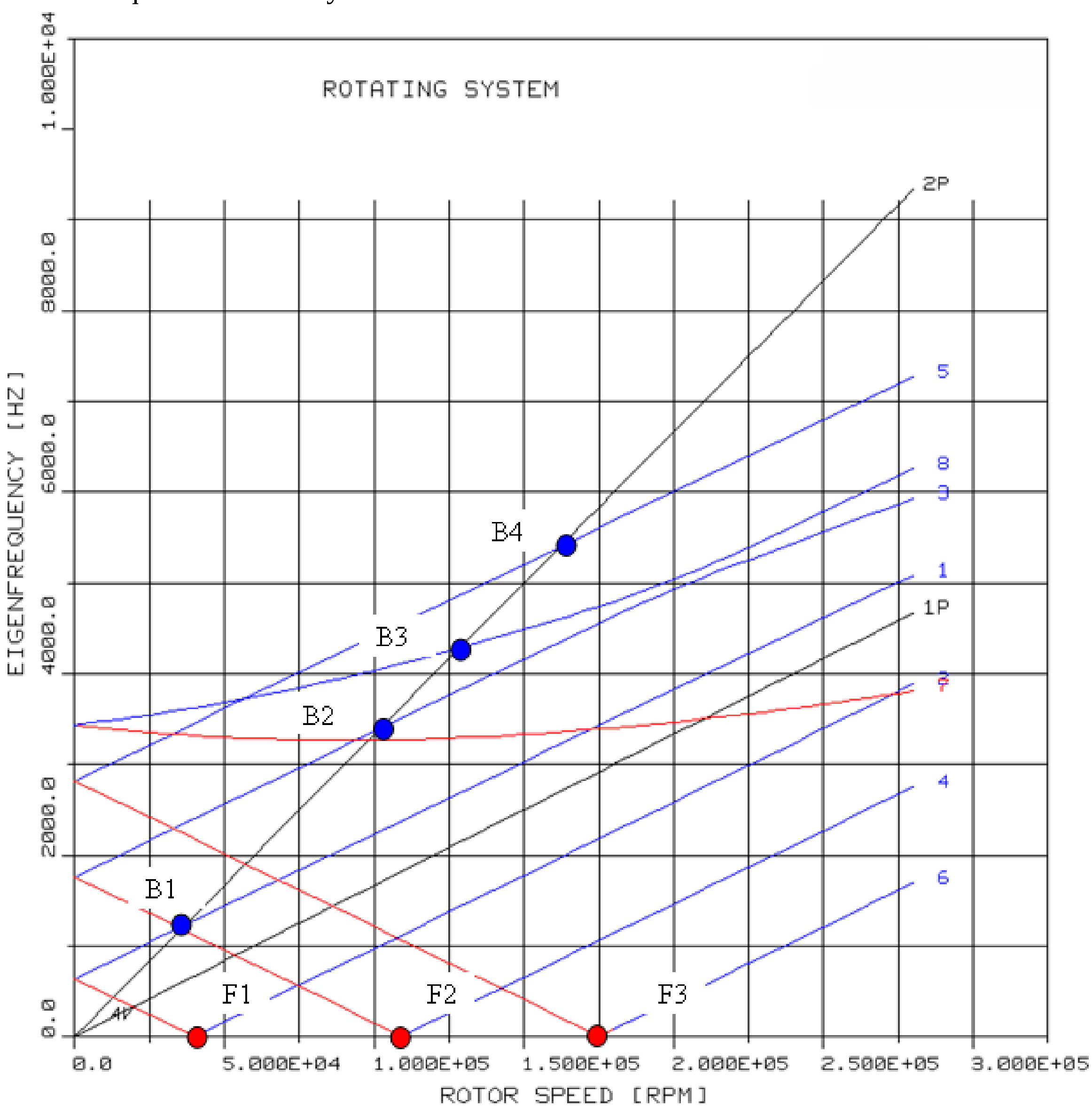


Fig. 78 Critical speeds for forward and backward whirl calculated in the rotating system.

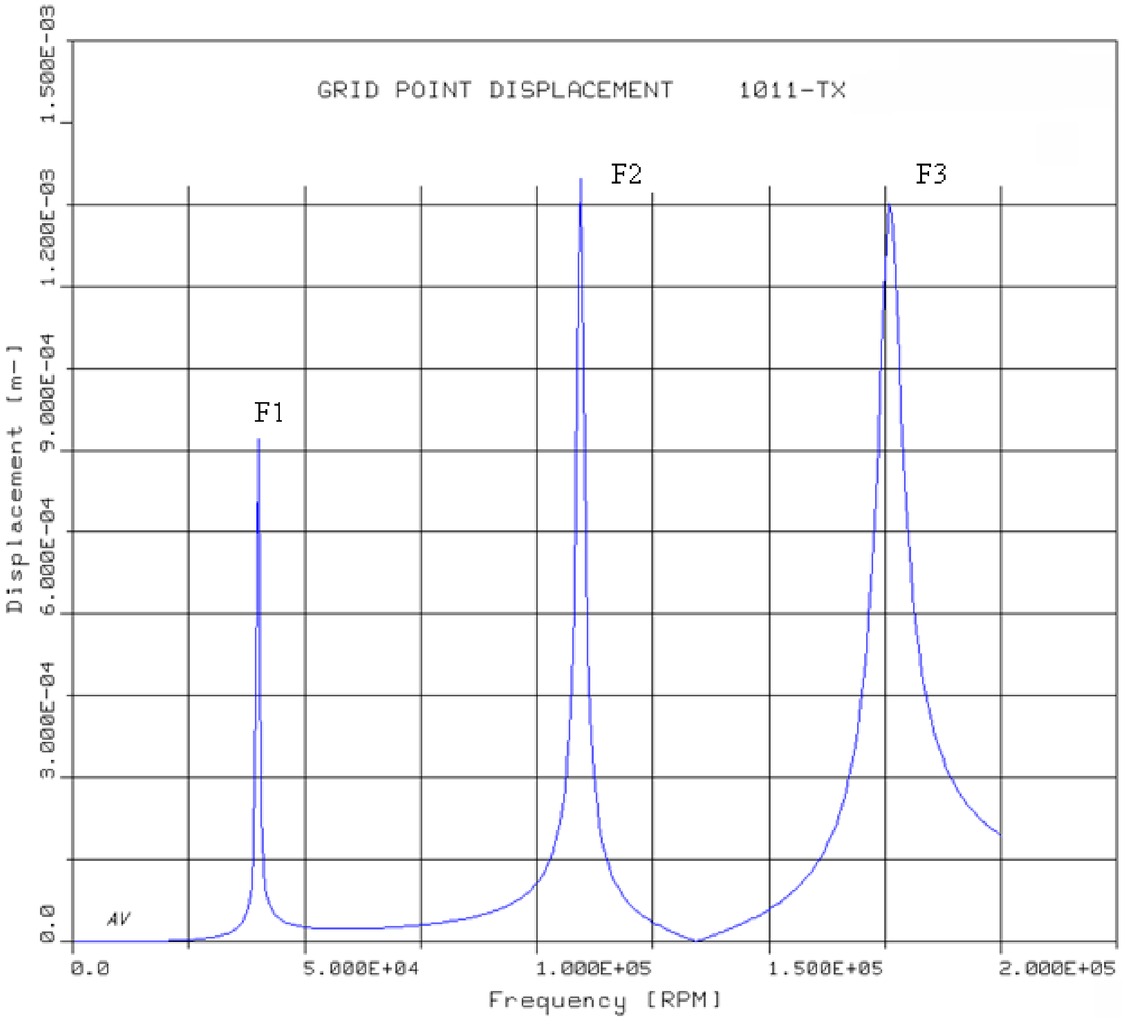


Fig. 79 Forwards whirl resonance peaks calculated with response analysis using backward or forward excitation at 0P in the rotating system.

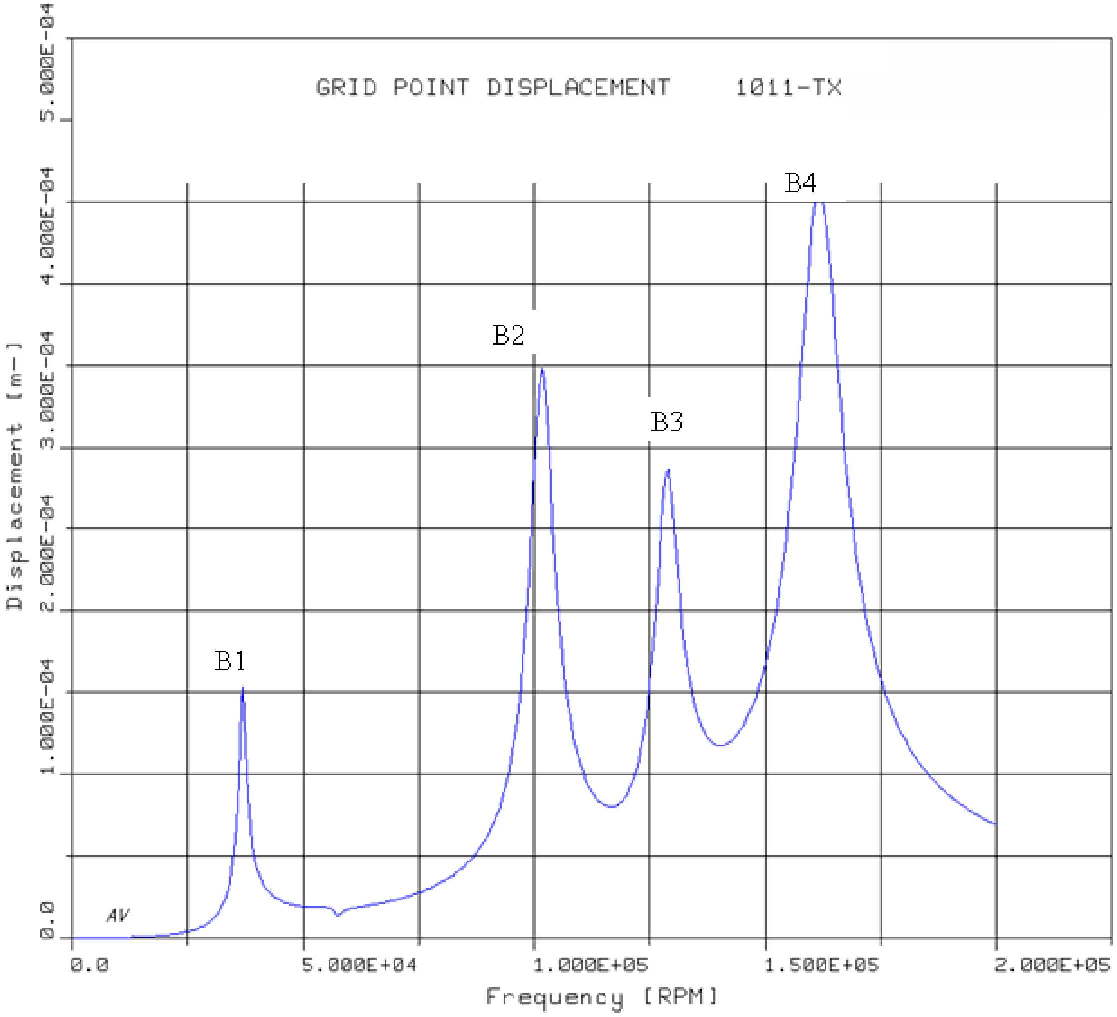


Fig. 80 Backwards whirl resonance peaks calculated with response analysis using forward excitation at 2P in the rotating system.

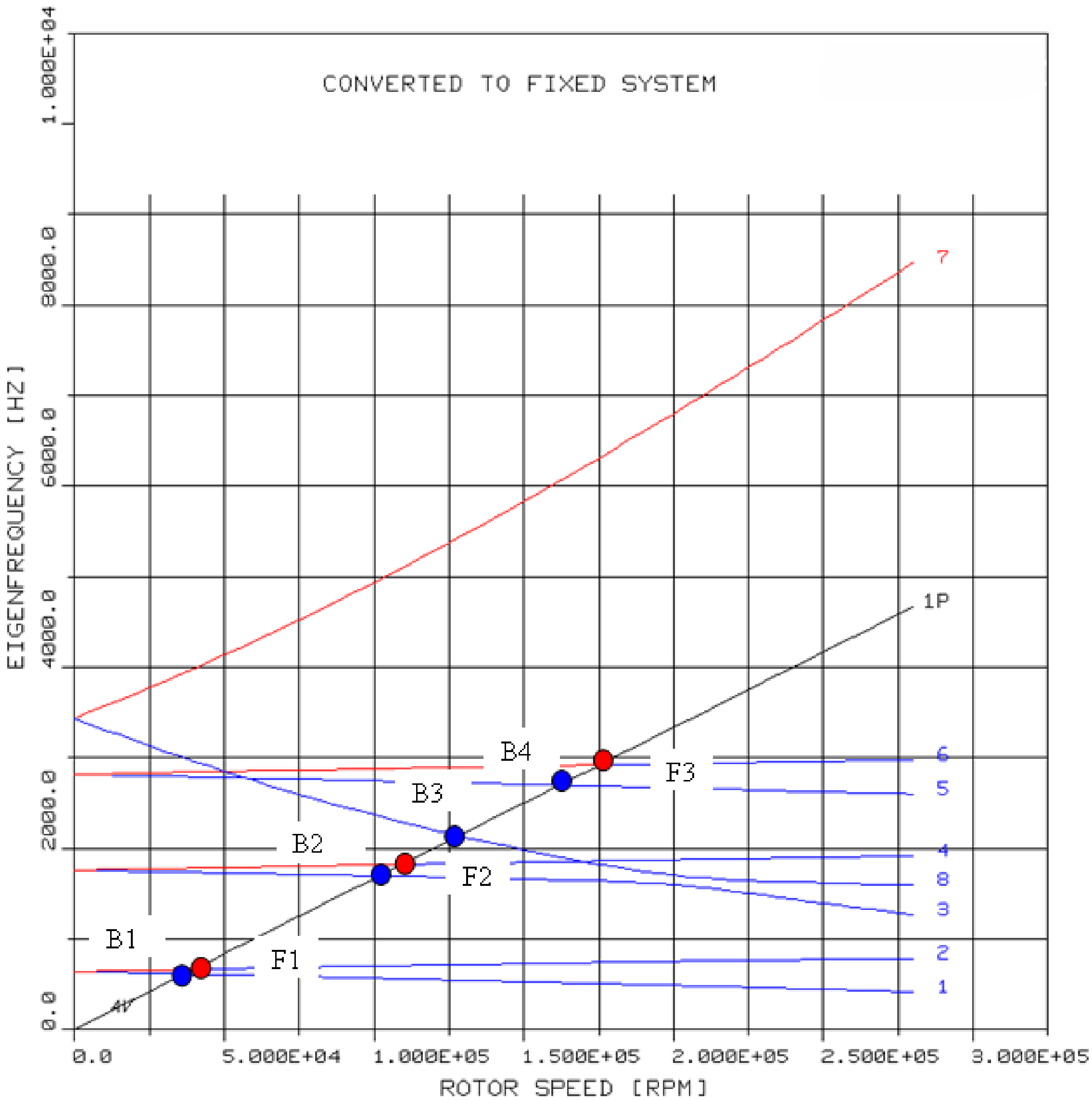


Fig. 81 Critical speeds for forward (red) and backward (blue) whirl calculated in the rotating system and converted to the fixed reference system.

```
0
            D E T E C T I O N   O F   R E S O N A N C E S   A N D   I N S T A B I L I T I E S

                          A N A L Y S I S   I N   R O T A T I N G   S Y S T E M



 FORWARD WHIRL RESONANCE         SOLUTION      ROTOR SPEED     WHIRL
                                  NUMBER          RPM        DIRECTION

                                       2      3.97729E+04    FORWARD
                                       4      1.09333E+05    FORWARD
                                       6      1.75299E+05    FORWARD


 BACKWARD WHIRL RESONANCE        SOLUTION      ROTOR SPEED     WHIRL
                                  NUMBER          RPM        DIRECTION

                                       1      3.67319E+04    BACKWARD
                                       3      1.01659E+05    BACKWARD
                                       5      1.61801E+05    BACKWARD
                                       8      1.28376E+05    BACKWARD


 INSTABILITIES                   SOLUTION      ROTOR SPEED     WHIRL
                                  NUMBER          RPM        DIRECTION

 START                                 2      4.40015E+04    FORWARD
 START                                 4      1.73213E+05    FORWARD
```

Table 41 Resonance points calculated by SOL 110

7.2.2 Asynchronous Analysis

The asynchronous analysis is done for one rotor speed and for different excitations frequencies defined on the FREQ1 entry. That means a vertical line in the Campbell diagram is analyzed as shown in Fig. 82 for the backward whirl resonances. A response curve is shown in Fig. 83. The curves for the forward whirl are shown in Fig. 84 and Fig. 85, respectively.

The response peak B3 in Fig. 82 is slightly below the critical speed B2 for synchronous option in Fig. 78. Hence the peak B2 in Fig. 80 must be slightly higher than the peak B3 in Fig. 83 which is also the case.

The response analysis is done in the rotating reference system. The frequencies are $\pm\Omega$ apart from those in the non-rotating Campbell diagram.

The damping of modes is shown in Fig. 86. Solution number 2 gets unstable above 44000 RPM. In a transient analysis, this would show up as a diverging solution. The real eigenvalues are shown in Fig. 87.

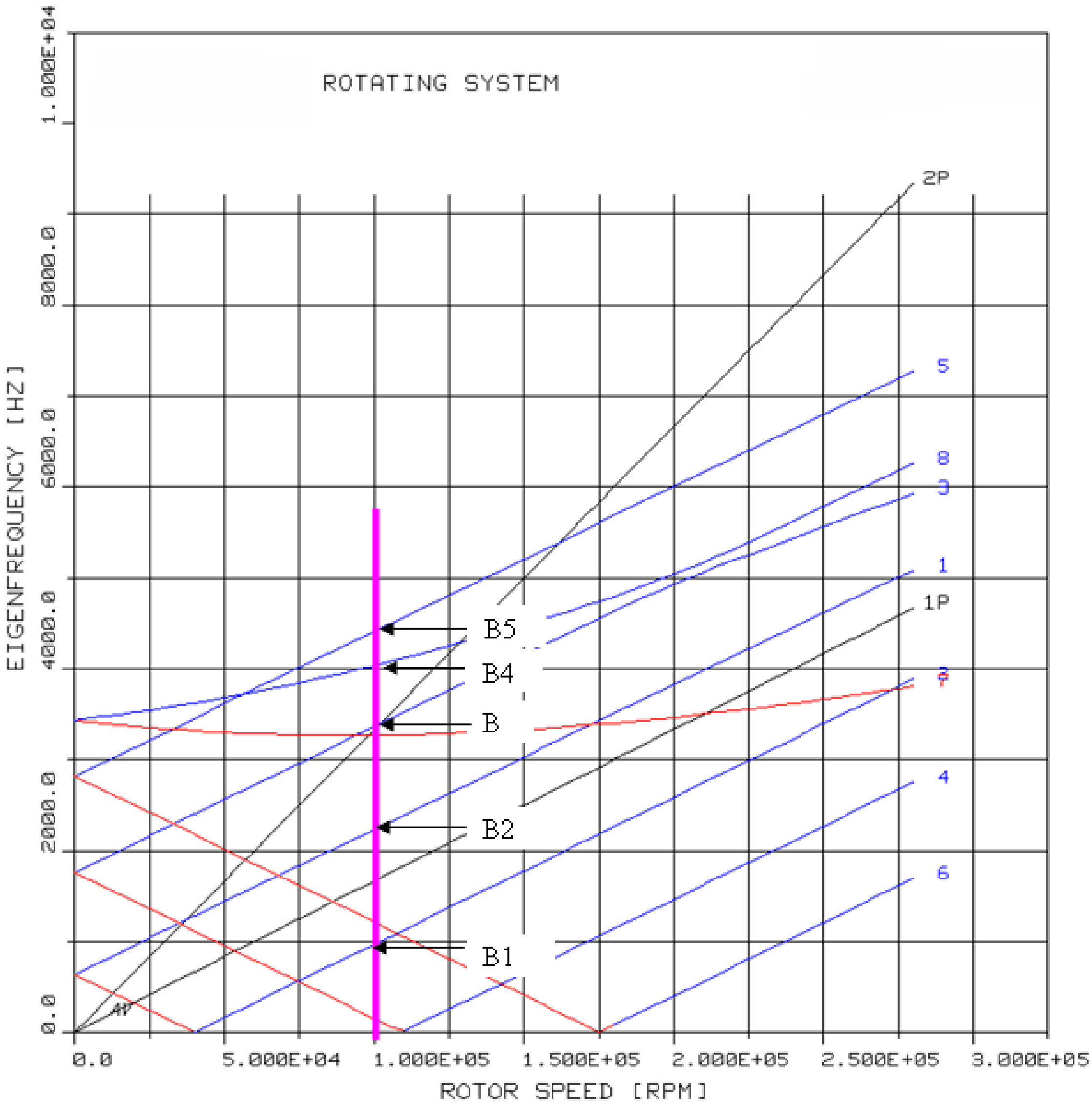


Fig. 82 Asynchronous analysis: Backward whirl resonances in the rotating reference system. Crossing with blue lines.

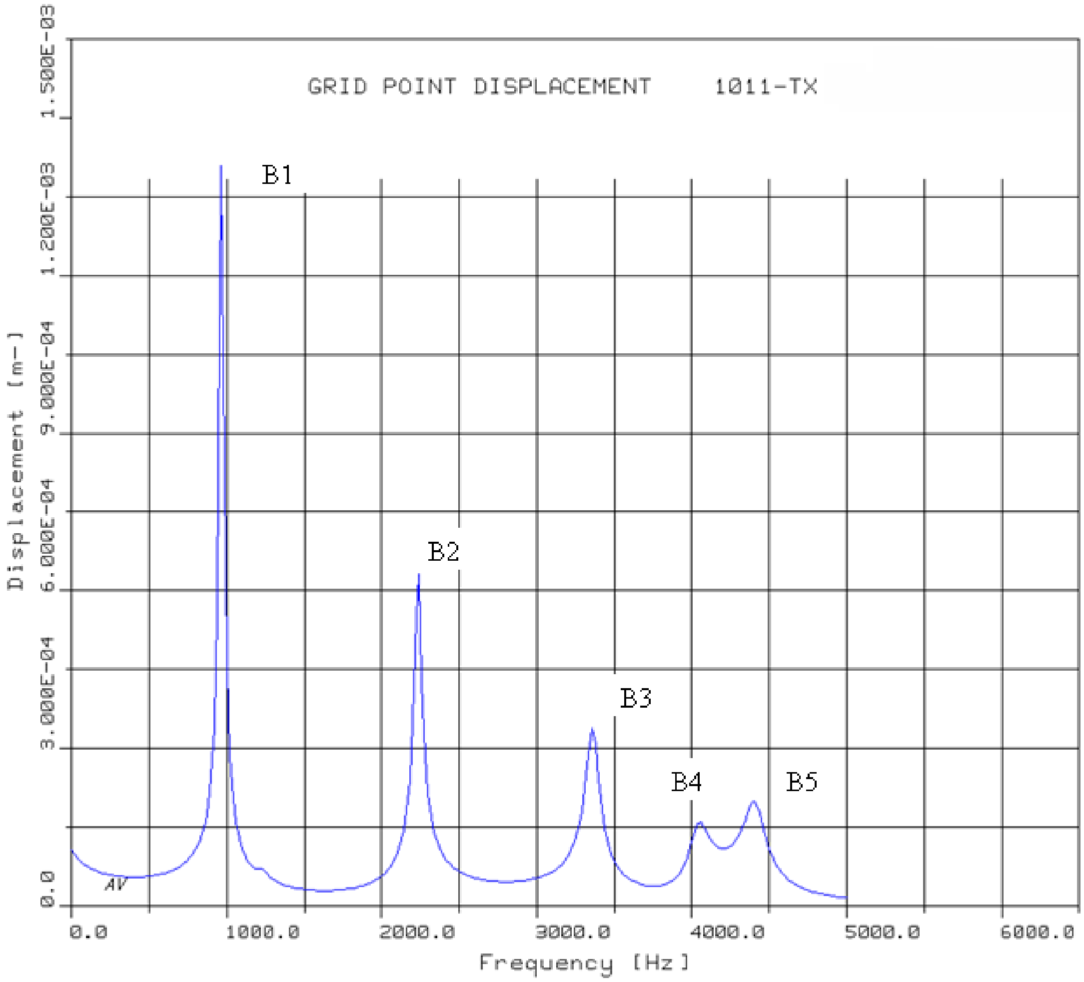


Fig. 83 Backward whirl response at 100000 RPM

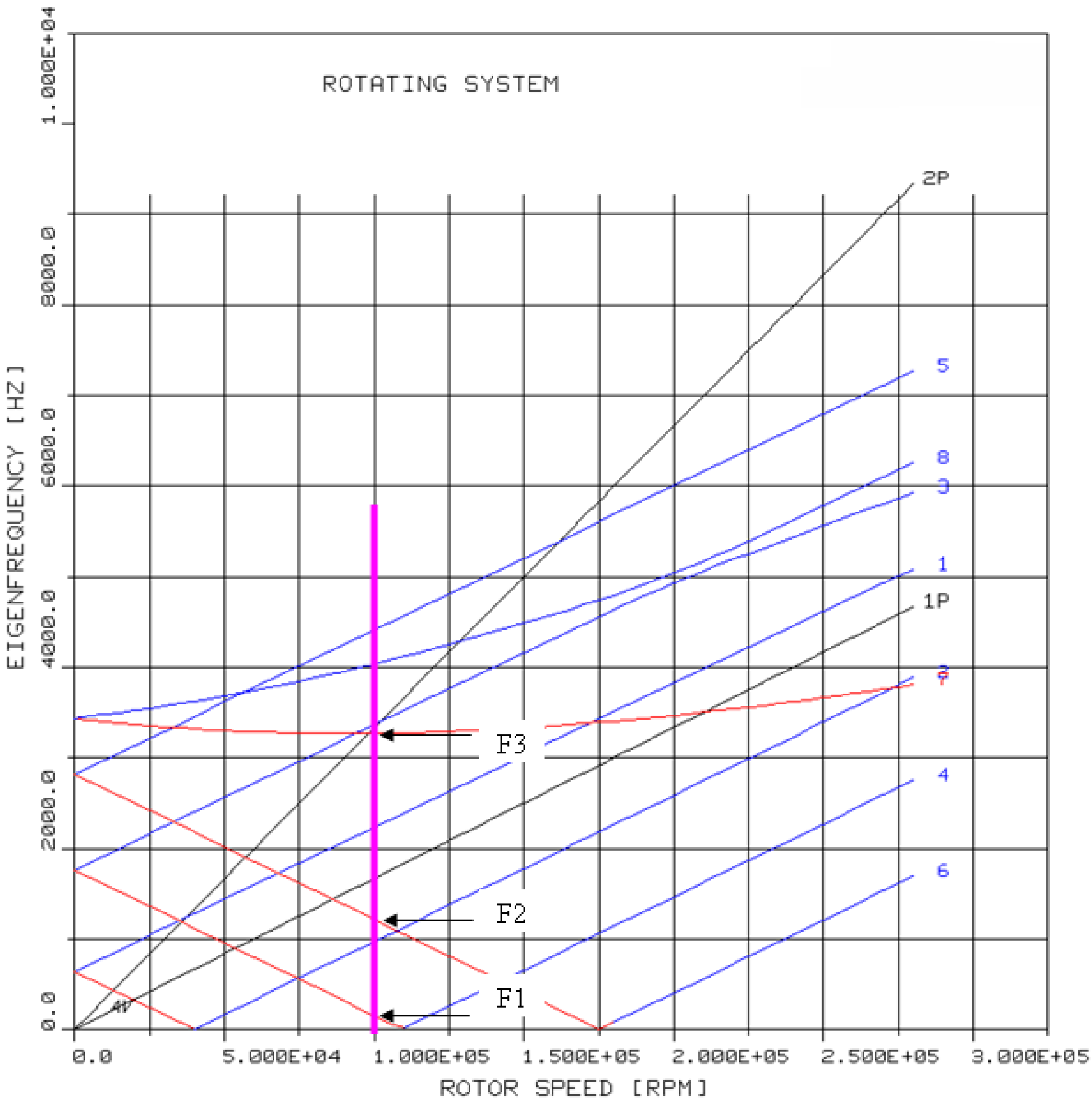


Fig. 84 Asynchronous analysis: Forward whirl resonances in the rotating reference system. Crossing with red lines.

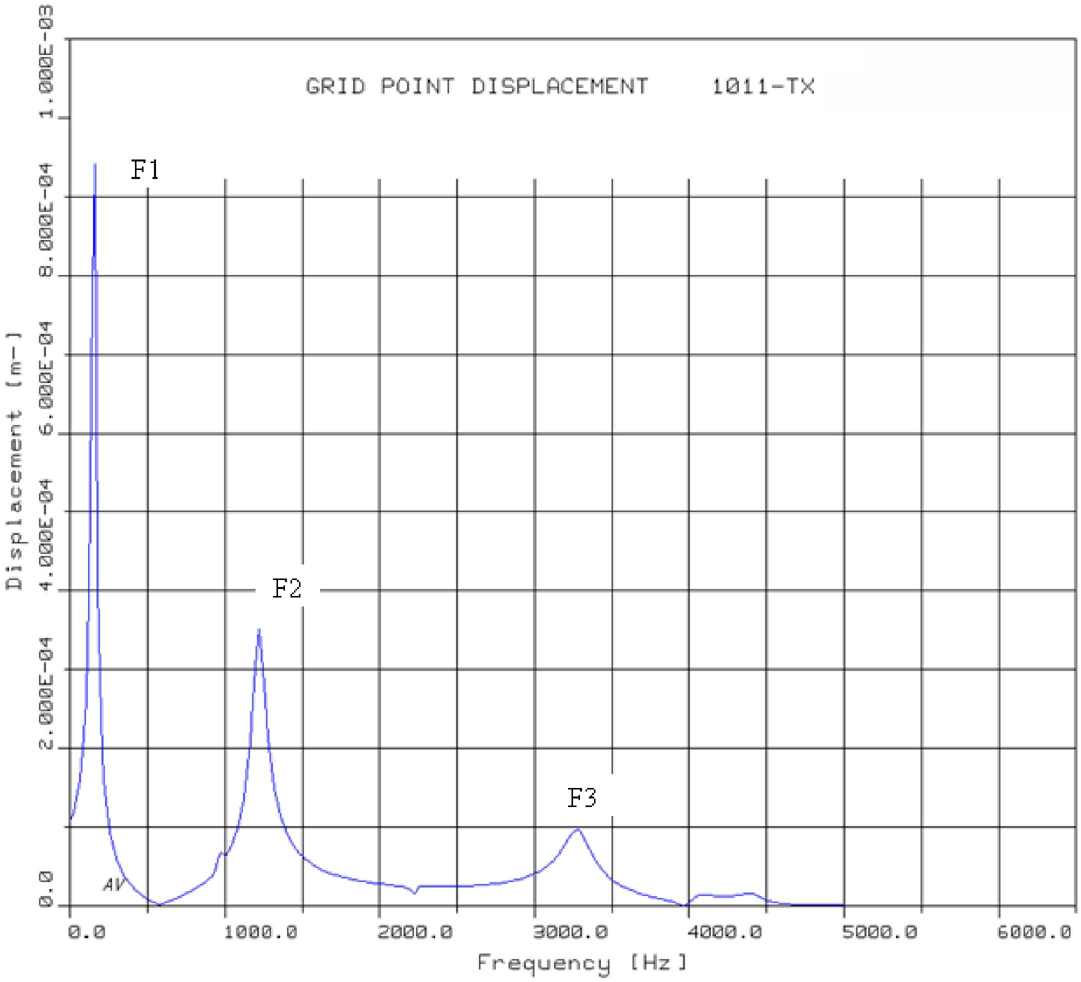


Fig. 85 Forward whirl response at 100000 RPM

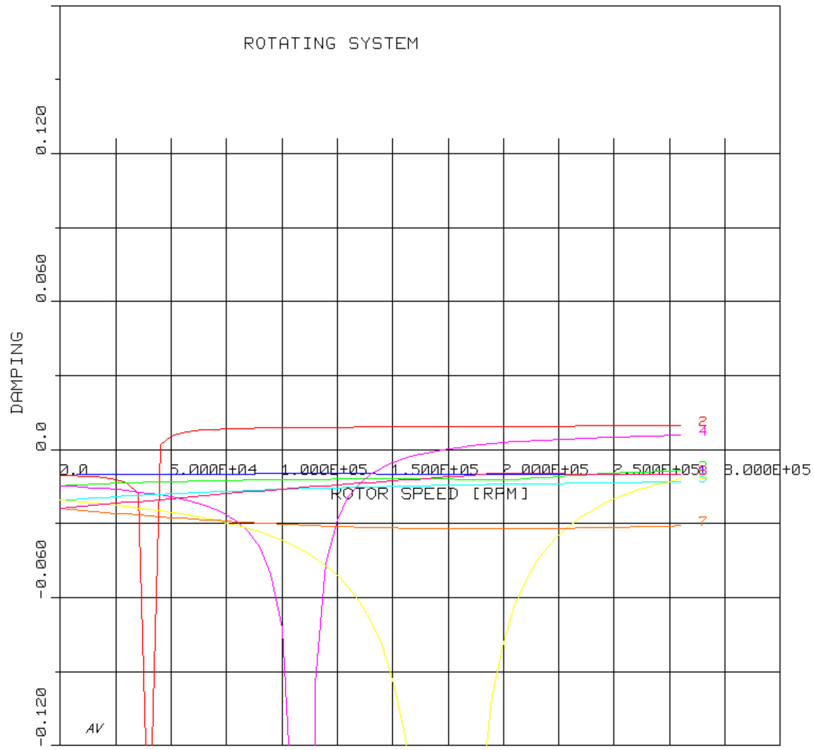


Fig. 86 Damping of modes

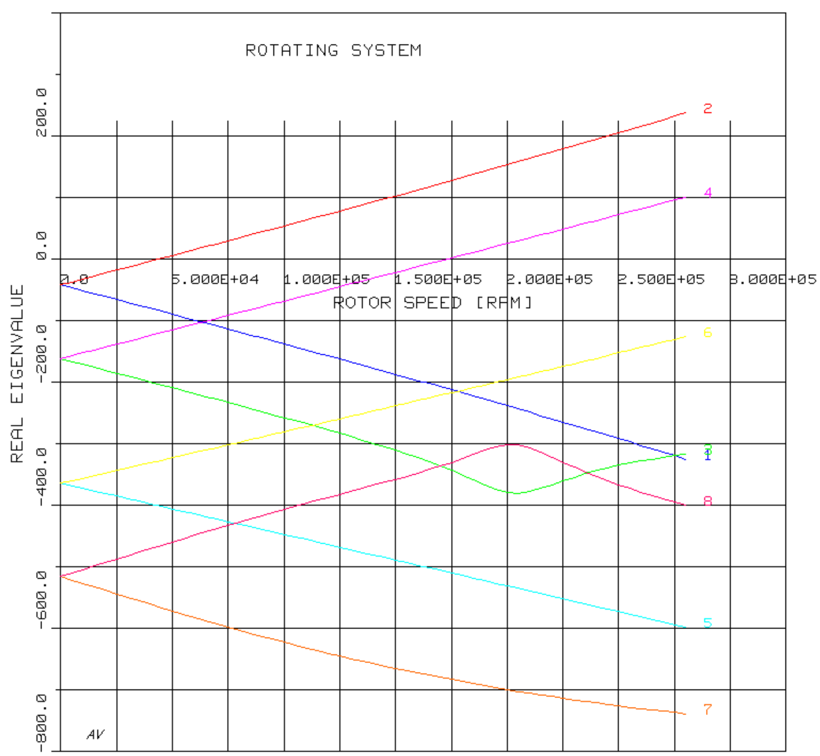


Fig. 87 Real eigenvalues

CHAPTER

8

Rotor Dynamics Examples Transient Response

8 Transient Response Examples

The model rotor067.dat is used as an example. It is a rotating cylinder idealized with bar elements. Asynchronous and synchronous cases were tested and are discussed in the following sections.

8.1 Asynchronous Analysis

The asynchronous analysis is done for a fixed rotor speed and with a linearly varying excitation function. The rotor dynamic input file is shown in Table 42. The time function is defined in the include file sincos-500.dat which has 50,000 values. A part of the file is shown in Table 43. The curve with ID 121 is the sine function and the curve 122 is the cosine function. The excitation is a forward whirl motion with the sine component in the x-direction and the cosine component in the y-direction. The frequency is linearly varying from 0 to 500 Hz. On the ROTORD entry, a rotor speed of 300 Hz is defined and the SYNC flag is set to zero. Because ETYPE=1, the excitation force is obtained by defining a mass unbalance, which the software multiplies by Ω^2 in order to obtain the force. The EORDER is set to 1.0.

```
$       SID     RSTART  RSTEP   NUMSTEP REFSYS  CMOUT   RUNIT   FUNIT
ROTORD  99      300.0   5.      1       FIX     -1.0    HZ      HZ      +ROT0
$       ZSTEIN  ORBEPS  ROTPRT  SYNC    ETYPE   EORDER
+ROT0   NO      1.0E-6  3       0       1       1.0                     +ROT1
$       RID1    RSET1   RSPEED1 RCORD1  W3-1    W4-1    RFORCE1
+ROT1   1       11      1.0     1       400.0   400.0
$
ROTORG  11      1001    THRU    1899
$
$       sid     s       s1      rload           rload
DLOAD   100     1.0-3   1.0     101     1.0     102
$
$               darea   delay   type    tabled
TLOAD1  101     131             0       121
TLOAD1  102     132             0       122
$
$       sid     grid    dof
DAREA   131     1899    1       1.0
DAREA   132     1899    2       1.0
$
TSTEP   201     50000   0.0002
$
include 'sincos-500.dat'
$
include 'mod-067a.dat'
$
```

Table 42 Input data for asynchronous rotor dynamic analysis for a fixed rotor speed of 300 Hz using 50000 time steps of 0.0002 seconds.

```
$       0 to 500 Hz in 10 seconds
$       sinus
tabled1 121                                                             +tbl1000
+tbl1000 0.      0.      2.000-4 6.283-6 4.000-4 2.513-5 6.000-4 5.655-5+tbl1001
+tbl1001 8.000-4 1.005-4 1.000-3 1.571-4 1.200-3 2.262-4 1.400-3 3.079-4+tbl1002
$ etc
+tb13500endt
$       cosinus
tabled1 122                                                             +tbl5000
+tbl5000 0.      1.      2.000-4 1.      4.000-4 1.      6.000-4 1.     +tbl5001
$ etc.
+tb17499 9.9992 -0.80947 9.9994 -0.31012 9.9996  0.30754 9.9998  0.80845+tb17500
+tb17500endt
```

Table 43 Part of the include file with the excitation functions

The displacement results from the analysis are shown in Fig. 88 for when the excitation is increased from 0 to 500 Hz in 10 seconds. There is a clear resonance peak at 1 second which is equivalent to 50 Hz. This is the forward whirl resonance of the translation. Fig. 89 shows the same item but now the excitation is increased from 0 to 500 Hz in one second. The maximum amplitudes now occur after 0.1 second and the rotor is not really in resonance because the structure does not have enough time to respond and the resonance point is quickly passed. The maximum amplitude in the slowly increasing case is 1.5 mm and in the fast case only 0.5 mm. The result of a frequency response analysis is shown in Fig. 90. The magnitude of the response is around 5.1 mm. In this case, the structure is in equilibrium in resonance. This amplitude is the maximum amplitude obtained for a very slow sweep through a 50 Hz excitation. A transient response with a constant excitation frequency of 51.08 Hz (see Table 44) is shown in Fig. 91. This corresponds to the critical speed for translation. The maximum amplitude of 5.07 mm is reached after approximately 2 seconds when starting from initial conditions of zero. This means that the structure needs 2 seconds to reach steady-state condition at this resonance point. Plots for the acceleration are shown in Fig. 92 and Fig. 93, respectively. The time function of the tilting rotation of the shaft is shown in Fig. 94 together with the plot of the excitation frequency over time and the Campbell diagram. The maximum amplitudes are reached at around 6.4 second. This corresponds to a 320 Hz excitation frequency. In the Campbell diagram the resonance frequency is around 320 Hz for a rotor speed of 300 Hz.

```
TLOAD2  101     131             0       0.      10.     51.081  0.0
TLOAD2  102     132             0       0.      10.     51.081  90.0
```

Table 44 TLOAD2 entries for constant frequency excitation

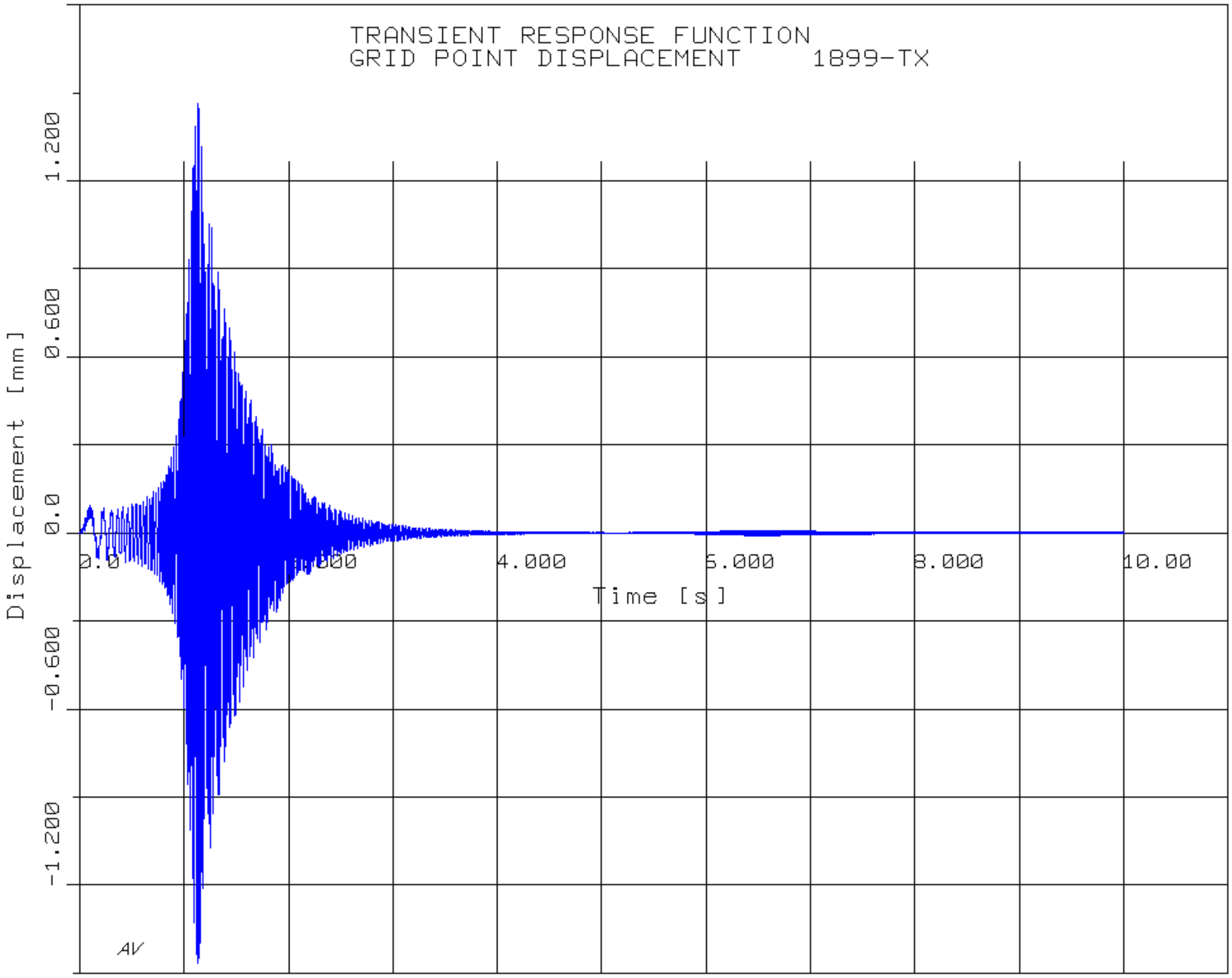


Fig. 88 Displacement response of the translation when the excitation frequency is increasing from 0 to 500 Hz in 10 seconds and passing through the critical speed

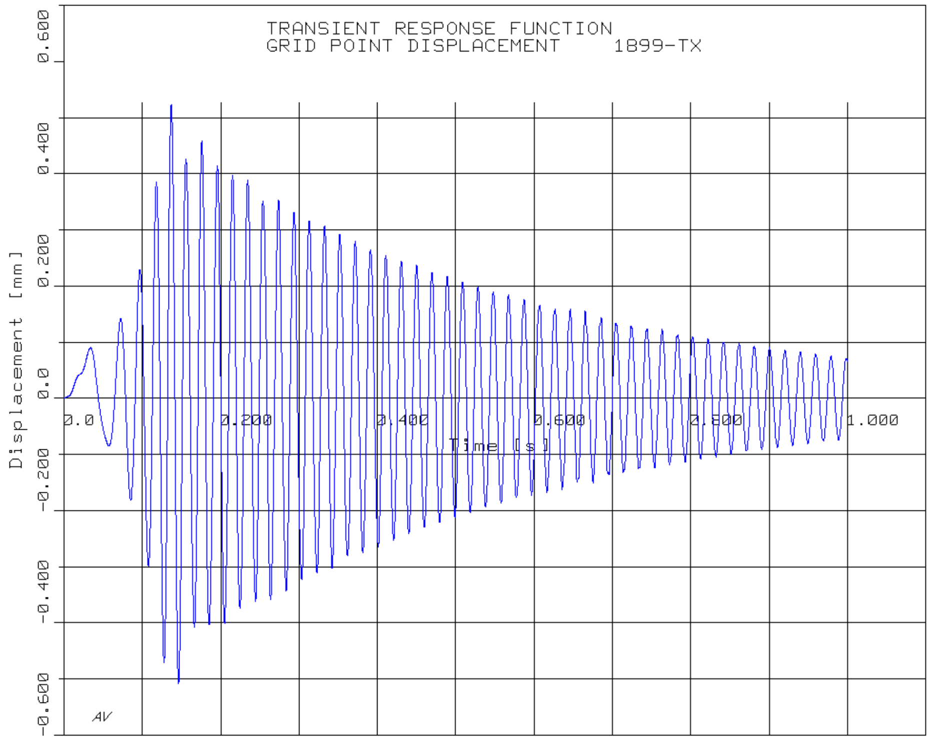


Fig. 89 Displacement response of the translation when the excitation is accelerating from 0 to 500 Hz in 1 second and passing through the critical speed

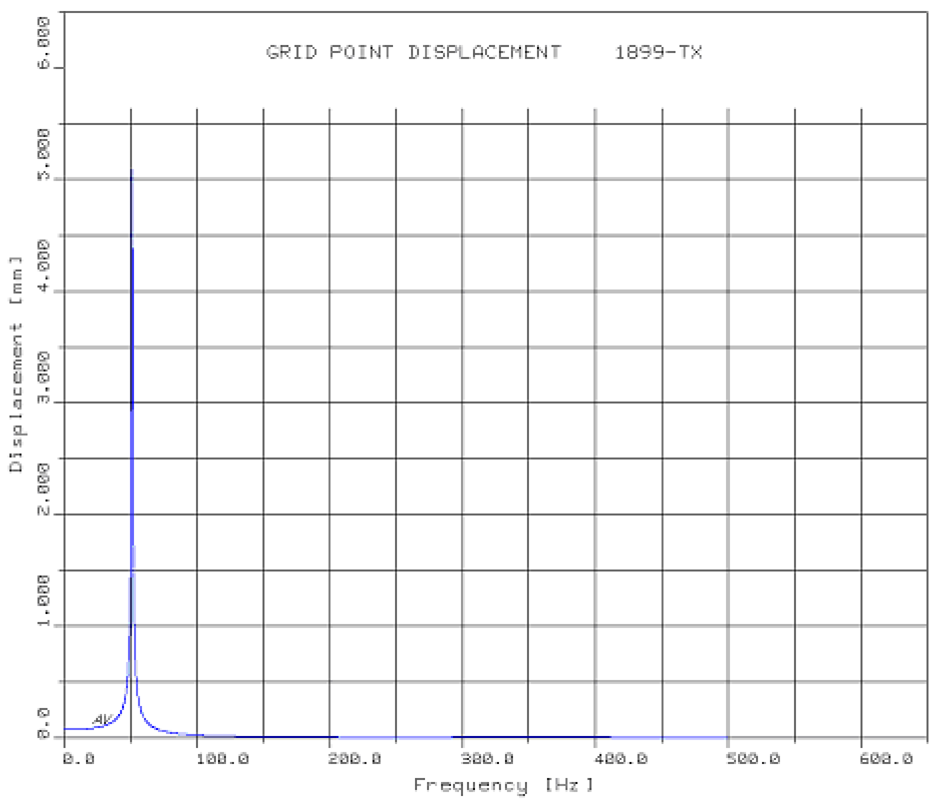


Fig. 90 Magnitude of the displacement from the frequency response analysis

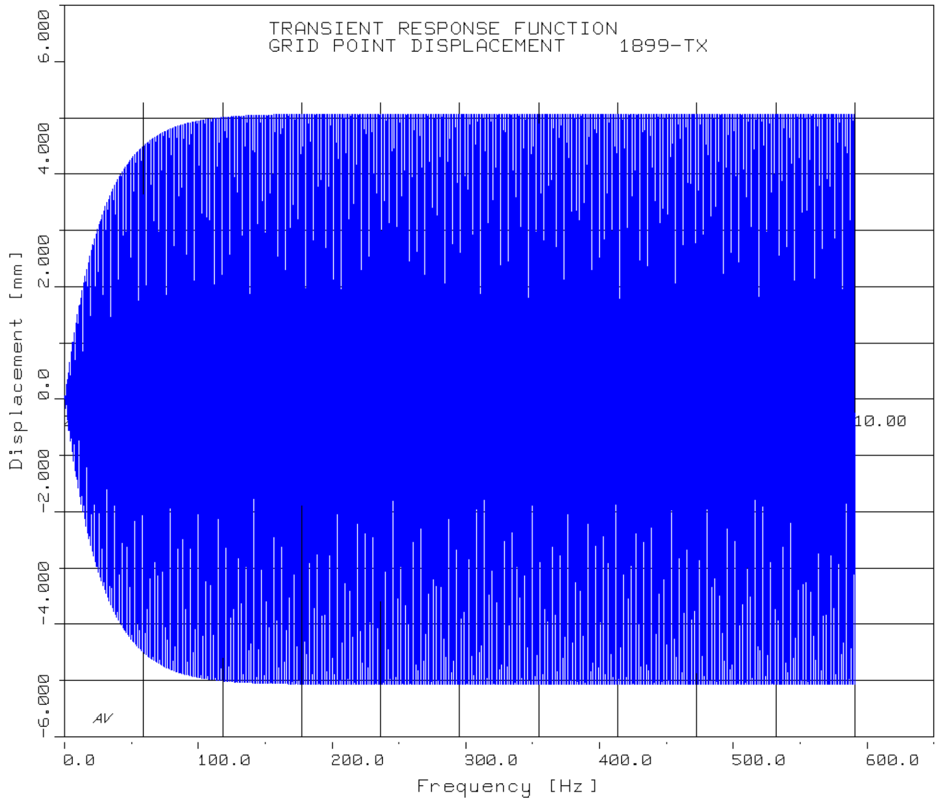


Fig. 91 Transient analysis with 51.08 excitation frequency

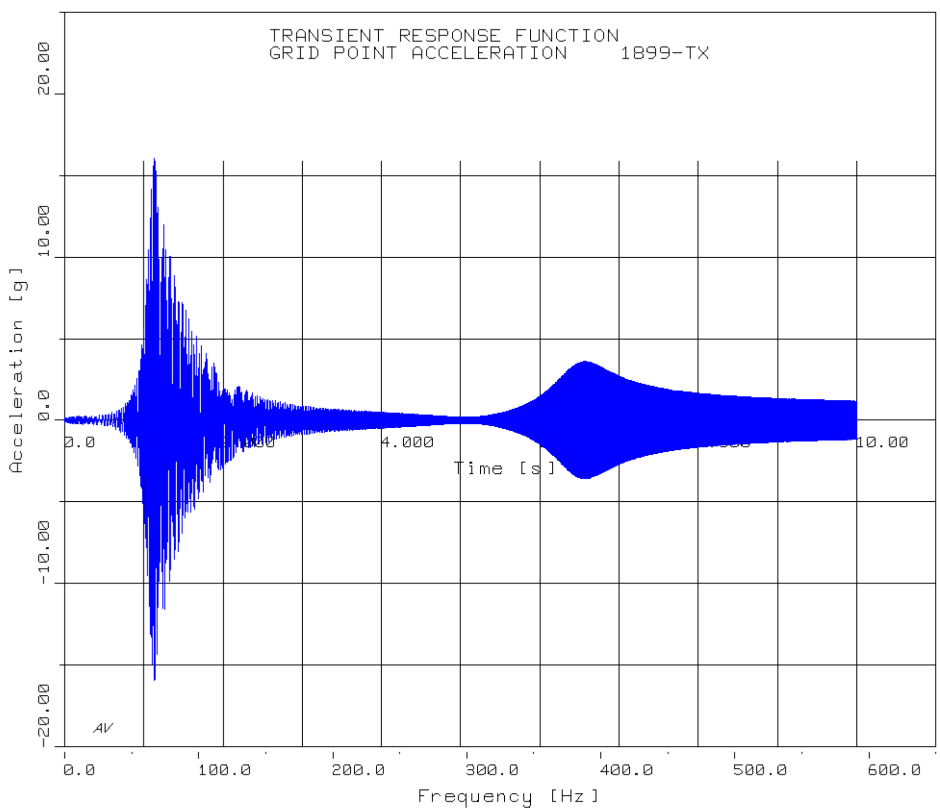


Fig. 92 Acceleration response for the slow excitation frequency case

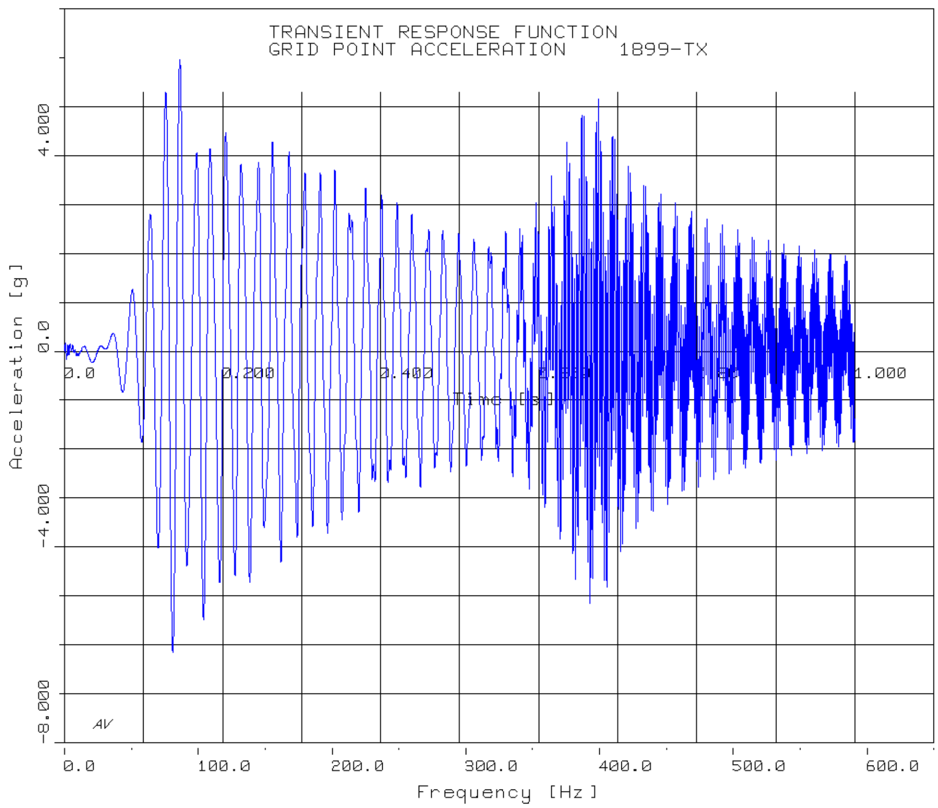


Fig. 93 Acceleration response for the fast excitation frequency case

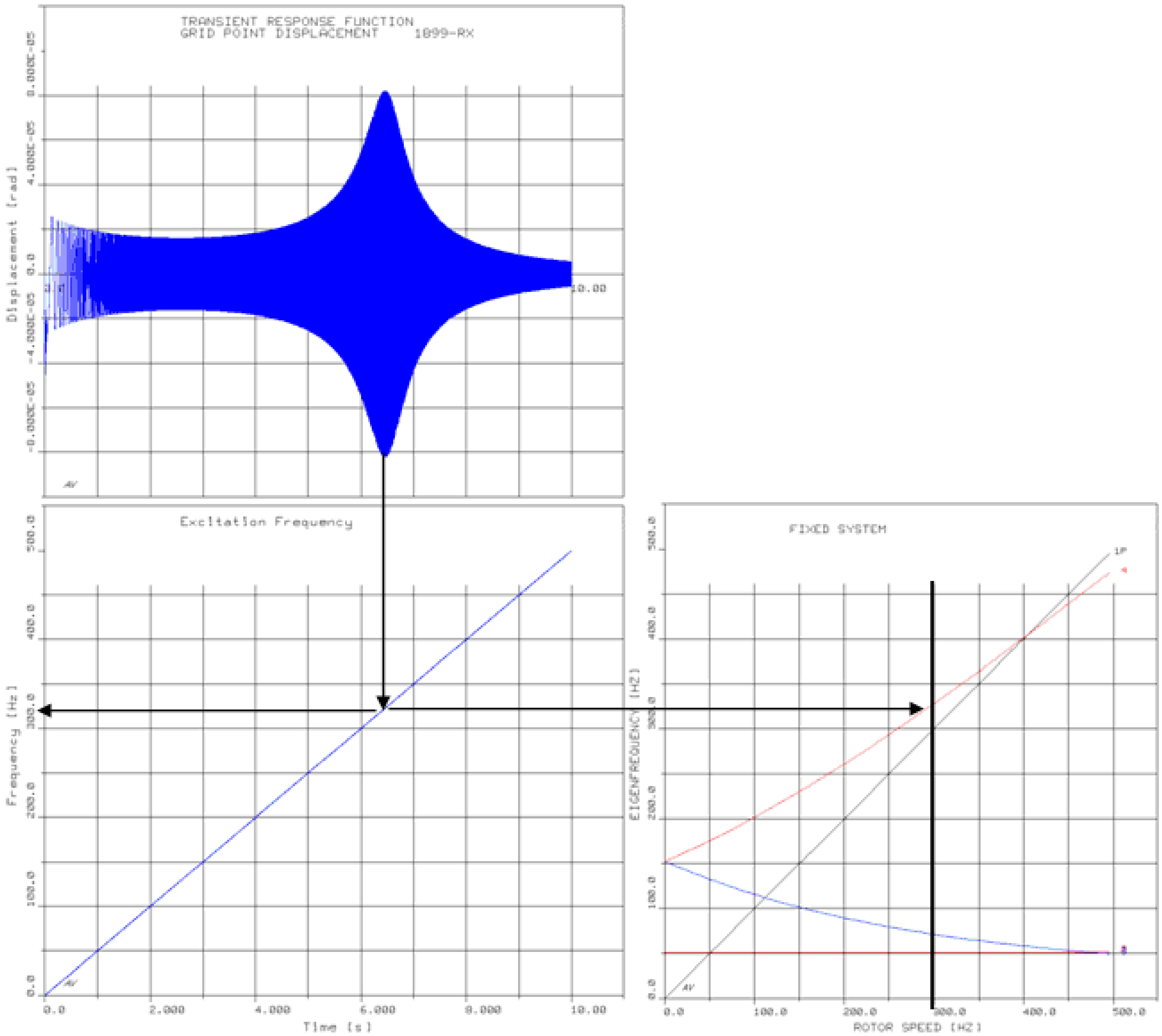


Fig. 94 Response of the tilting motion

8.2 Synchronous Analysis

The number of time steps on the TSTEP entry must be equal to NUMSTEP on the ROTORD entry. The rotor speed must be defined as the range up to 500 Hz. It is not possible to start at zero rotor speed because then the excitation force will be zero and the program will stop. Now, the SYNC flag must be set to one. The input is shown in Table 45.

```
$
$       SID     RSTART  RSTEP   NUMSTEP REFSYS  CMOUT   RUNIT   FUNIT
ROTORD  99      0.01    0.01    50000   FIX     -1.0    HZ      HZ      +ROT0
$       ZSTEIN  ORBEPS  ROTPRT  SYNC    ETYPE   EORDER
+ROT0   NO      1.0E-6  3       1       1       1.0                     +ROT1
$       RID1    RSET1   RSPEED1 RCORD1  W3-1    W4-1    RFORCE1
+ROT1   1       11      1.0     1       400.0   400.0
$
TSTEP   201     50000   0.0002
$
```

Table 45 Input for synchronous analysis

The results of the synchronous analysis are shown in Fig. 95. The excitation is now along the 1P line in the Campbell diagram. Now, the rotor speed is equal (synchronous) to the excitation frequency and the simulation shows how the rotor behaves when the critical speed is passed. The crossing of the 1P line with the forward tilting mode is under a shallow angle. Therefore, the resonance peak is less pronounced as shown in Fig. 96. Here, ETYPE = 0 which means that the force is entered directly, and is not dependent on the rotor speed. Also a fast simulation was used (0 to 1000 Hz in 2 seconds). The occurrence of the peak is not exactly where it would be expected from the Campbell diagram. A simulation of 10 seconds is shown in Fig. 97. The main frequency is at 250 Hz. When this is multiplied with transfer function showing a strong resonance, mainly the resonance peak is seen. When the resonance peak is weaker, a combination of the input signal with peak at 250 Hz and the resonance at 400 is found. In Fig. 97 the resulting peak of the output signal is around 350 Hz.

By changing the sign of the cosine part of excitation in the y-direction, a backward whirl excitation can be simulated as shown in Table 46. This may not be physically meaningful. The resonance of the tilting mode is shown in Fig. 98.

```
$       sid     grid    dof
DAREA   131     1899    1       1.0
$     backward
DAREA   132     1899    2       -1.0
```

Table 46 Backward whirl excitation

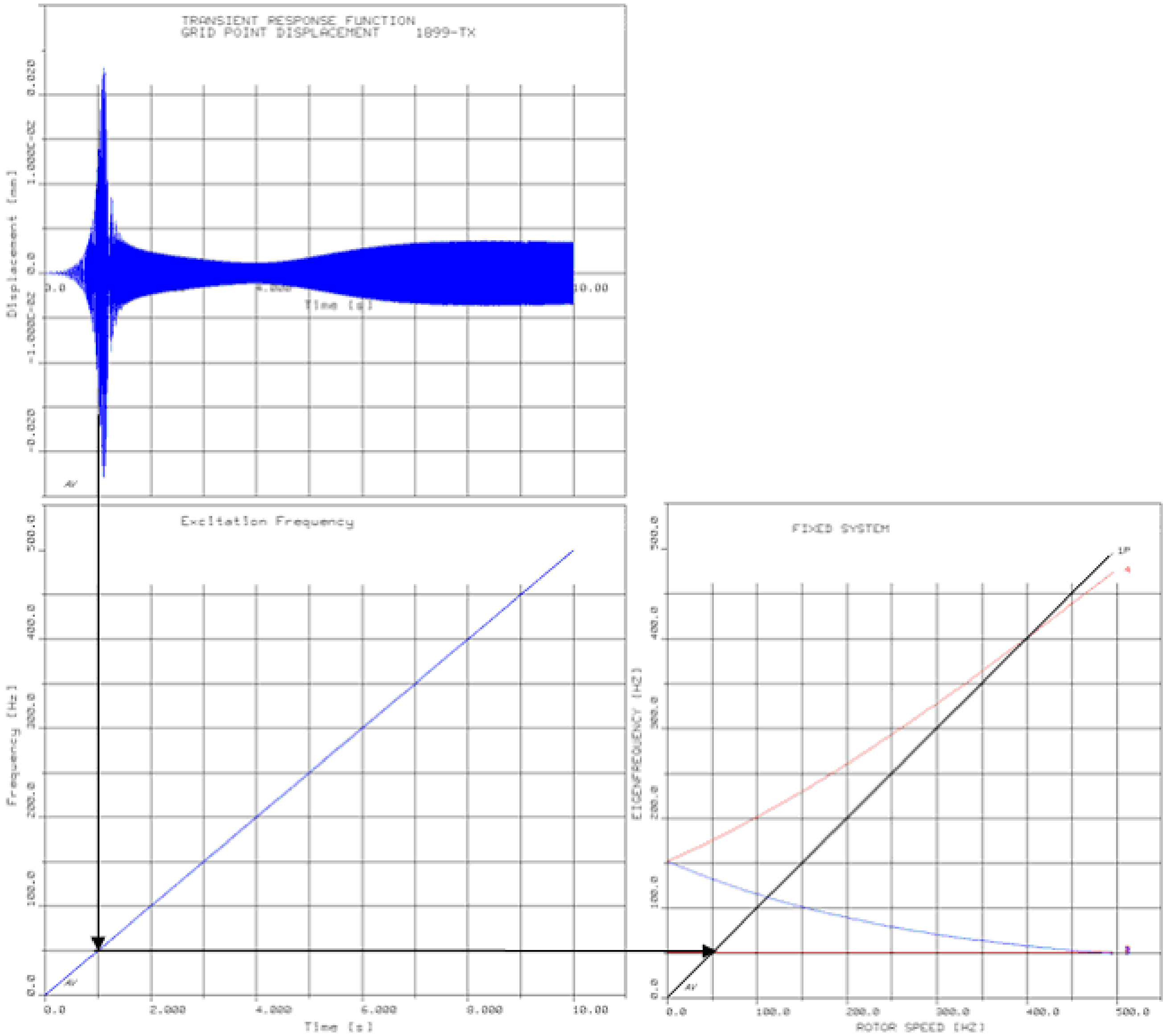


Fig. 95 Running through the translation peak at around 50 Hz

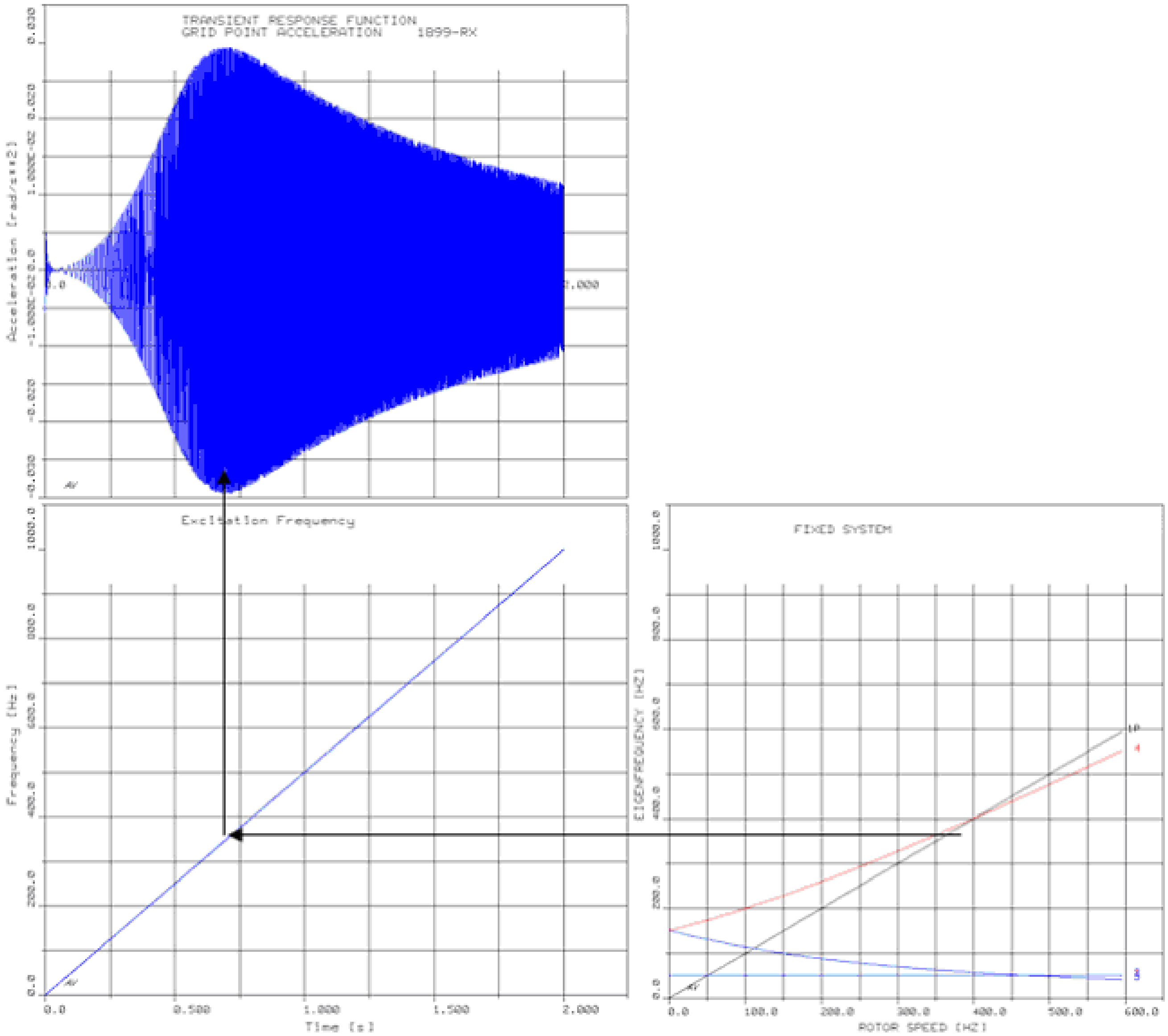


Fig. 96 ETYPE = 0, Running from 0 to 1000 Hz in 2 seconds

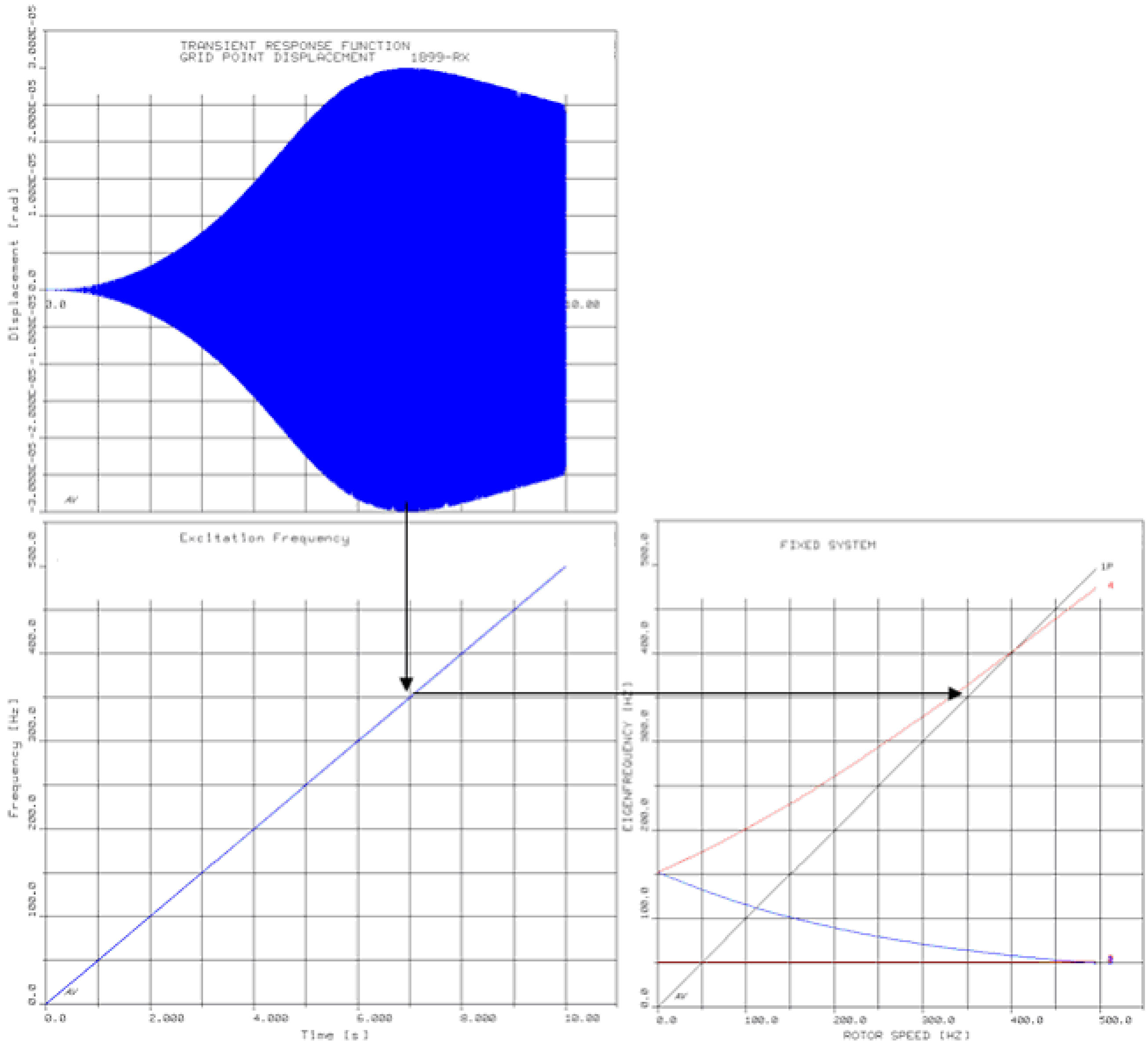


Fig. 97 Synchronous analysis with resonance of tilting mode

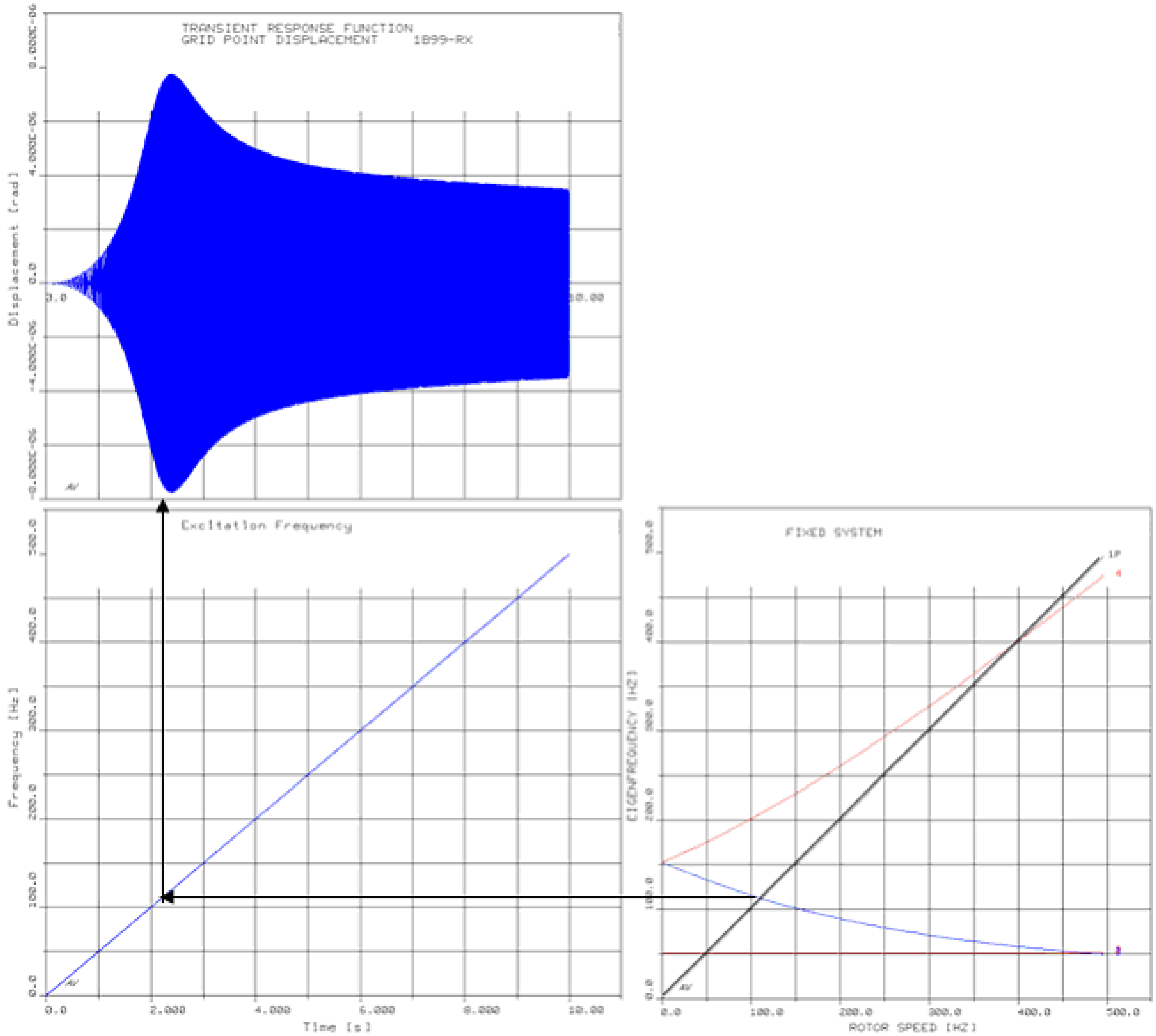


Fig. 98 Running through the backward tilting mode

CHAPTER

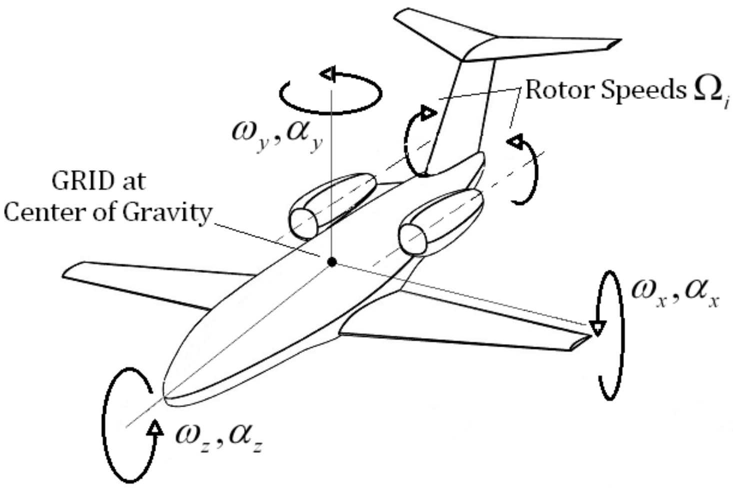
9

Example of a Maneuver Load Analysis

9 Maneuver Load Analysis Example

An application of gyroscopic forces is an aircraft maneuvering during flight. The aircraft rotates about the three axes passing through its center of gravity (CG). The example in the figure has a coordinate system located at the CG. The X, Y, and Z axis of this system are aligned such that the maneuvers loads, defined with RFORCE, RFORCE1, or RFORCE2 bulk entries are pitch (ω_x, α_x), yaw (ω_y, α_y), and roll (ω_z, α_z).

The multiple rotors in this example would each be defined with an individual ROTORD entry, each with unique speeds corresponding to the values specified in the RSTART and RSPEED fields.



The static forces acting on the model in this example include the following.

- Gyroscopic forces that result from the pitching and yawing motion. The gyroscopic forces act at the grid points that define the rotors.
- Damping forces that result from structural damping in the rotors. The damping forces act at the grid points that define the rotors.
- Inertia forces that result from the pitching, rolling, and yawing motion. These inertia force act at all grid points.
- Gravity force that act at all grid points.

```
RMETHOD=99
SUBCASE 1
   LOAD=11
BEGIN BULK
$            SID  RSTART   RSTEP NUMSTEP  REFSYS   CMOUT   RUNIT   FUNIT
ROTORD        99 10000.0     1.0       1     FIX      1.     RPM     HZ+
$         ZSTEIN  ORBEPS  ROTPRT
+             NO  1.0E-5       3                                       +
$            RID    RSET  RSPEED   RCORD      W3      W4  RFORCE  BRGSET
+              1       1     1.0       1                              1+
+              2       2     1.0       1                              2+
+              3       3     1.4       2                              3
$CG location
GRID, 999, 0, 36.42, 0.12, 12.3
$Unit Translational Acceleration Loads (g's)
GRAV, 1, 0, 386., 1., 0., 0.
GRAV, 2, 0, 386., 0., 1., 0.
GRAV, 3, 0, 386., 0., 0., 1.

$Unit Rotational Velocity Loads (rad/sec)
RFORCE, 11, 999, .159, 1., 0., 0.
RFORCE, 12, 999, .159, 0., 1., 0.
RFORCE, 13, 999, .159, 0., 0., 1.

$Unit Rotational Acceleration Loads (rad/sec/sec)
RFORCE, 21, 999, 0., 1., 0., 0., ,+
+, .159
RFORCE, 22, 999, 0., 0., 1., 0., ,+
+, .159
RFORCE, 23, 999, 0., 0., 0., 1., ,+
+, .159

$Combined and scaled unit loads

LOAD, 11, 1., .7, 1, 1.1, 2, .6, 3,+
+, .14, 11, 4.2, 12, .17, 13, .36, 21,+
+, 6.4, 22, 1.7, 23
```

CHAPTER

10

Example of a Model with two Rotors analyzed with all Methods

10 Example of a Model with two Rotors analyzed with all Methods

The rotor dynamic options in NX Nastran 7 were extensively tested for a model with two rotors. The following items were studied:

1. Damping
2. Use of ROTORB entries (necessary for analysis in the rotating system)
3. W3R and W4R parameters
4. Relative rotor speed
5. Complex eigenvalues, modal method
6. Complex eigenvalues, direct method
7. Frequency response modal method
8. Frequency response direct method
9. Transient response modal method
10. Transient response direct method

The results were analyzed step by step and the solutions were all consistent.

10.1 Model

The model consists of two different rotors as shown in Fig. 99. The rotors are uncoupled. Therefore the results must be identical to analyses with the appropriate single models. Bearings are attached to both ends of the shafts and are modeled with CELAS1/PELAS for the stiffness and CDAMP1/PDAMP for the viscous damping. RBE2 elements connect the bearings to the shafts.

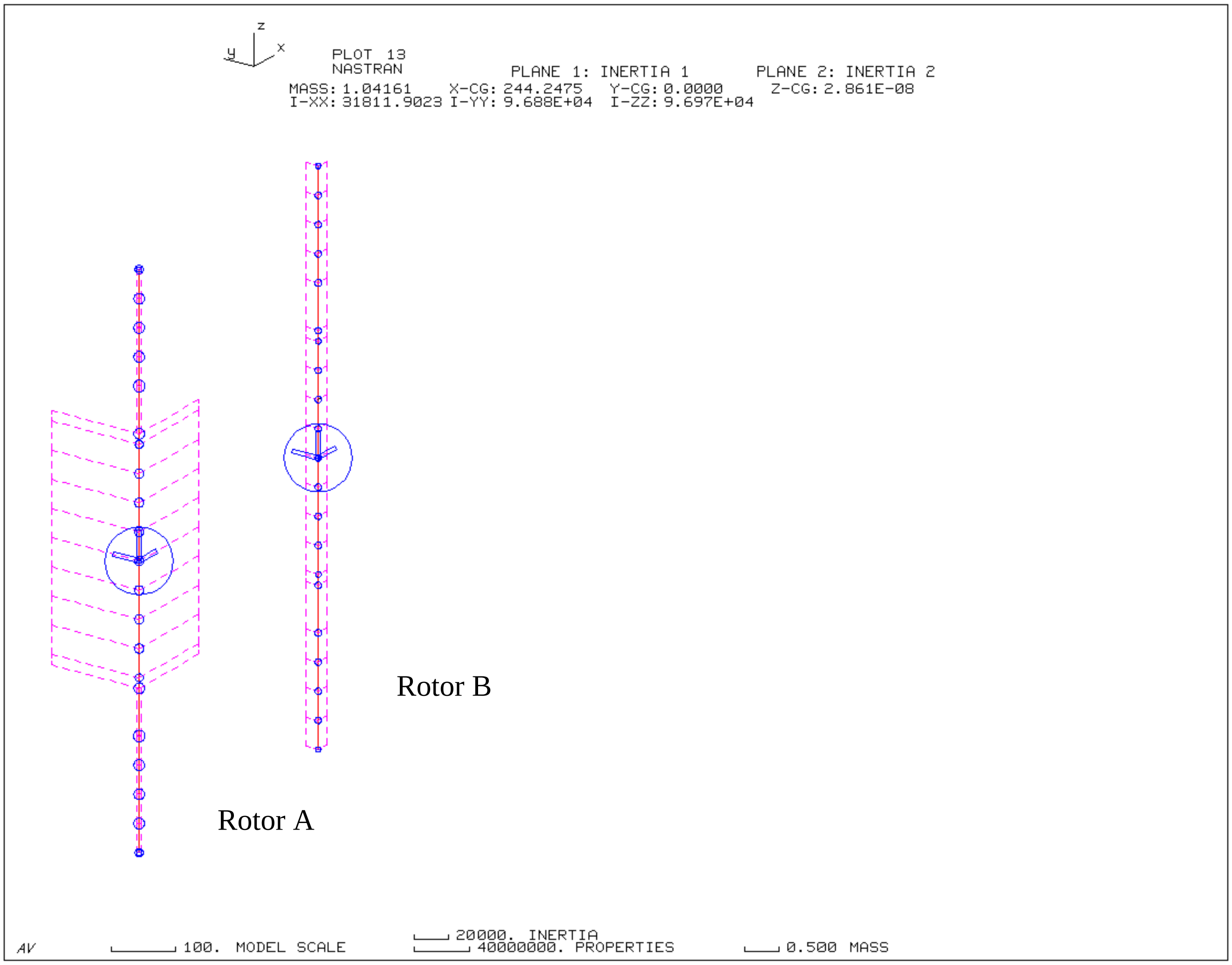


Fig. 99 Rotor model

10.2 Modes

The modes are listed in Table 47 and shown in Fig. 100 through Fig. 109. The shaft of rotor A is thinner than that of rotor B. Therefore, the shaft torsion and extension modes appear. They are not relevant for rotor dynamic analysis.

Mode	Eigenfrequency	Mode shape
1	34.06293	Shaft torsion, rotor A
2	43.55074	Translation y, rotor A
3	43.55074	Translation x, rotor A
4	57.26161	Translation y, rotor B
5	57.26161	Translation x, rotor B
6	77.77828	Tilt about x-axis, rotor B
7	77.77828	Tilt about y-axis, rotor B
8	92.30737	Tilt about x-axis, rotor A
9	92.30737	Tilt about y-axis, rotor A
10	391.6217	Shaft extension, rotor A

Table 47 Eigenfrequencies and modes

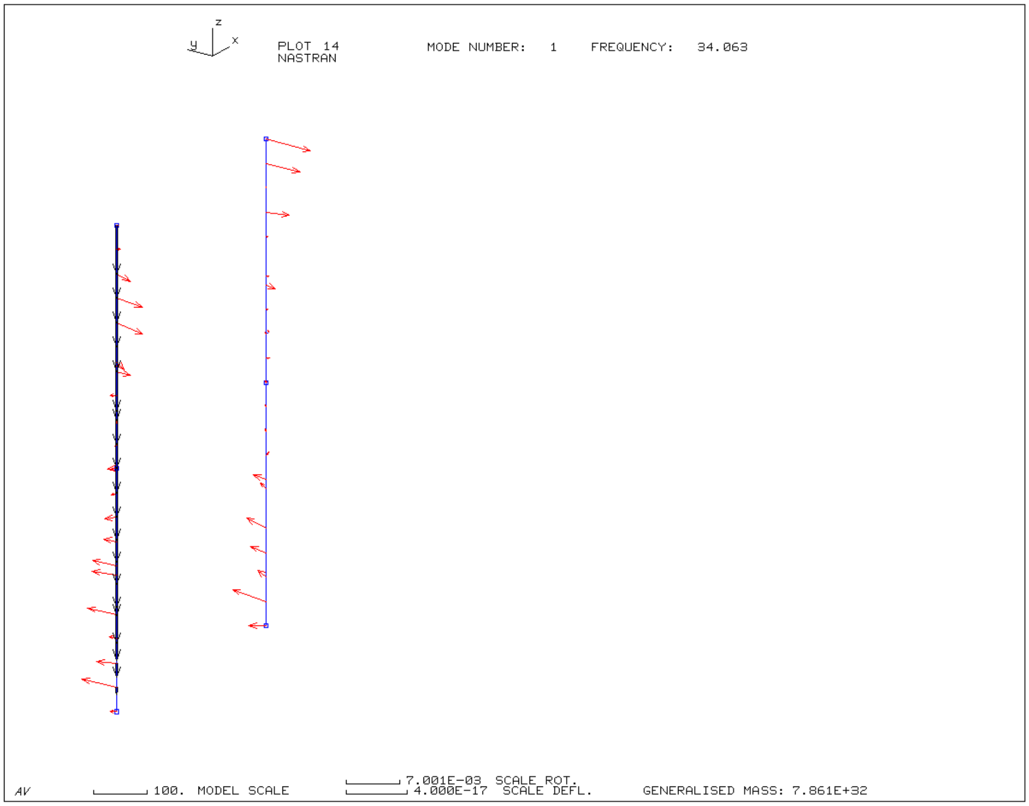


Fig. 100 Shaft torsion rotor A

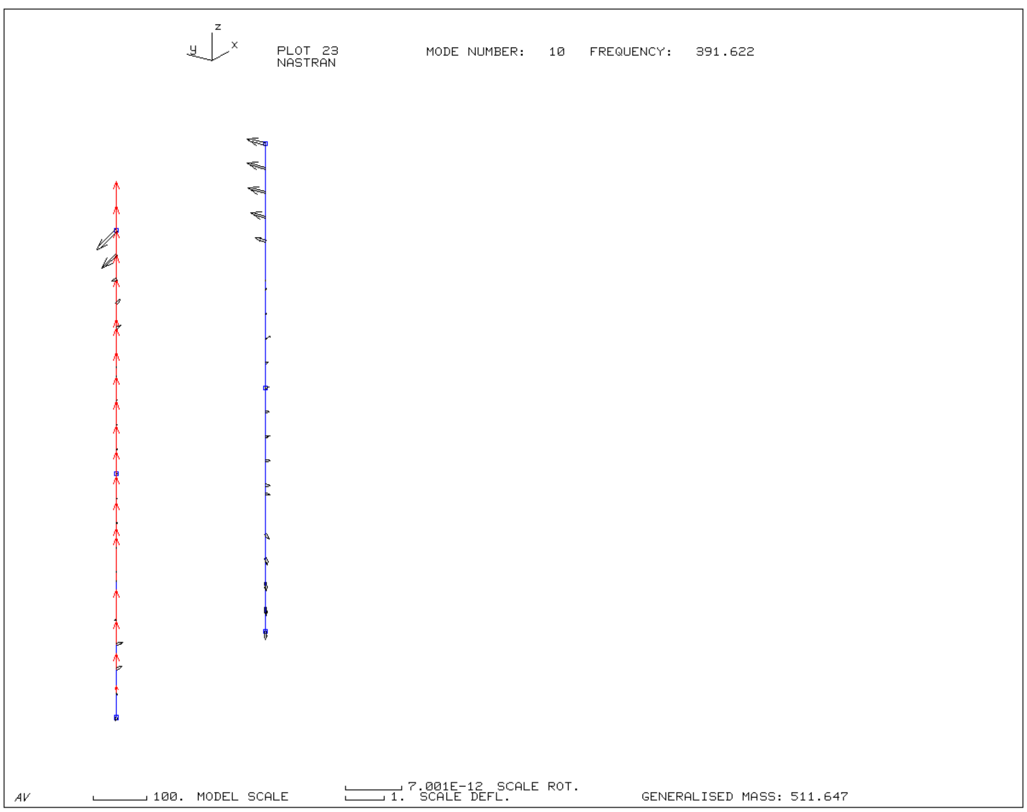


Fig. 101 Shaft extension rotor A

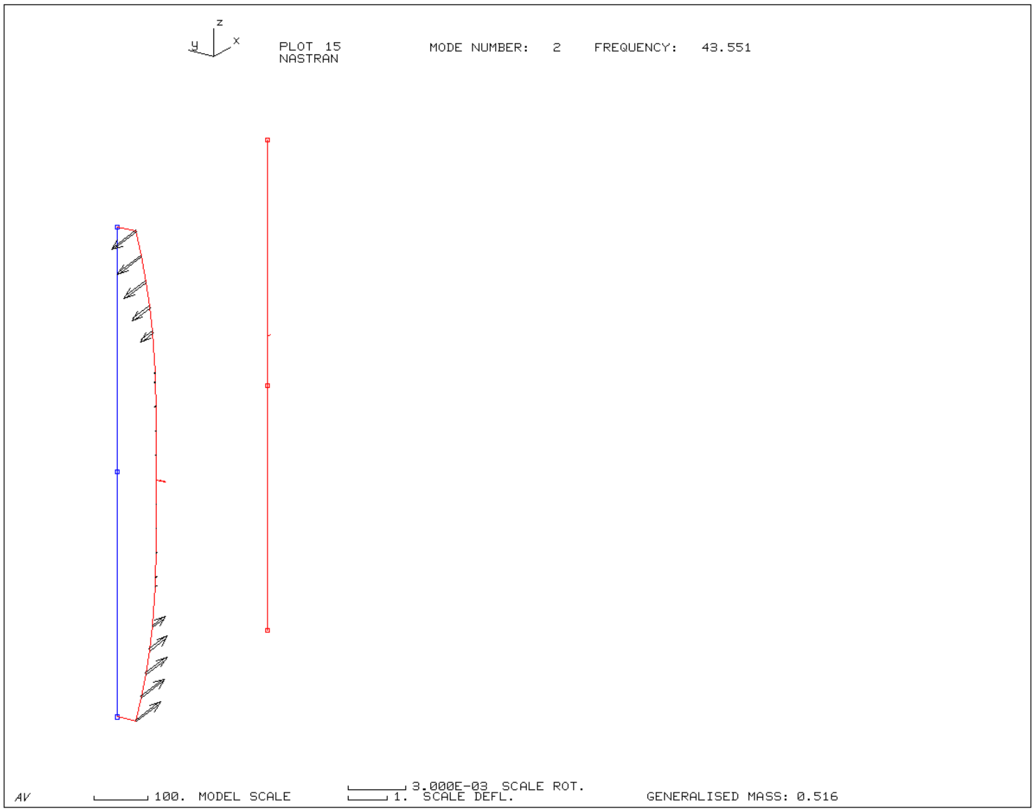


Fig. 102 Translation y, rotor A

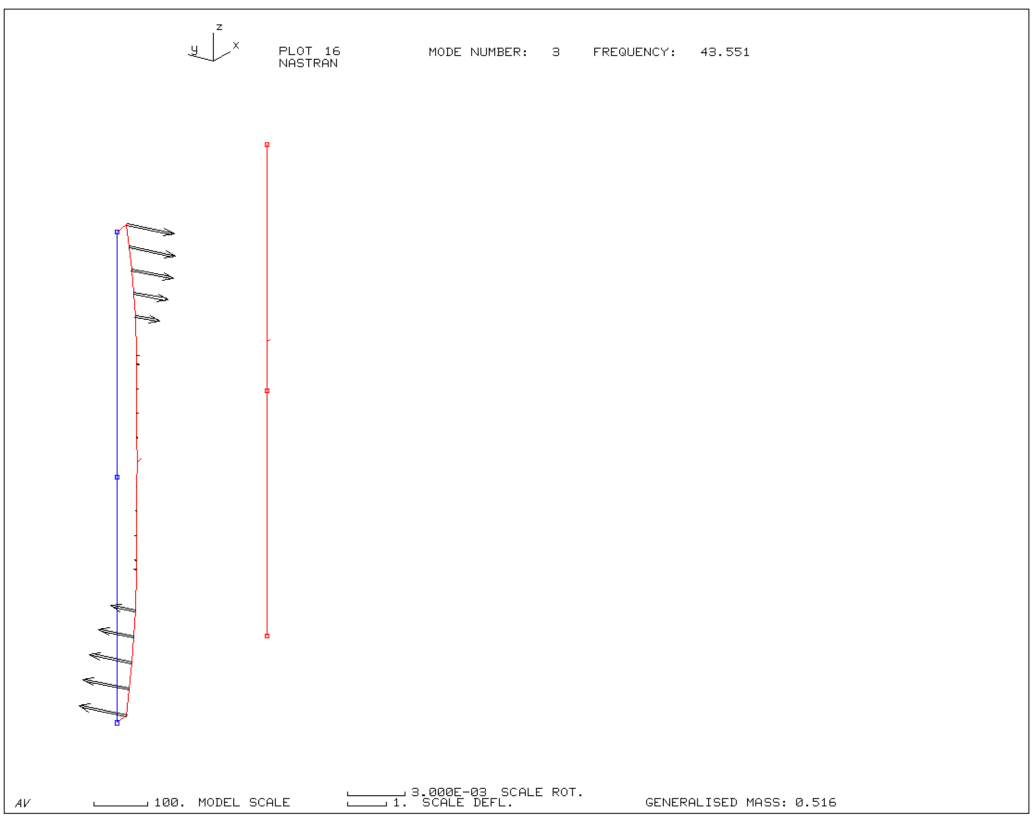


Fig. 103 Translation x, rotor A

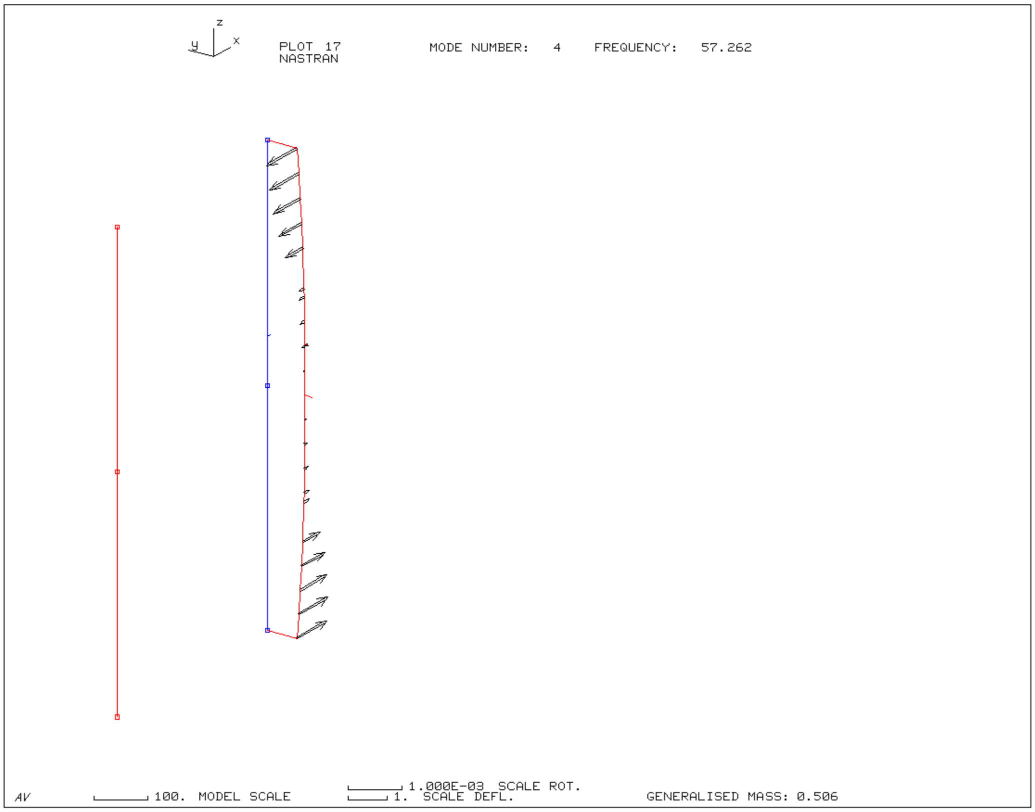


Fig. 104 Translation y, rotor B

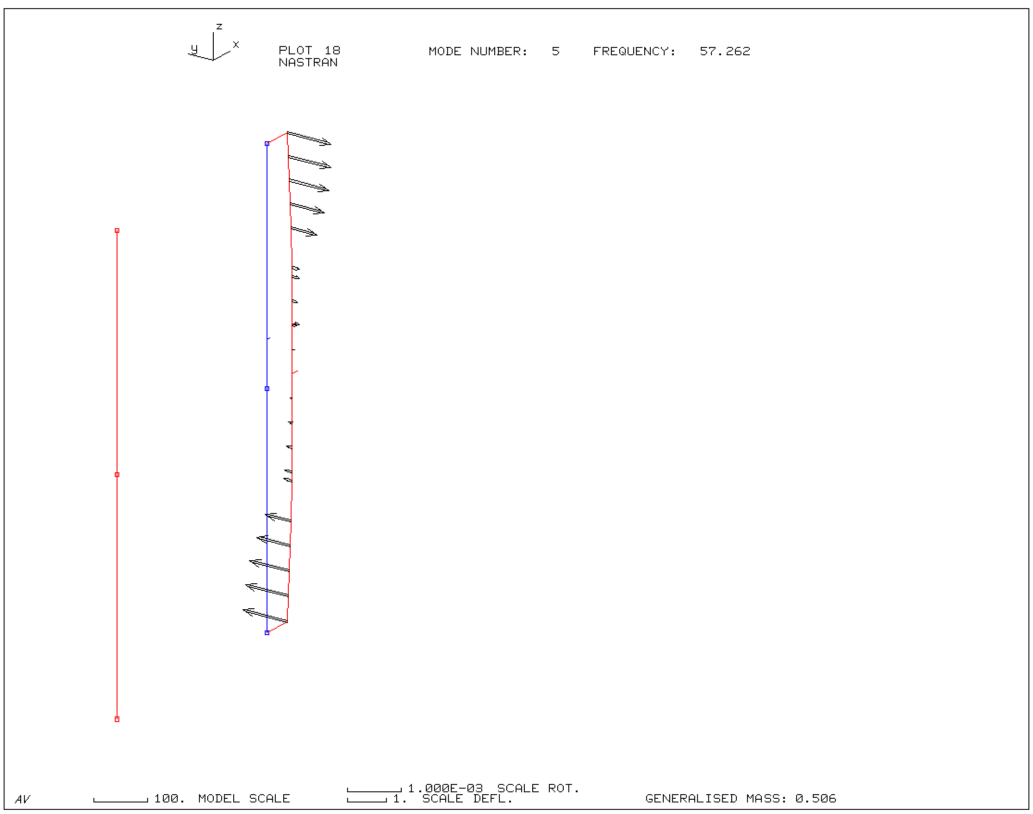


Fig. 105 Translation x, rotor B

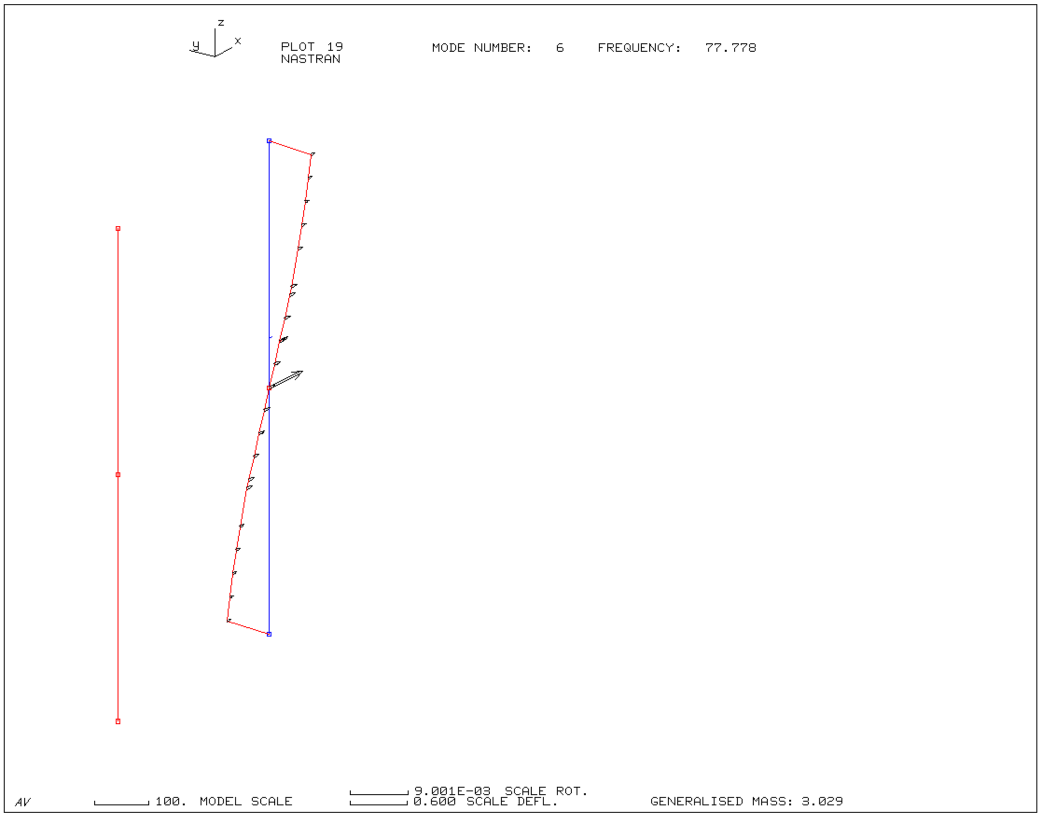


Fig. 106 Tilt about x-axis, rotor B

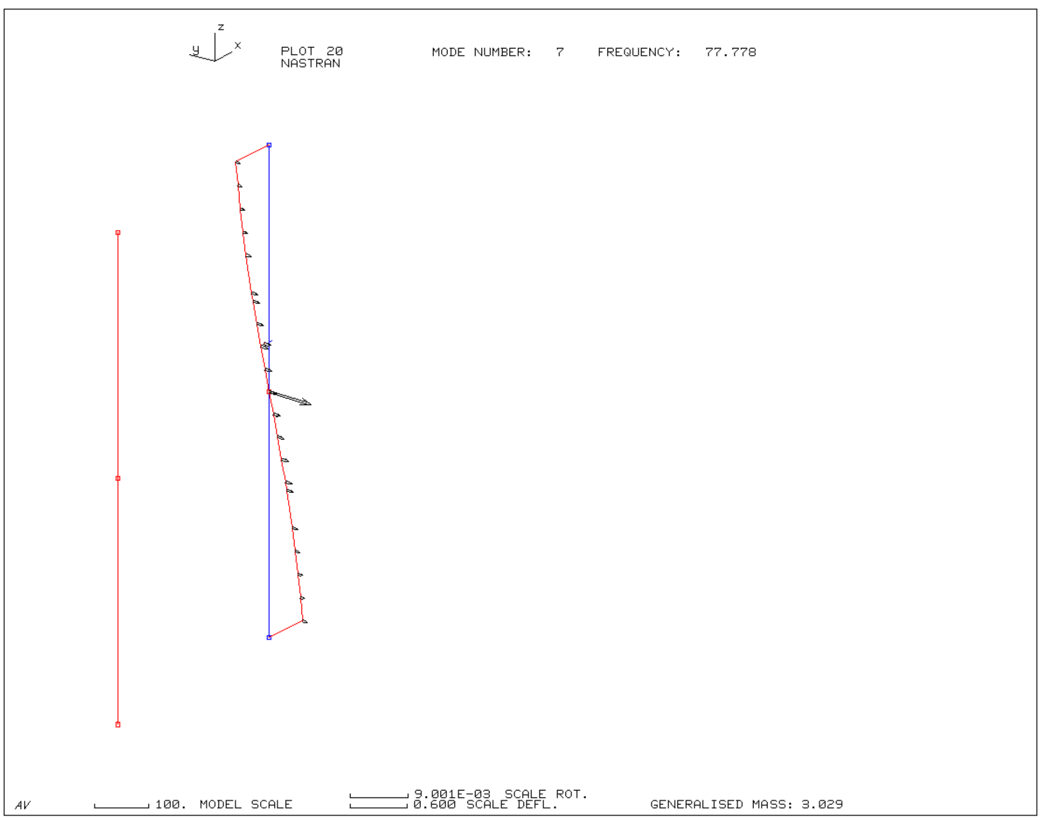


Fig. 107 Tilt about y-axis, rotor B

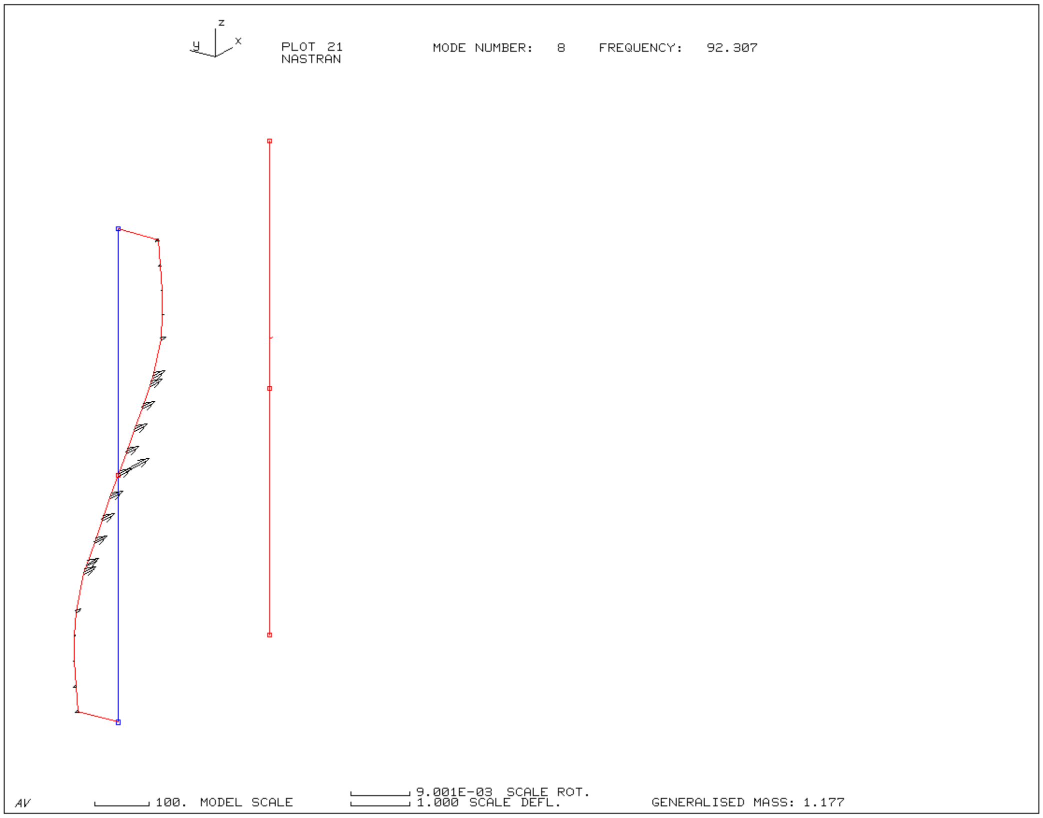


Fig. 108 Tilt about x-axis, rotor A

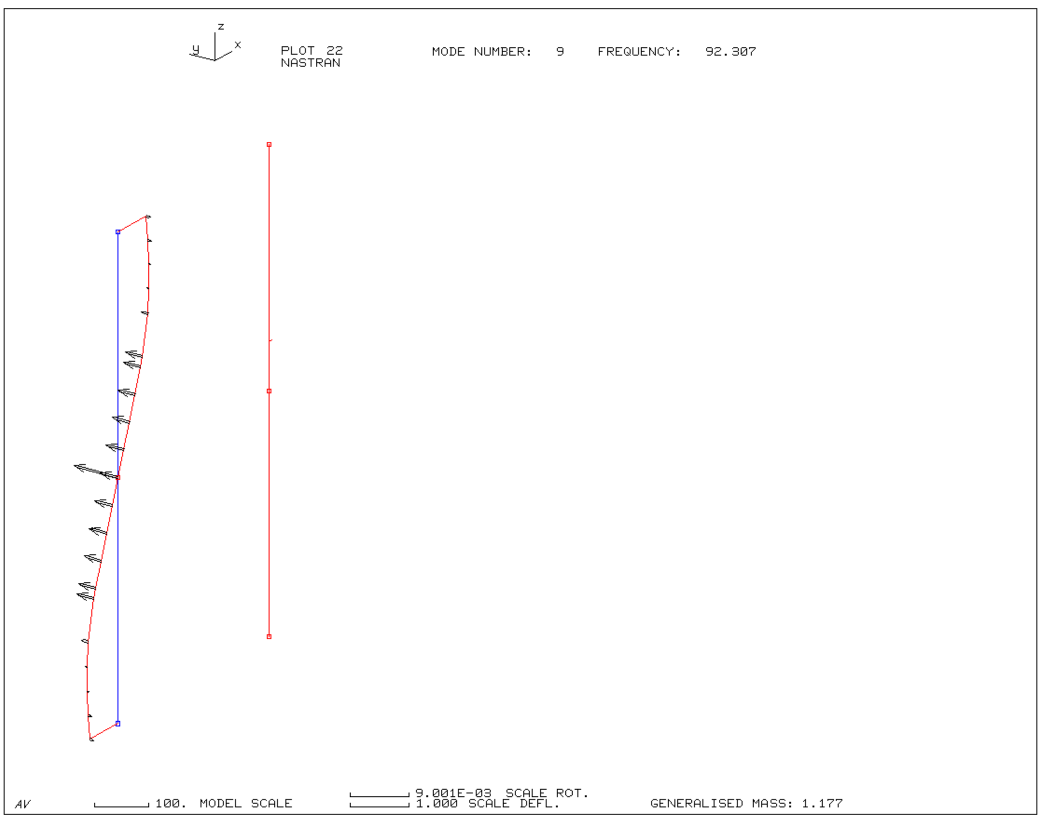


Fig. 109 Tilt about y-axis, rotor A

10.3 Complex Eigenvalues

The following mnemonics are used for the test models:

r220abxyz_sol	two rotors, rotating system
r221abxyz_sol	two rotors, fixed system

a	rotor A defined in the include file	mod_120a.dat
b	rotor B defined in the include file	mod_120a.dat

x:	configuration of the damping values
y:	damping of the MAT1 entries in percent
z:	damping on PARAM G in percent

Example:

Model **r220ab444_110**: CDAMP configuration 4, GE=0.04 on MAT1 entry and PARAM,G,0.04, rotating system and solution 110.

The damping is transformed by the eigenfrequencies unless the character 'w' appears in the model name.

Model	Description
r220ab000_110	Two rotors, zero damping, rotating system
r220ab100_110	Two rotors, CELAS damping only on the bearings
r220ab200_110	Two rotors, CELAS damping on bearings and translation motion in the rotors
r220ab300_110	Two rotors, CELAS damping on bearings and tilt motion in the rotors
r220ab400_110	Two rotors, CELAS damping on bearings, tilt and translation in the rotors
r220ab140_110	Two rotors, CELAS damping on the bearings, MAT1 damping 4%
r220ab104_110	Two rotors, CELAS damping on the bearings, PARAM G damping 4%
r220ab444_110	Two rotors, CELAS damping, MAT1 4% and PARAM G damping 4%
r120a444_110	Rotor A with all damping types
r120b444_110	Rotor B with all damping types
r220ab444w_110	Two rotors, all damping types with W3R and W4R
r120a444w_110	Rotor A with all damping types and with W3R and W4R
r120b444w_110	Rotor B with all damping types and with W3R and W4R
r221ab000_110	Two rotors, zero damping, fixed system

Table 48 Files used in the test sequence

10.4 Damping

The damping is divided into two parts:

1. Internal damping acting in the rotor part.
2. External damping acting on the bearings.

Damping can be defined by

4. CDAMP/PDAMP
5. GE on (for example) MAT1 entry
6. PARAM G

A loss factor on the CELAS entry could also be used, but it is not used here.

Configuration	Bearings		Rotor translation		Rotor tilt	
	Rotor A	Rotor B	Rotor A	Rotor B	Rotor A	Rotor B
0	0	0	0	0	0	0
1	30	8	0	0	0	0
2	30	8	60	80	0	0
3	30	8	0	0	8.0E+5	4.0+5
4	30	8	60	80	8.0E+5	4.0+5

Table 49 Damping values for the viscous damping elements PDAMP

The MAT1 damping acts on the appropriate structural elements.

The PARAM G damping acts on the whole structure, both the rotor and the bearings, including the CELAS elements.

10.4.1 Model without Damping

Model: r220ab000_110 - Two rotors with equal speed calculated with SOL 110 in the rotating system. The Campbell diagram is shown in Fig. 110. The results are converted to the fixed system as shown in Fig. 111. The real parts are shown in Fig. 112. The values are numerical zeroes. The whirl direction can be calculated in the rotating system even if there is no damping. The results for same model calculated in the fixed system are shown in Fig. 113. Here, the whirl direction of the translation modes cannot be calculated because there are no rotor dynamic forces acting. Therefore, the results of the translation modes cannot be converted to the rotating system as shown in Fig. 114.

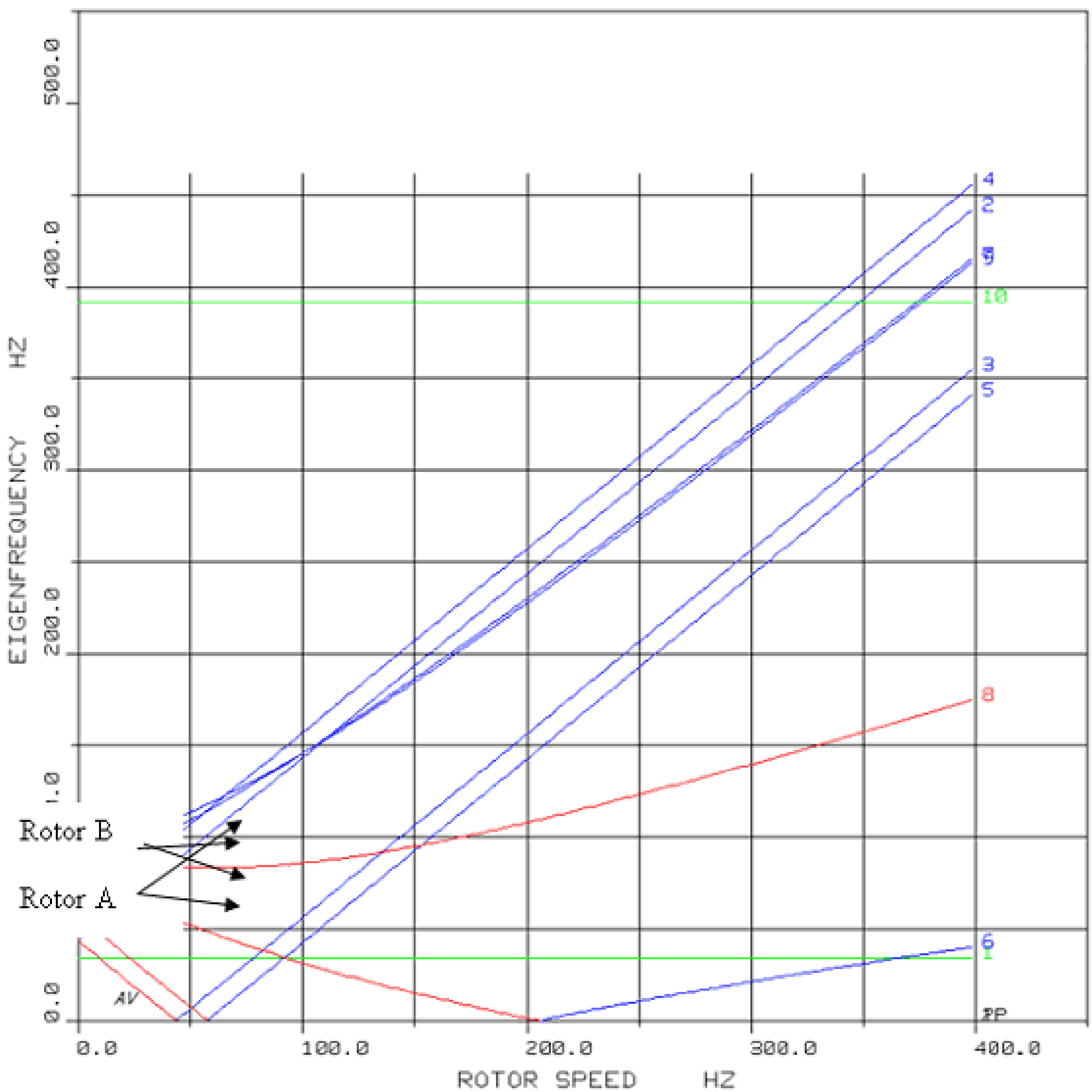


Fig. 110 Campbell diagram in the rotating system for both rotors. The green lines are shaft torsion and extension for rotor A

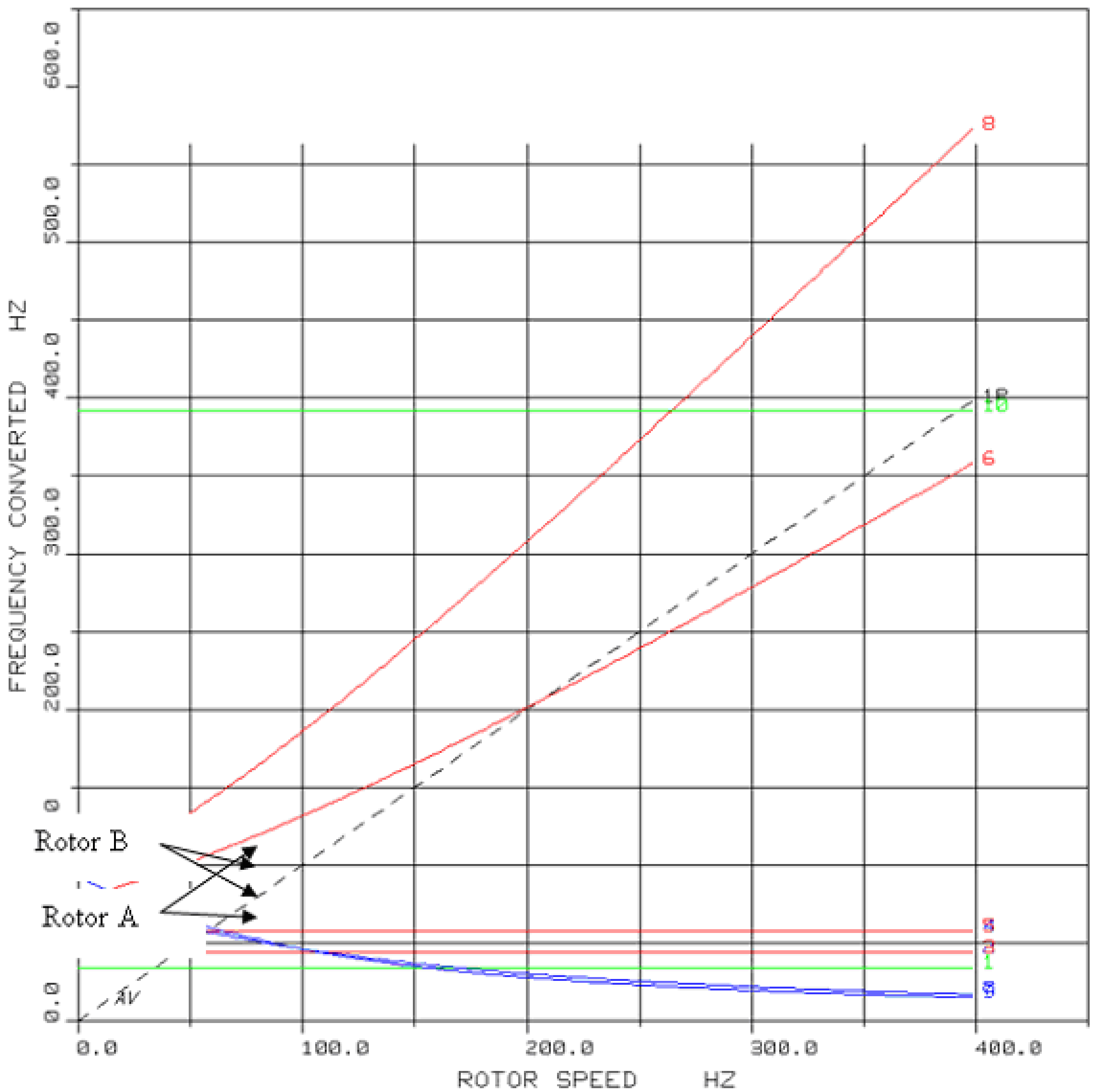


Fig. 111 Campbell diagram in the fixed system for both rotors. The green lines are shaft torsion and extension for rotor A

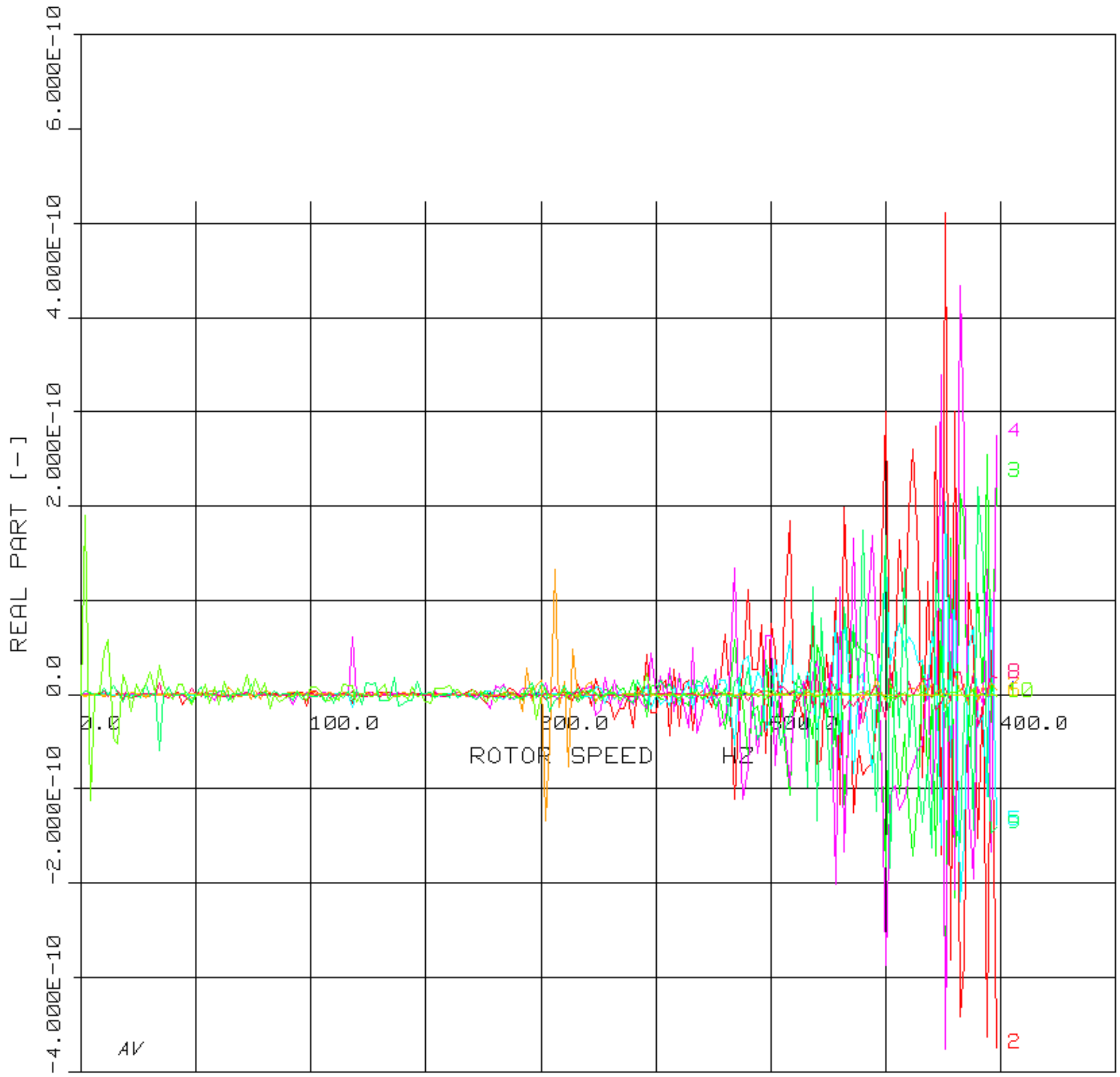


Fig. 112 Real part. Because there is no damping, the real part is practically zero

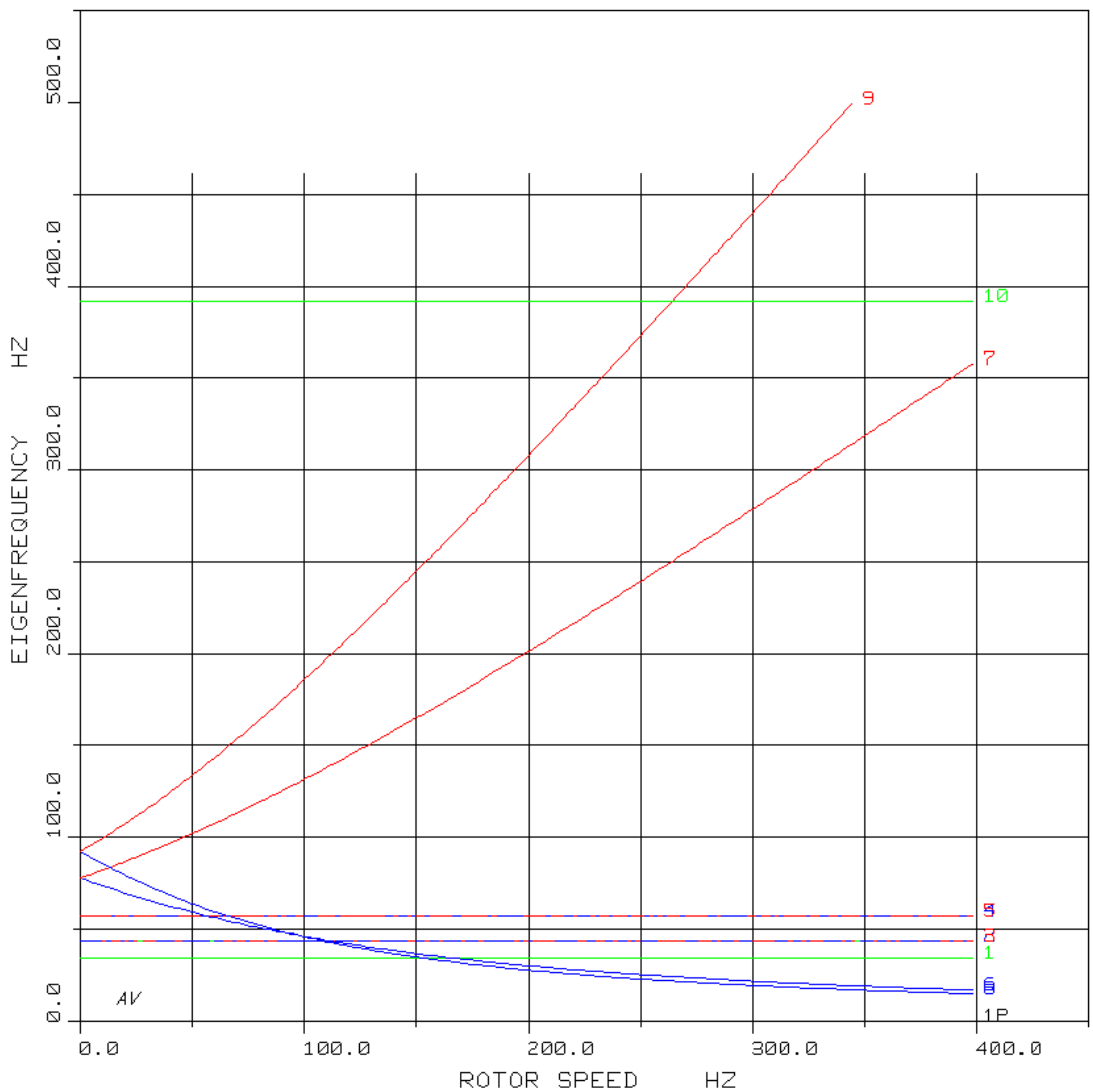


Fig. 113 Analysis in the fixed system. Whirl direction not found for the translation modes.

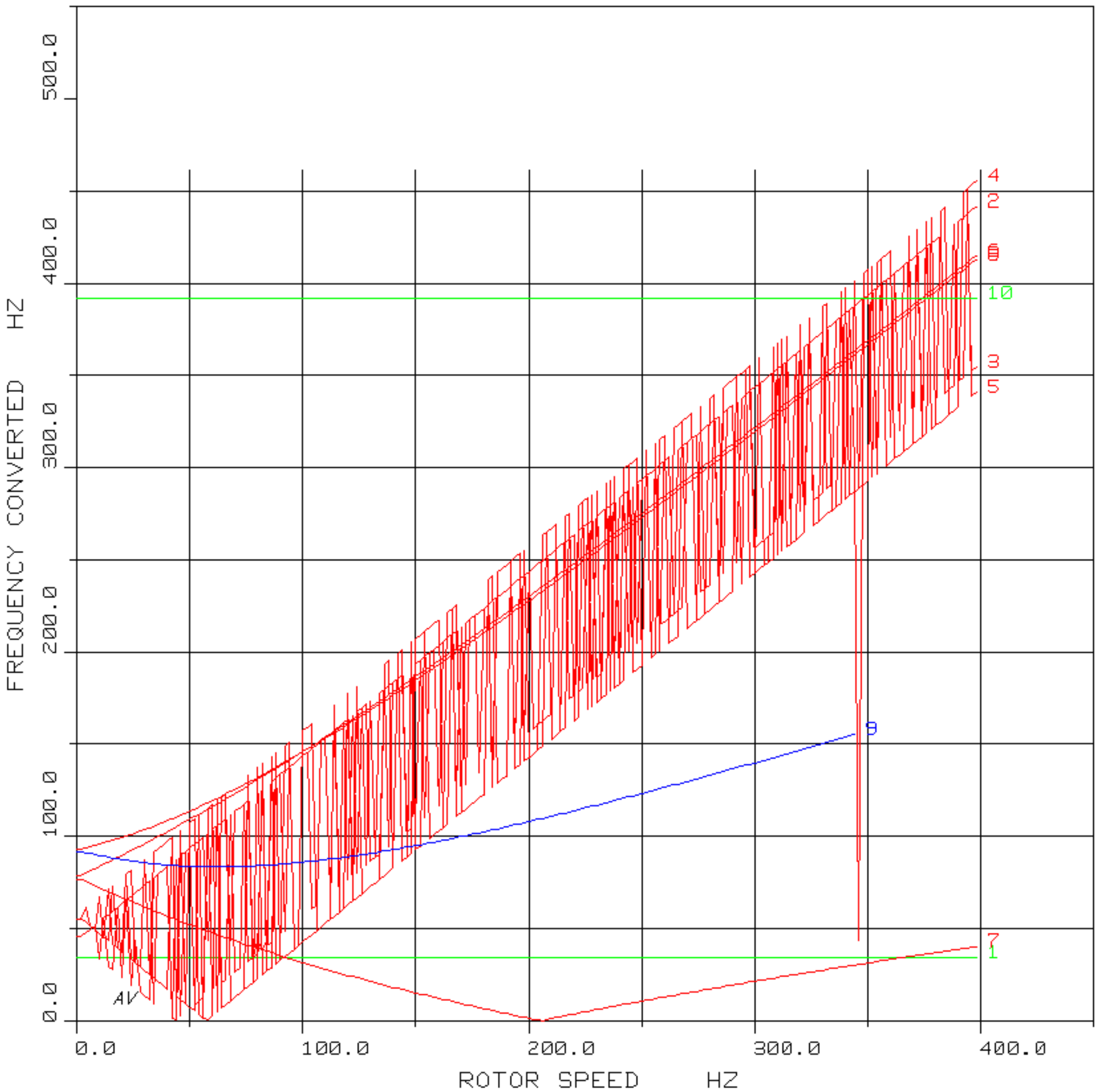


Fig. 114 Analysis in the fixed system. Results converted to the rotating system.

10.4.2 Damping in the Fixed System

In the model r220ab100, damping from CDAMP elements of the bearings is included. The rotors are stable. The damping curves are shown in Fig. 115.

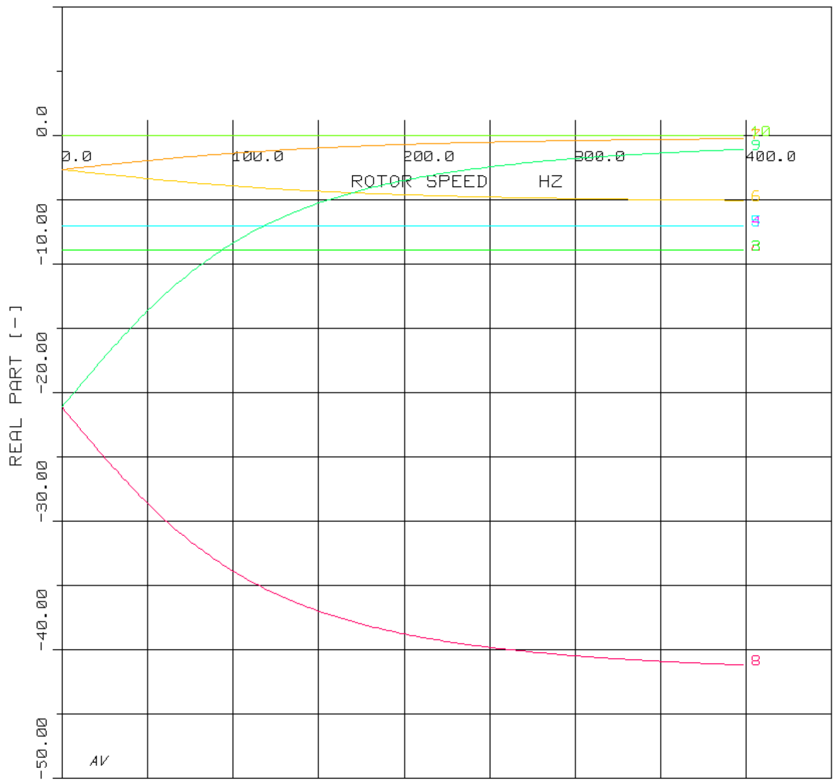


Fig. 115 Real part. Damping is acting only on the bearings. The system is stable

10.4.3 Damping in the Rotors

Activating the CDAMP elements for rotor translation leads to unstable translation modes as shown in Fig. 116. The instability speeds are above the critical speeds. Adding damping to the tilting modes only, the system remains stable as shown in Fig. 117. The case of translation and tilting damping is shown in Fig. 119.

The damping from CDAMP elements act directly as viscous damping. The material damping defined on the MAT1 entries are normally accounted for as the imaginary parts of the stiffness matrix. In rotor dynamic analysis the damping from the MAT1 entries are converted to viscous damping with the eigenvalues. The case of MAT1 damping combined with damping in the bearings is shown in Fig. 120. Also in this case, the rotors become unstable.

The damping from PARAM G is acting on the whole structure and the CELAS elements of the bearings. The results with PARAM G damping of 4% are shown in Fig. 121. The results with all damping types are shown in Fig. 122.

The damping factors calculated with Simcenter Nastran are shown in Table 50. These are given as fraction of critical equivalent viscous damping and are half the values of the structural damping factors. The first column in the table is from analysis in the rotating system, the second column were calculated in the fixed system. The values are identical.

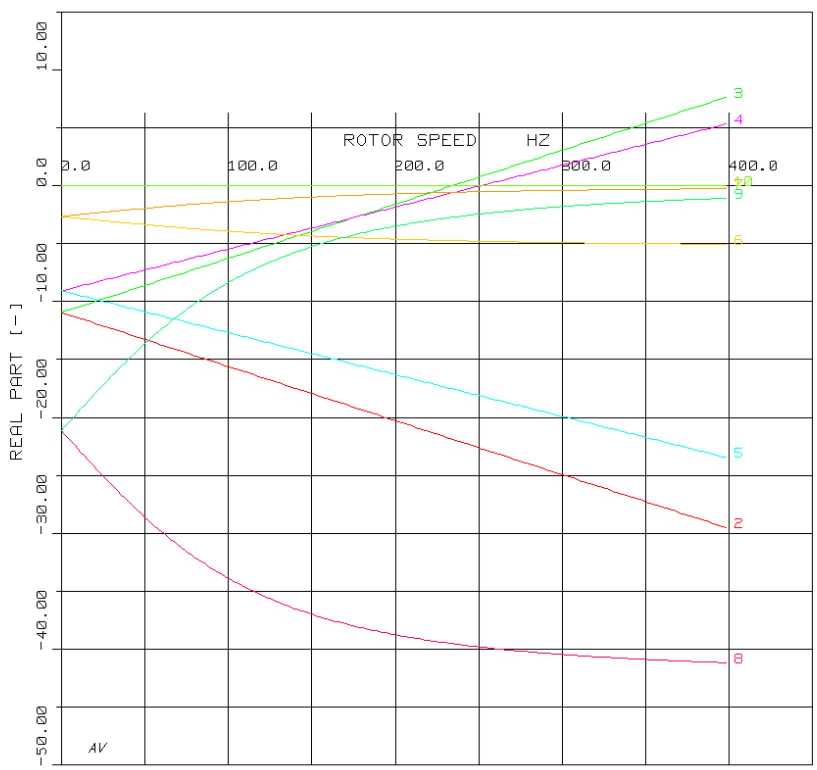


Fig. 116 Real part, Configuration 2, Translation damping in rotors. Both rotors get unstable

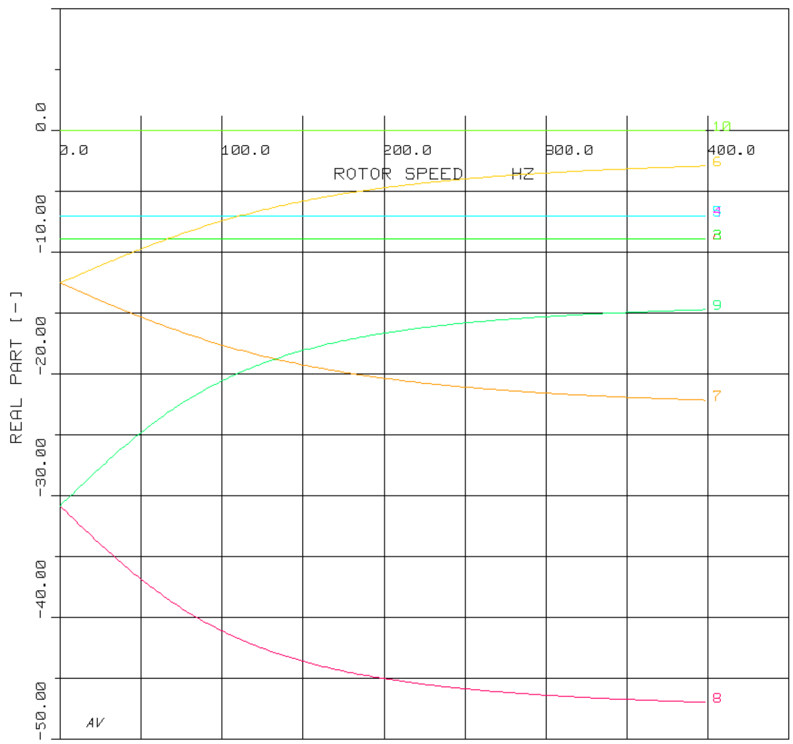


Fig. 117 Real part, Tilt damping in rotors. System is stable

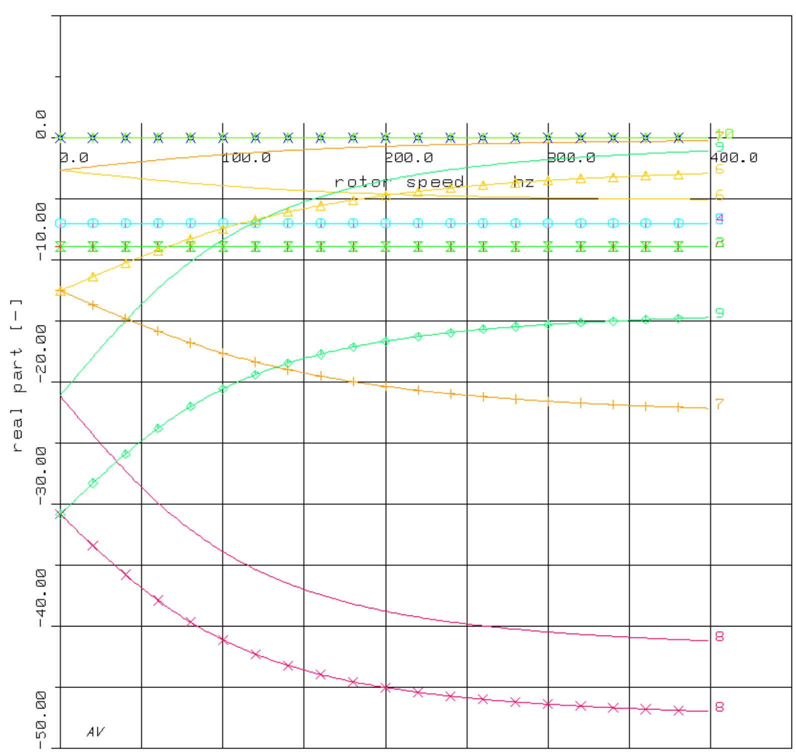


Fig. 118 Influence of rotor tilt damping (symbols) compared to the case of damping only on bearings

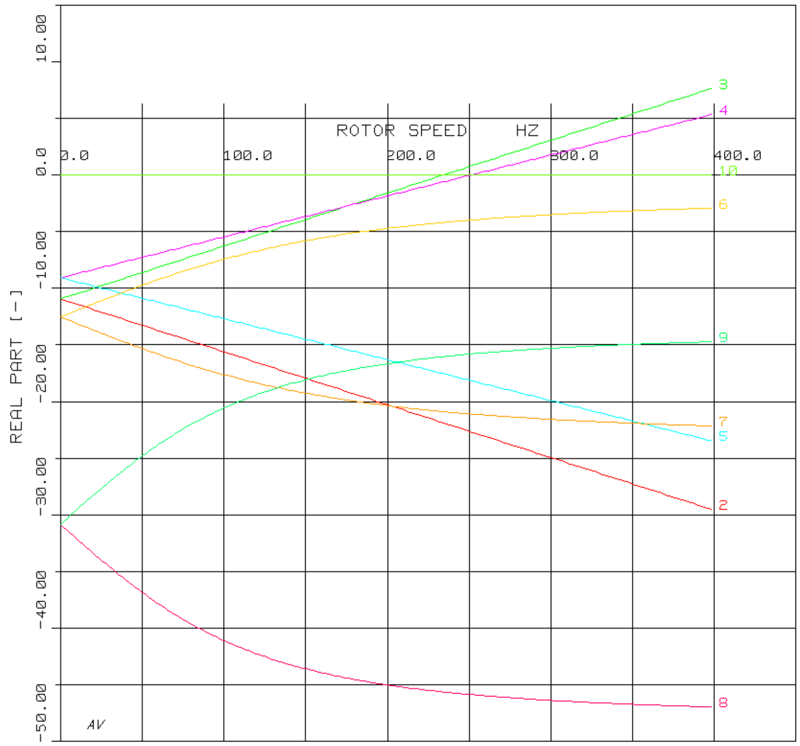


Fig. 119 Real part, Tilt and translation damping in rotors.

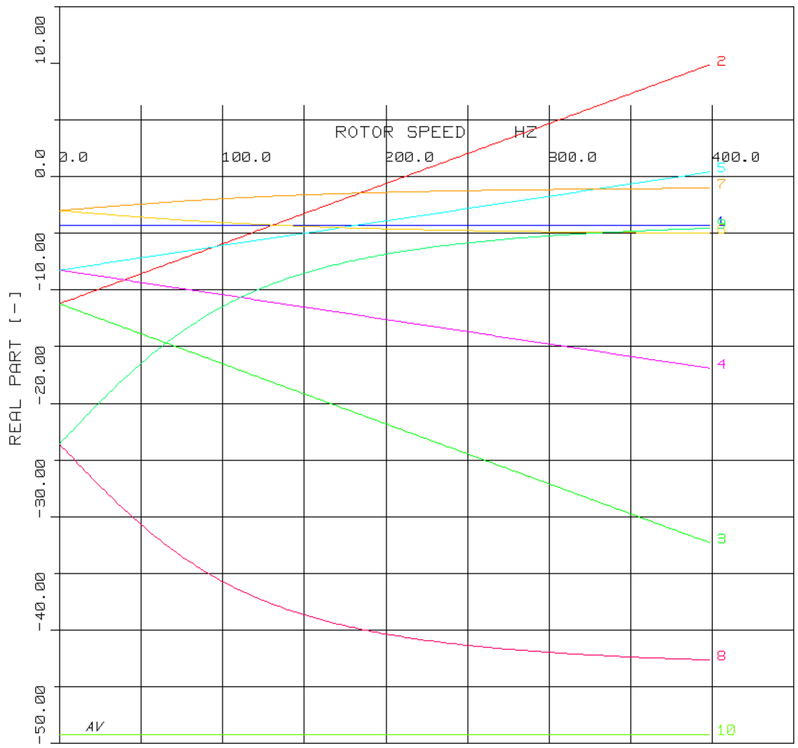


Fig. 120 Real part, MAT1 structural damping of 4% (2% viscous damping) in the rotors

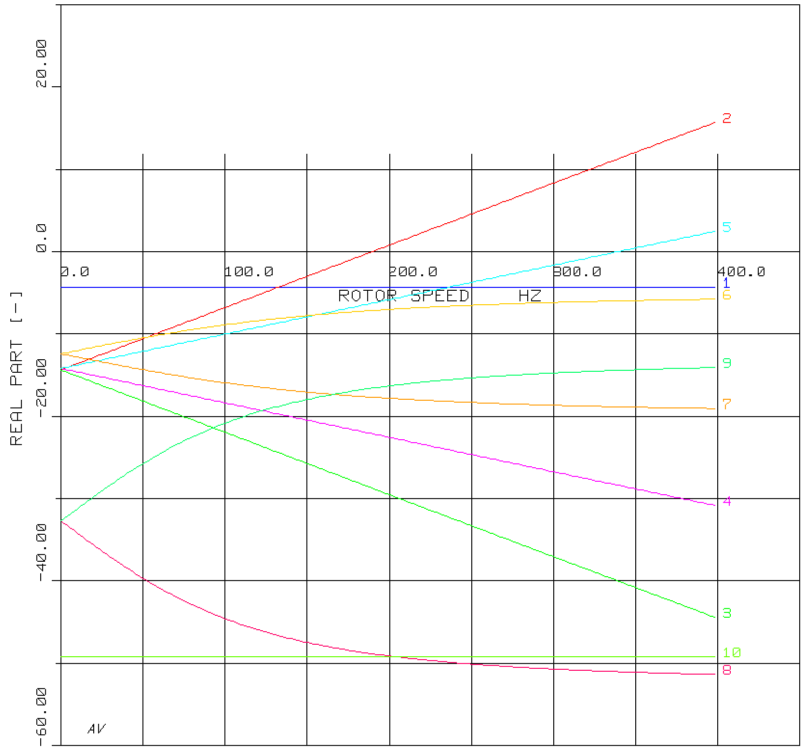


Fig. 121 Real part, PARAM G structural damping of 4% (2% viscous damping) in the whole model

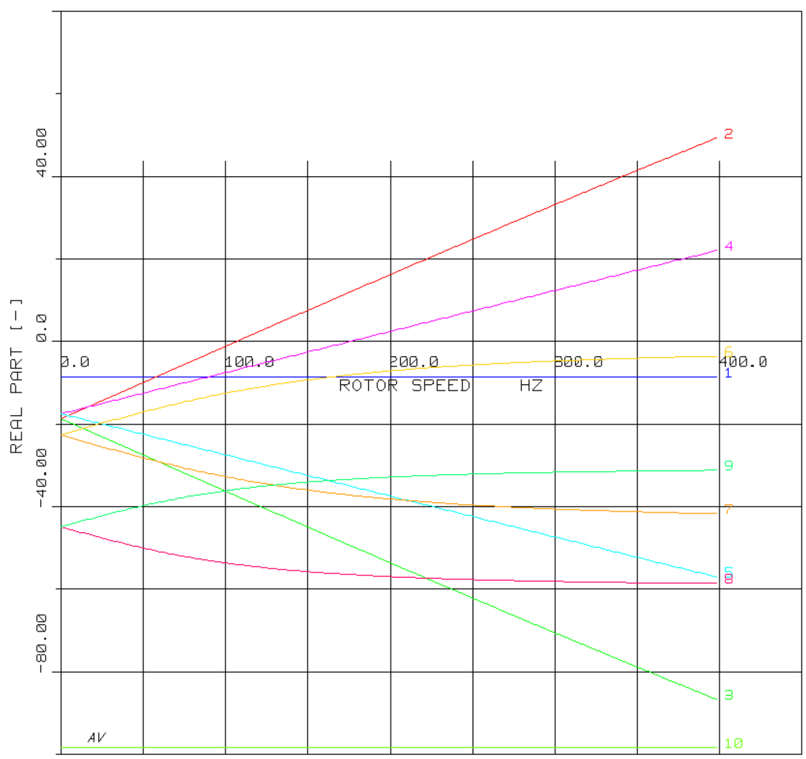


Fig. 122 Real part. PDAMP on bearings and rotors, MAT1=0.04 and PARAM G=0.04

```
 ^^^                                        ^^^
 ^^^ INTERNAL DAMPING FACTORS MAT1          ^^^ INTERNAL DAMPING FACTORS MAT1


 MATRIX K4HHDPD                             MATRIX K4HHDPD

 (   1)           1                         (   1)           1

    1   2.0000E-02                             1   2.0000E-02
    2   8.4491E-03                             2   8.4491E-03
    3   8.4491E-03                             3   8.4491E-03
    4   3.4614E-03                             4   3.4614E-03
    5   3.4614E-03                             5   3.4614E-03
    6   7.0596E-04                             6   7.0596E-04
    7   7.0596E-04                             7   7.0596E-04
    8   4.2592E-03                             8   4.2592E-03
    9   4.2592E-03                             9   4.2592E-03
   10   2.0000E-02                            10   2.0000E-02


 ^^^                                        ^^^
 ^^^ INTERNAL DAMPING FACTORS PARAM G       ^^^ INTERNAL DAMPING FACTORS PARAM G


 MATRIX K5HHDPD                             MATRIX K5HHDPD

 (   1)           1                         (   1)           1

    1   2.0000E-02                             1   2.0000E-02
    2   1.2075E-02                             2   1.2075E-02
    3   1.2075E-02                             3   1.2075E-02
    4   6.6622E-03                             4   6.6622E-03
    5   6.6622E-03                             5   6.6622E-03
    6   1.7272E-02                             6   1.7272E-02
    7   1.7272E-02                             7   1.7272E-02
    8   1.5812E-02                             8   1.5812E-02
    9   1.5812E-02                             9   1.5812E-02
   10   2.0000E-02                            10   2.0000E-02


 ^^^                                        ^^^
 ^^^ INTERNAL DAMPING FACTORS FROM CDAMP    ^^^ INTERNAL DAMPING FACTORS FROM
CDAMP


 MATRIX BHHDPD                              MATRIX BHHDPD

 (   1)           1                         (   1)           1

    1   3.2412E-35                             1   3.2412E-35
    2   7.4409E-03                             2   7.4409E-03
    3   7.4409E-03                             3   7.4409E-03
    4   5.7579E-03                             4   5.7579E-03
    5   5.7579E-03                             5   5.7579E-03
    6   2.0239E-02                             6   2.0239E-02
    7   2.0239E-02                             7   2.0239E-02
    8   1.6751E-02                             8   1.6751E-02
```

```
   9   1.6751E-02                           9   1.6751E-02
  10   5.0201E-22                          10   5.0201E-22


 ^^^                                       ^^^
 ^^^ EXTERNAL DAMPING FACTORS CDAMP        ^^^ EXTERNAL DAMPING FACTORS CDAMP


 MATRIX BHHSPD                             MATRIX BHHSPD

 (  1)          1                          (  1)          1

   1   2.7751E-35                           1   2.7751E-35
   2   3.2530E-02                           2   3.2530E-02
   3   3.2530E-02                           3   3.2530E-02
   4   1.9451E-02                           4   1.9451E-02
   5   1.9451E-02                           5   1.9451E-02
   6   5.4042E-03                           6   5.4042E-03
   7   5.4042E-03                           7   5.4042E-03
   8   3.6436E-02                           8   3.6436E-02
   9   3.6436E-02                           9   3.6436E-02
  10   1.6056E-21                          10   1.6056E-21


 ^^^                                       ^^^
 ^^^ EXTERNAL DAMPING FACTORS FROM PARAM G  ^^^ EXTERNAL DAMPING FACTORS FROM
PARAM G


 MATRIX K5HHSPD                            MATRIX K5HHSPD

 (  1)          1                          (  1)          1

   1   3.0758E-35                           1   3.0758E-35
   2   7.9252E-03                           2   7.9252E-03
   3   7.9252E-03                           3   7.9252E-03
   4   1.3338E-02                           4   1.3338E-02
   5   1.3338E-02                           5   1.3338E-02
   6   2.7282E-03                           6   2.7282E-03
   7   2.7282E-03                           7   2.7282E-03
   8   4.1882E-03                           8   4.1882E-03
   9   4.1882E-03                           9   4.1882E-03
  10   1.0289E-22                          10   1.0289E-22


 ^^^                                       ^^^
 ^^^ TOTAL INTERNAL DAMPING FACTORS        ^^^ TOTAL INTERNAL DAMPING FACTORS


 MATRIX DHHDPD                             MATRIX DHHDPD

 (  1)          1                          (  1)          1

   1   4.0000E-02                           1   4.0000E-02
   2   2.7965E-02                           2   2.7965E-02
   3   2.7965E-02                           3   2.7965E-02
   4   1.5881E-02                           4   1.5881E-02
   5   1.5881E-02                           5   1.5881E-02
```

```
   6   3.8217E-02                            6   3.8217E-02
   7   3.8217E-02                            7   3.8217E-02
   8   3.6822E-02                            8   3.6822E-02
   9   3.6822E-02                            9   3.6822E-02
  10   4.0000E-02                           10   4.0000E-02


^^^                                        ^^^
^^^ TOTAL EXTERNAL DAMPING FACTORS         ^^^ TOTAL EXTERNAL DAMPING FACTORS


MATRIX DHHSPD                              MATRIX DHHSPD

(  1)          1                           (  1)          1

   1   5.8509E-35                            1   5.8509E-35
   2   4.0455E-02                            2   4.0455E-02
   3   4.0455E-02                            3   4.0455E-02
   4   3.2789E-02                            4   3.2789E-02
   5   3.2789E-02                            5   3.2789E-02
   6   8.1324E-03                            6   8.1324E-03
   7   8.1324E-03                            7   8.1324E-03
   8   4.0625E-02                            8   4.0625E-02
   9   4.0625E-02                            9   4.0625E-02
  10   1.7085E-21                           10   1.7085E-21
```

Table 50 Damping factors calculated by the program

10.5 Model with two Rotors compared to uncoupled Analysis of the individual Rotors

The analysis with all damping types was analyzed for the single rotors and compared to the results obtained with both rotors.

Model with two rotors	r220ab444
Model with rotor A	r120a444
Model with rotor B	r120b444

The Campbell diagram is shown in Fig. 123 and the real parts in Fig. 124. The results are identical.

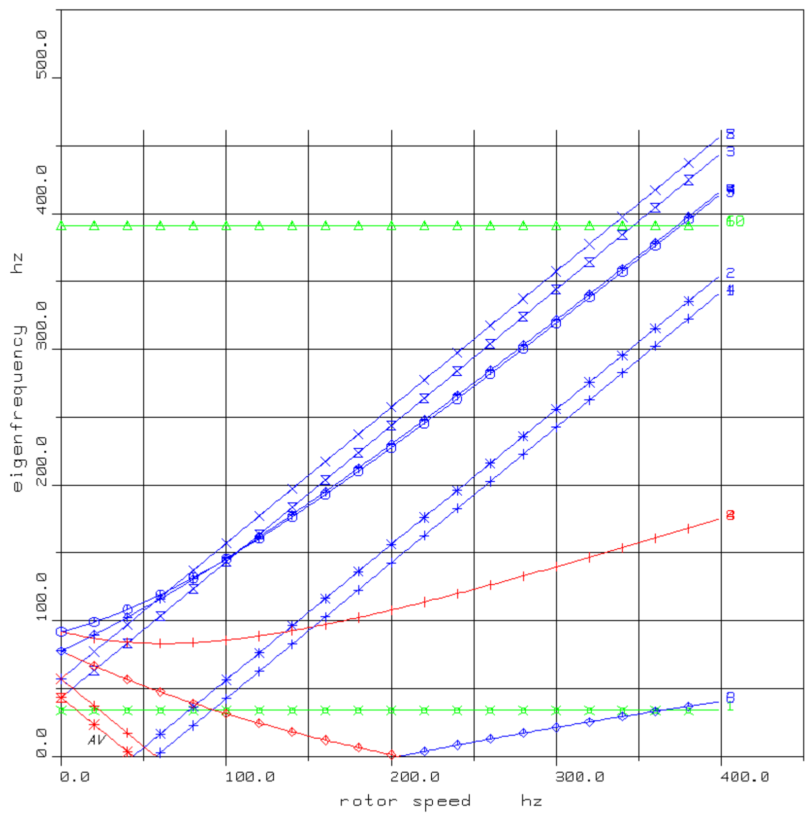


Fig. 123 Analysis with two rotors compared to two single analyses of each rotor (symbols)

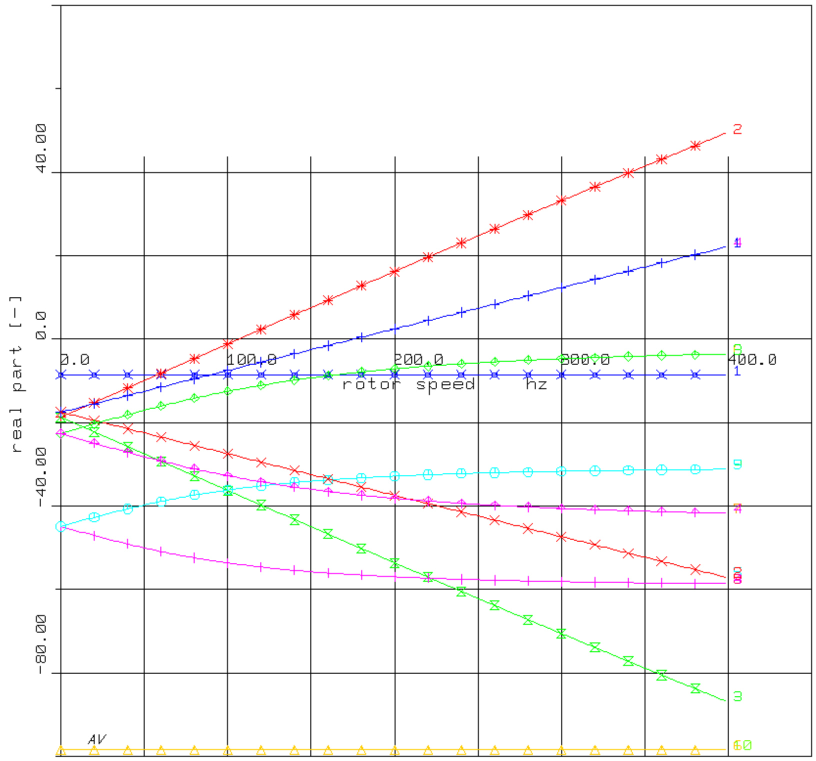


Fig. 124 Real parts: Analysis with two rotors compared to two single analyses of each rotor (symbols)

10.5.1 Analysis in the Rotating and the Fixed System

The model r220ab444 was analyzed in the fixed system with the model r221ab444. The results in the rotating system are shown in Fig. 125 and in the fixed system in Fig. 126. The eigenfrequencies are identical. The real eigenvalues shown in Fig. 127 are also identical.

The possible resonances with the 1P line are shown in Fig. 128 for the fixed system. The crossings of the forward whirl with the red lines at 43.5, 57.2 and 205 Hz are shown with arrows pointing up. The backward whirl crossings with the blue lines at 43.5, 57.2 (blue lines are coincident with the horizontal red lines), 57.0 and 69.4 Hz are shown as arrows pointing down. The program calculates the crossing points as shown in Table 51. They are consistent with the crossing points shown in Fig. 128. In addition to the asynchronous analysis, a synchronous analysis is done. The comparison between the synchronous and asynchronous results is shown in Table 52. The results are almost identical. The difference is due to the fact that the synchronous analysis is done without damping.

The resonances in the rotating system are shown in Fig. 129. The crossings with the abscissa (0P line) are shown as arrows pointing up at 43.5, 57.2 and 205 Hz. The crossings of the backward whirl modes, shown as blue lines, with the 2P line are shown by arrows pointing to the left at 43.5, 57.0, 57.2 and 59.4 Hz. The crossing points calculated by the program are shown in Table 53. They are consistent with the values in Fig. 129. The crossings with the linear modes are not calculated for the rotating system. In the synchronous analysis, the crossings with the 2P line at 17 and 195 Hz are listed. The comparison between the synchronous and asynchronous results is shown in Table 54. Also here, the results are practically identical. The calculation of the crossing points is dependent on the step size of the rotor speed. With large step size, the crossing points may be inaccurate and crossings with the abscissa may be missing.

The instabilities due to internal damping are shown in Fig. 130. A comparison of the results for the fixed and rotating system is provided in Table 56. The results are identical.

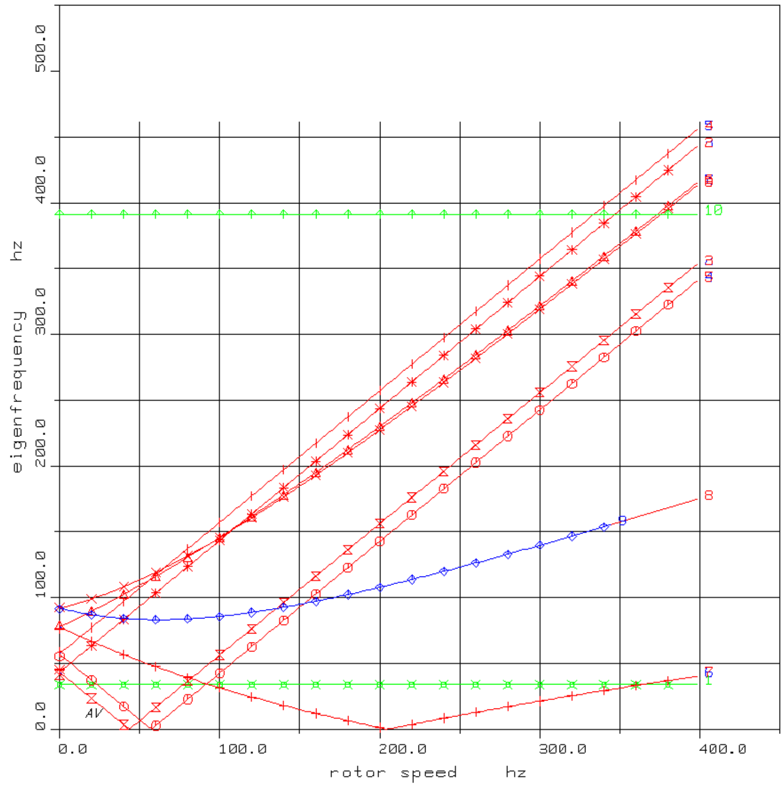


Fig. 125 Eigenfrequencies in the rotating system. Comparison of analyses in rotating and fixed system

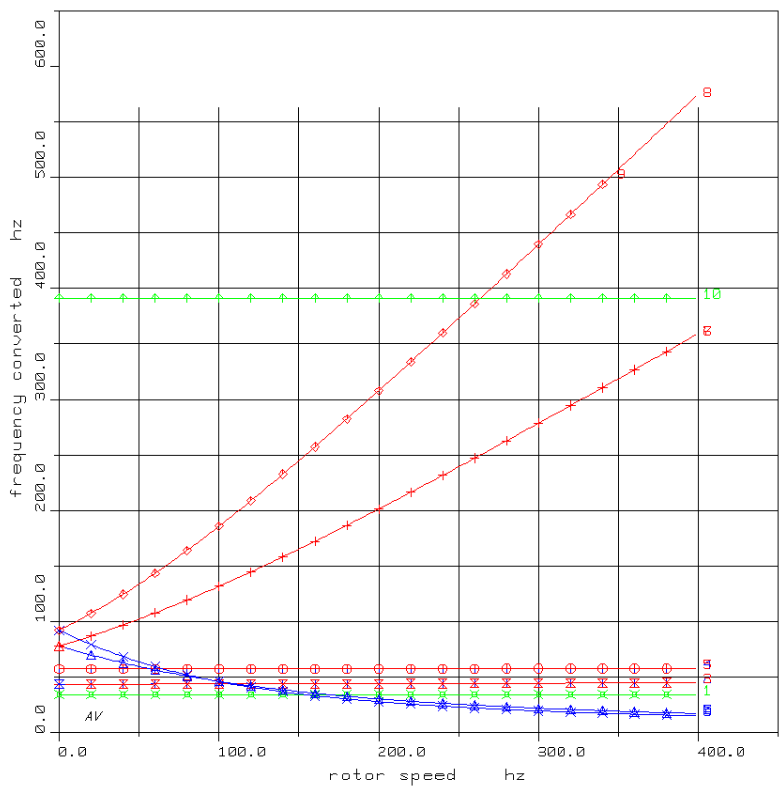


Fig. 126 Eigenfrequencies in the fixed system. Comparison of analyses in rotating and fixed system

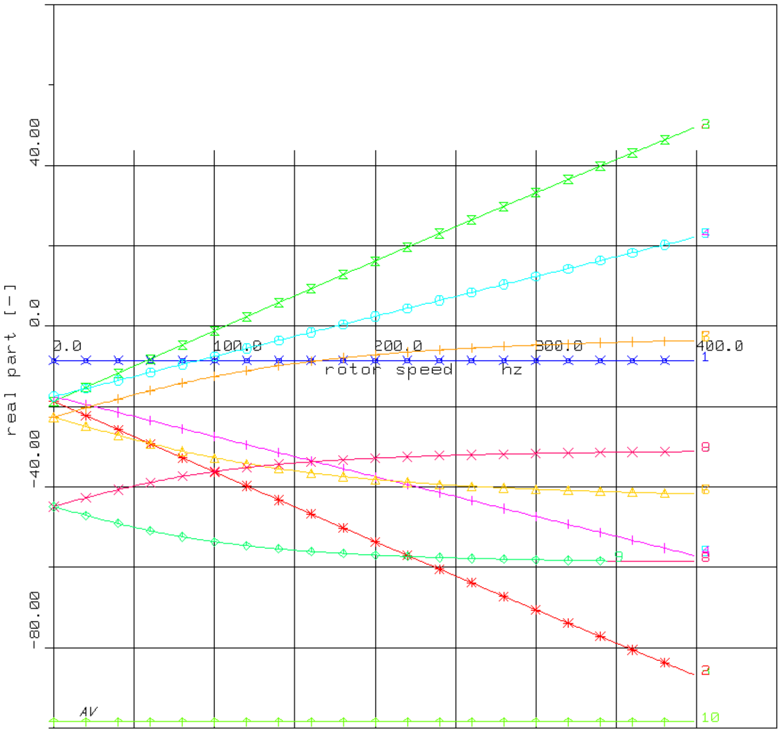


Fig. 127 Real eigenvalues. Comparison of analyses in rotating and fixed system

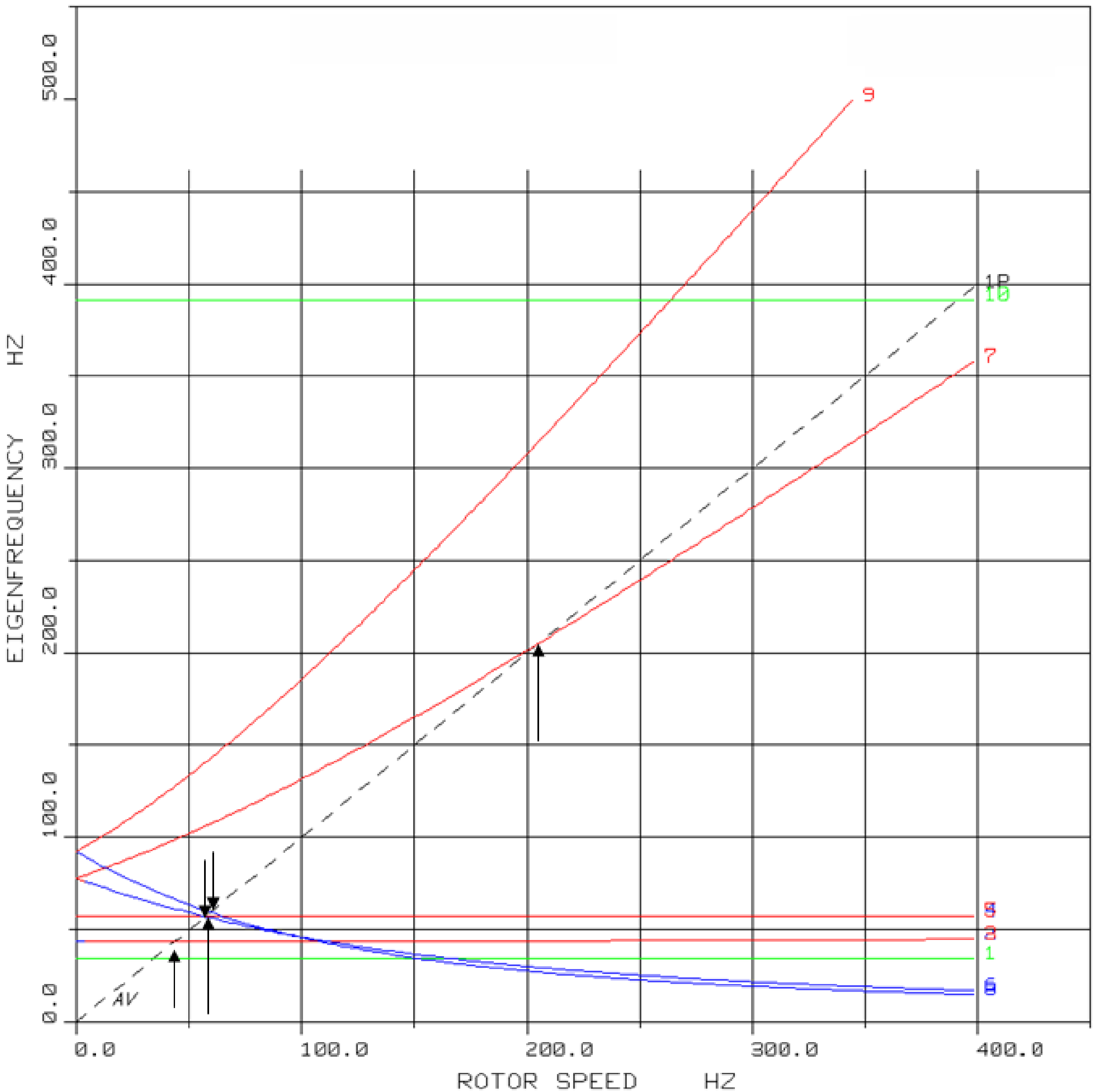


Fig. 128 Crossings with 1P line for fixed system analysis

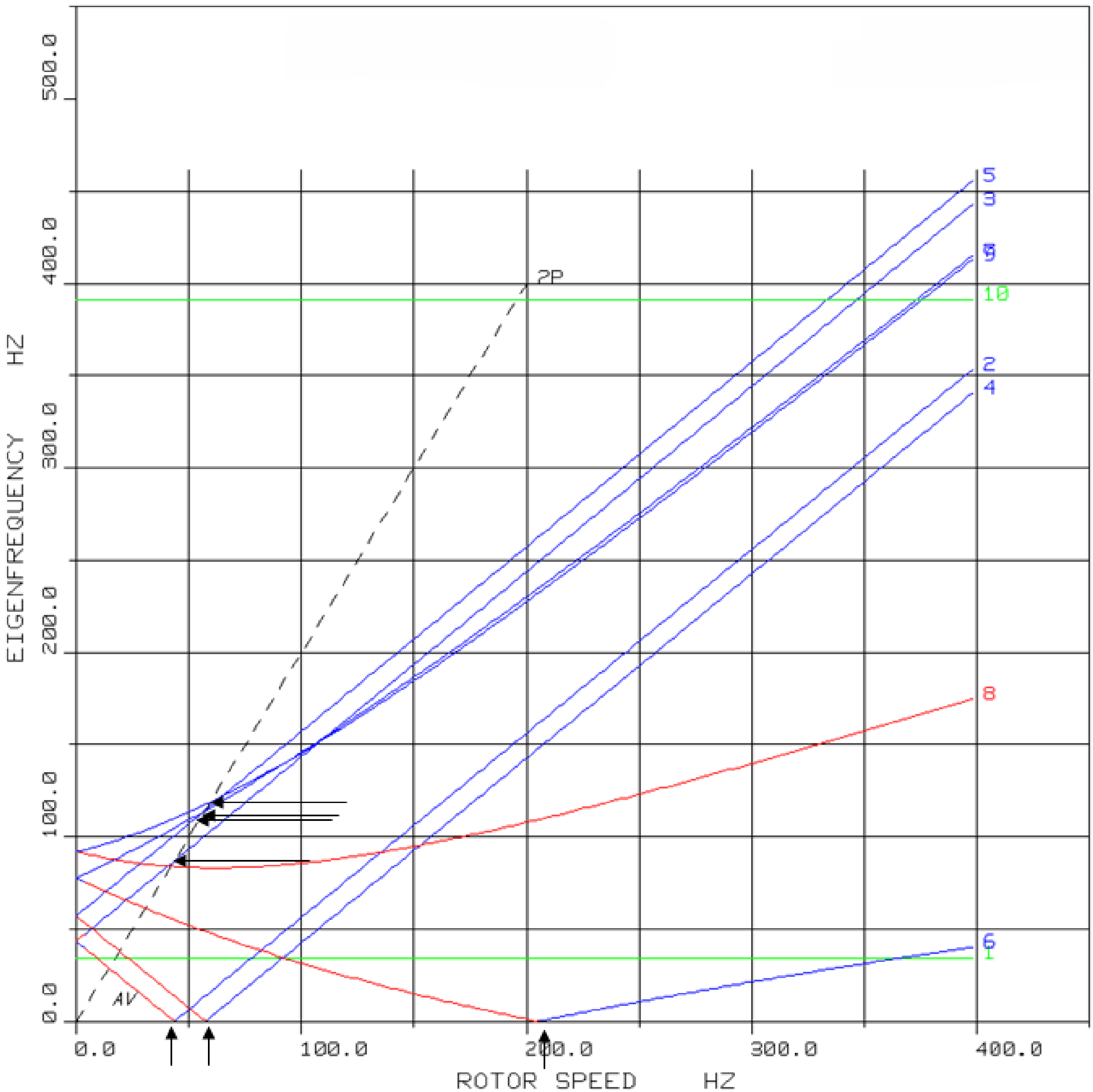


Fig. 129 Crossings with 0P and 2P line for fixed system analysis

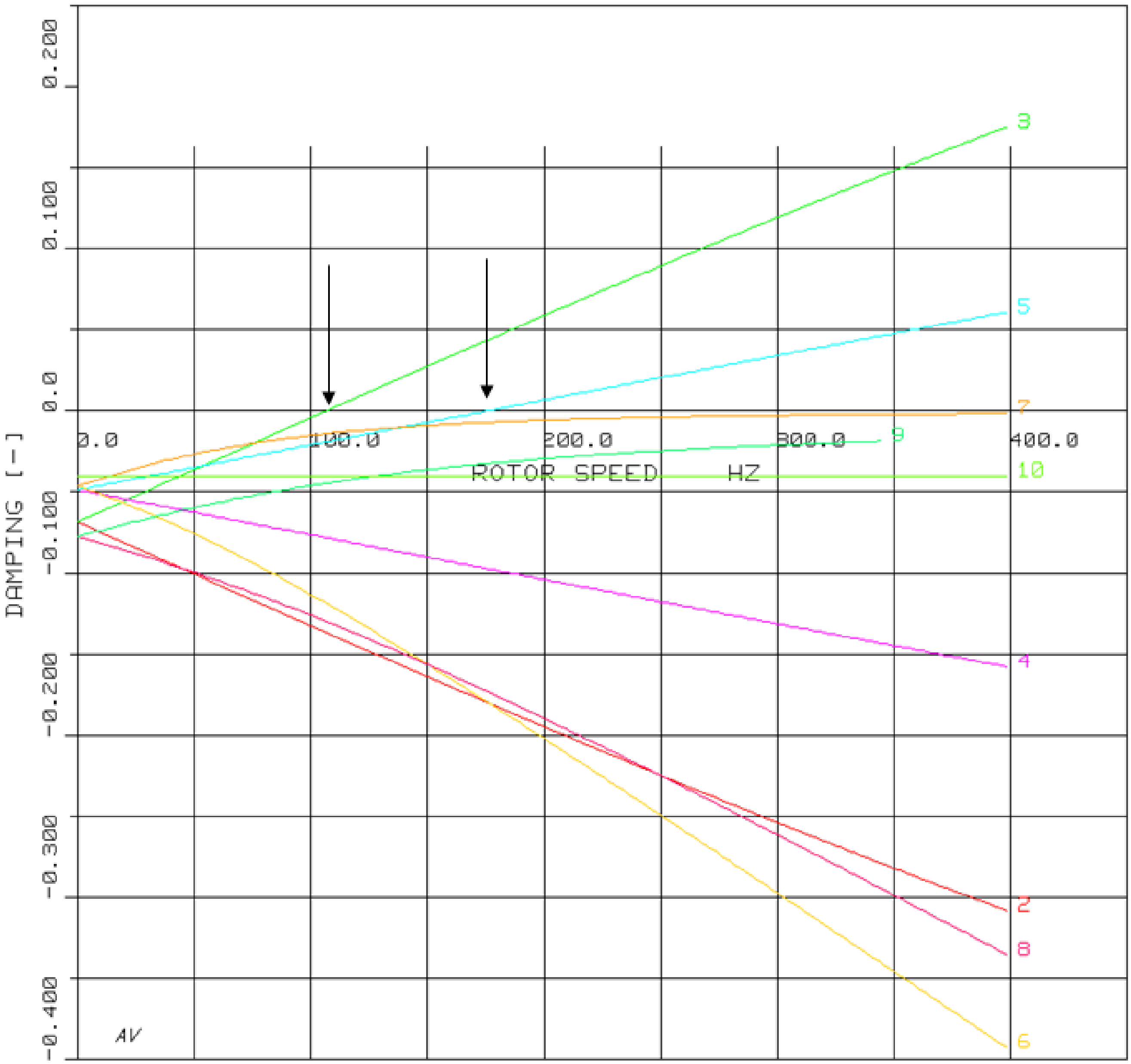


Fig. 130 Damping of model in fixed system. Instabilities at 106.6 and 175.5 Hz

```
D E T E C T I O N   O F   R E S O N A N C E S   A N D   I N S T A B I L I T I E S
                    A N A L Y S I S   I N   F I X E D   S Y S T E M


 WHIRL RESONANCE                SOLUTION      ROTOR SPEED     WHIRL
                                 NUMBER           HZ        DIRECTION


                                ROTOR NUMBER       1     RELATIVE SPEED
1.00000E+00

                                      1       3.40357E+01    LINEAR
                                      2       4.34658E+01    BACKWARD
                                      3       4.34658E+01    FORWARD
                                      4       5.72010E+01    BACKWARD
                                      5       5.72010E+01    FORWARD
                                      6       5.70330E+01    BACKWARD
                                      7       2.04963E+02    FORWARD
                                      8       5.93875E+01    BACKWARD
                                     10       3.91308E+02    LINEAR

                                ROTOR NUMBER       2     RELATIVE SPEED
1.00000E+00

                                      1       3.40357E+01    LINEAR
                                      2       4.34658E+01    BACKWARD
                                      3       4.34658E+01    FORWARD
                                      4       5.72010E+01    BACKWARD
                                      5       5.72010E+01    FORWARD
                                      6       5.70330E+01    BACKWARD
                                      7       2.04963E+02    FORWARD
                                      8       5.93875E+01    BACKWARD
                                     10       3.91308E+02    LINEAR


 INSTABILITIES                  SOLUTION      ROTOR SPEED     WHIRL
                                 NUMBER           HZ        DIRECTION

 START                                3       1.06553E+02    FORWARD
 START                                5       1.75484E+02    FORWARD


 CRITICAL SPEEDS FROM SYNCHRONOUS ANALYSIS
                                SOLUTION      ROTOR SPEED     WHIRL
                                 NUMBER           HZ        DIRECTION

                                      1       3.40630E+01    LINEAR
                                      2       4.35508E+01    BACKWARD
                                      3       4.35508E+01    FORWARD
                                      4       5.70888E+01    BACKWARD
                                      5       5.72617E+01    FORWARD
                                      6       5.72617E+01    BACKWARD
                                      7       5.95621E+01    BACKWARD
                                      9       2.05076E+02    FORWARD
                                     10       3.91622E+02    LINEAR
                                     11       3.40630E+01    LINEAR
                                     12       4.35508E+01    BACKWARD
                                     13       4.35508E+01    FORWARD
```

14	5.70888E+01	BACKWARD
15	5.72617E+01	FORWARD
16	5.72617E+01	BACKWARD
17	5.95621E+01	BACKWARD
19	2.05076E+02	FORWARD
20	3.91622E+02	LINEAR

Table 51 Resonance output for rotor in fixed system

```
                              WHIRL RESONANCE

 SOLUTION       ROTOR SPEED     WHIRL
  NUMBER             HZ       DIRECTION


 ROTOR NUMBER       1     RELATIVE SPEED  1.00000E+00

                    ASYNCHRONOUS                      SYNCHRONOUS
       1       3.40357E+01    LINEAR    1       3.40630E+01    LINEAR
       2       4.34658E+01    BACKWARD  2       4.35508E+01    BACKWARD
       3       4.34658E+01    FORWARD   3       4.35508E+01    FORWARD
       4       5.72010E+01    BACKWARD  6       5.72617E+01    BACKWARD
       5       5.72010E+01    FORWARD   5       5.72617E+01    FORWARD
       6       5.70330E+01    BACKWARD  4       5.70888E+01    BACKWARD
       7       2.04963E+02    FORWARD   9       2.05076E+02    FORWARD
       8       5.93875E+01    BACKWARD  7       5.95621E+01    BACKWARD
      10       3.91308E+02    LINEAR   10       3.91622E+02    LINEAR

 ROTOR NUMBER       2     RELATIVE SPEED  1.00000E+00

       1       3.40357E+01    LINEAR   11       3.40630E+01    LINEAR
       2       4.34658E+01    BACKWARD 12       4.35508E+01    BACKWARD
       3       4.34658E+01    FORWARD  13       4.35508E+01    FORWARD
       4       5.72010E+01    BACKWARD 16       5.72617E+01    BACKWARD
       5       5.72010E+01    FORWARD  15       5.72617E+01    FORWARD
       6       5.70330E+01    BACKWARD 14       5.70888E+01    BACKWARD
       7       2.04963E+02    FORWARD  19       2.05076E+02    FORWARD
       8       5.93875E+01    BACKWARD 17       5.95621E+01    BACKWARD
      10       3.91308E+02    LINEAR   20       3.91622E+02    LINEAR
```

Table 52 Whirl resonances calculated in the fixed system in asynchronous and synchronous analysis

```
D E T E C T I O N   O F   R E S O N A N C E S   A N D   I N S T A B I L I T I E S
                   A N A L Y S I S   I N   R O T A T I N G   S Y S T E M


 FORWARD WHIRL RESONANCE        SOLUTION      ROTOR SPEED     WHIRL
                                 NUMBER           HZ        DIRECTION

                                      2       4.34655E+01    FORWARD
                                      4       5.72009E+01    FORWARD
                                      6       2.04895E+02    FORWARD


 BACKWARD WHIRL RESONANCE       SOLUTION      ROTOR SPEED     WHIRL
                                 NUMBER           HZ        DIRECTION


                                ROTOR NUMBER       1     RELATIVE SPEED
1.00000E+00

                                      3       4.34658E+01    BACKWARD
                                      5       5.72010E+01    BACKWARD
                                      7       5.70330E+01    BACKWARD
                                      9       5.93875E+01    BACKWARD

                                ROTOR NUMBER       2     RELATIVE SPEED
1.00000E+00

                                      3       4.34658E+01    BACKWARD
                                      5       5.72010E+01    BACKWARD
                                      7       5.70330E+01    BACKWARD
                                      9       5.93875E+01    BACKWARD


 INSTABILITIES                  SOLUTION      ROTOR SPEED     WHIRL
                                 NUMBER           HZ        DIRECTION

 START                                2       1.06553E+02    FORWARD
 START                                4       1.75484E+02    FORWARD


 CRITICAL SPEEDS FROM SYNCHRONOUS ANALYSIS
                                SOLUTION      ROTOR SPEED     WHIRL
                                 NUMBER           HZ        DIRECTION

                                      1       4.35508E+01    FORWARD
                                      2       5.72617E+01    FORWARD
                                      3       2.05076E+02    FORWARD
                                      4       1.70315E+01    LINEAR
                                      5       4.35508E+01    BACKWARD
                                      6       5.70888E+01    BACKWARD
                                      7       5.72617E+01    BACKWARD
                                      8       5.95621E+01    BACKWARD
                                      9       1.95811E+02    LINEAR
                                     10       4.35508E+01    FORWARD
                                     11       5.72617E+01    FORWARD
                                     12       2.05076E+02    FORWARD
                                     13       1.70315E+01    LINEAR
```

14	4.35508E+01	BACKWARD
15	5.70888E+01	BACKWARD
16	5.72617E+01	BACKWARD
17	5.95621E+01	BACKWARD
18	1.95811E+02	LINEAR

Table 53 Resonance output for rotor in rotating system

```
FORWARD WHIRL RESONANCE
SOLUTION      ROTOR SPEED     WHIRL
 NUMBER           HZ       DIRECTION
              ASYNCRHONOUS                 SYNCHRONOUS                    SYNCHRONOUS
     2        4.34655E+01    FORWARD     1  4.35508E+01    FORWARD     10  4.35508E+01    FORWARD
     4        5.72009E+01    FORWARD     2  5.72617E+01    FORWARD     11  5.72617E+01    FORWARD
     6        2.04895E+02    FORWARD     3  2.05076E+02    FORWARD     12  2.05076E+02    FORWARD

BACKWARD WHIRL RESONANCE
SOLUTION      ROTOR SPEED     WHIRL
 NUMBER           HZ       DIRECTION


ROTOR NUMBER       1     RELATIVE SPEED  1.00000E+00
              ASYNCRHONOUS                 SYNCHRONOUS                    SYNCHRONOUS
     3        4.34658E+01    BACKWARD    5  4.35508E+01    BACKWARD    12  2.05076E+02    FORWARD
     5        5.72010E+01    BACKWARD    7  5.72617E+01    BACKWARD    12  2.05076E+02    FORWARD
     7        5.70330E+01    BACKWARD    6  5.70888E+01    BACKWARD    12  2.05076E+02    FORWARD
     9        5.93875E+01    BACKWARD    8  5.95621E+01    BACKWARD    12  2.05076E+02    FORWARD

ROTOR NUMBER       2     RELATIVE SPEED  1.00000E+00
              ASYNCRHONOUS                 SYNCHRONOUS                    SYNCHRONOUS
     3        4.34658E+01    BACKWARD   14  4.35508E+01    BACKWARD    12  2.05076E+02    FORWARD
     5        5.72010E+01    BACKWARD   16  5.72617E+01    BACKWARD    12  2.05076E+02    FORWARD
     7        5.70330E+01    BACKWARD   15  5.70888E+01    BACKWARD    12  2.05076E+02    FORWARD
     9        5.93875E+01    BACKWARD   17  5.95621E+01    BACKWARD    12  2.05076E+02    FORWARD
```

Table 54 Whirl resonances calculated in the rotating system in asynchronous and synchronous analysis

FIXED SYSTEM		ROTATING SYSTEM		
ASYNC	SYNC	ASYNC	SYNC	
4.34658E+01	4.35508E+01	4.34658E+01	4.35508E+01	BACKWARD
4.34658E+01	4.35508E+01	4.34655E+01	4.35508E+01	FORWARD
5.72010E+01	5.72617E+01	5.72010E+01	5.72617E+01	BACKWARD
5.72010E+01	5.72617E+01	5.72009E+01	5.72617E+01	FORWARD
5.70330E+01	5.70888E+01	5.70330E+01	5.70888E+01	BACKWARD
2.04963E+02	2.05076E+02	2.04895E+02	2.05076E+02	FORWARD
5.93875E+01	5.95621E+01	5.93875E+01	5.95621E+01	BACKWARD

Table 55 Comparison of critical speed calculated in the fixed and rotating system in synchronous and asynchronous analysis

INSTABILITIES	SOLUTION NUMBER	ROTOR SPEED HZ	WHIRL DIRECTION
FIXED SYSTEM			
START	3	1.06553E+02	FORWARD
START	5	1.75484E+02	FORWARD
ROTATING SYSTEM			
START	2	1.06553E+02	FORWARD
START	4	1.75484E+02	FORWARD

Table 56 Instabilities found in the fixed and the rotating system

10.5.2 The Parameters W3R and W4R

Because the real eigenvalues are not calculated in the direct solutions, and because in the rotor dynamic solutions the complex stiffness matrix is not used, representative values of the frequency must be defined. They must be defined for each rotor individually. The values for W3R and W4R are in radians per second: $W3 = 2\pi\ f_3$ and $W4 = 2\pi\ f_4$ where f_3 and f_4 are representative frequencies in Hz. They are used for the following damping types:

W3 PARAM G damping
W4 MAT1 damping

If the values are zero in the modal solutions, the eigenvalues are used. If the values are zero in the direct solutions, the damping is set to zero.

Also in this case, analyses with the individual rotors were compared to the results with two rotors in the model:

Model with two rotors	r220ab444w
Model with rotor A	r120a444w
Model with rotor B	r120b444w

The following values were used:

Rotor A	500	approximately 80 Hz
Rotor B	600	approximately 95 Hz

The Campbell diagram is shown in Fig. 131 and the real parts in Fig. 132. The results are identical.

The eigenfrequencies for the two rotor model with and without W3R and W4R are shown in Fig. 133. The results are practically identical because the damping has only a small influence on the frequencies. The real parts are shown in Fig. 134. Because the stiffness proportional damping is divided by the W3R and W4R values, the damping for the modes with high frequency are too high. See Curve 10 in Fig. 134. The damping of the modes with lower eigenfrequencies will be too low. The real parts are shown with a different scale in Fig. 135. The results are similar, but somewhat different. The critical speeds are higher with W3R and W4R factors because the damping is lower.

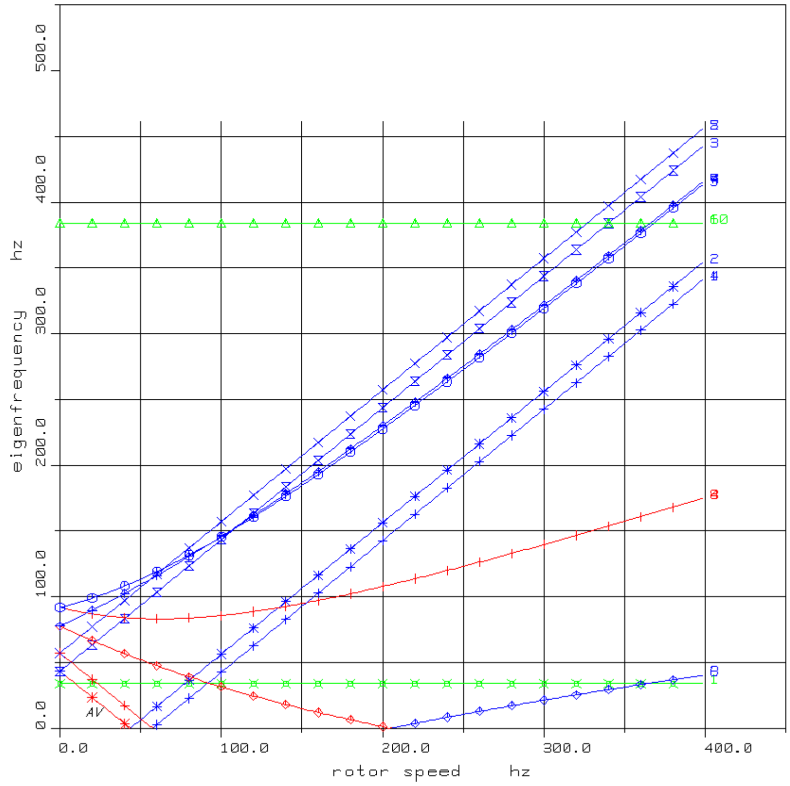


Fig. 131 W3R and W4R. Analysis with two rotors compared to two single analyses of each rotor (symbols)

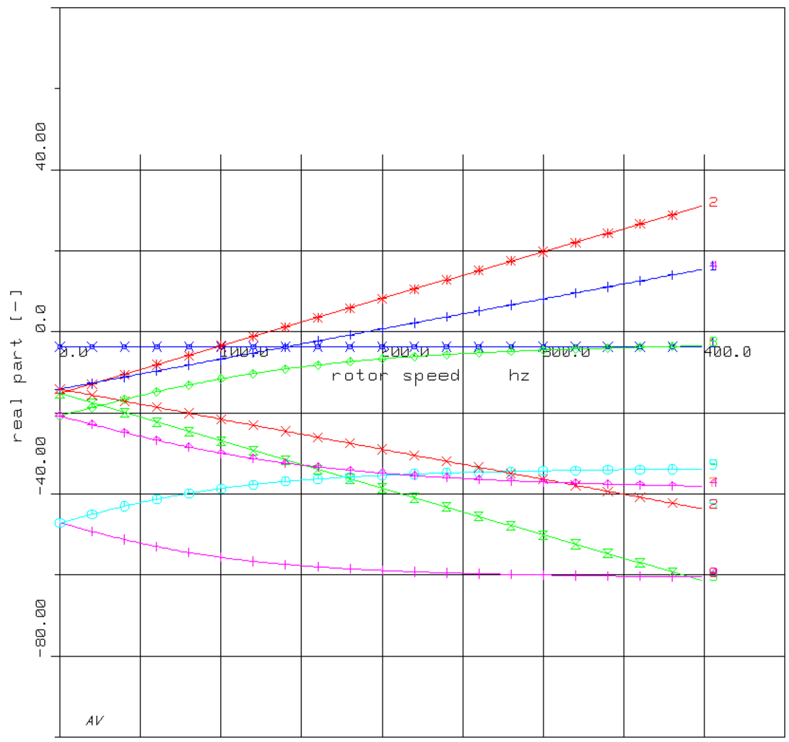


Fig. 132 W3R and W4R. Analysis with two rotors compared to two single analyses of each rotor (symbols)

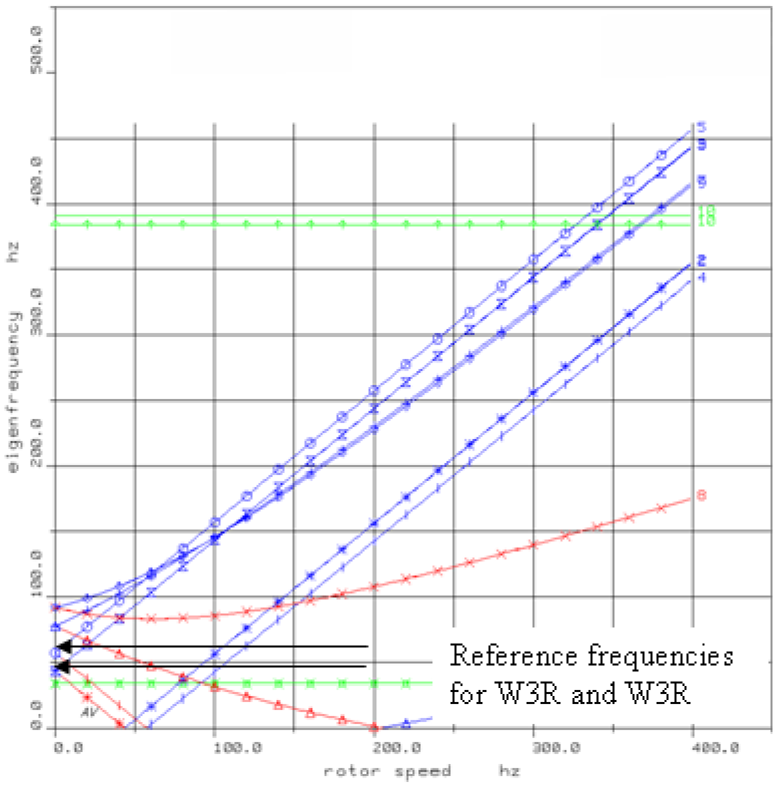


Fig. 133 SOL 110 without W3R and W4R compared to analysis with W3R and W4R (symbols)

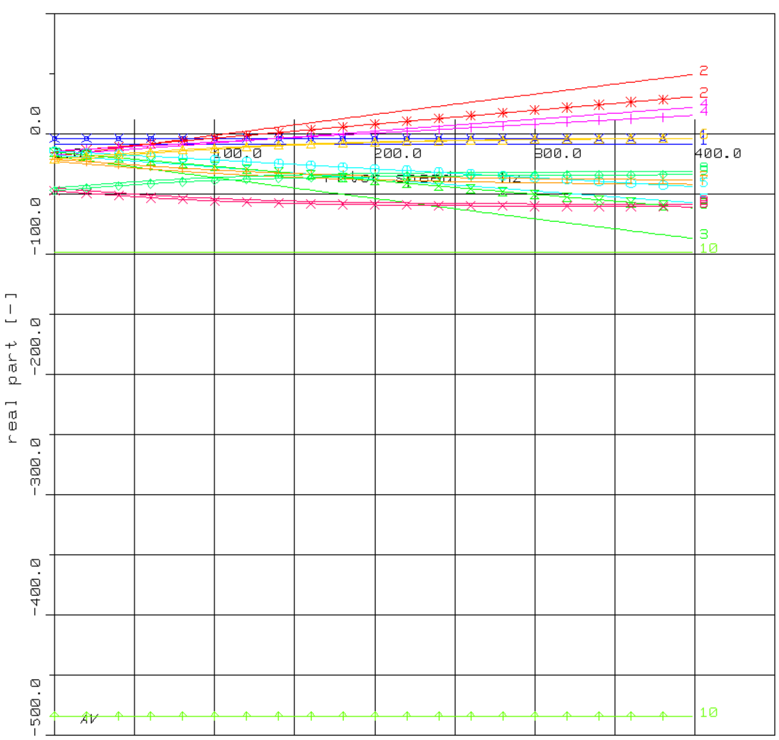


Fig. 134 SOL 110 without W3R and W4R compared to analysis with W3R and W4R (symbols)

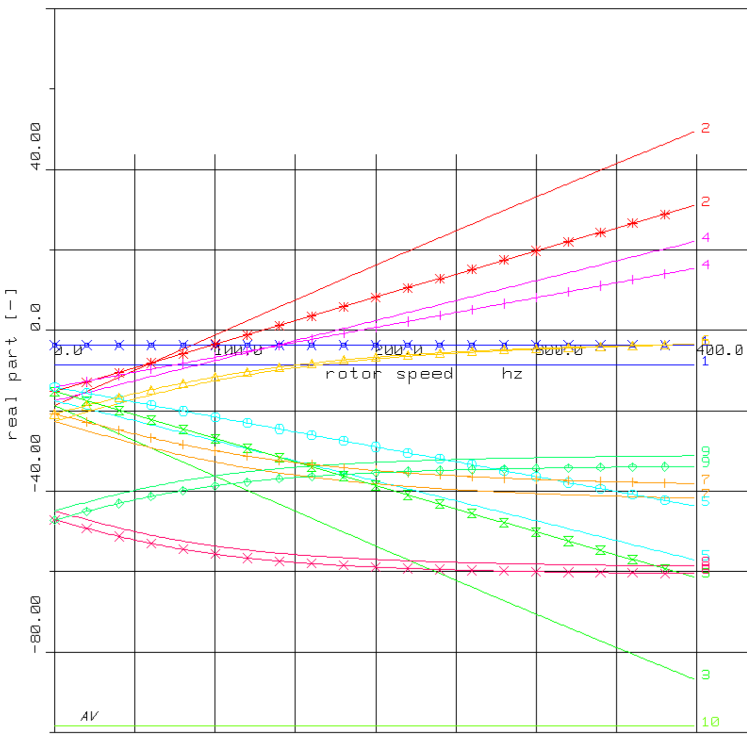


Fig. 135 SOL 110 without W3R and W4R compared to analysis with W3R and W4R (symbols)

10.5.3 Analysis with the Direct Method SOL 107

The analyses were also made with the direct method. The comparison between the fixed and the rotating system is shown in Fig. 136 for the eigenfrequencies and in Fig. 137 for the real eigenvalues. The results are practically identical. There is a small difference in the frequency of the forward frequency of the tilting mode of rotor A, probably due to numerical problems. There are also some numerical problems with the linear mode around 380 Hz.

A comparison between the results of SOL 110 (modal) and SOL 107 (direct) is provided in Fig. 138 for the eigenfrequencies and in Fig. 139 for the real part of the eigenvalues. The damping is shown in Fig. 140. There are small differences due to truncation errors in the modal analysis. The results with the number of real modes in SOL 110 increased to 20 instead of 10 are shown in Fig. 141. Here the frequencies are much closer than in Fig. 138. The same comparisons between SOL 110 and SOL 107 are shown in Fig. 142 and Fig. 143 for the eigenfrequencies and the real part, respectively. Also here, the difference is due to truncation errors in the modal analysis.

With SOL 107 there may be difficulties in obtaining the correct complex modes. Therefore, the whirling direction may be wrong. This can be seen in Fig. 142 for some lines which shift in colour for the direct solution. This can be overcome by selecting the single vector Lanczos method with the following command:

NASTRAN SYSTEM(108)=2 $

With this option, the eigenvectors and the whirl directions are correct as shown in Fig. 144. Also, the eigenfrequencies converted to the fixed system are correct as shown in Fig. 145. In this case, all complex eigenvalues were found. Fig. 146 shows the result of the mode tracking with the solutions as symbols. With the default method (Block Lanczos) the whirl directions are not correct as shown in Fig. 147. Therefore, the conversion to the fixed system does not work as shown in Fig. 148. There are also missing solutions between 200 and 216 Hz rotor speed. The solutions are shown in Fig. 149 where the symbols are missing in this range.

For the analysis in the fixed system there may be numerical difficulties with the default method. Fig. 150 shows the eigenfrequencies with SYSTEM(108)=2. The damping is shown in Fig. 151 and the converted frequencies in Fig. 152. The solutions are correct and all solutions were found for all rotor speeds as shown in Fig. 153. With the default method shown in Fig. 154 the mode tracking stops at 60 Hz rotor speed because the solutions are missing. The mode tracking resumes the curves after this speed as new solutions. The conversion to the rotating system works as shown in Fig. 155, but the imaginary parts of the complex modes are very small. The damping curves are shown in Fig. 156. Here there is a slot in the curves at 60 Hz rotor speed. The solutions are shown in Fig. 157. It can be seen that solutions are missing at 60 Hz rotor speed.

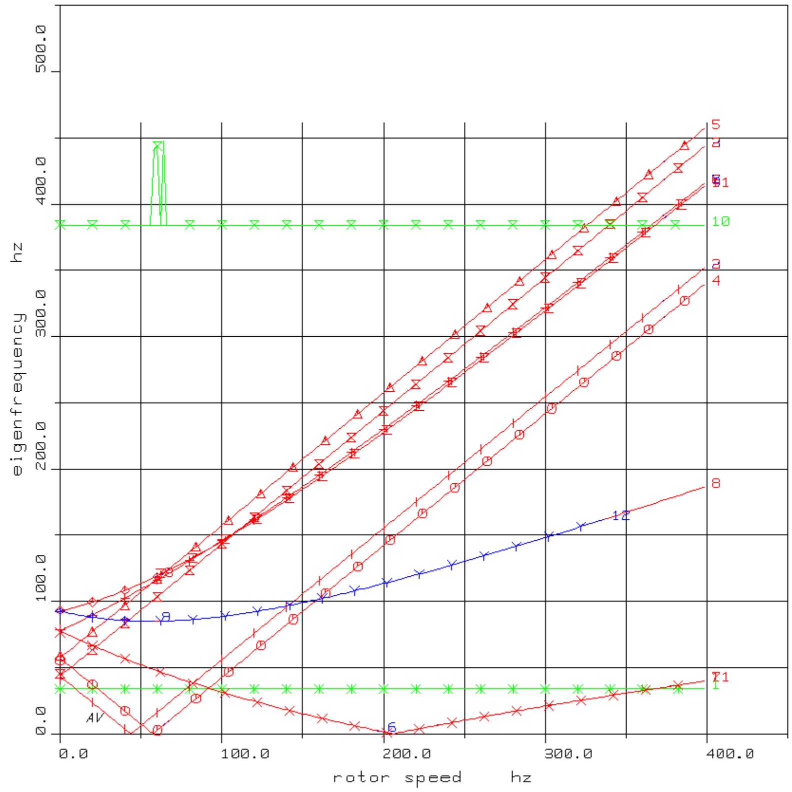


Fig. 136 SOL 107 in the rotating system. Comparison of analyses in rotating and fixed system

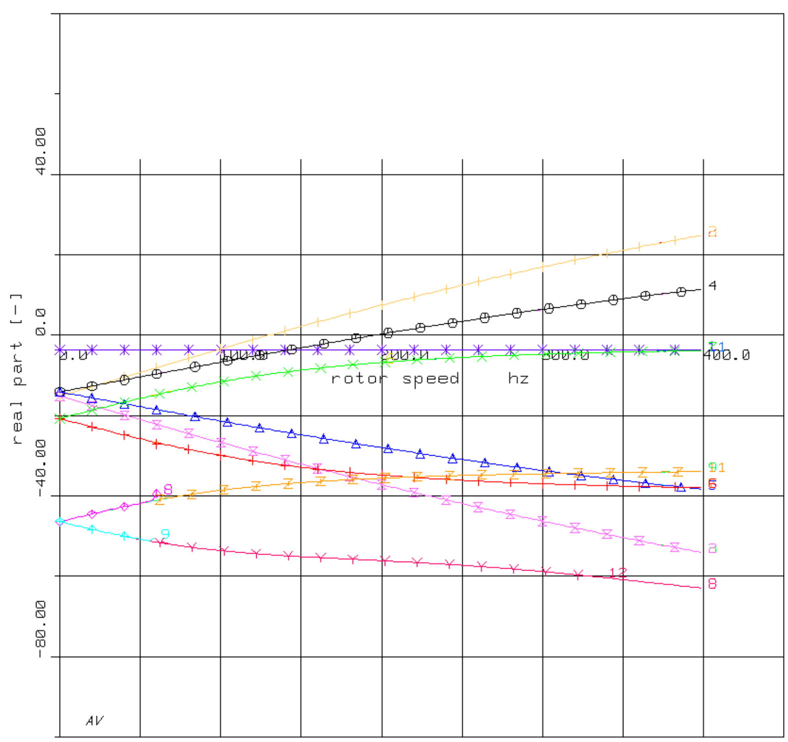


Fig. 137 Real eigenvalues, SOL 107. Comparison of analyses in rotating and fixed system

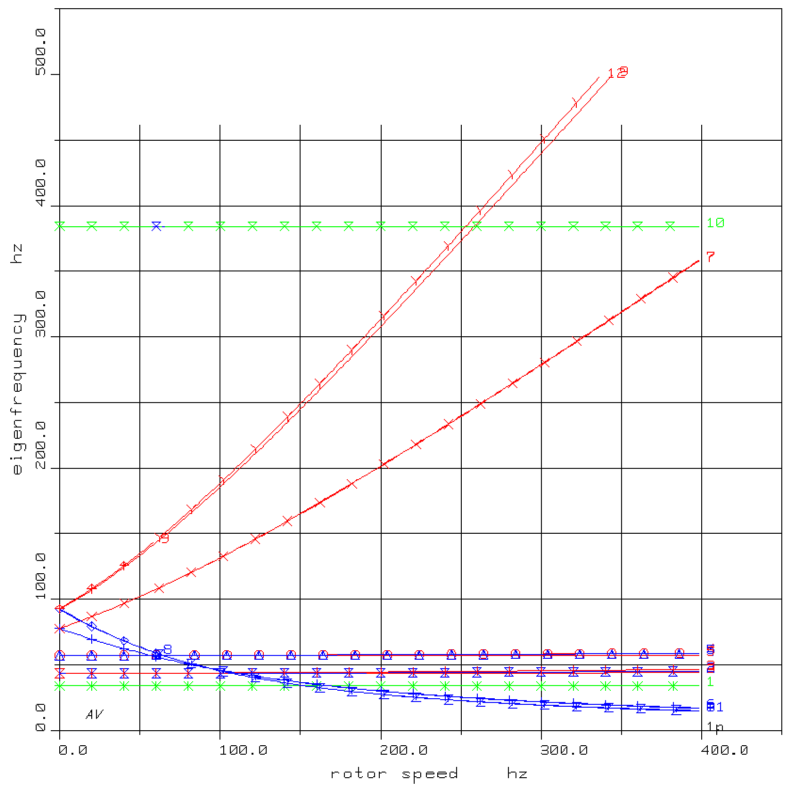


Fig. 138 Eigenfrequencies, comparison of SOL 110 and 107 (symbols) in the fixed system

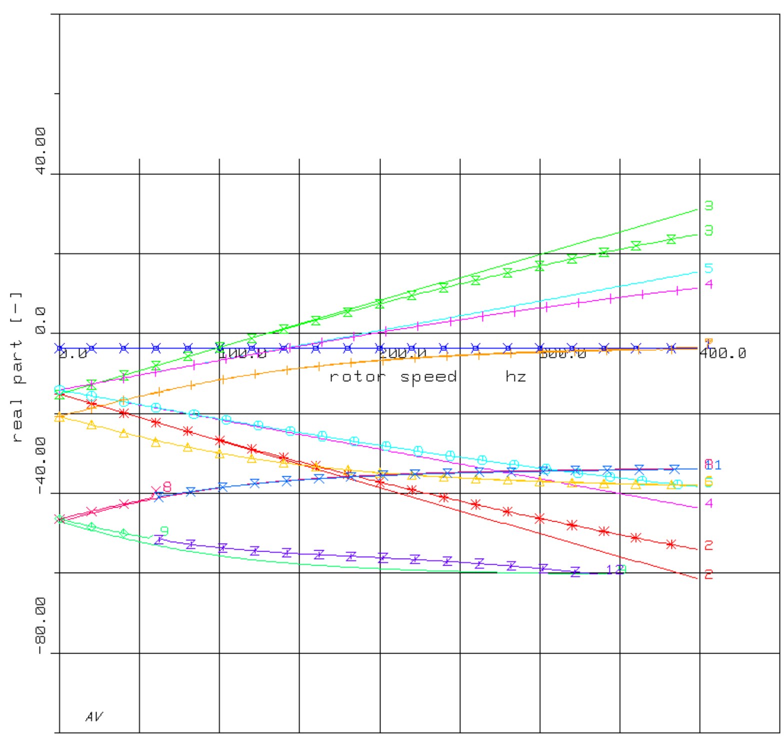


Fig. 139 Real eigenvalues, comparison of SOL 110 and 107 (symbols) in the fixed system

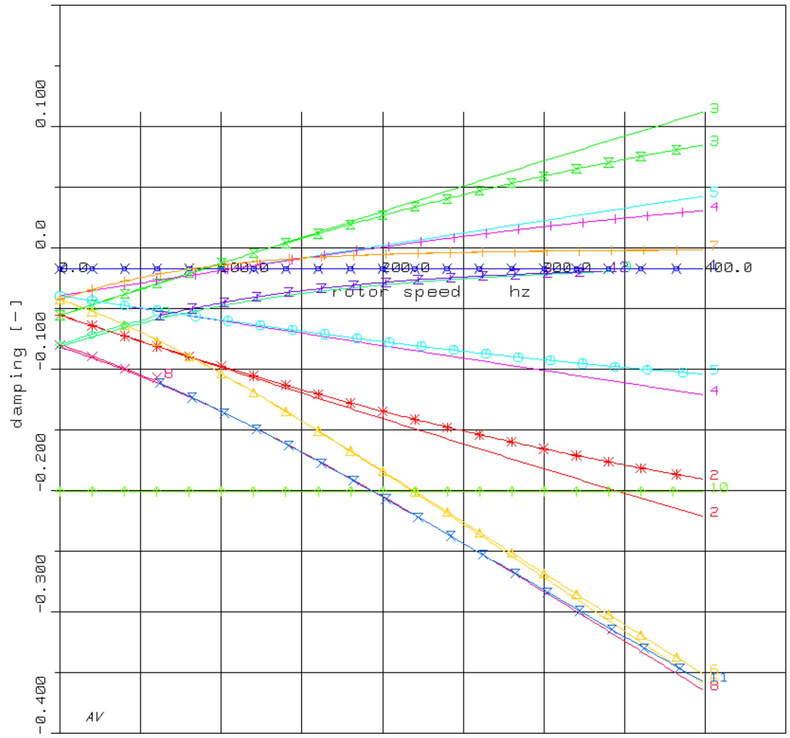


Fig. 140 Damping, comparison of SOL 110 and 107 (symbols) in the fixed system

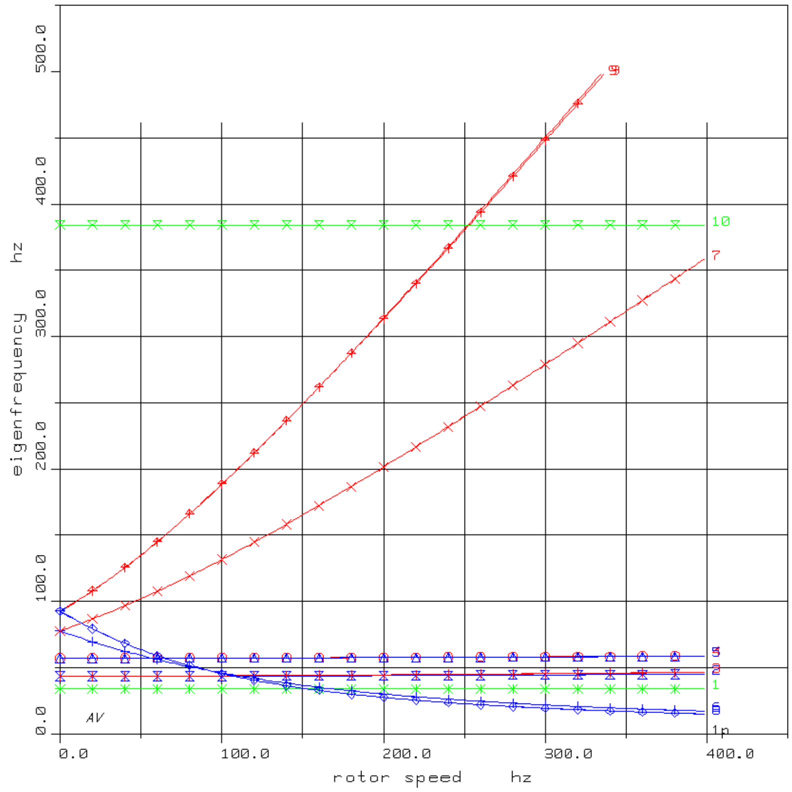


Fig. 141 Eigenfrequencies, comparison of SOL 110 with 20 modes and 107 (symbols) in the fixed system

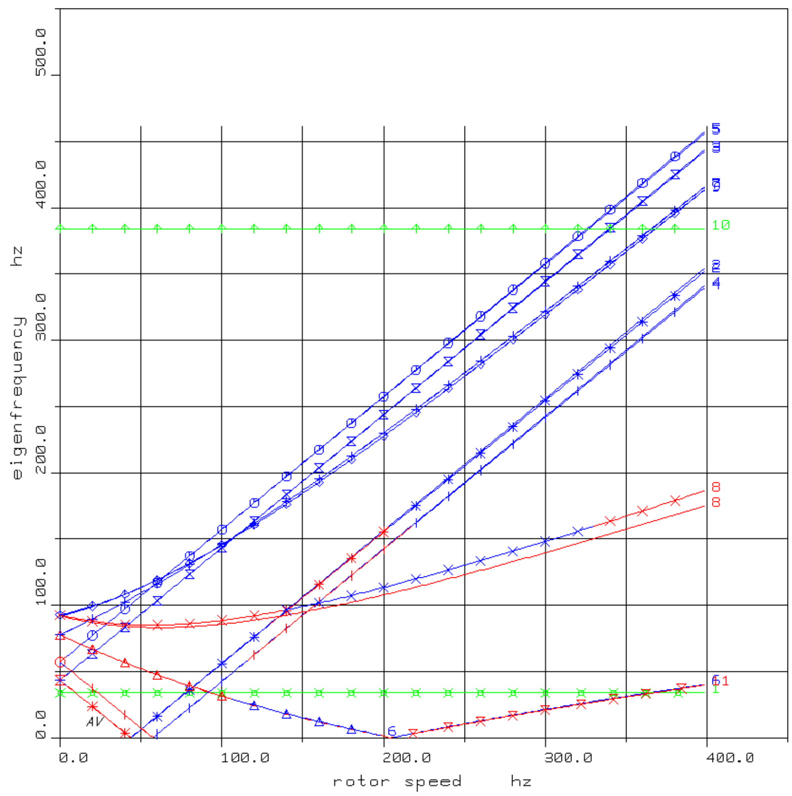


Fig. 142 Eigenfrequencies, comparison of SOL 110 and 107 (symbols) in the rotating system

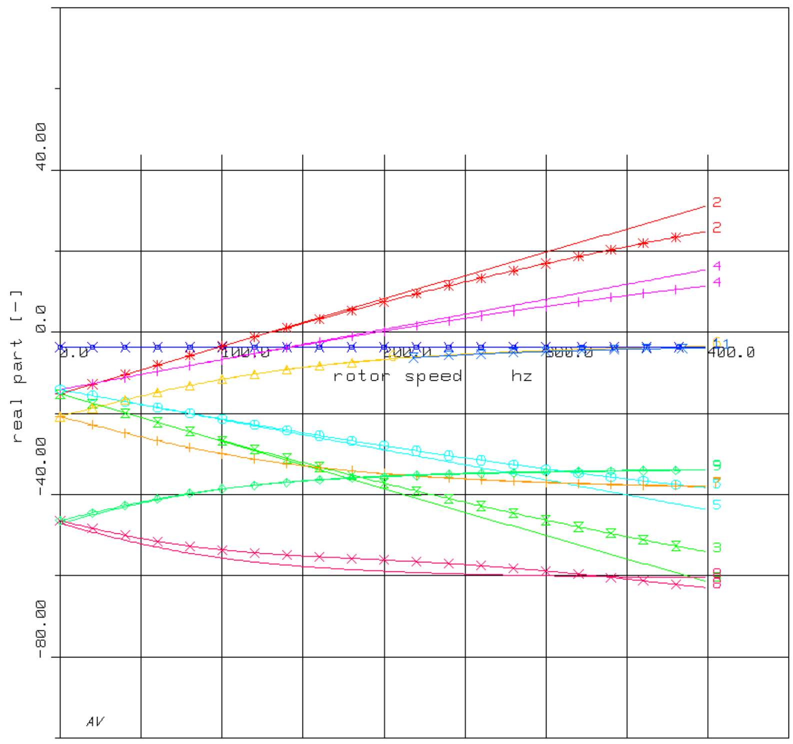


Fig. 143 Real eigenvalues, comparison of SOL 110 and 107 (symbols) in the rotating system

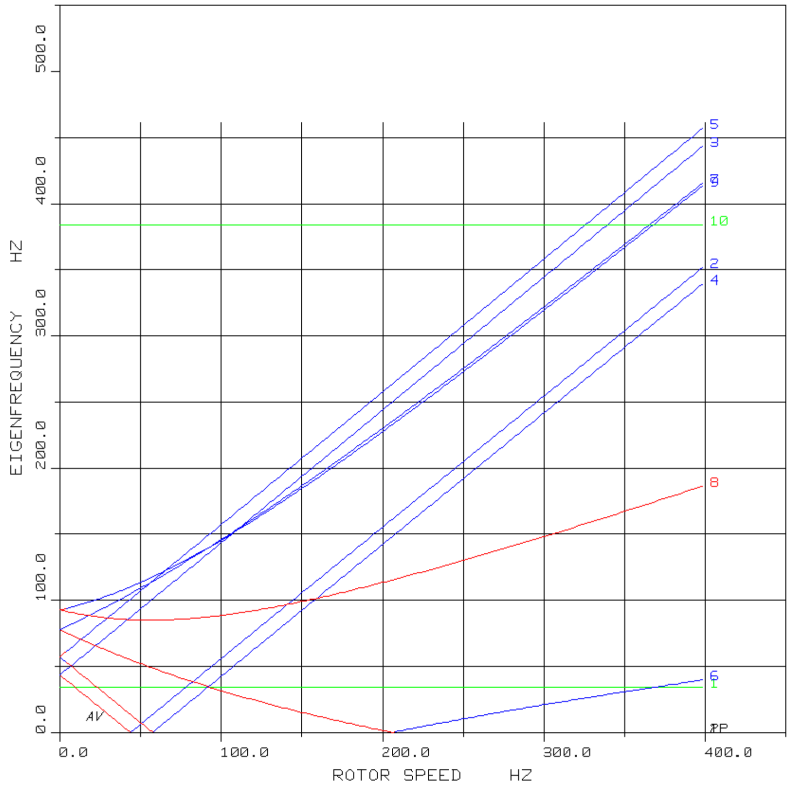


Fig. 144 Rotating system, SOL 107 with SYSTEM(108)=2

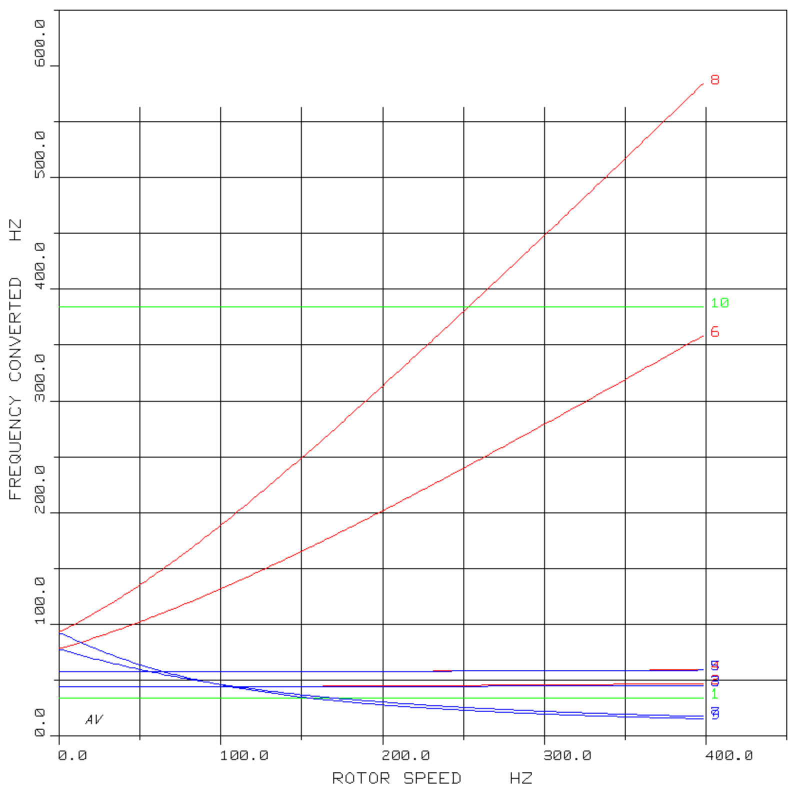


Fig. 145 Rotating system, SOL 107 with SYSTEM(108)=2. Converted eigenfrequencies

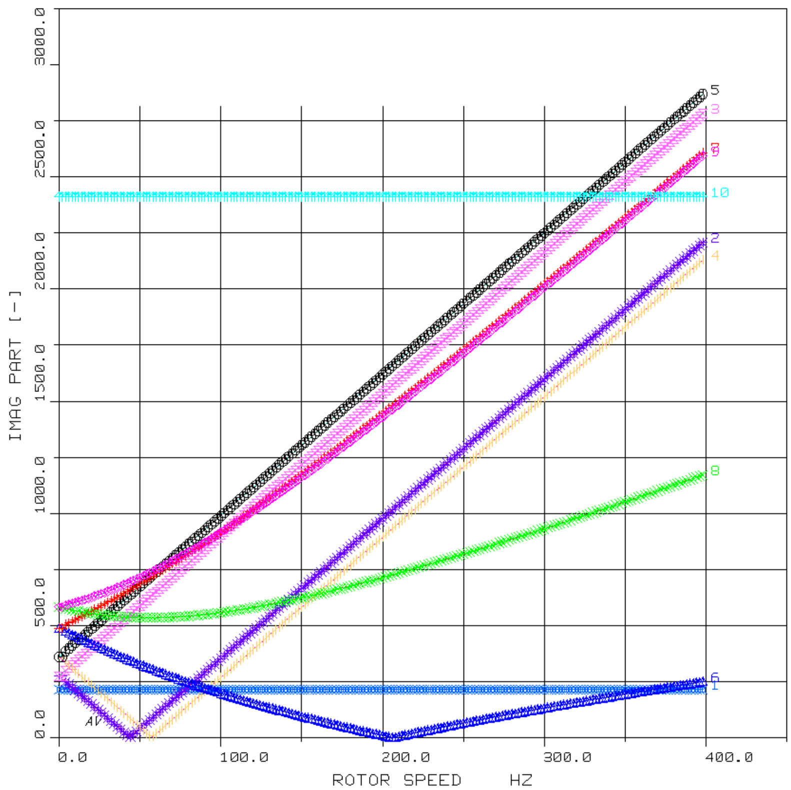


Fig. 146 Rotating system, SOL 107 with SYSTEM(108)=2. All solutions found for all speeds

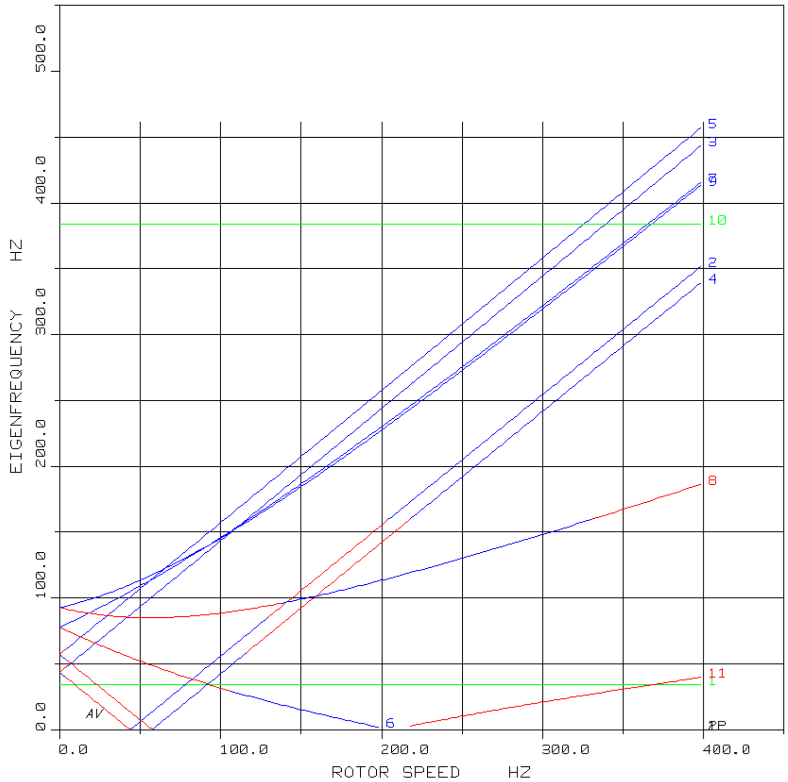


Fig. 147 Rotating system, SOL 107. Solution 6 is missing between 200 and 216 RPM

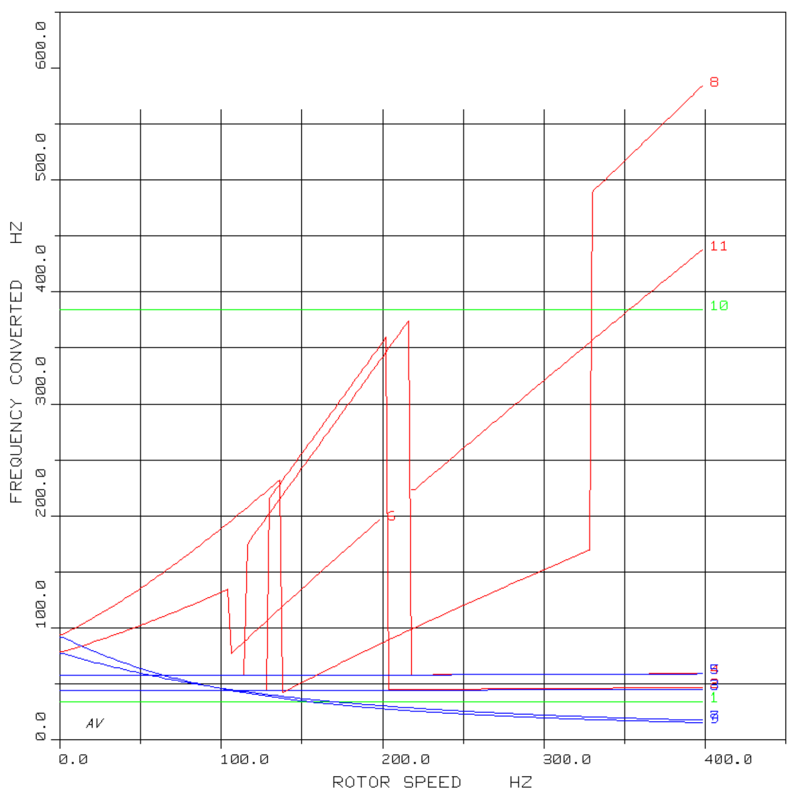


Fig. 148 Rotating system, SOL 107. Eigenvectors are not correct and whirl direction is wrong

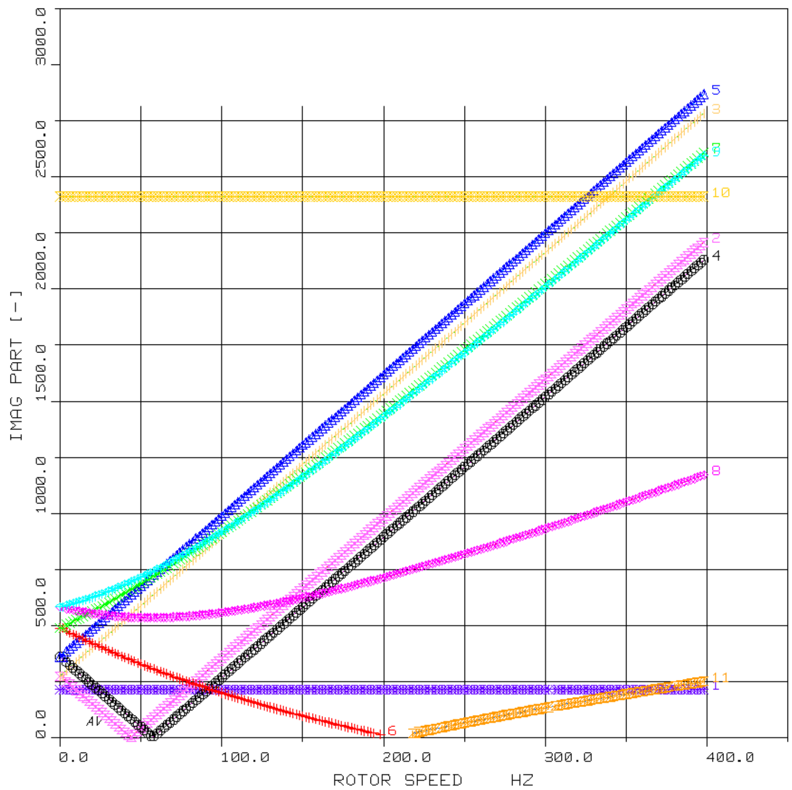


Fig. 149 Missing solutions

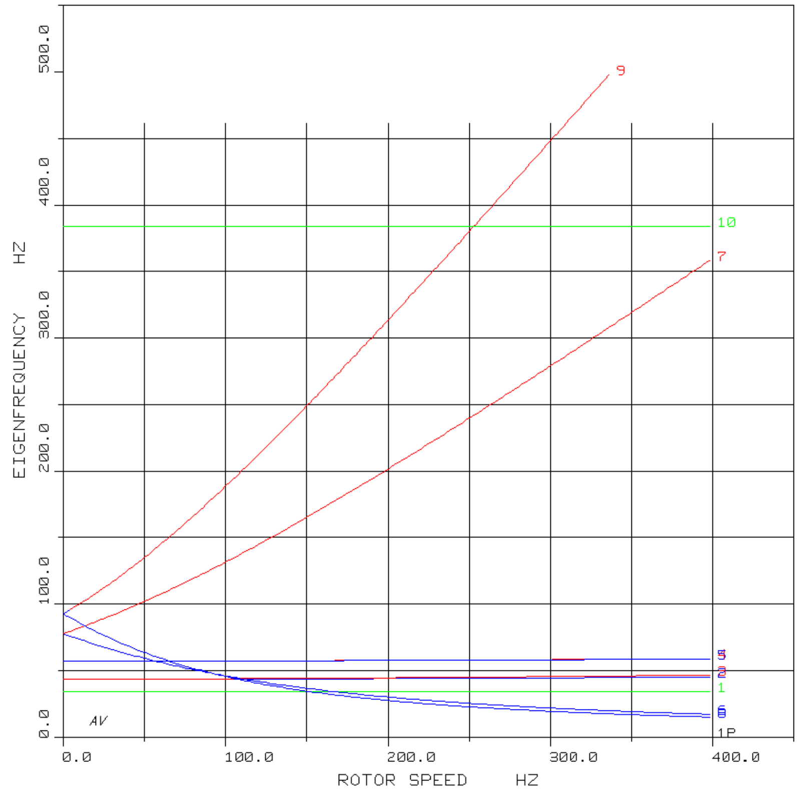


Fig. 150 Eigenfrequencies, SYSTEM(108)=2

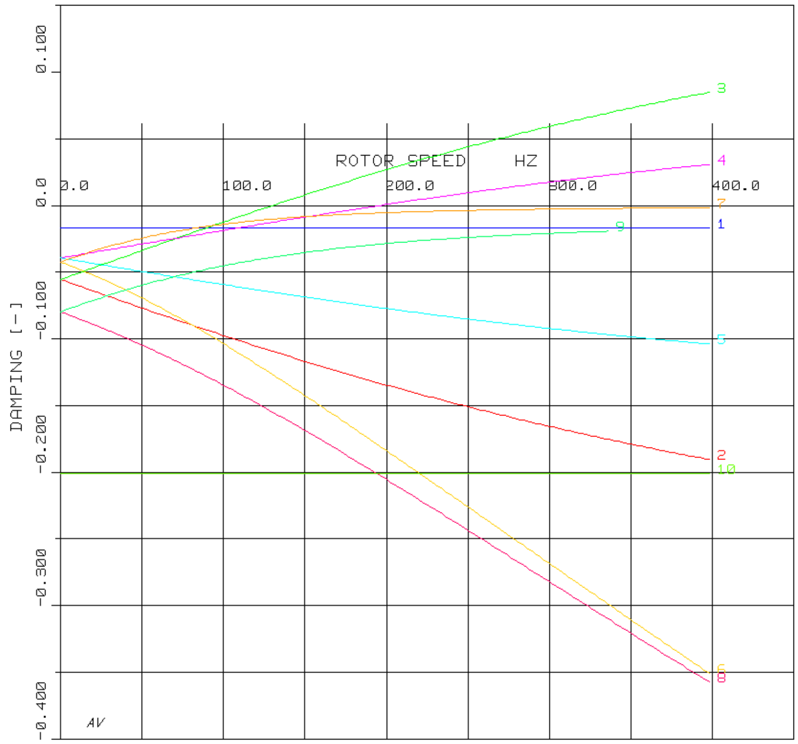


Fig. 151 Damping, SYSTEM(108)=2

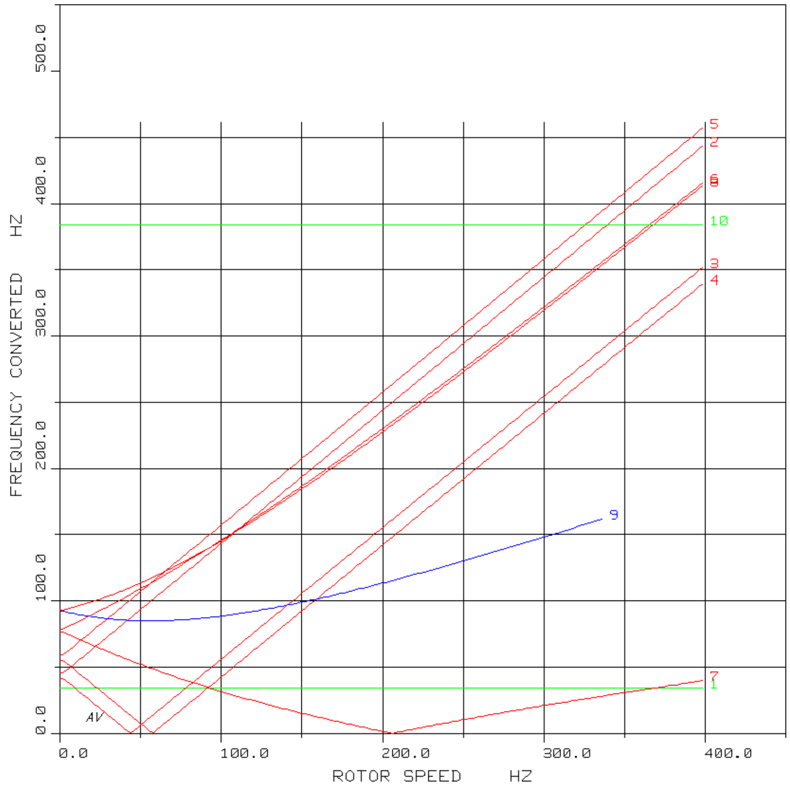


Fig. 152 Converted frequencies, SYSTEM(108)=2

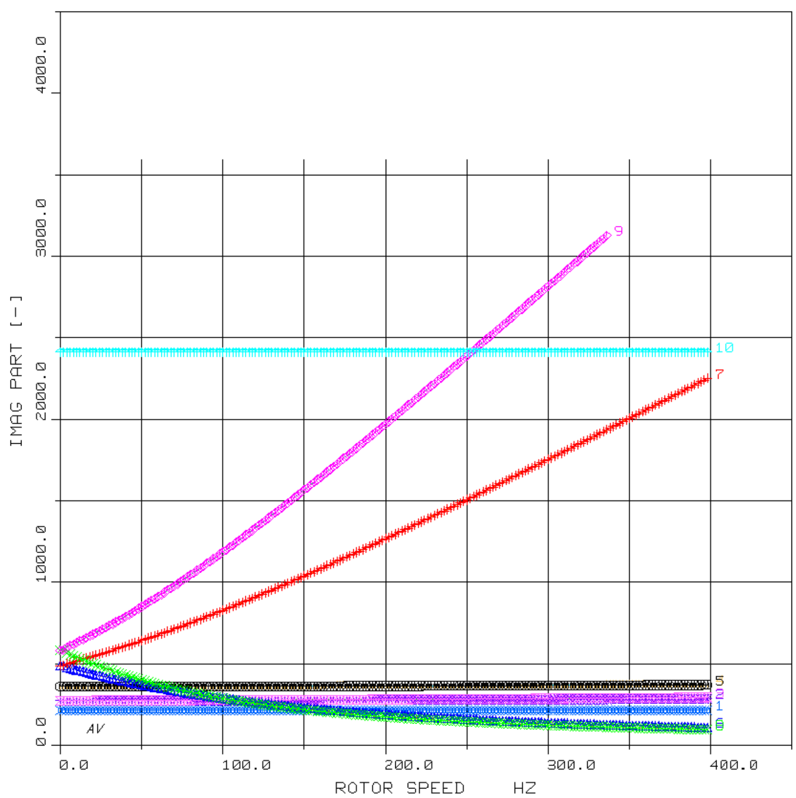


Fig. 153 All solutions found for all speeds, SYSTEM(108)=2

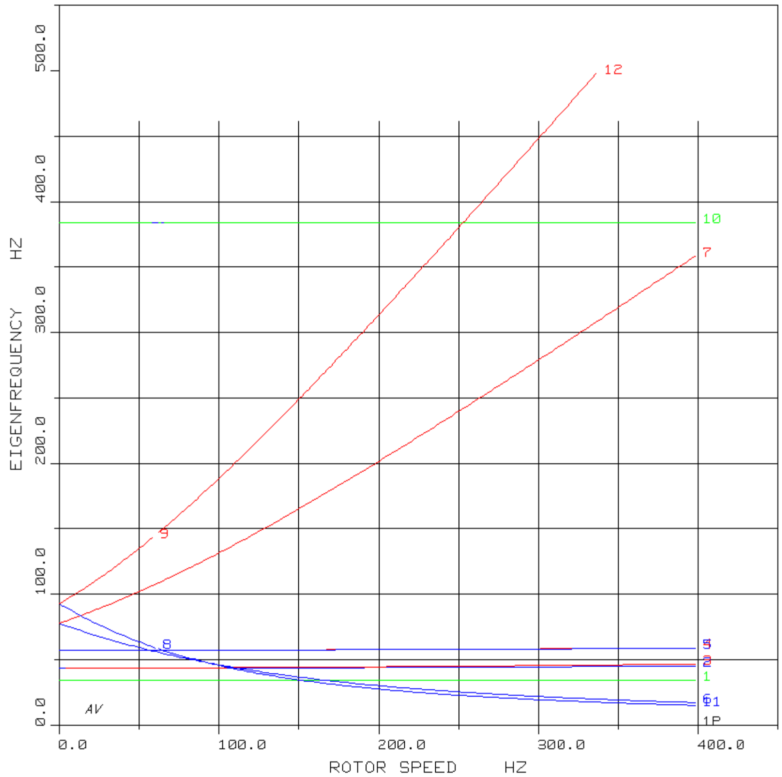


Fig. 154 Eigenfrequencies, Solutions missing at 60 RPM

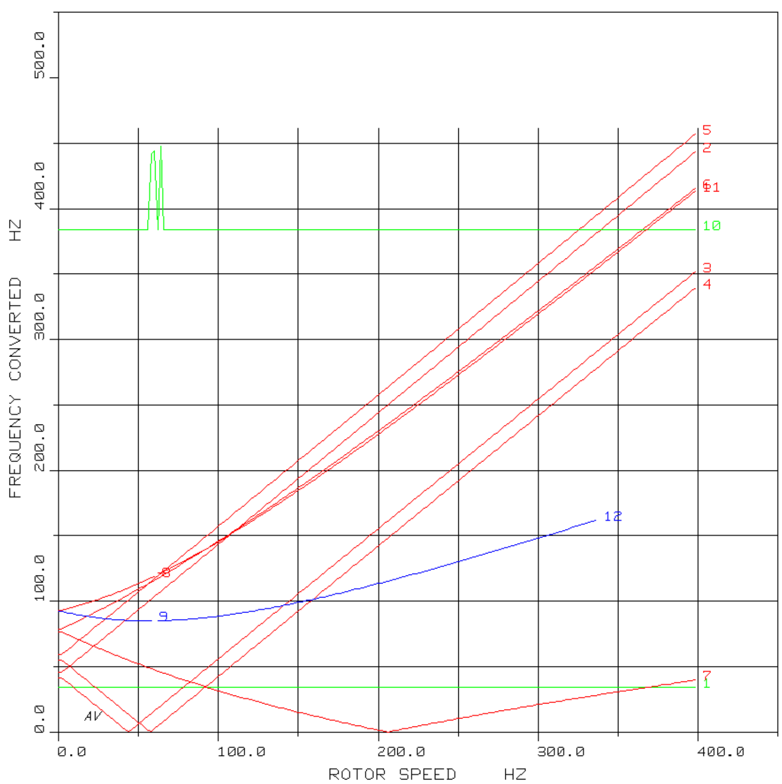


Fig. 155 Converted to rotating system

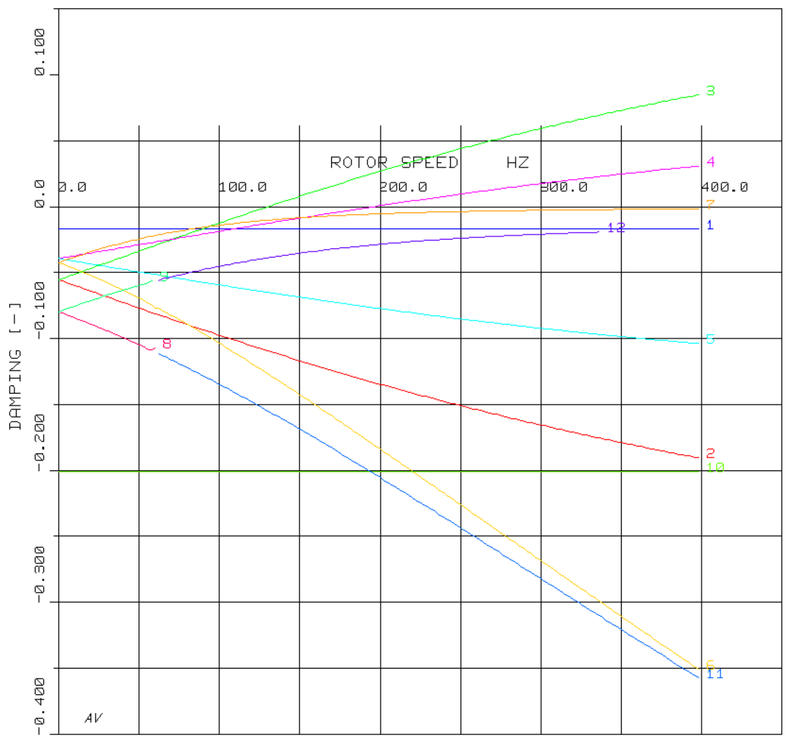


Fig. 156 Damping

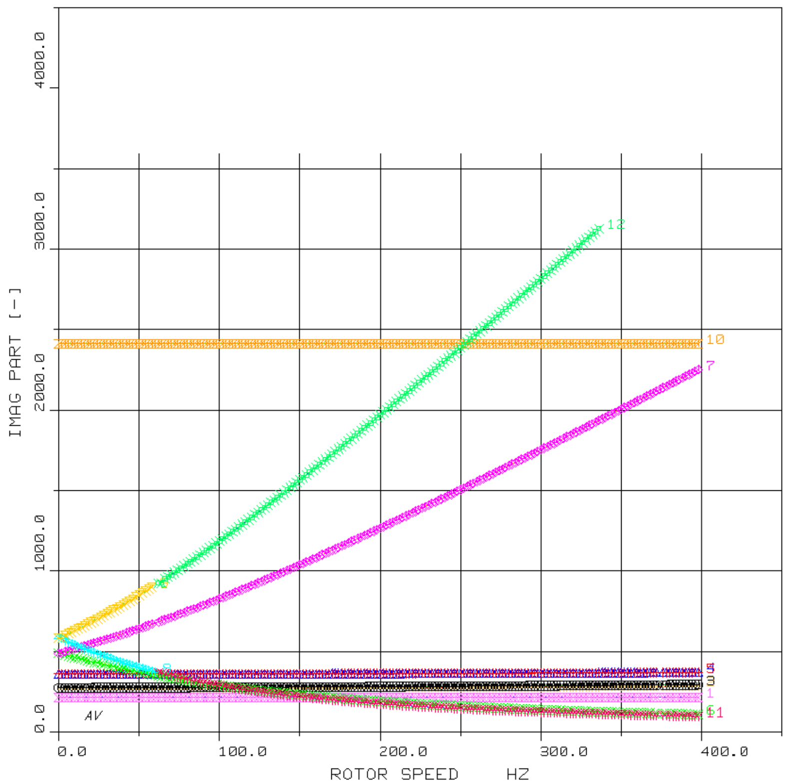


Fig. 157 Solutions missing at 60 RPM

10.6 Relative Rotor Speed

The model was also analyzed with different relative rotor speeds and compared to the equivalent single rotor models.

10.6.1 Single Rotor Models

The Campbell diagram for rotor A with different relative rotor speeds are shown in Fig. 158 with speeds of 1.2 (denoted by fast), 0.8 (denoted by slow) compared to the reference case of factor 1.0. The rotor speeds are relative to the speed entered on the ROTORD entry. Scaling the speeds by the appropriate factors, the curves of Fig. 159 are obtained. The eigenfrequencies are identical. The damping curves are shown in Fig. 160 and the scaled values in Fig. 161. The scaled damping values are identical. Similar curves for the real part of the eigenvalues are shown in Fig. 162 and Fig. 163. This rotor has no critical speed for the forward tilting mode.

A similar analysis was done for rotor B. The eigenfrequencies are shown in Fig. 164 together with the excitation lines for each rotor. The critical speed of the forward tilting mode is 205.1 Hz for the reference case, 170.9 Hz (170.9 x 1.2 = 205.1) for the fast rotor and 256.3 Hz (256.3 x 0.8 = 205.0) for the slow rotor. Scaling the speed, all rotors have the critical speed at 205.1 Hz as shown in Fig. 165. The 1P line is the frequency equal to the reference speed defined on the ROTORD entry. Looking at the crossing with this line, the critical speed for the reference case is at 205.1 Hz shown with the horizontal arrow in Fig. 164. The fast rotor has no crossing with the reference speed defined on the ROTORD 1P line and the fast rotor has a crossing at 140 Hz. The excitation of the rotor is, however, in the real rotor speed. A synchronous analysis with SOL 111 with the slow rotor and with RSPEED = 0.8 and EORDER = 1.0 (excitation order) in Fig. 166 shows a peak at 140 Hz, but here the excitation is faster than the real rotor speed. With respect to the real rotor speed, the excitation is 1/0.8=1.25. A synchronous analysis with EORDER = 0.8 is shown in Fig. 167. Here the peak is at 256 Hz as it should be. Because it is physically the same rotor, the critical speed is in reality at 205 Hz as shown in Fig. 165. In the output, the values of Fig. 164 are written out. They are listed in Table 47. These are the critical speeds with respect to the to the reference speed defined on the ROTORD l speed. This is useful for models with several rotors running at different speeds.

The different models are:

r121a444w_110	RSPEED = 1.0
r121aws1_110	RSPEED = 1.2
r121aws2_110	RSPEED = 0.8
r121b444w_110	RSPEED = 1.0
r121bws1_110	RSPEED = 1.2
r121bws2_110	RSPEED = 0.8

```
CRITICAL SPEEDS FROM SYNCHRONOUS ANALYSIS
                       SOLUTION      ROTOR SPEED     WHIRL
                        NUMBER           HZ        DIRECTION

 RSPEED = 0.8
                             1       7.13610E+01    BACKWARD
                             2       7.15770E+01    BACKWARD
                             3       7.15770E+01    FORWARD
                             4       2.56345E+02    FORWARD

RSPEED = 1.0
                             1       5.70888E+01    BACKWARD
                             2       5.72616E+01    LINEAR
                             3       5.72616E+01    LINEAR
                             4       2.05076E+02    FORWARD

RSPEED = 1.2
                             1       4.75740E+01    BACKWARD
                             2       4.77180E+01    LINEAR
                             3       4.77180E+01    LINEAR
                             4       1.70897E+02    FORWARD
```

Table 57 Critical speeds for different relative speed of the rotor

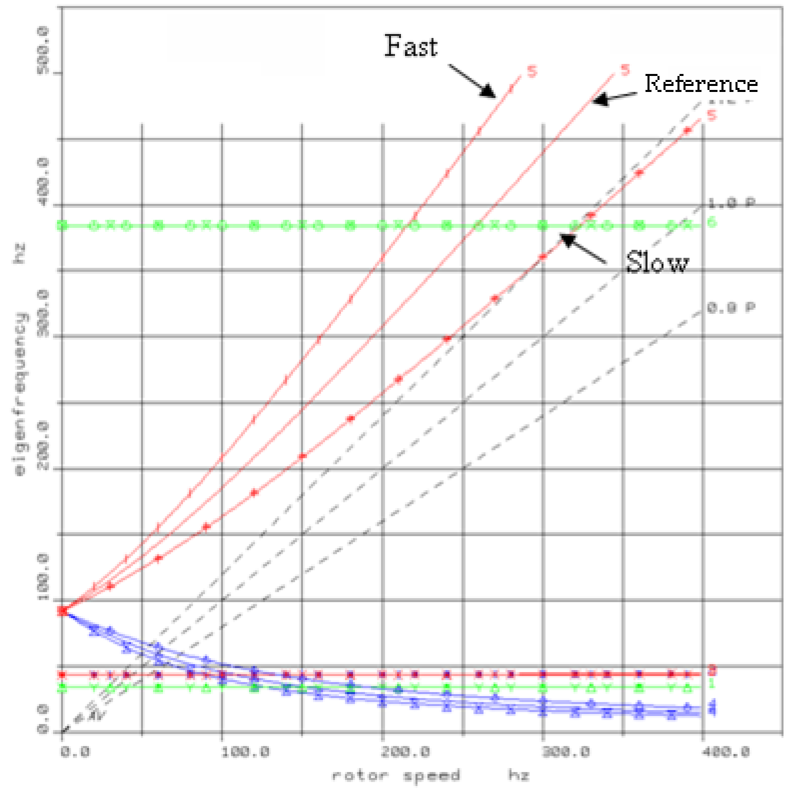


Fig. 158 Rotor A with relative speeds of 0.8, 1.0 and 1.2

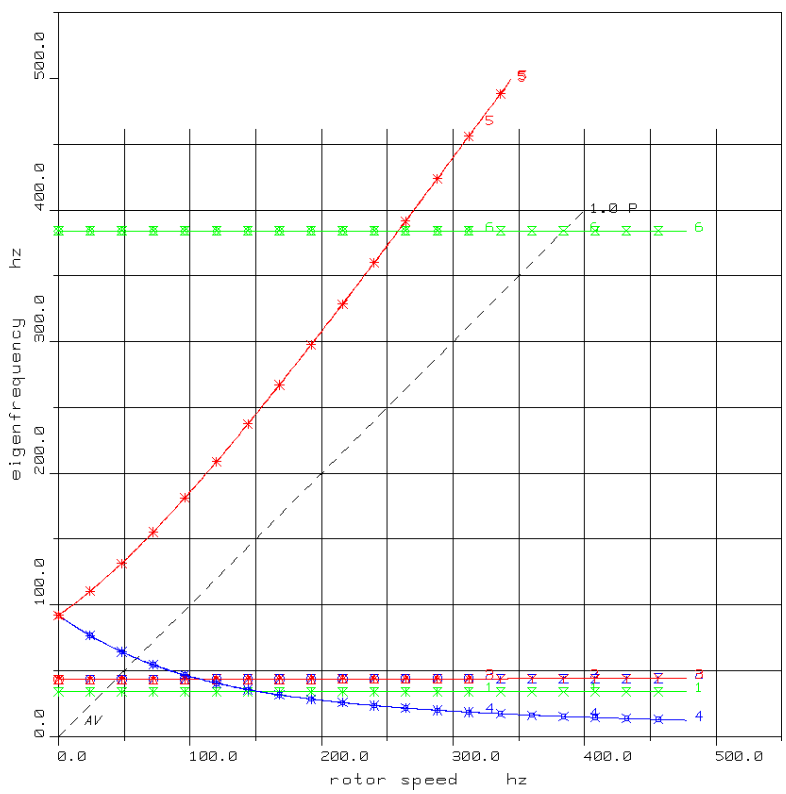


Fig. 159 Rotor A with relative speeds of 0.8, 1.0 and 1.2, scaled to to the reference speed defined on the ROTORD speed

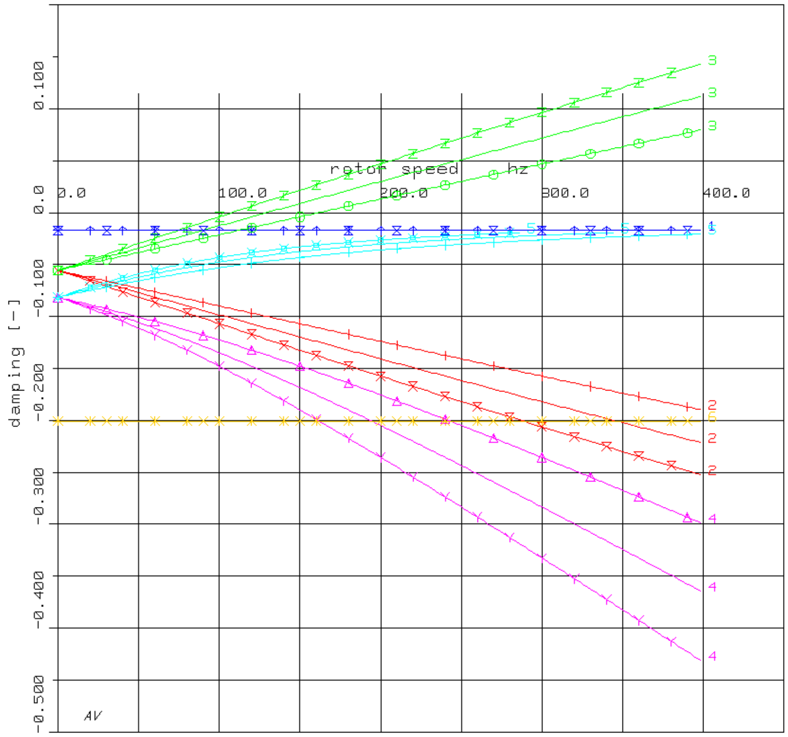


Fig. 160 Damping, Rotor A with relative speeds of 0.8, 1.0 and 1.2

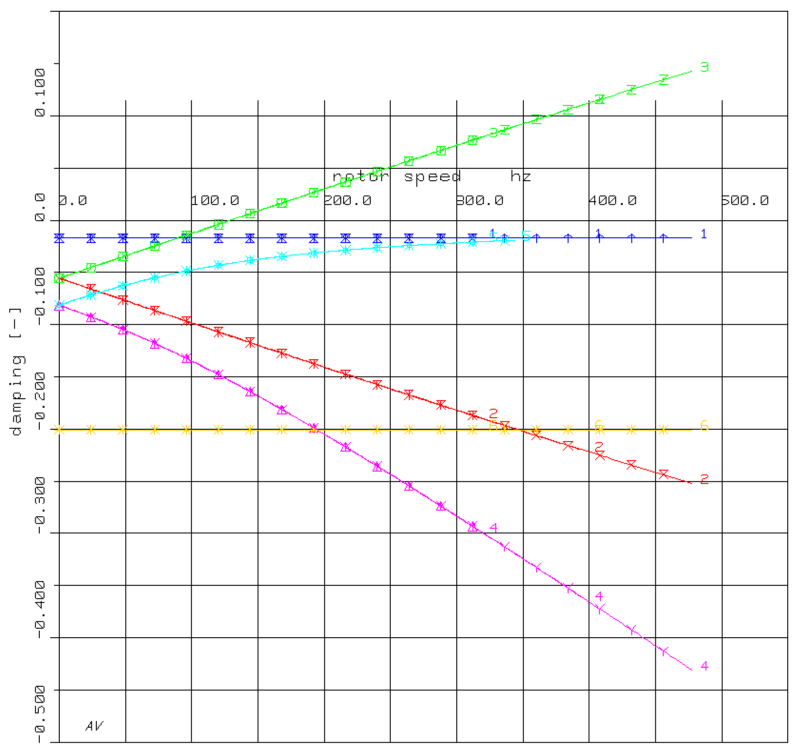


Fig. 161 Damping, relative speeds of 0.8, 1.0 and 1.2, scaled to to the reference speed defined on the ROTORD speed

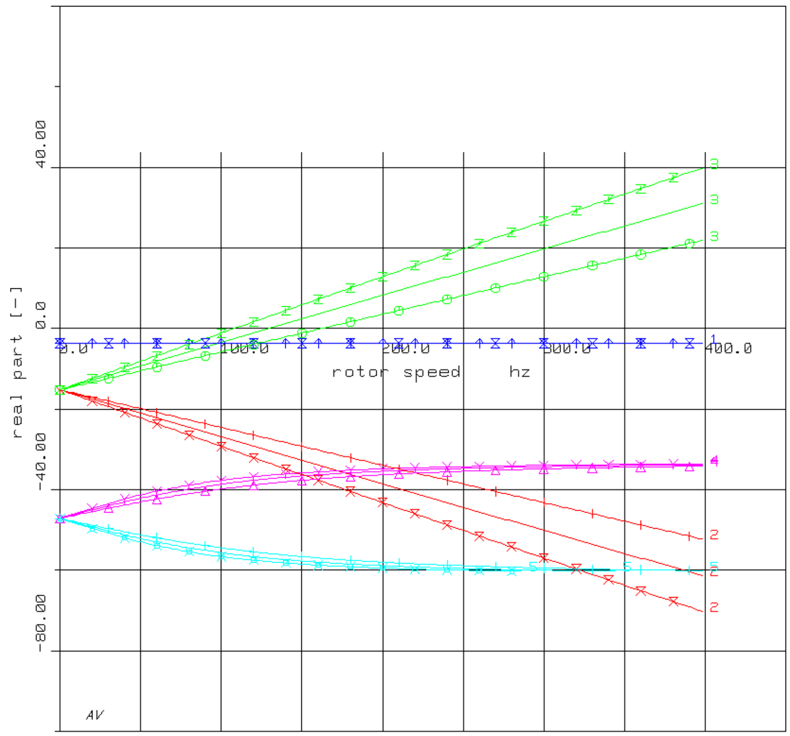


Fig. 162 Real part, relative speeds of 0.8, 1.0 and 1.2

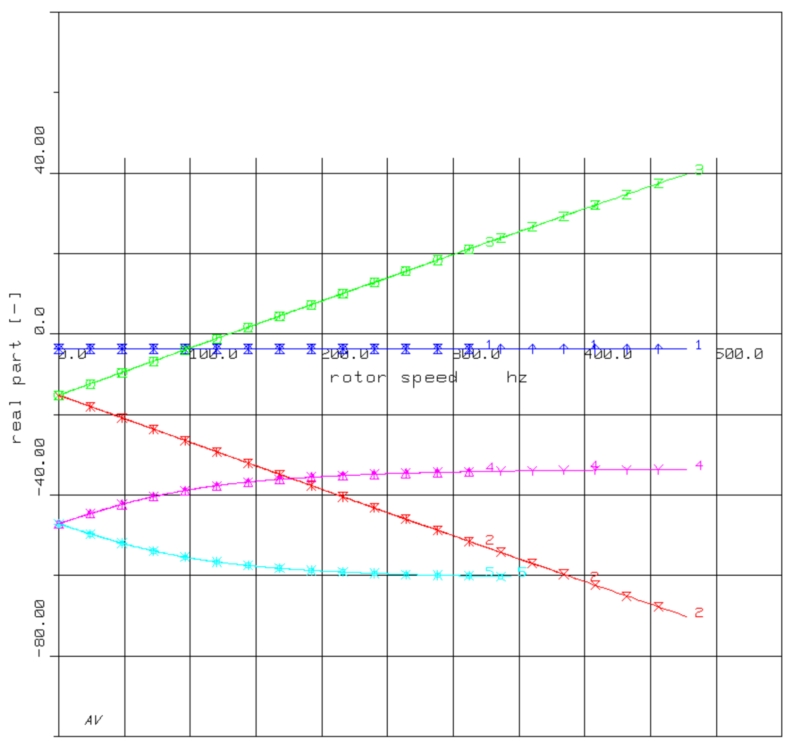


Fig. 163 Real part, relative speeds of 0.8, 1.0 and 1.2, scaled to to the reference speed defined on the ROTORD speed

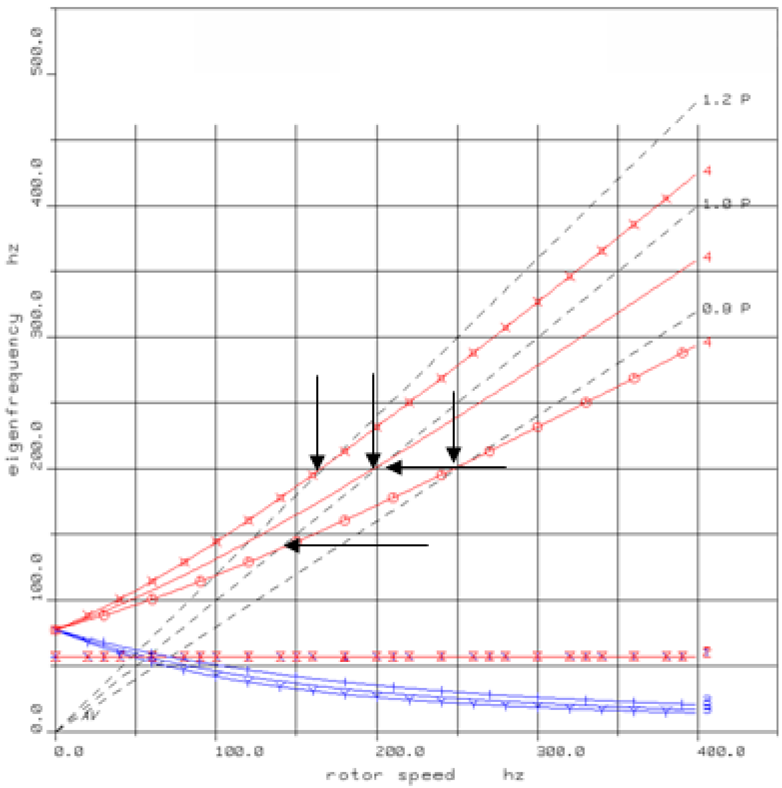


Fig. 164 Rotor B with relative speeds of 0.8, 1.0 and 1.2 and excitation lines

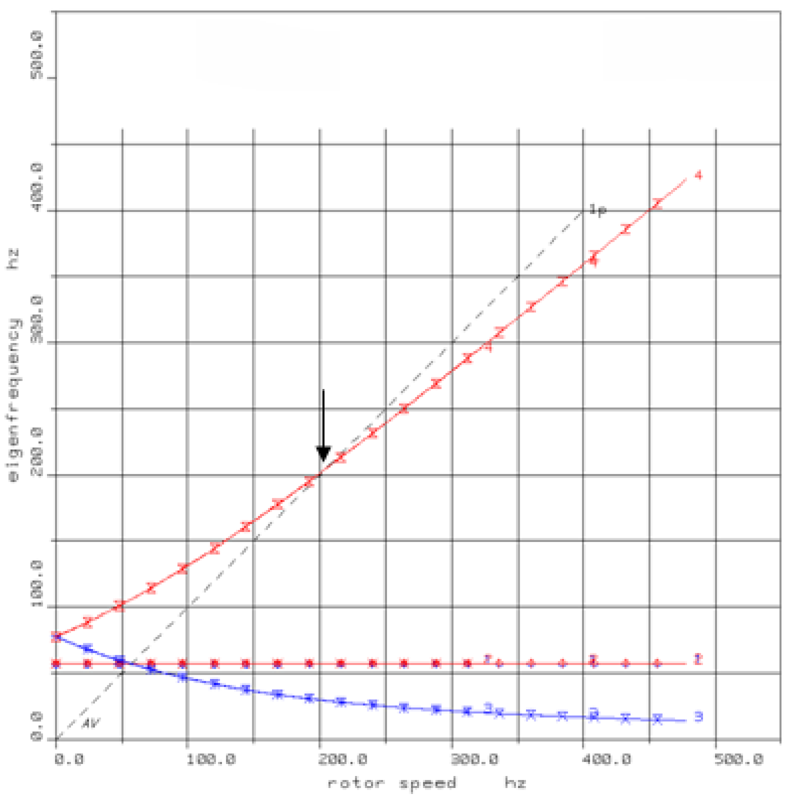


Fig. 165 Rotor B with relative speeds of 0.8, 1.0 and 1.2 scaled to to the reference speed defined on the ROTORD speed

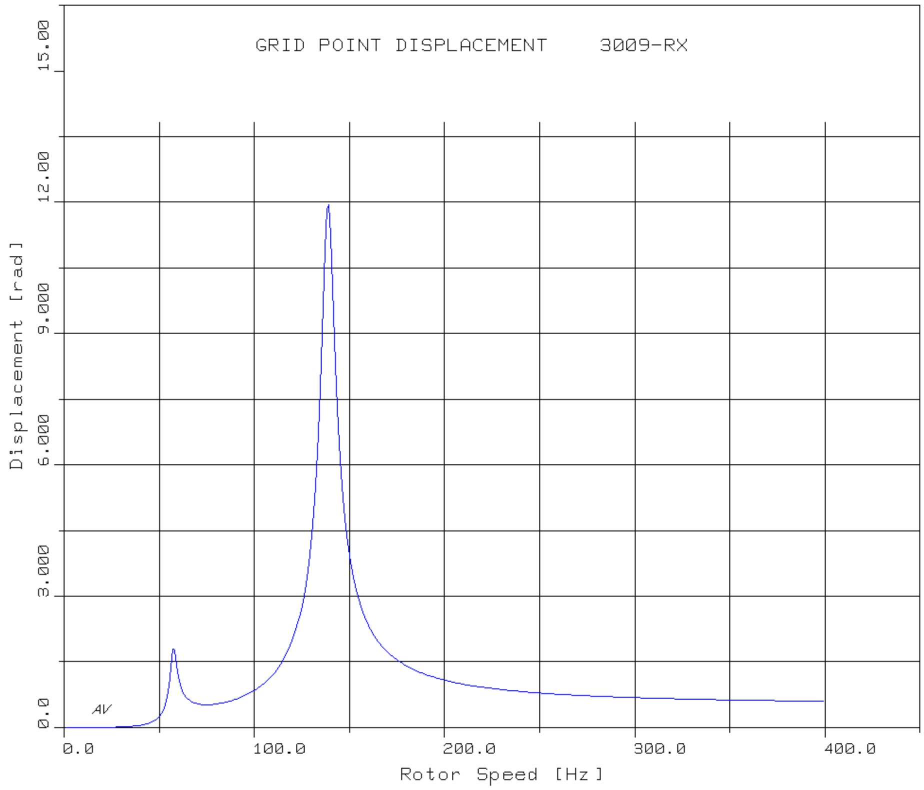


Fig. 166 Slow rotor RSPEED = 0.8 and EORDER = 1.0

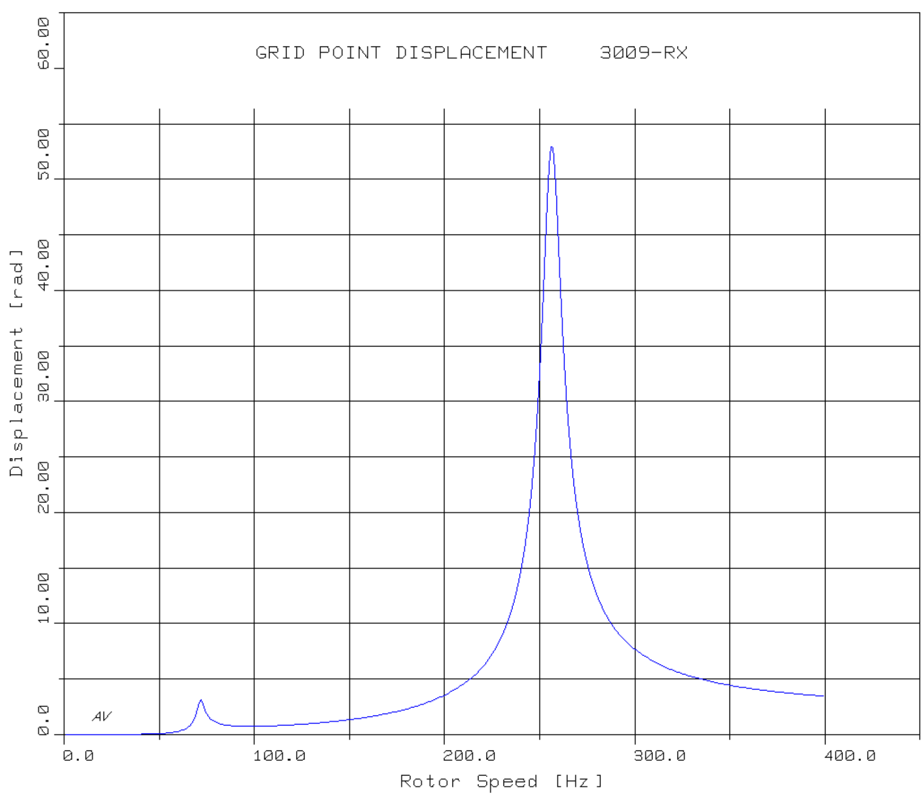


Fig. 167 Slow rotor RSPEED = 0.8 and EORDER = 0.8

10.6.2 Models with two Rotors

Scaling the speed of Rotor A with 0.8 and Rotor B with 1.2, the results in Fig. 168 are obtained. The real parts are shown in Fig. 169. The results of the analysis with two rotors are identical to those obtained by the individual rotors. The case of Rotor A running at 1.2 and rotor B at 0.8 relative speed are shown in Fig. 170 and Fig. 171, respectively. Also in this case the results are identical. A similar analysis was made in the rotating structure. The eigenfrequencies are shown in Fig. 172 and the real part of the eigenvalues in Fig. 173.

With multiple rotors with different speeds, the solution cannot be transformed from the fixed to the rotating system and vice versa. This is because for the transformation, the reference speed defined on the ROTORD l rotor speed is used, but the individual rotor speeds are different. After the solution the program does not know which solutions belong to which rotor. For coupled systems the modes may be mixed. This is the case for coaxial rotors. The conversion is calculated, but the plots are not relevant. The real eigenvalues calculated in the fixed and the rotating systems are identical as shown in Fig. 174.

For this case the analyses were done with SOL 107. The comparison for the fixed system is shown in Fig. 175 for the eigenfrequencies and in Fig. 176 for the real parts of the eigenvalues. Similar analyses for the rotating system are shown in Fig. 177 for the eigenfrequencies and in Fig. 178 for the real parts of the eigenvalues. The slight differences are due to truncation errors in the modal method.

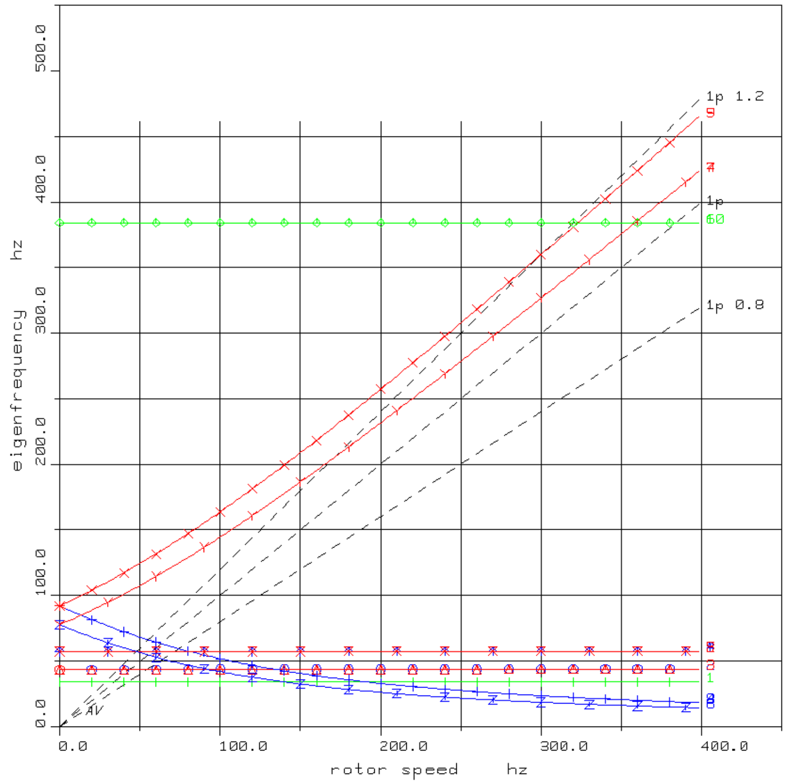


Fig. 168 Rotor A with 0.8 and rotor B with 1.2 compared to the individual analyses (with symbols)

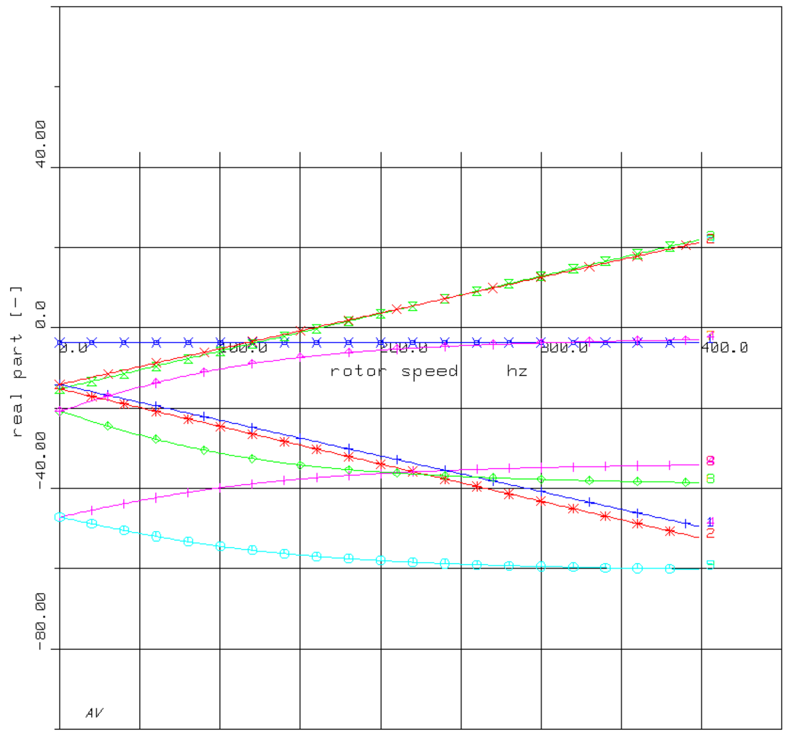


Fig. 169 Real part, Rotor A with 0.8 and rotor B with 1.2 compared to the individual analyses (with symbols)

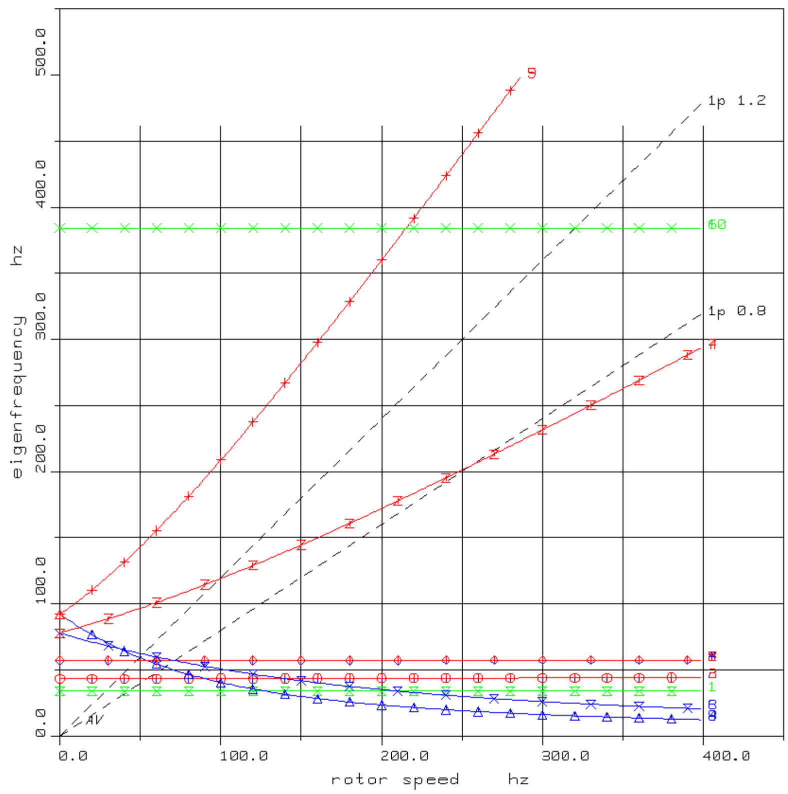


Fig. 170 Rotor A with 1.2 and rotor B with 0.8 compared to the individual analyses (with symbols)

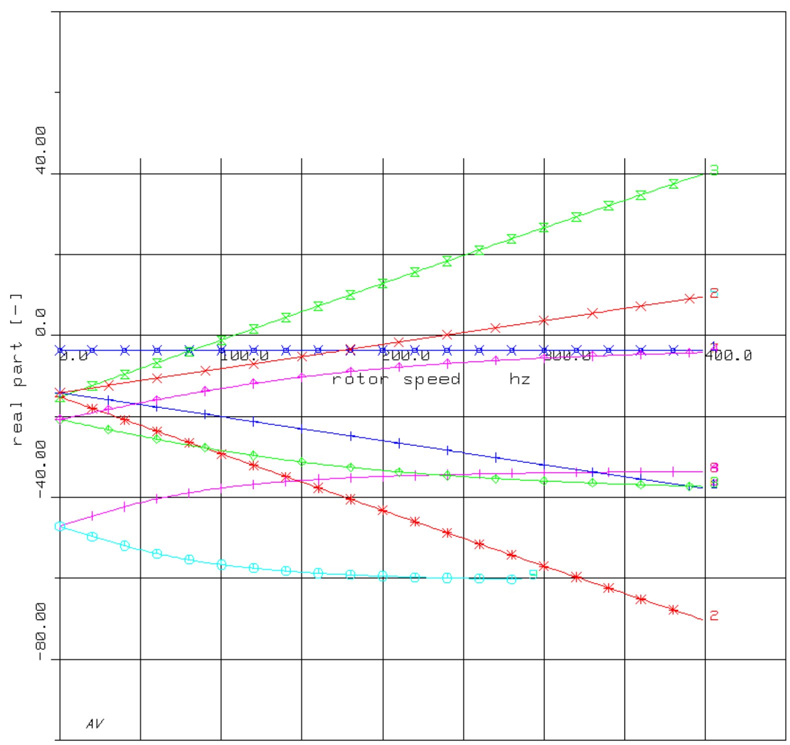


Fig. 171 Real part, Rotor A with 1.2 and rotor B with 0.8 compared to the individual analyses (with symbols)

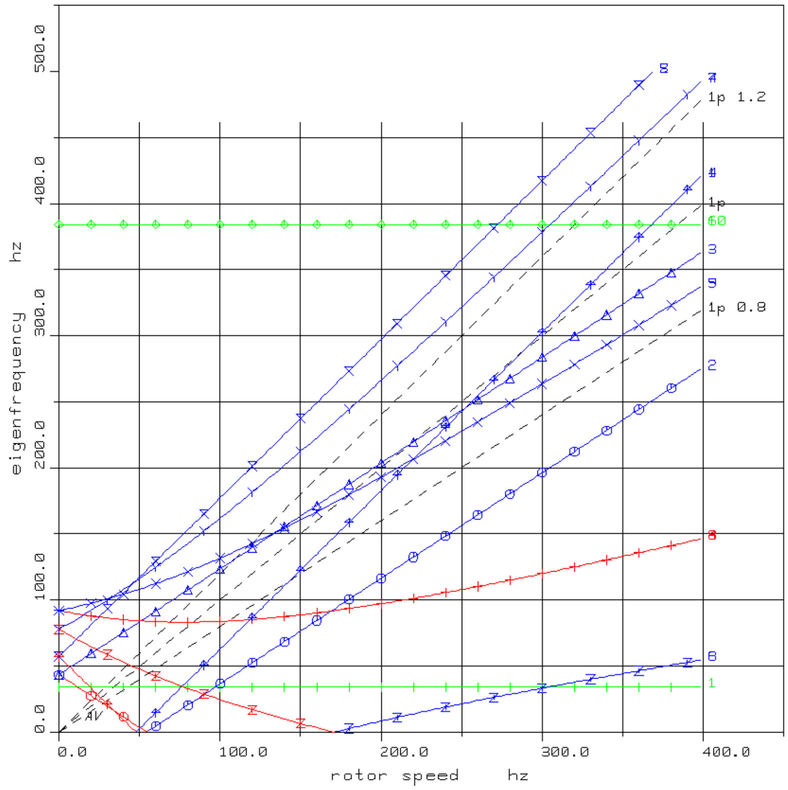


Fig. 172 Rotating system. Rotor A: 0.8, rotor B: 1.2 compared to the individual analyses (with symbols)

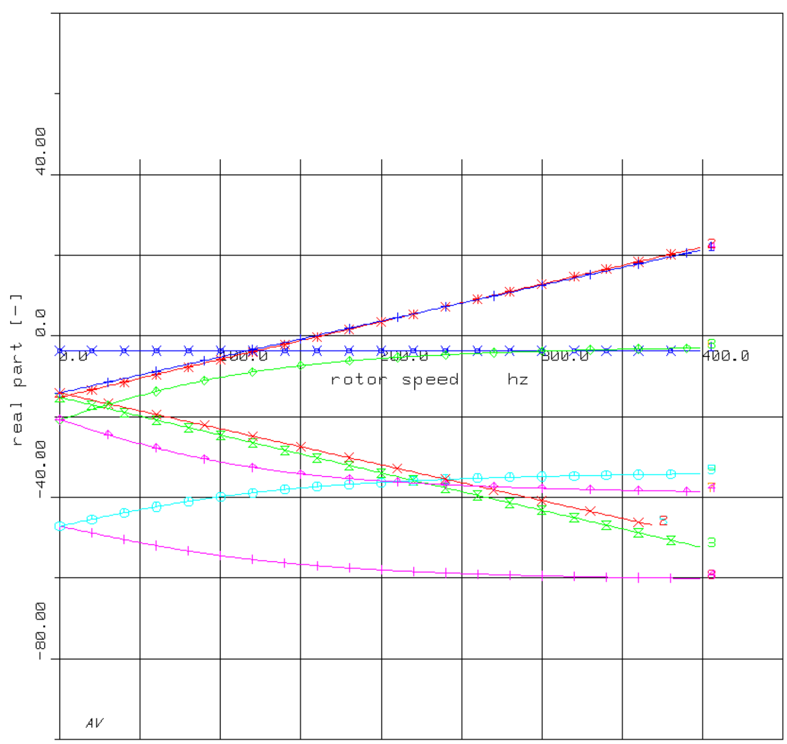


Fig. 173 Rotating system. Rotor A: 0.8, rotor B: 1.2 compared to the individual analyses (with symbols)

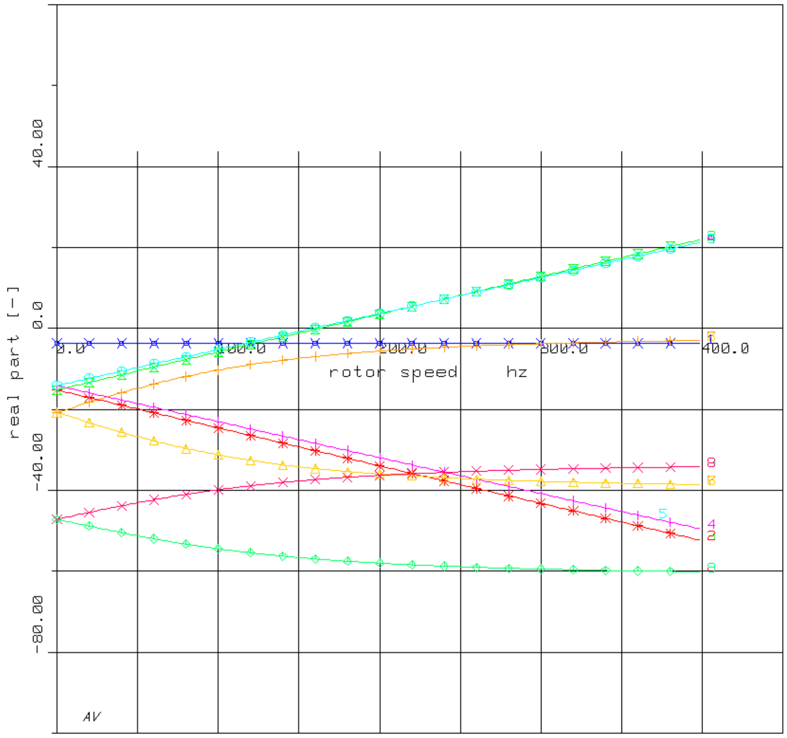


Fig. 174 Real eigenvalues. Rotor A: 0.8, rotor B: 1.2 calculated in the fixed and rotating system (symbols)

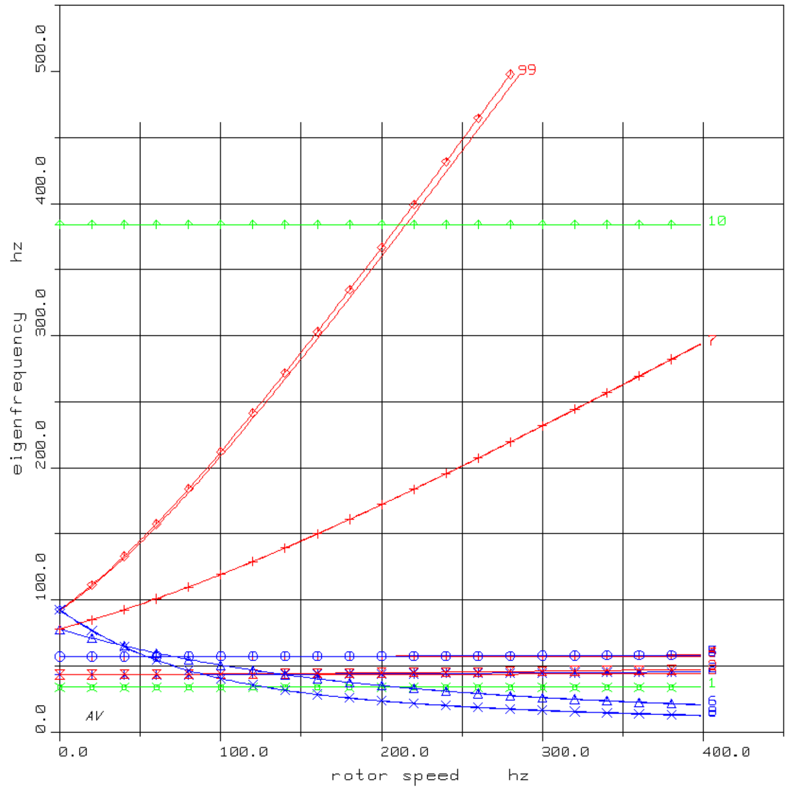


Fig. 175 Eigenfrequencies of modal and direct (symbols) solution in the fixed system. A: 1.2, B: 0.8

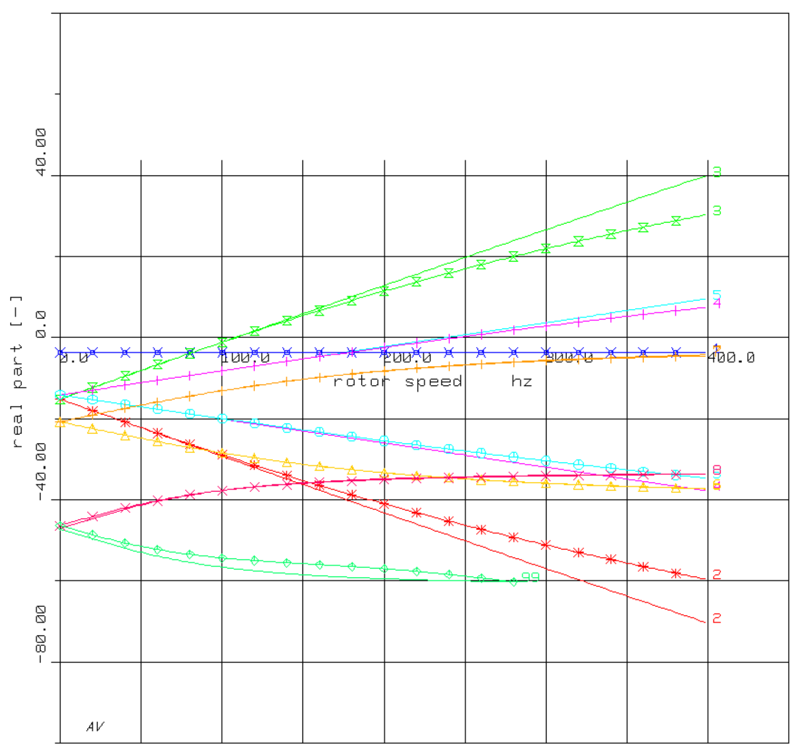


Fig. 176 Real eigenvalues of modal and direct (symbols) solution in the fixed system. A: 1.2, B: 0.8

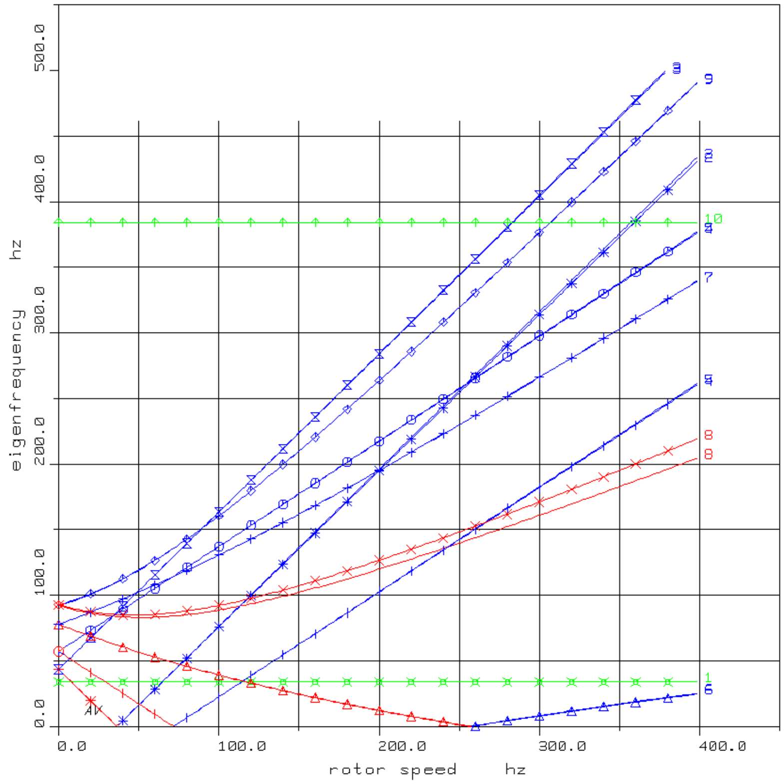


Fig. 177 Rotating system. Modal and direct (symbols) solution in the fixed system. A: 1.2, B: 0.8

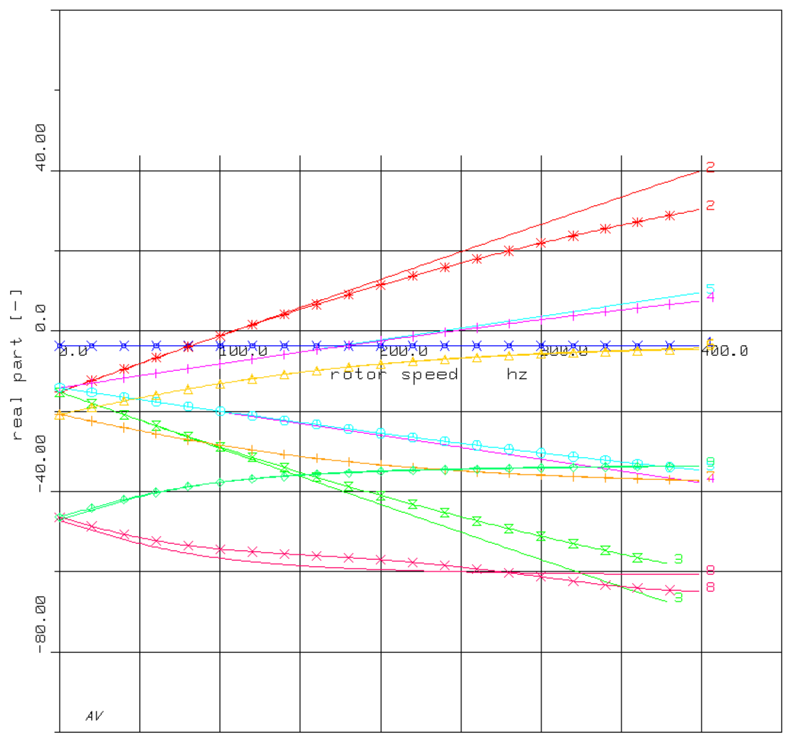


Fig. 178 Rotating system. Modal and direct (symbols) solution in the fixed system. A: 1.2, B: 0.8

10.7 Frequency Response Analysis

For the model with two rotors, an asynchronous response analysis in the frequency domain was conducted. The analyses were made at a rotor speed of 200 Hz as indicated in the Campbell diagram (Fig. 179) and the damping diagram (Fig. 180). The damping of the bearings was increased in order to avoid resonances of unstable solutions. In the models, the relative speed of rotor A was 1.2 and rotor B 0.8.

10.7.1 Modal Solution SOL 111

In the frequency response analysis the real and imaginary part of the displacement and rotation (tilt) of both rotors is calculated. The results are shown in Fig. 181 through Fig. 184. The blue lines represent the magnitude. The response can now be analyzed around the peak using the slope of the real part, the imaginary part, the width of the magnitude peak (half power), or the Nyquist method. The magnitude is shown in Fig. 185 and the Nyquist plot in Fig. 186 of the translation peak of Rotor A. Usually, the results of the Nyquist method yield the best results. The frequencies and the damping values can now be compare to the Campbell diagram as shown in Fig. 187 and the damping diagram shown in Fig. 188. The results are in good agreement. In the modal solutions, 20 real modes are used.

10.7.2 Direct Solution SOL 108

The results of the direct solutions are slightly different from those of the modal solutions due to truncation errors in the modal formulation. The eigenfrequencies are close as shown in Fig. 189. The damping of the tilt modes is lower for the direct solution. Therefore, the peaks of the higher modes will be higher with the direct solution. The comparisons between the modal method (SOL 111) and the direct method (SOL 108) are shown in Fig. 191 through Fig. 194.

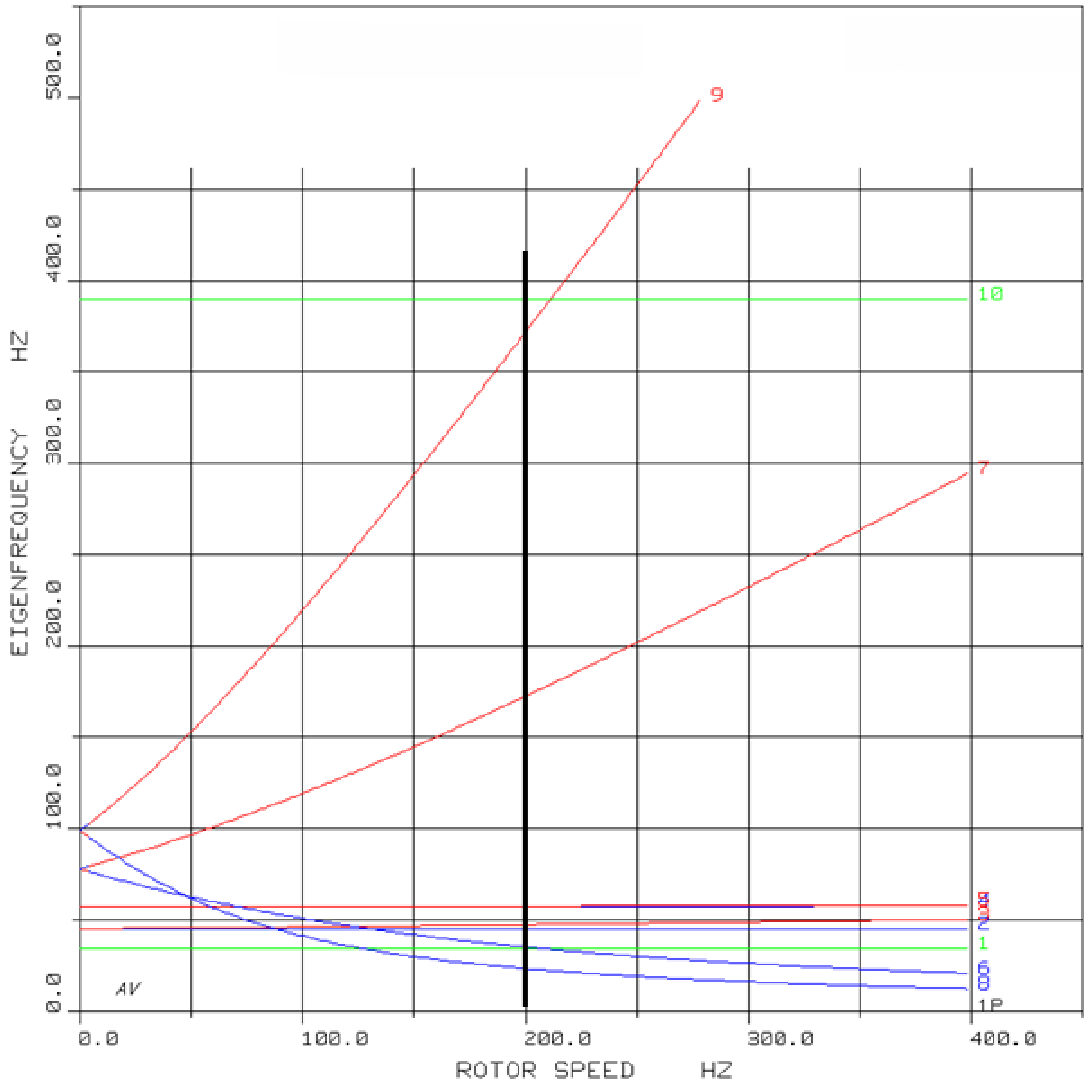


Fig. 179 Campbell diagram for the model with two rotors. A: 1.2, B: 0.8

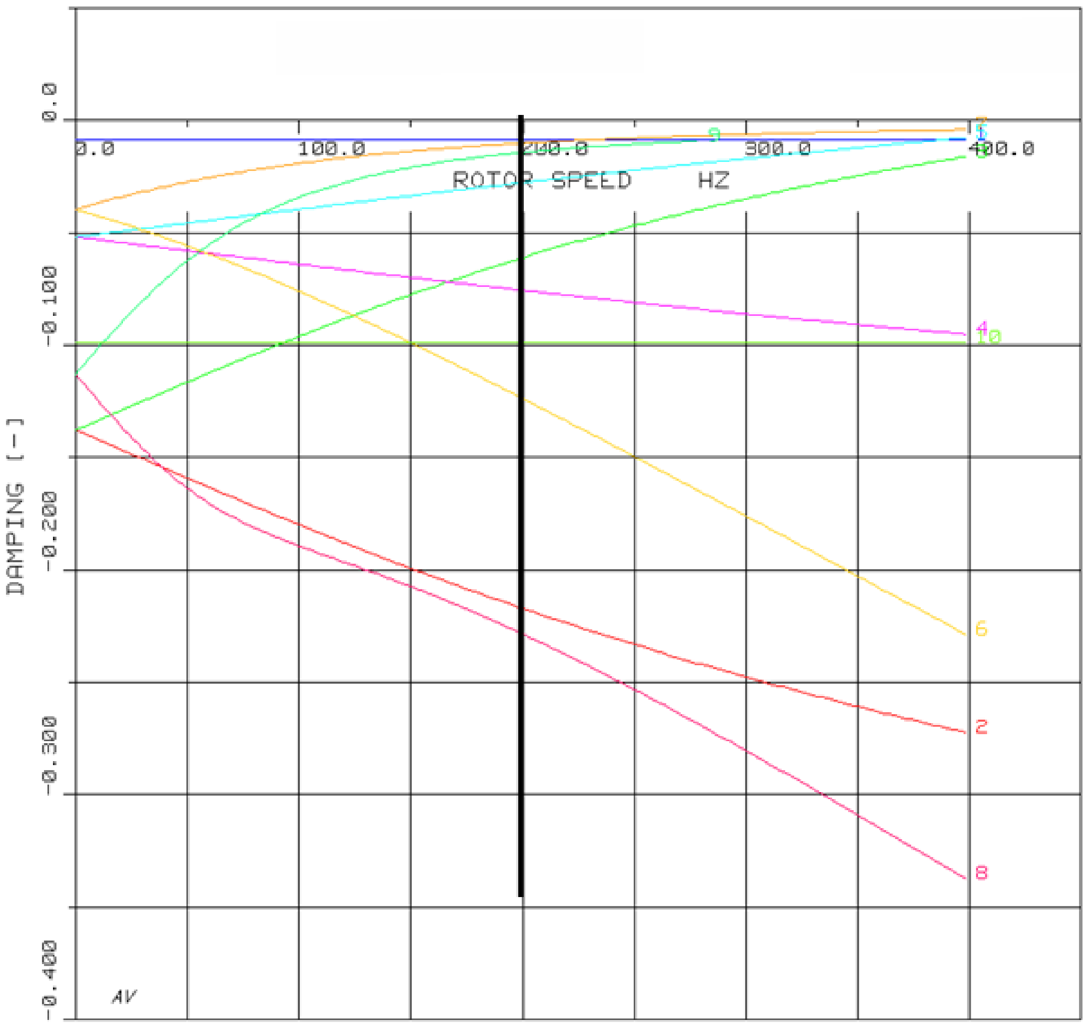


Fig. 180 diagram for the model with two rotors. A: 1.2, B: 0.8

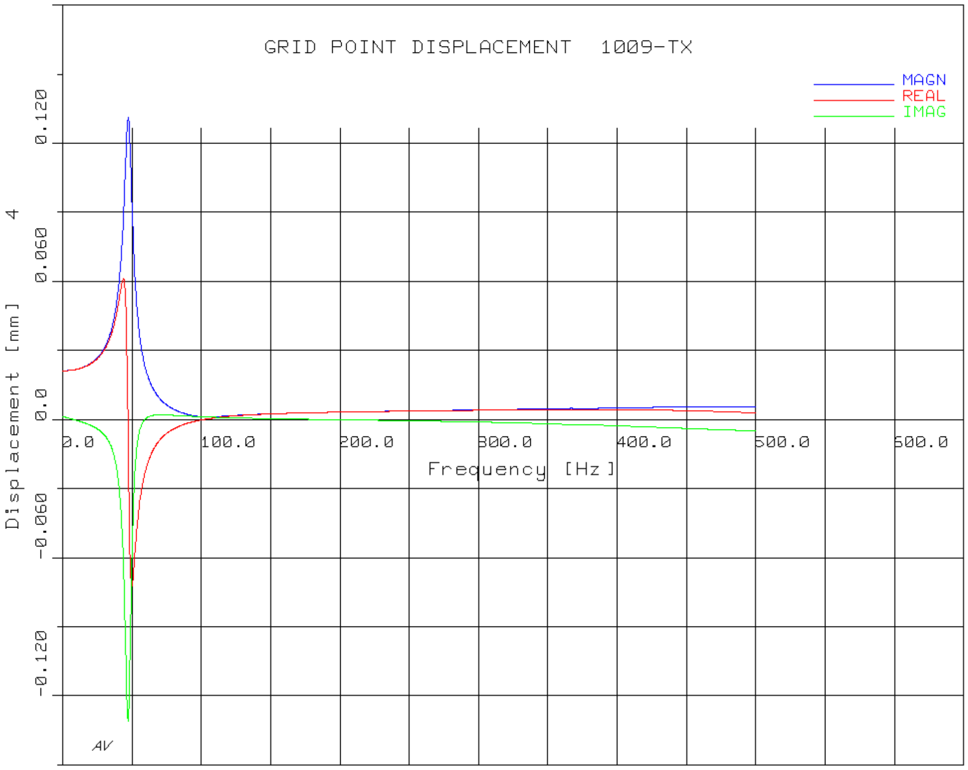


Fig. 181 Translation of rotor A

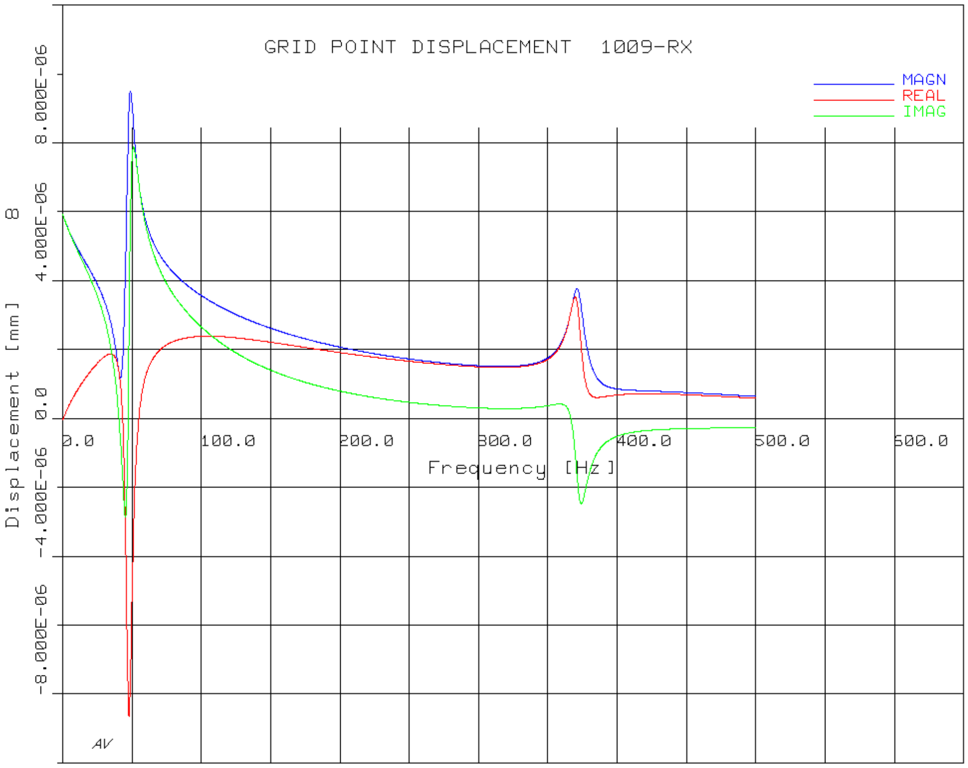


Fig. 182 Rotation of rotor A

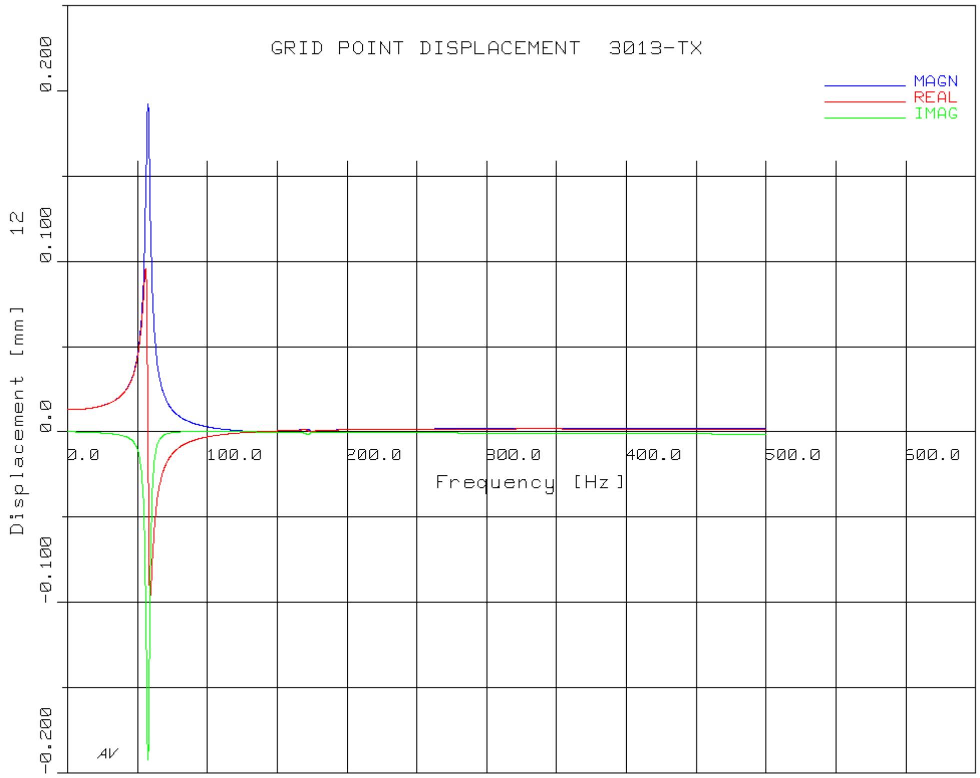


Fig. 183 Translation of rotor B

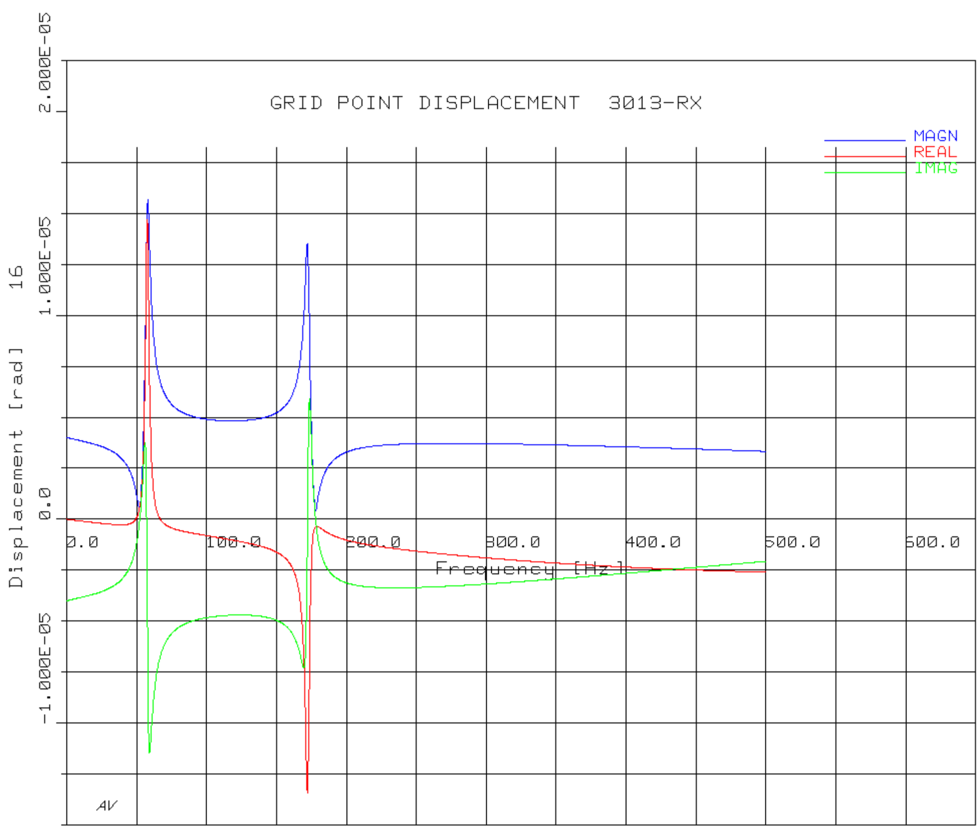


Fig. 184 Rotation of rotor B

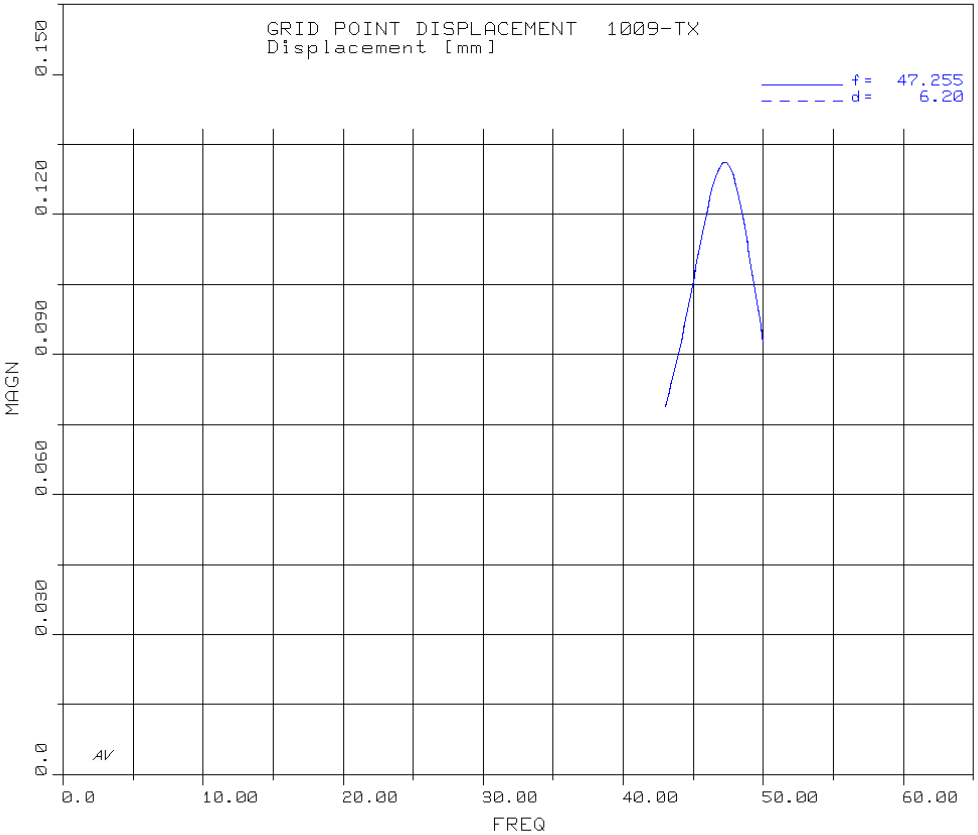


Fig. 185 Magnitude of the translation peak of rotor A

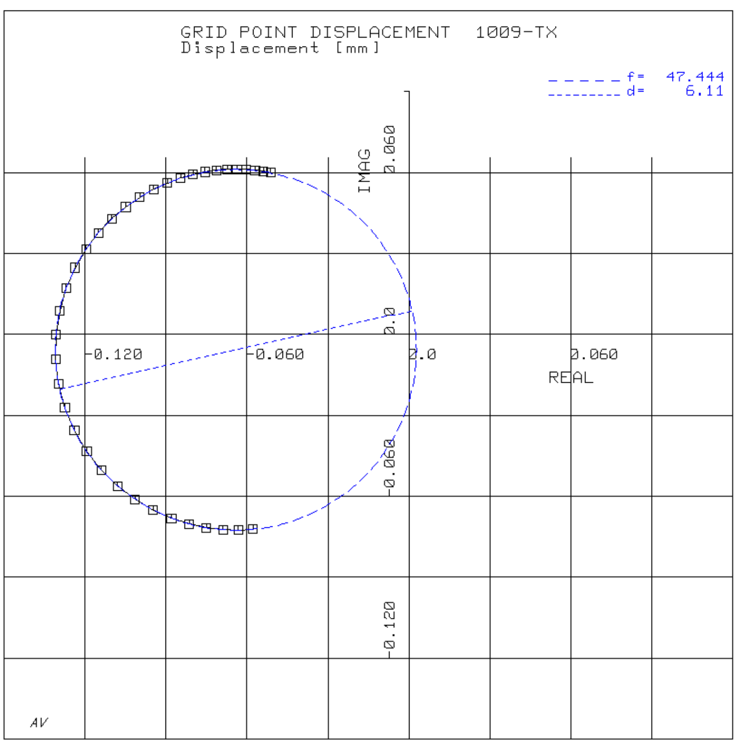


Fig. 186 Nyquist plot of the translation peak of rotor A

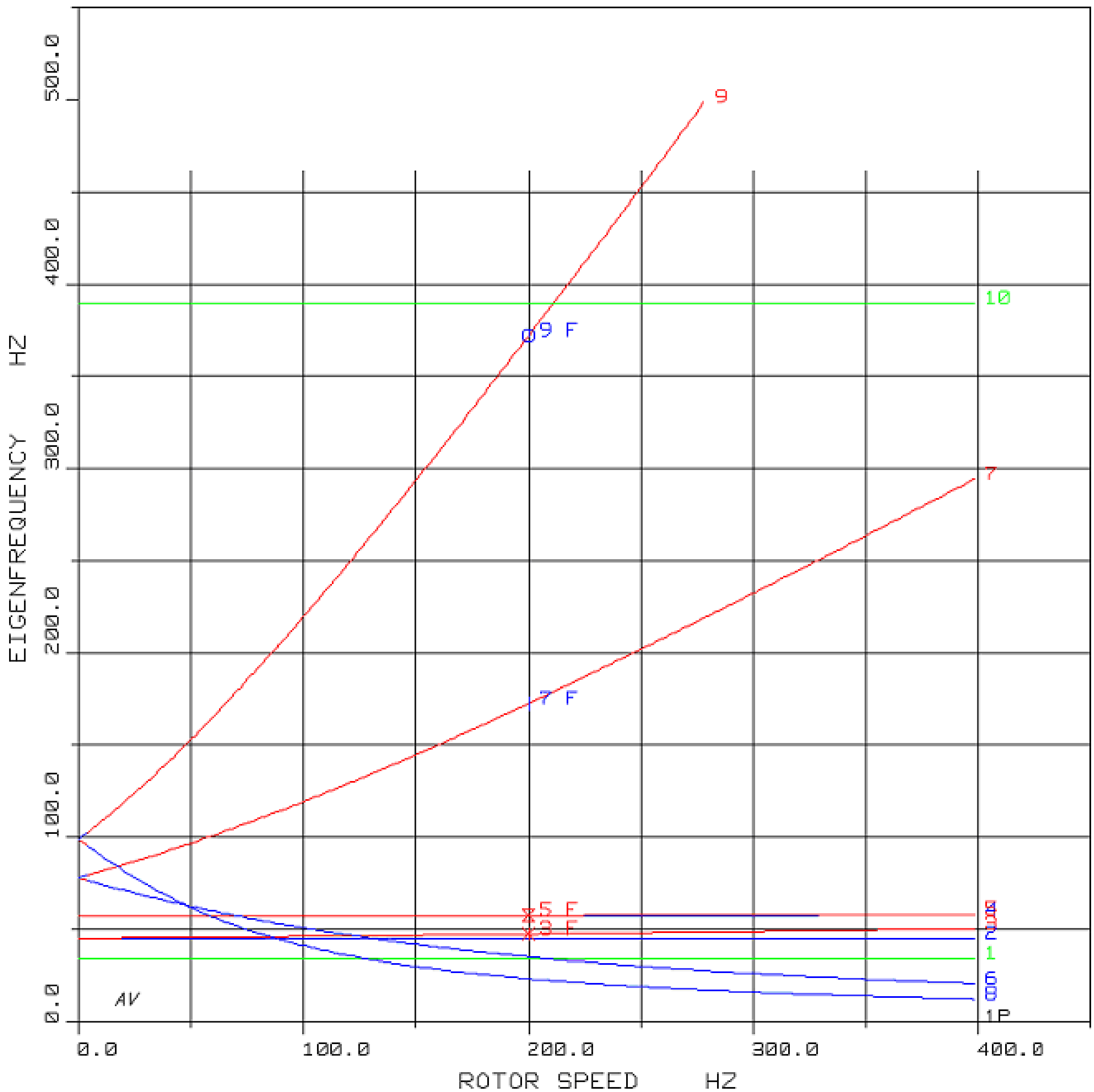


Fig. 187 Results from the frequency response analysis (forward whirl) compared to the Campbell diagram.

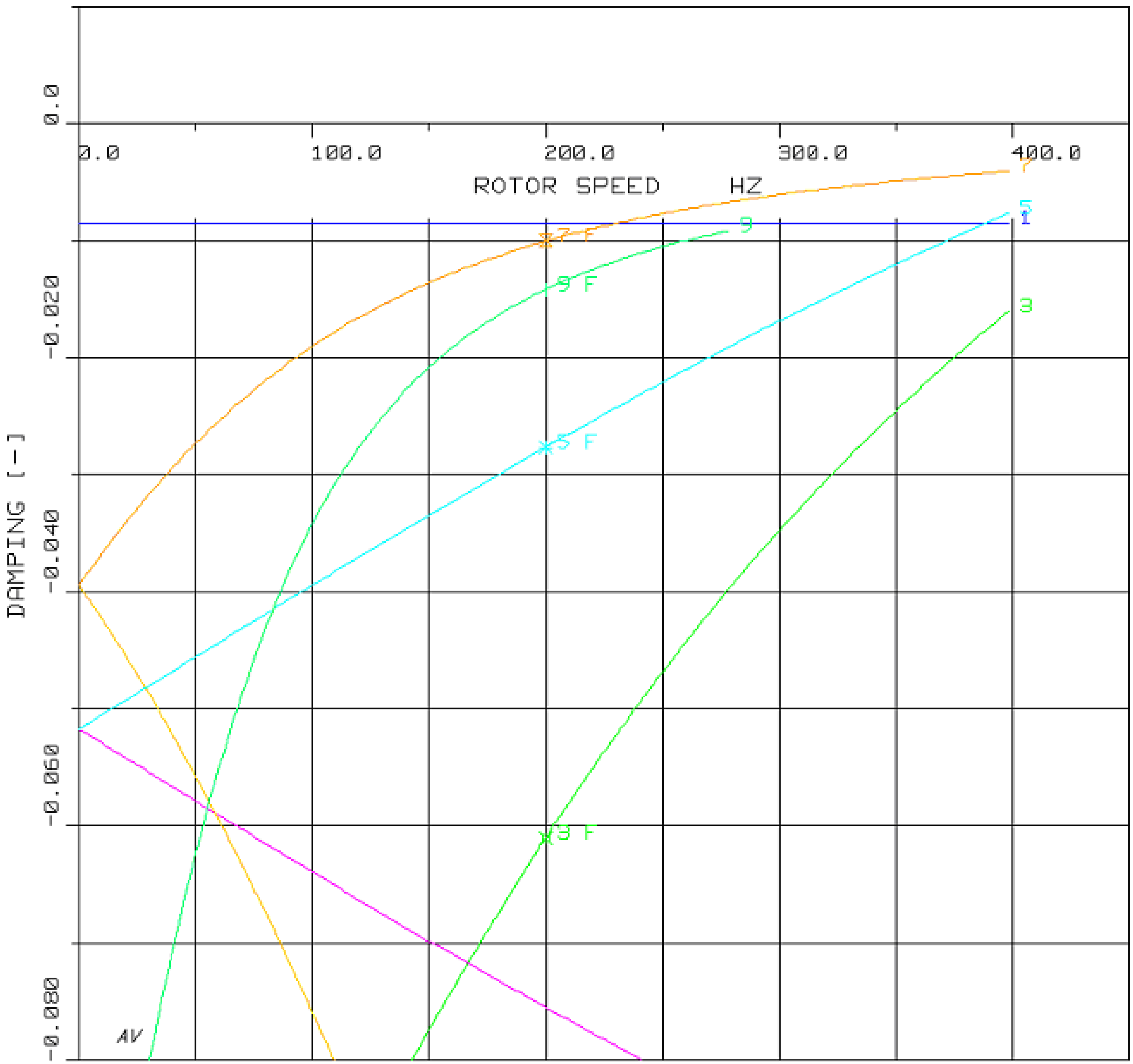


Fig. 188 Results from the frequency response analysis (forward whirl) compared to the damping diagram

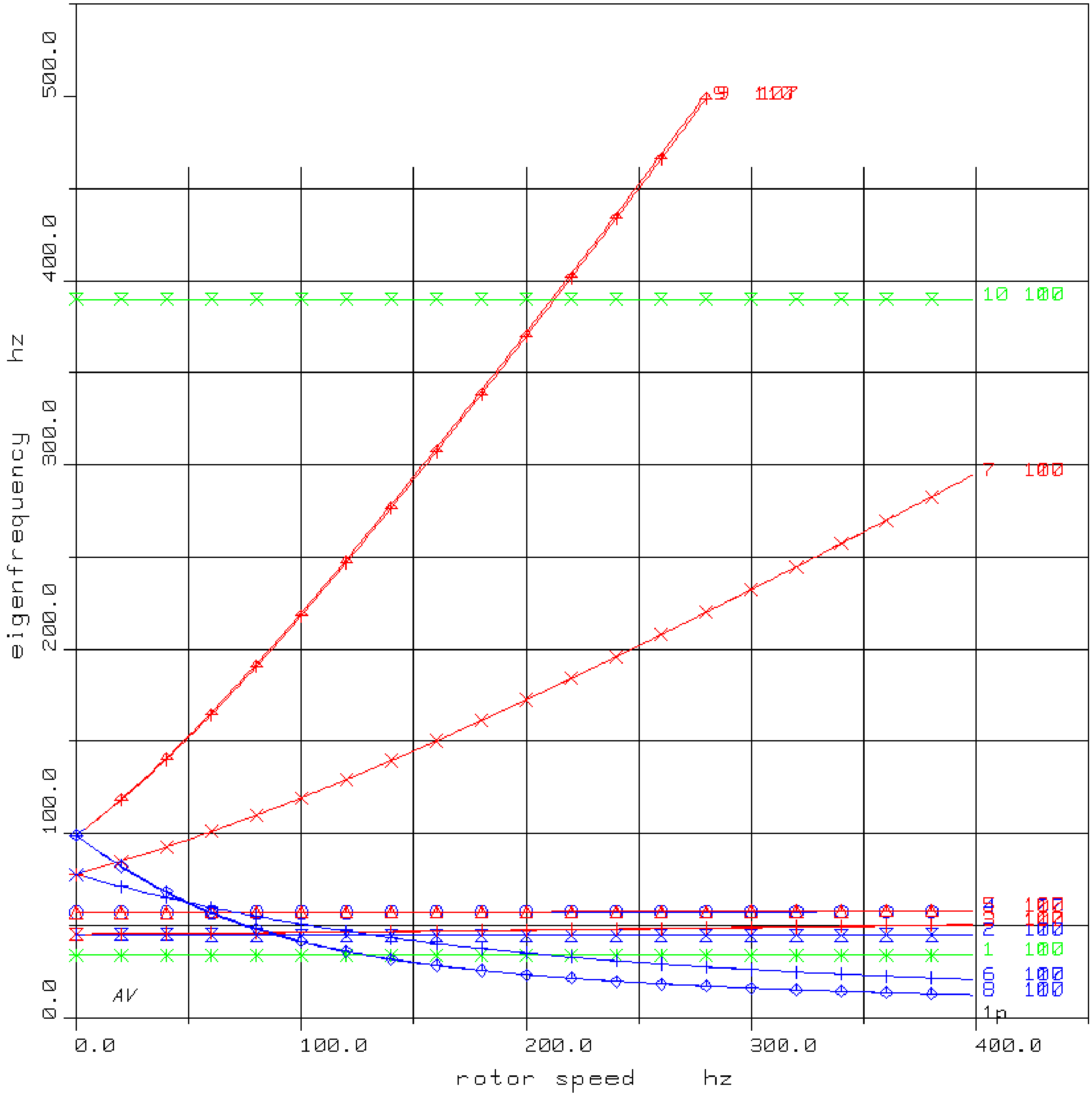


Fig. 189 Eigenfrequencies obtained with the modal and the direct method (symbols)

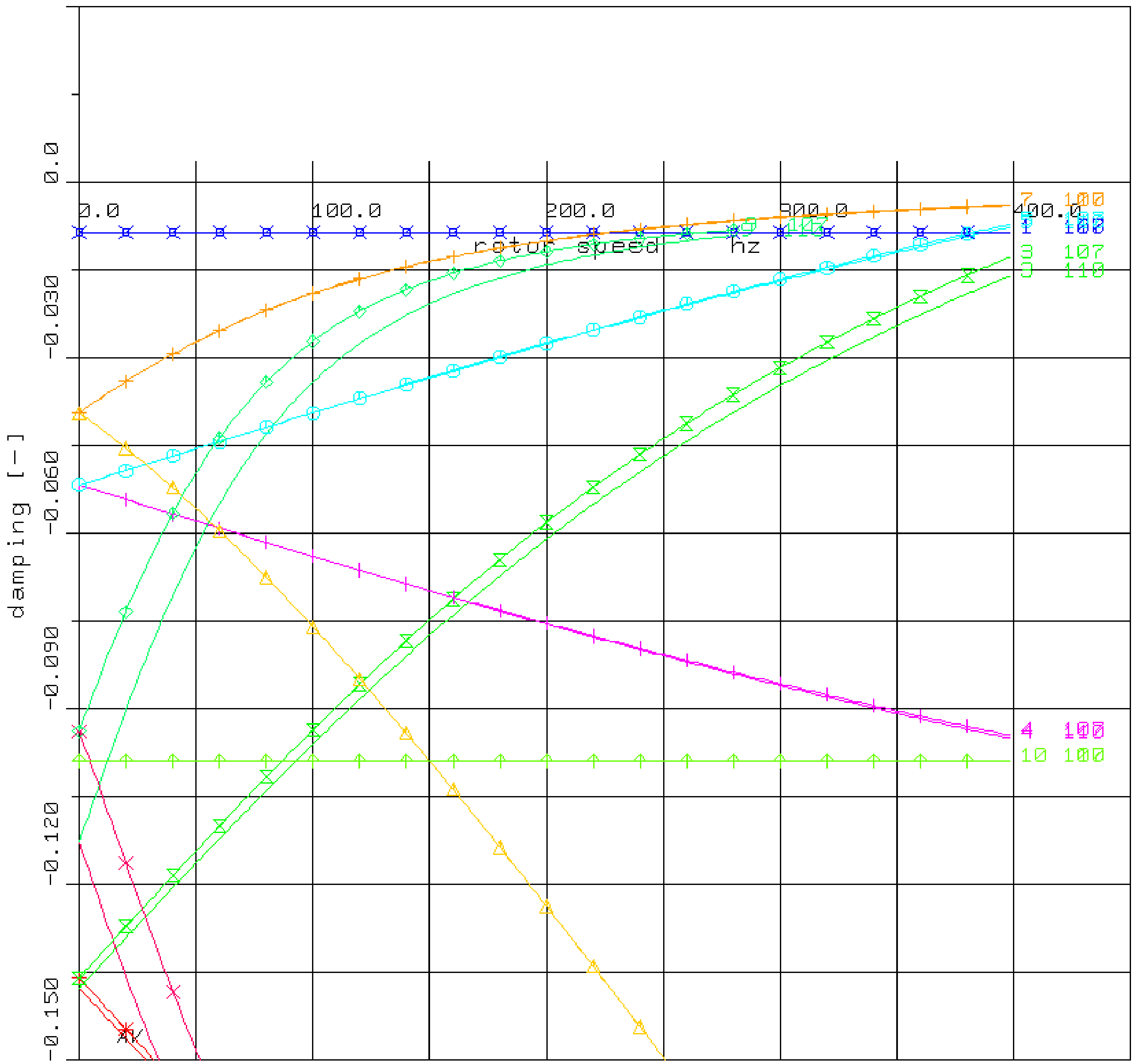


Fig. 190 Damping curves obtained with the modal and the direct method (symbols)

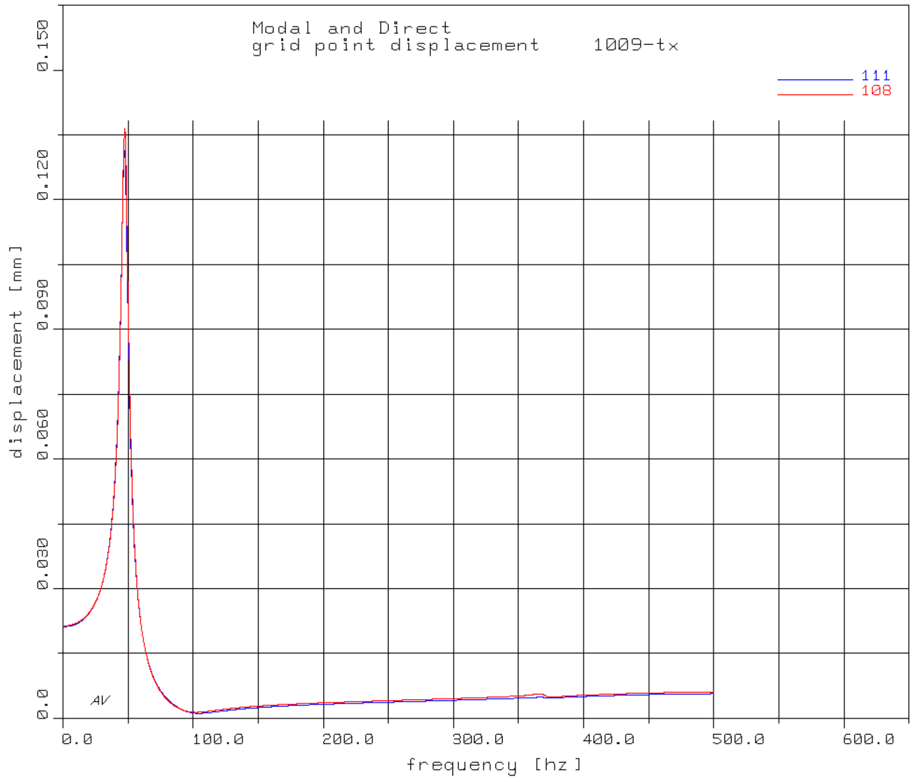


Fig. 191 Magnitude of translation, rotor A, modal (blue) and direct method (red)

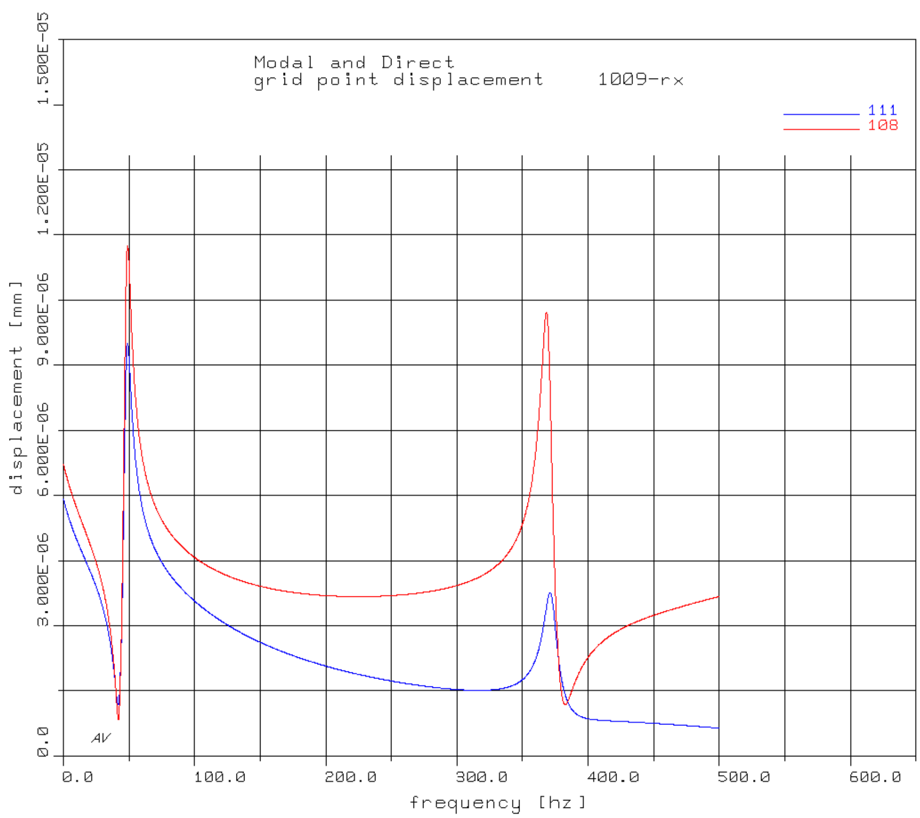


Fig. 192 Magnitude of rotation, rotor A, modal (blue) and direct method (red)

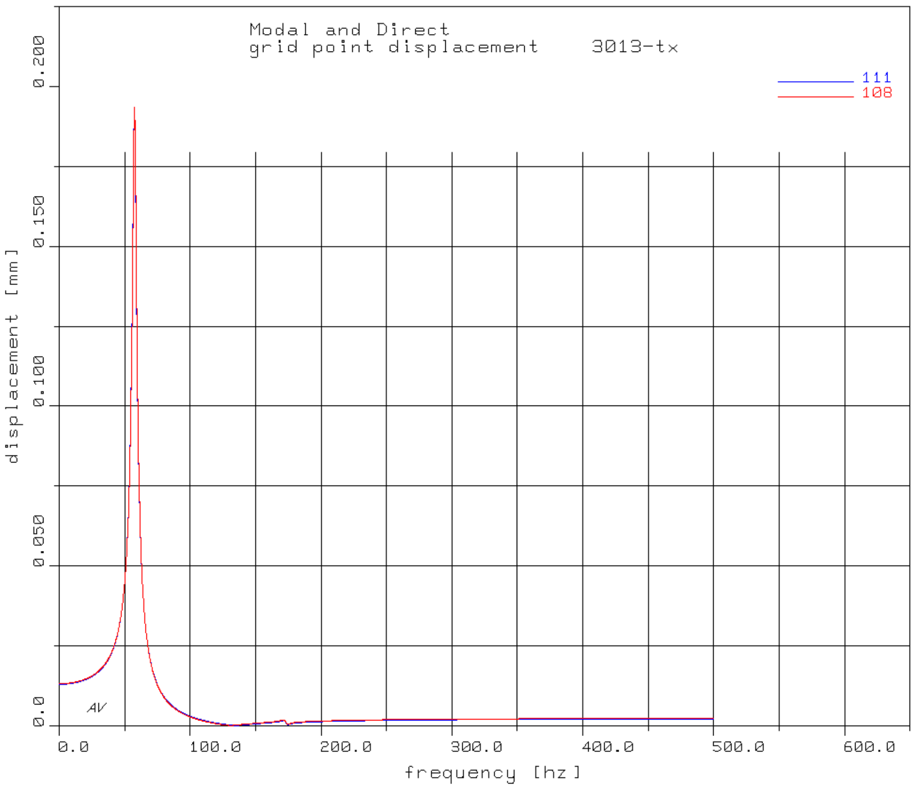


Fig. 193 Magnitude of translation, rotor B, modal (blue) and direct method (red)

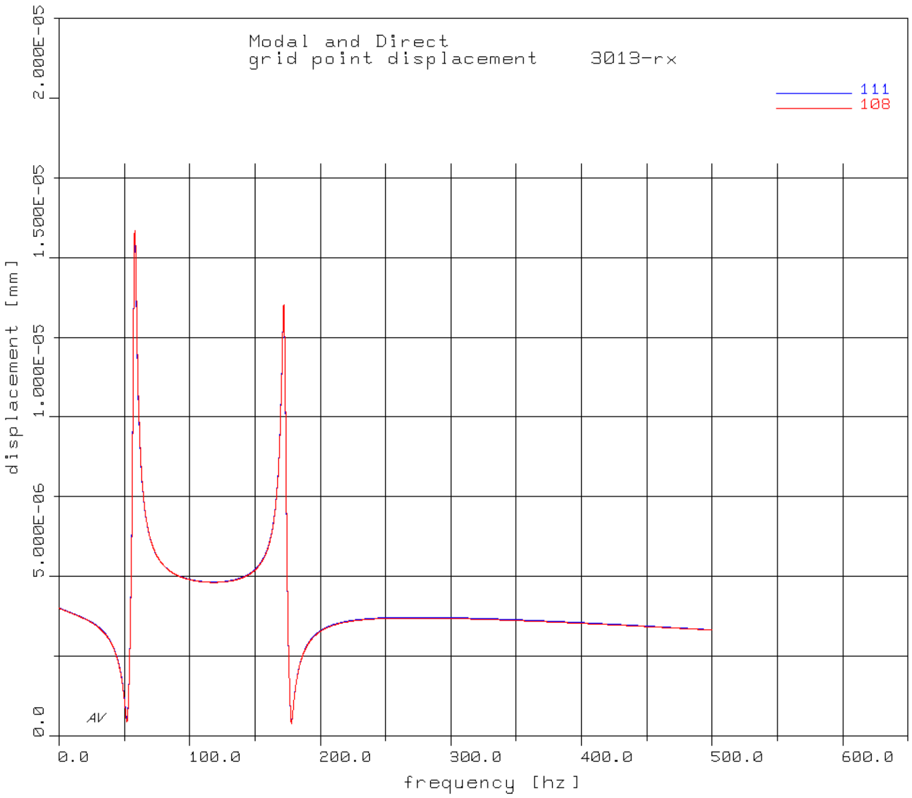


Fig. 194 Magnitude of rotation, rotor B, modal (blue) and direct method (red)

10.8 Transient response Analysis

Similar to the frequency response analysis, asynchronous analysis in the time domain at 200 Hz rotor speed were done. The excitation functions are sine and cosine functions with linearly increasing frequency shown in Fig. 195.

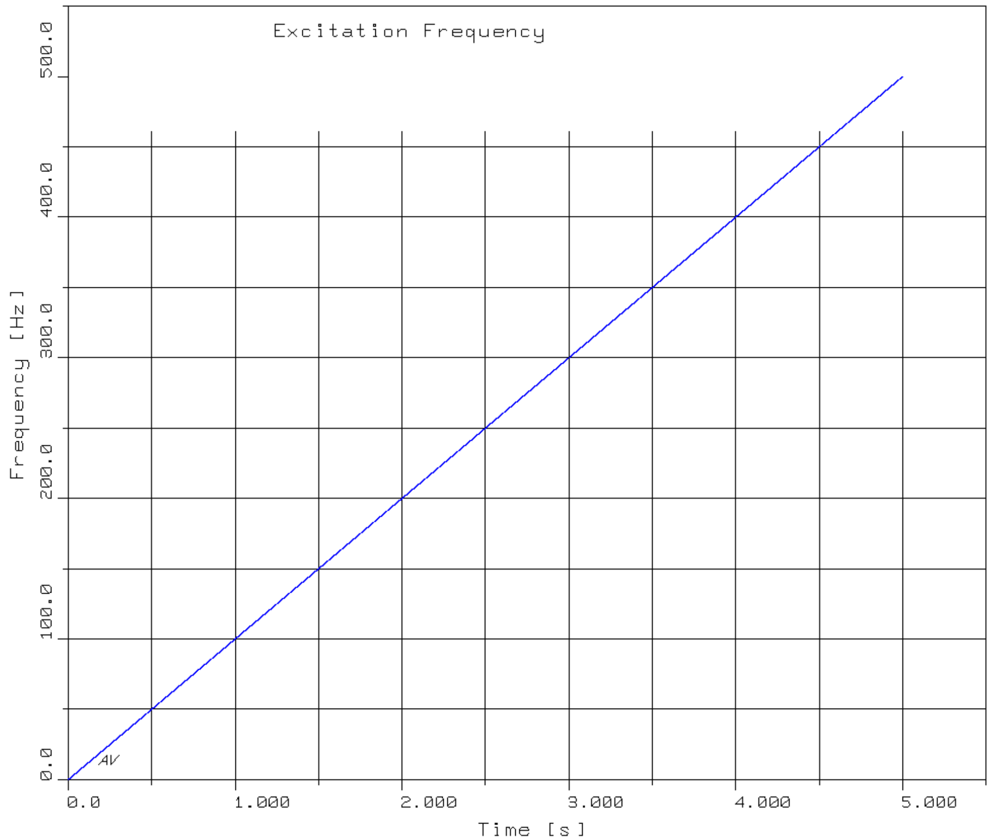


Fig. 195 Excitation frequency as function of time

10.8.1 Modal Method

Results of the modal analysis in time domain with forward whirl excitation are shown in Fig. 196 through Fig. 199. The regions with large amplitude match the resonances found in the Campbell diagram. A Fourier analysis was made and the amplitudes scaled to those of the frequency response analysis and compared in Fig. 200 through Fig. 203. The curves agree well. Only the response at the high frequency of the tilt motion of Rotor A (Fig. 201) is slightly different. In the transient analysis, the rotor is accelerating and the response is not steady-state as in the frequency response analysis.

An analysis was made with a short impulse as excitation when the rotor is rotating at 200 Hz speed. This will excite all modes. Laplace transformations of the results were made by calculating the Fourier transformation using an exponential window function. Then the frequency and damping of the peaks were found by the Nyquist method. The artificial damping due to the real exponent in the window function was subtracted from the calculated damping. The frequencies obtained are compared to the Campbell diagram in Fig. 204 and the damping values in Fig. 205.

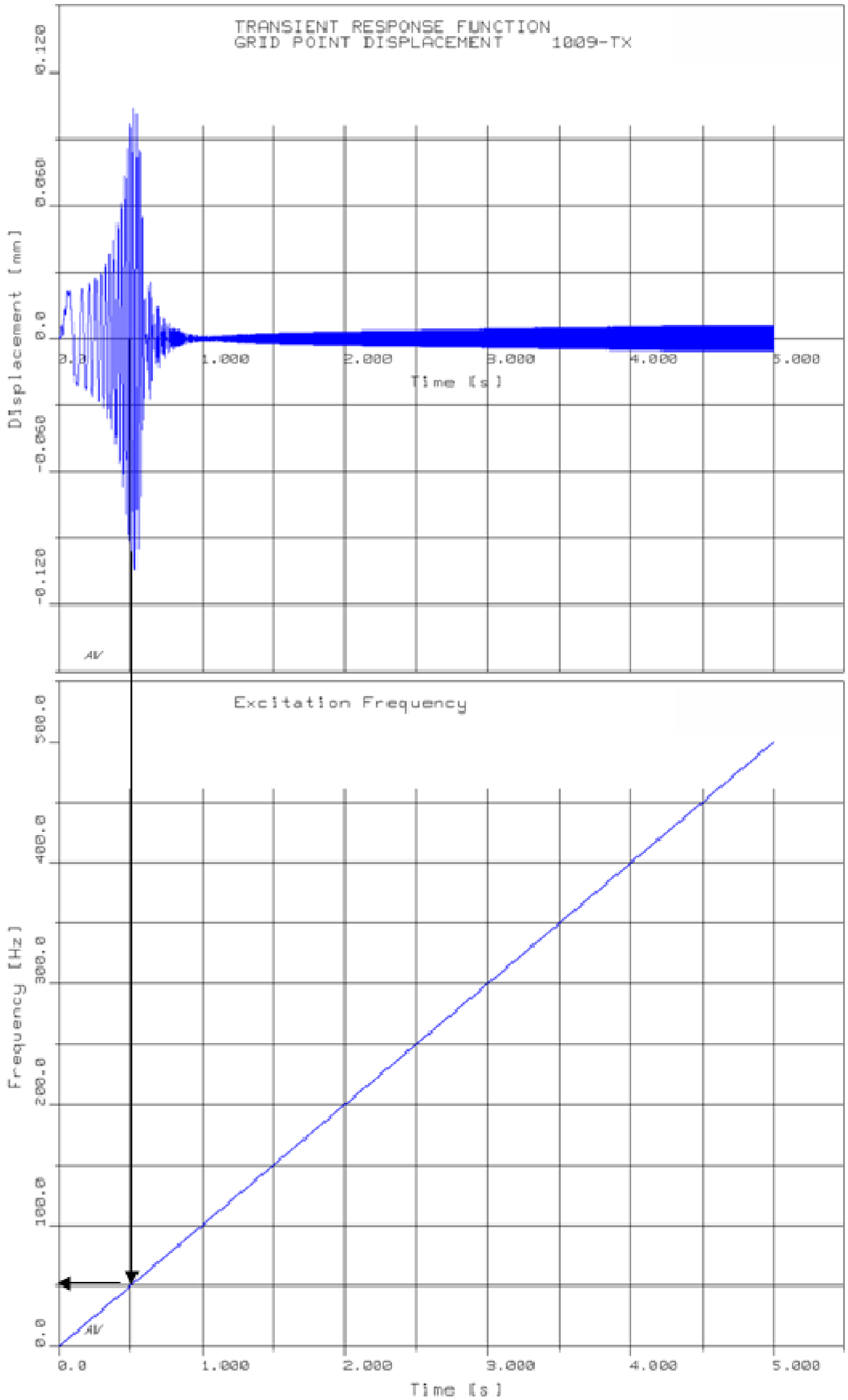


Fig. 196 Translation of rotor A

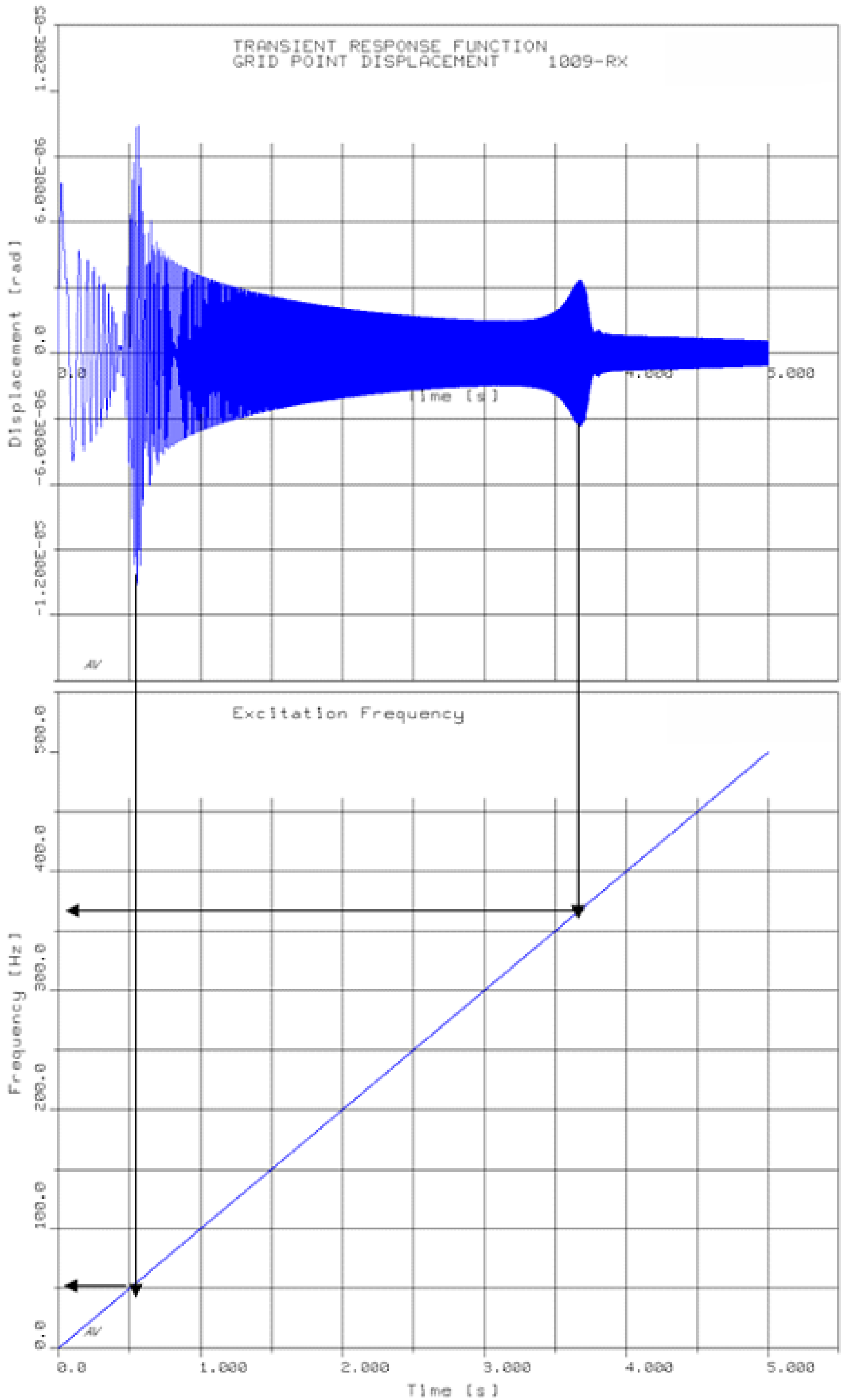


Fig. 197 Rotation (tilt) of rotor A

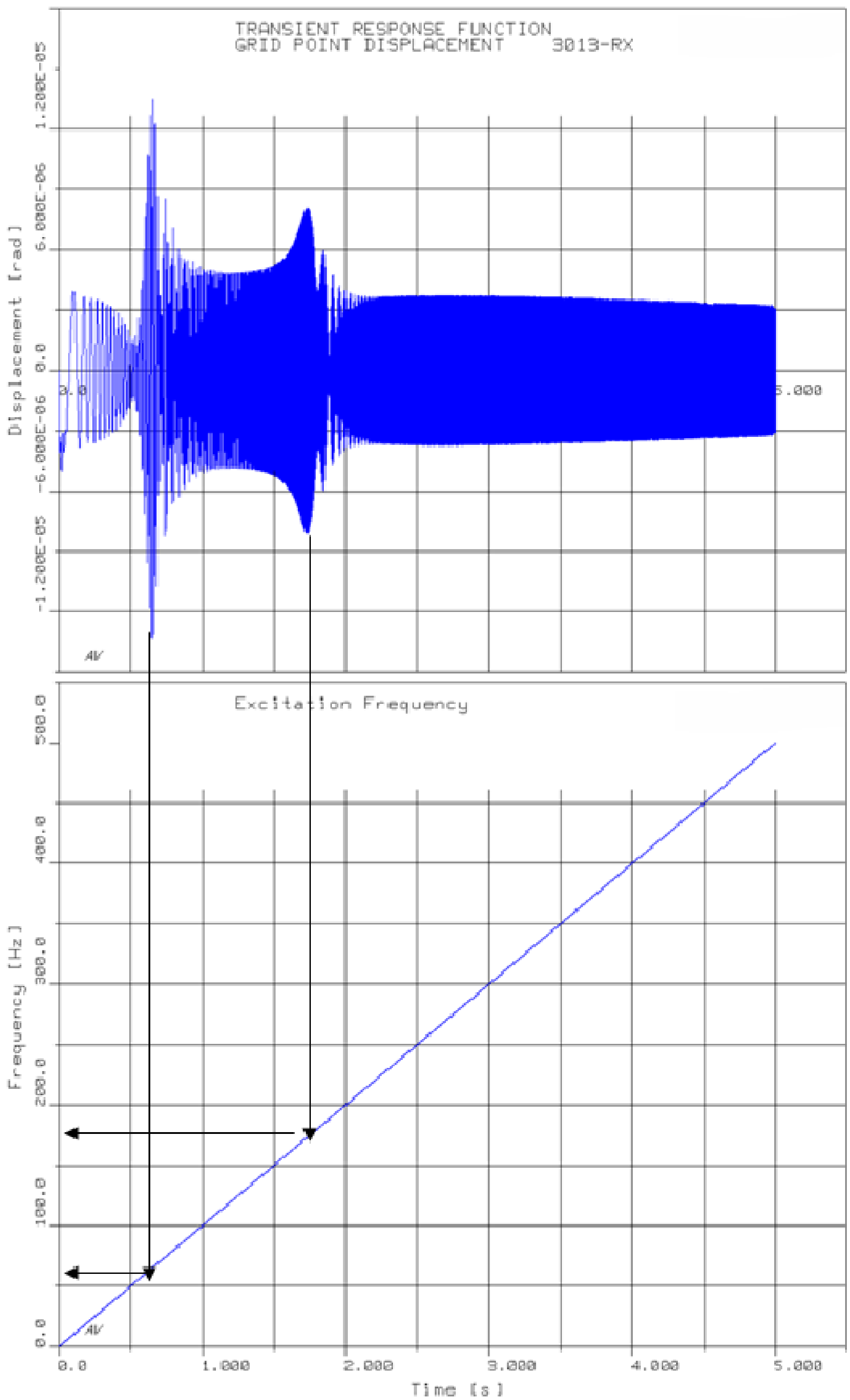


Fig. 198 Translation of rotor B

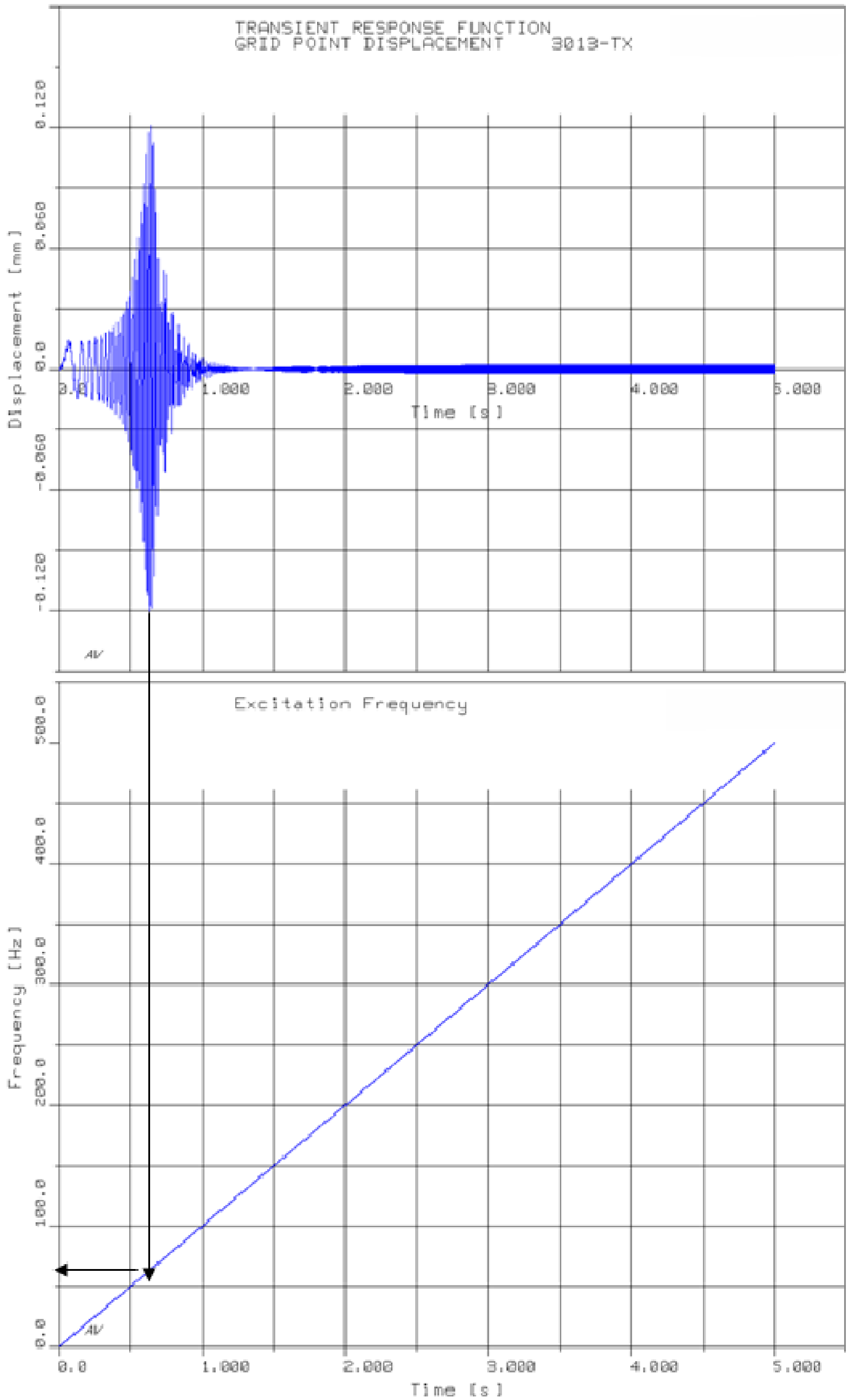


Fig. 199 Rotation (tilt) of rotor B

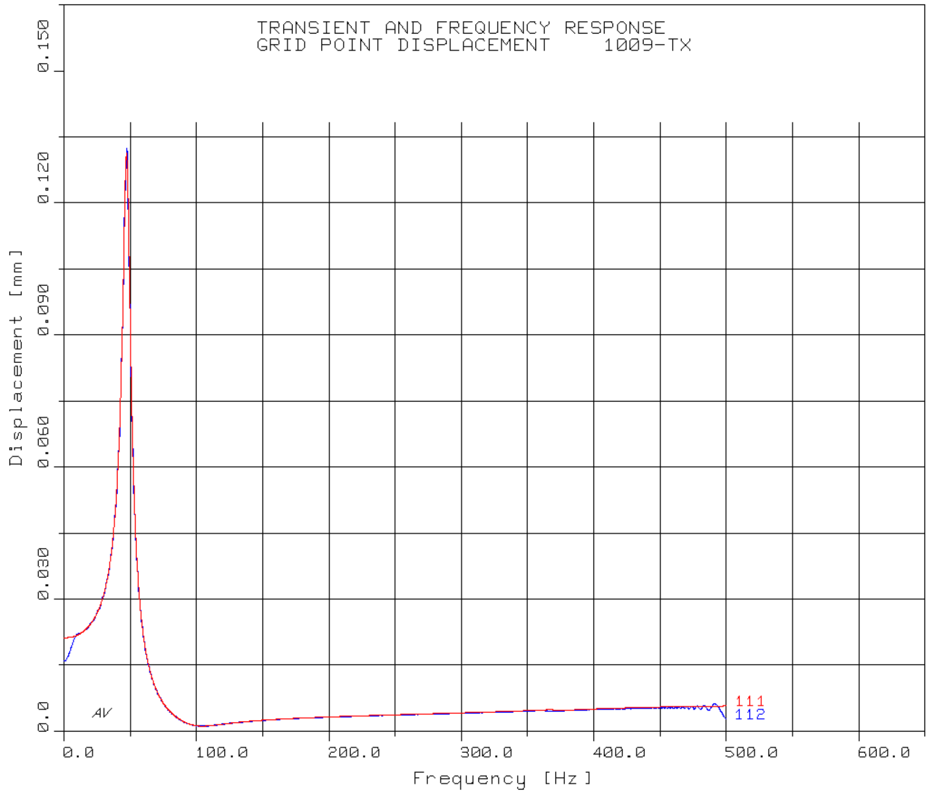


Fig. 200 Translation of rotor A. FFT of transient response and magnitude of frequency response

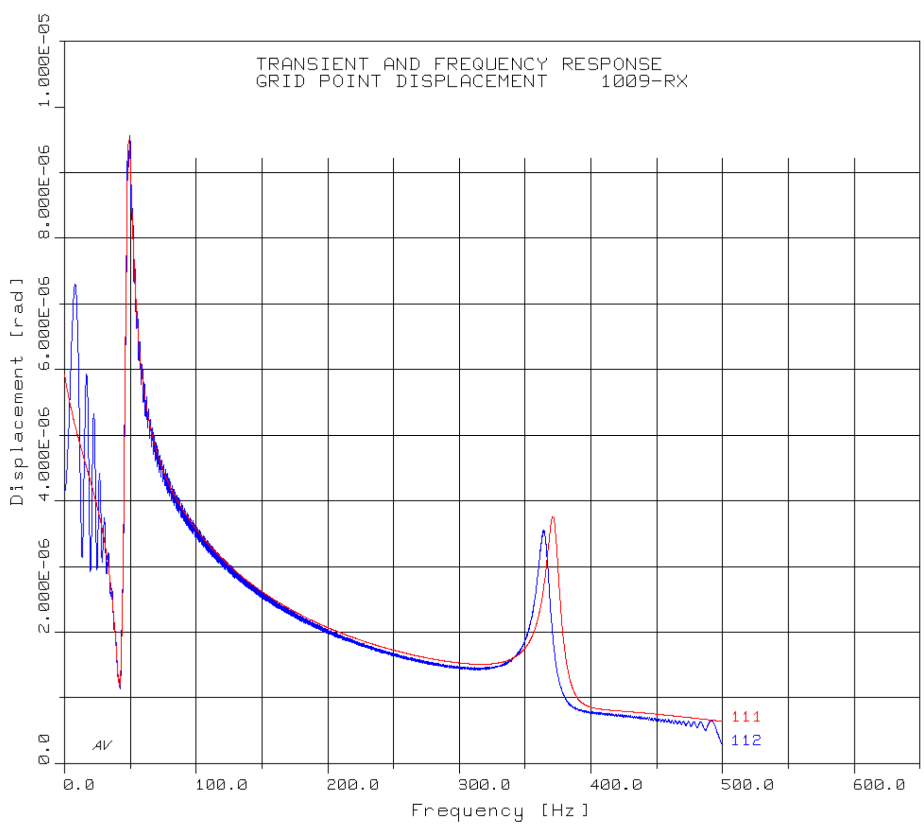


Fig. 201 Tilt of rotor A. FFT of transient response and magnitude of frequency response

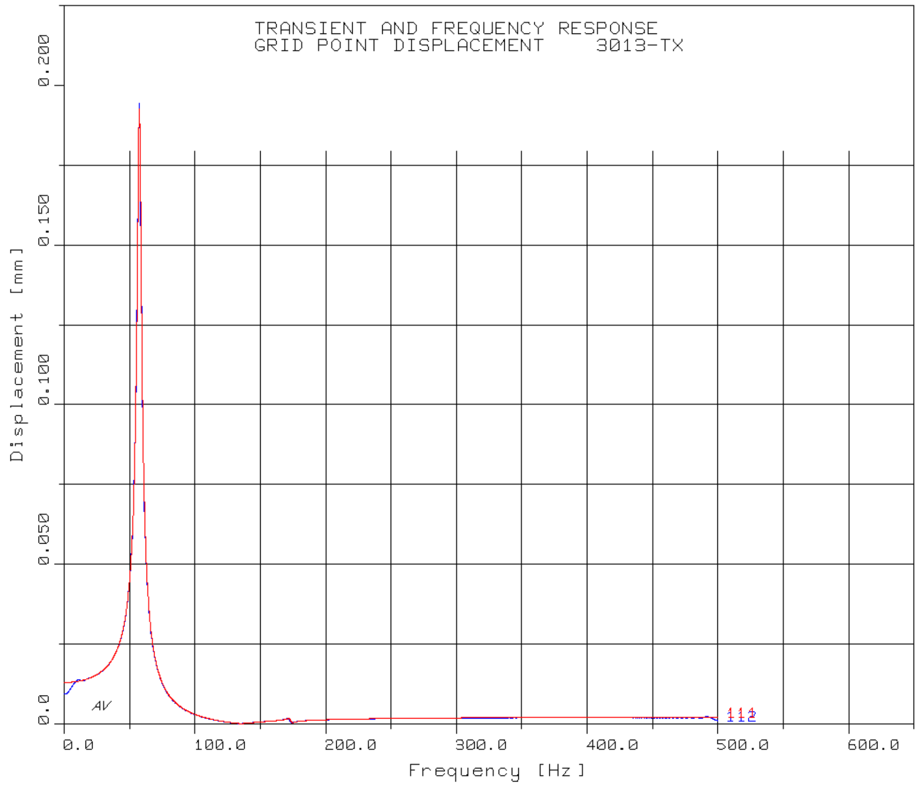


Fig. 202 Translation of rotor B. FFT of transient response and magnitude of frequency response

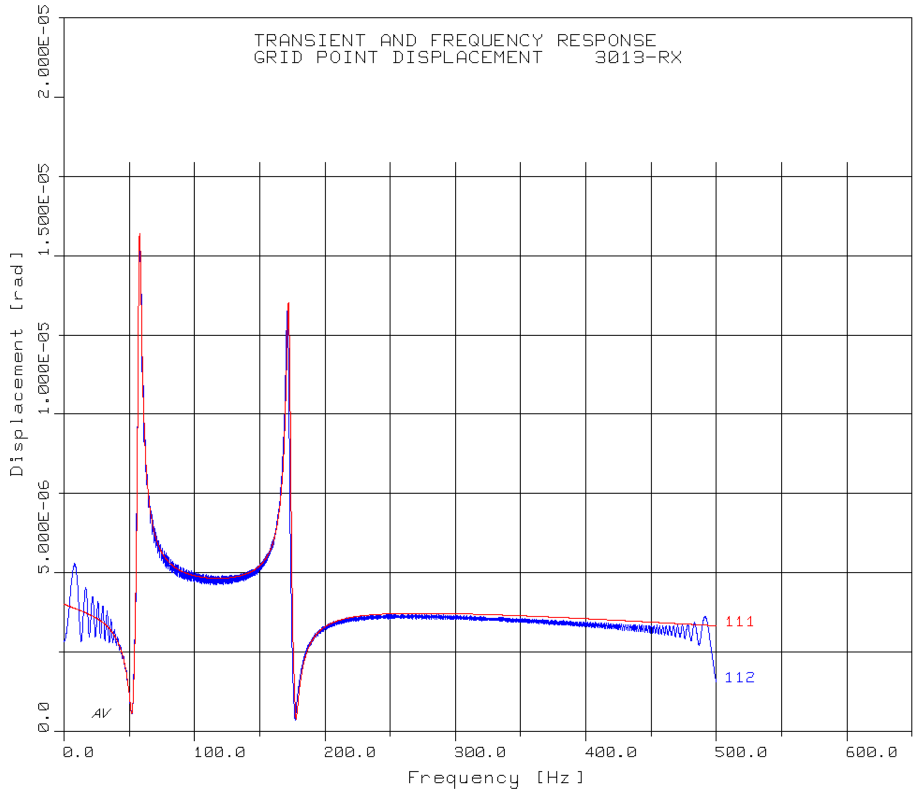


Fig. 203 Tilt of rotor B. FFT of transient response and magnitude of frequency response

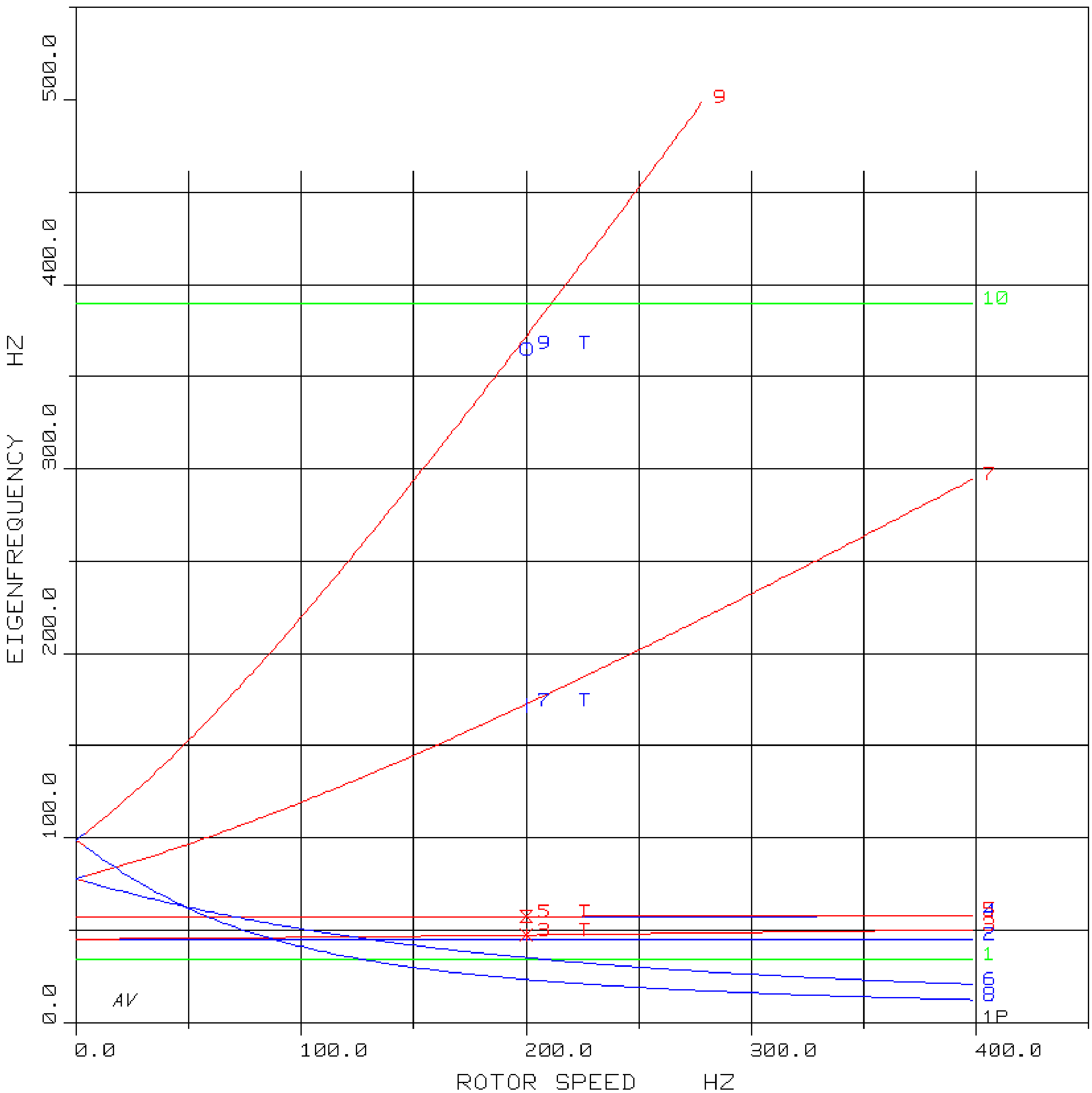


Fig. 204 Campbell diagram with frequencies calculated from the Laplace transformation and the Nyquist method of the transient analysis with impulse excitation

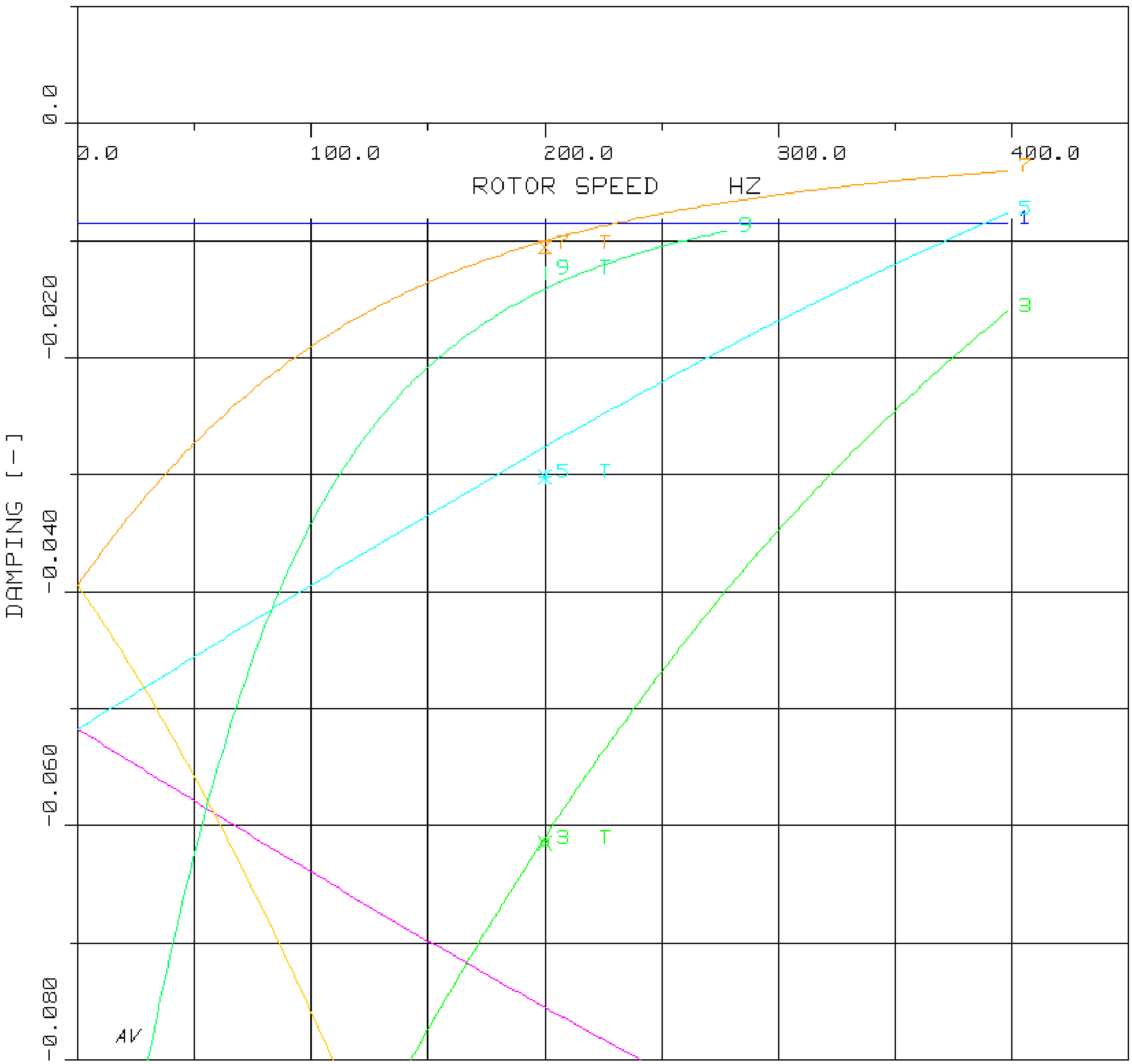


Fig. 205 Damping diagram with damping values calculated from the Laplace transformation and the Nyquist method of the transient analysis with impulse excitation

10.8.2 Direct Method

The transient response was also analyzed with the direct method in SOL 109 and compared to the results of the modal method. The results are compared in Fig. 206 through Fig. 209. The results are practically identical. Only the tilt of rotor A has higher amplitudes in the direct method. The reason is the low damping in the direct solution shown in Fig. 190. The same was also found in the frequency response analysis.

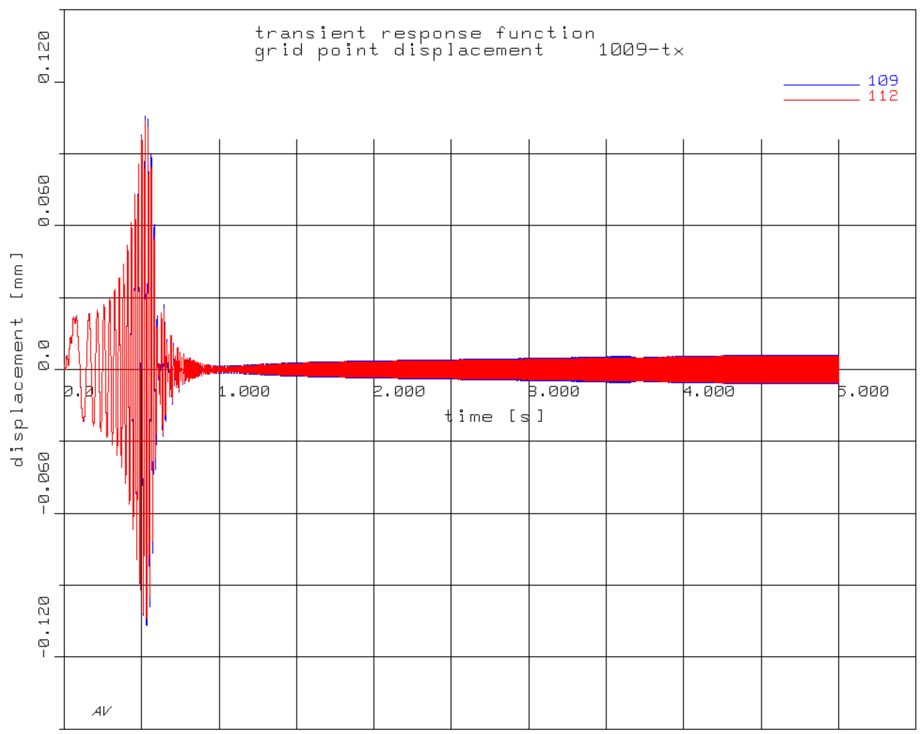


Fig. 206 Translation response of rotor A with direct method (blue) and modal method (red)

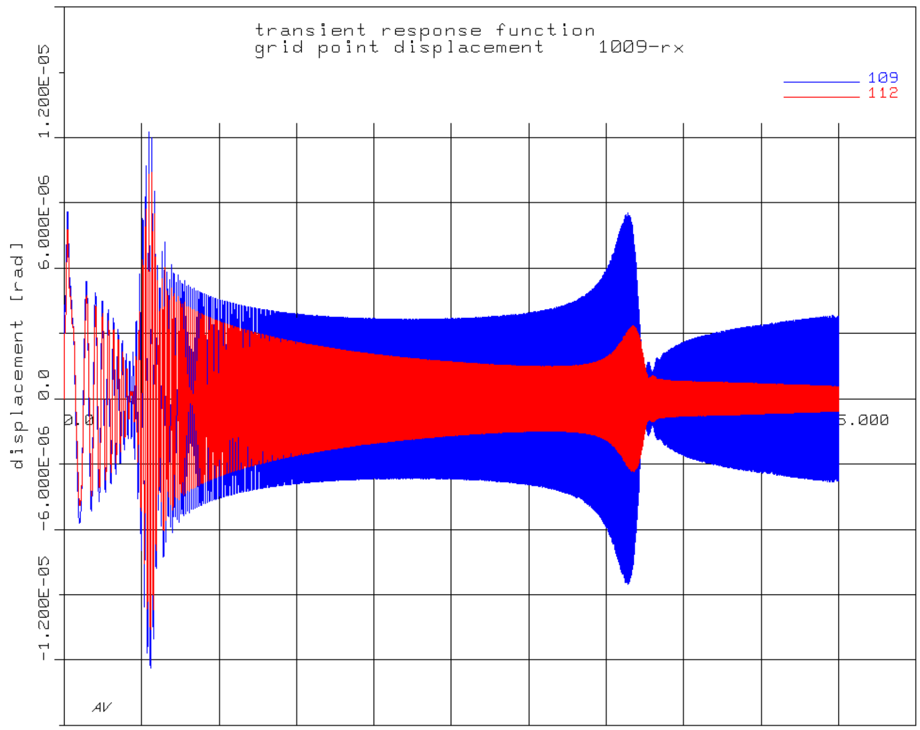


Fig. 207 Tilt response of rotor A with direct method (blue) and modal method (red)

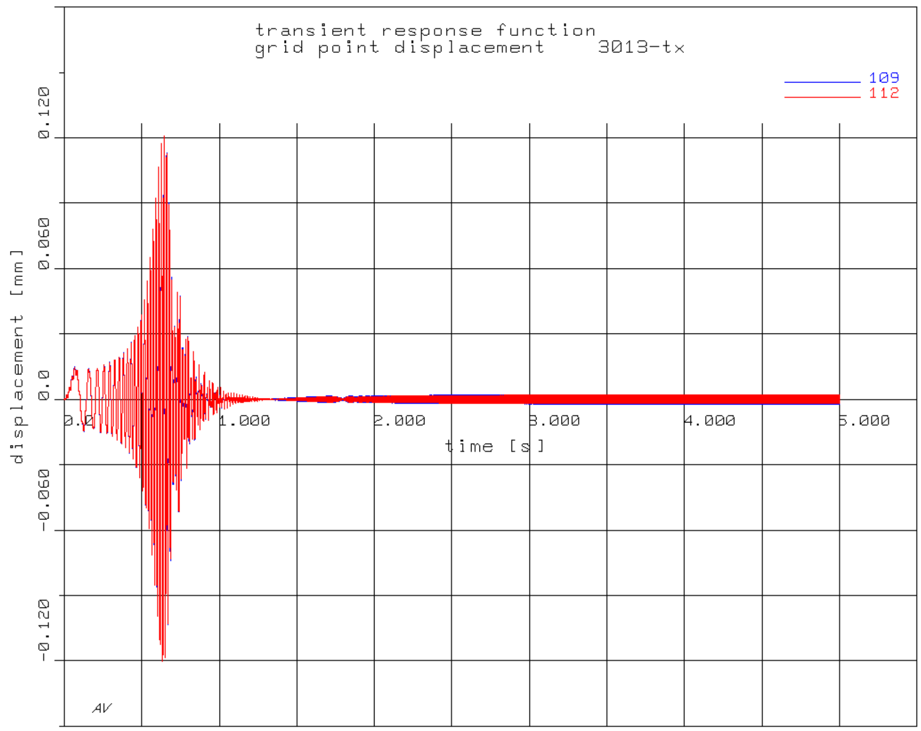


Fig. 208 Translation response of rotor B with direct method (blue) and modal method (red)

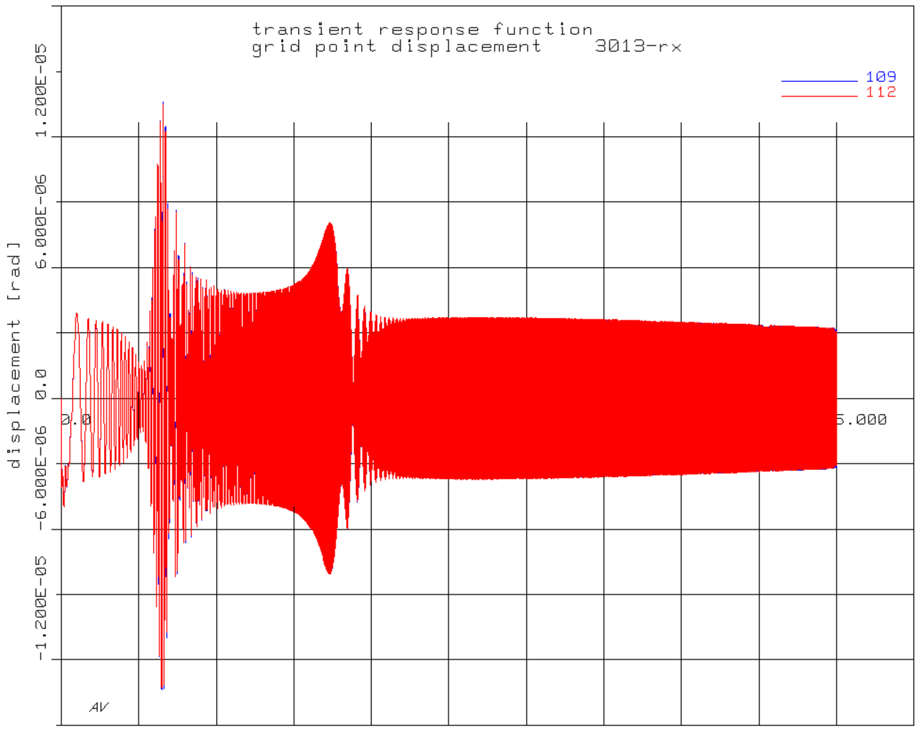


Fig. 209 Tilt response of rotor B with direct method (blue) and modal method (red)

10.9 Analysis of a Model with one Rotor

Analyses with a model with one rotor were done for all solutions. The basic models are:

r120bwd_xxx.dat	for the rotating system
r121bwd_xxx.dat	for the fixed system

The damping in the bearings was increased in order to avoid instabilities and the W3R and W4R parameters were used in order to compare the results of modal and direct methods.

10.9.1 Complex Modes

For the verification of the response analyses, the Campbell diagrams were established.

10.9.1.1 Fixed system

Campbell diagram r121bwd_110

The critical speeds are for forward whirl are:	57.0	translation
	205.0	tilt
Backward whirl:	57.0	translation
	57.0	tilt

The forward whirl motions are the red lines and the backward whirl the blue lines in Fig. 210. The eigenfrequencies of the forward and backward whirl translation modes are identical. The critical speeds are found at the crossing points with the 1P line denoted by A and B in the figure. Asynchronous analyses are done along the vertical lines denoted by X and Y at 57 and 205 Hz, respectively.

The damping curves are shown in Fig. 211. The damping values of the forward whirls are decreasing and the damping of the backward whirl modes are increasing with speed.

The resonance peaks are:

A	Translation forward whirl
B	Tilt forward whirl
C	205 Hz rotor speed, translation forward whirl
D	57 Hz rotor speed, tilt forward whirl
E	Tilt backward whirl
F	205 Hz rotor speed, tilt backward whirl
G	Translation backward whirl
H	205 Hz rotor speed, translation backward whirl

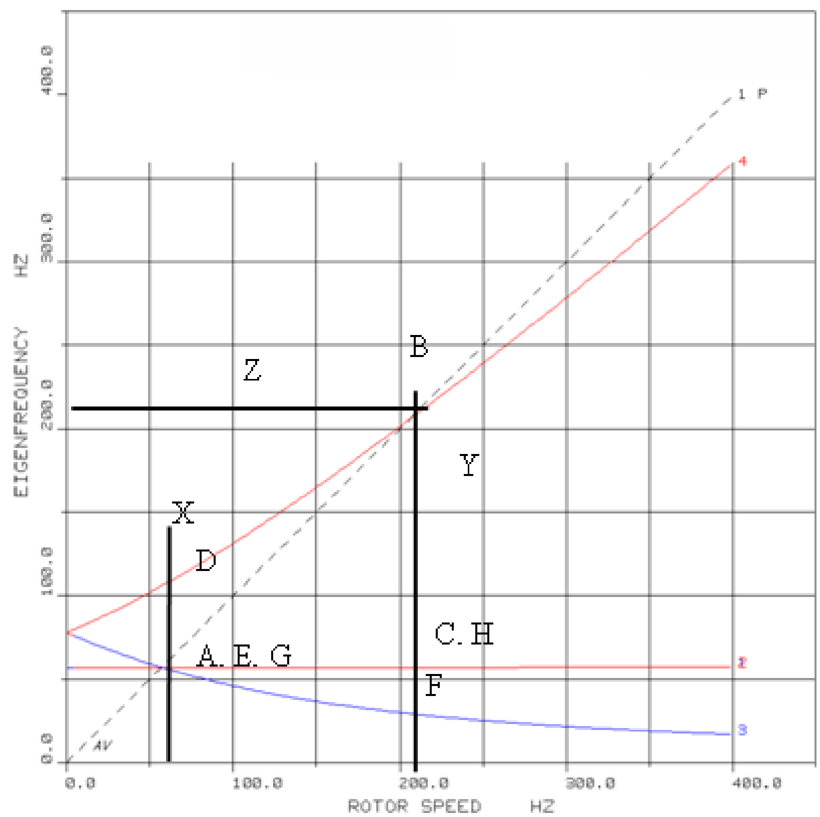


Fig. 210 Eigenfrequencies in the fixed system

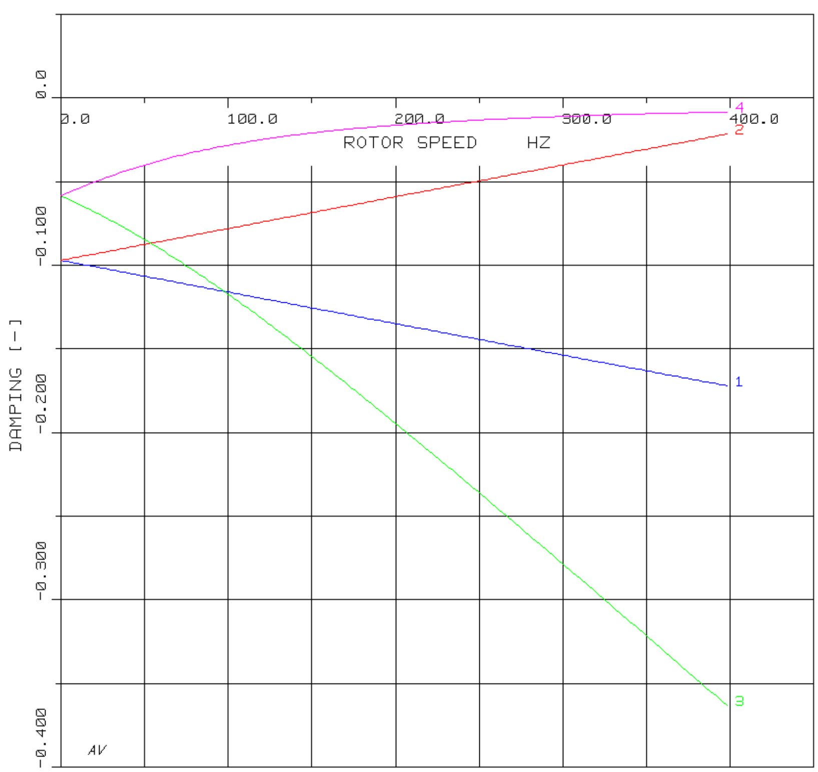


Fig. 211 Damping in the fixed system

10.9.2 Rotating System

The Campbell diagram in the rotating system is shown in Fig. 212 and the damping curves in Fig. 213. In the rotating system the eigenfrequencies of the forward whirl modes decrease and cross the abscissa at the speeds of 57 and 205 Hz. These are the forward whirl resonances. At these points the solutions with negative frequencies become positive and vice versa. After the crossing points the whirl directions are changed to backward whirl.

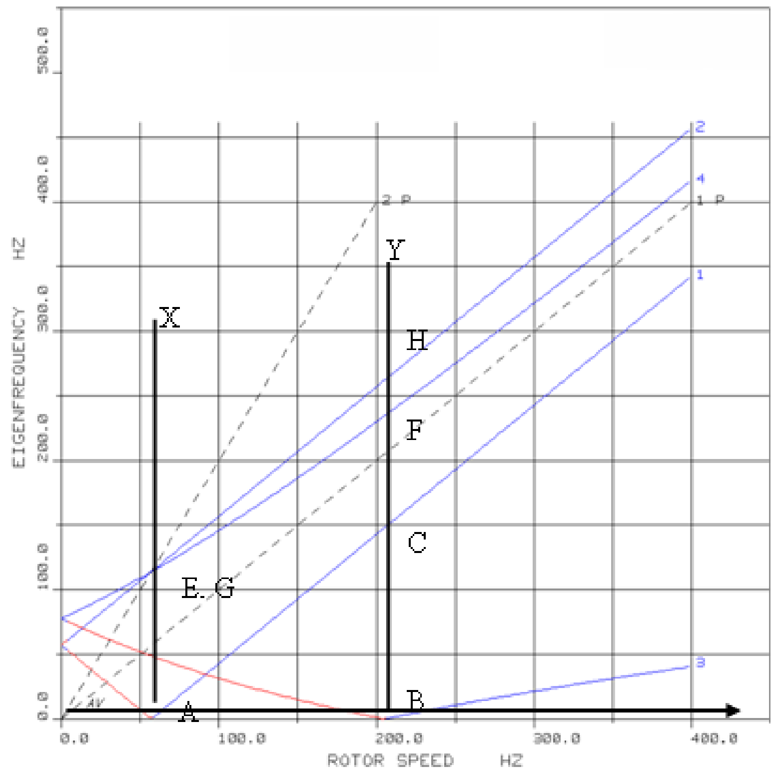


Fig. 212 Eigenfrequencies in the rotating system

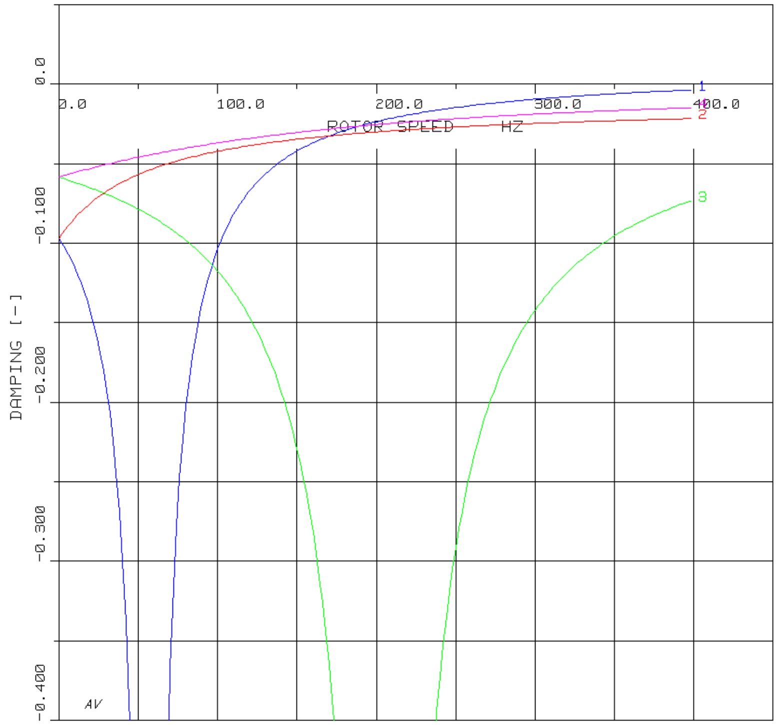


Fig. 213 Damping in the rotating system

10.9.3 Frequency Response Analyses

10.9.3.1 Fixed System

The results of the synchronous analysis of the forward whirl are shown in Fig. 214 for the translation and in Fig. 215 for the tilt motions respectively. The result of an asynchronous analysis with 57 Hz rotor speed is shown in Fig. 216. The peak value of 8 mm is identical to the peak value in the synchronous analysis. The peak for the tilting motion D is shown in Fig. 217. The result of an asynchronous analysis at 205 Hz rotor speed is shown in Fig. 218. In this case, the peak values of approximately 0.00109 are equal and occur at 205 Hz. The results of the synchronous and the asynchronous analyses are in agreement. The translation response is shown in Fig. 219. Here, the peak is very large but this case is not realistic because the unbalance force is in reality acting at 205 Hz and not at 57 Hz.

The results of a synchronous analysis with backward whirl excitation are shown in Fig. 220 and Fig. 221. Both resonances are at 57 Hz and the amplitudes are lower than those of the forward whirl because the damping of the backward whirl is higher. The results of an asynchronous analysis at 57 Hz rotor speed are shown in Fig. 222 and Fig. 223 for translation and tilt, respectively. The amplitudes are equal to those of the synchronous analysis.

The synchronous analysis was also done with the direct method with SOL 108. The results are shown as red lines in Fig. 224 and Fig. 225. For higher speeds, the results are different from the previous analyses where four real modes were used (blue lines). The modal solution was repeated with eight real modes accounted for. The results are shown with green lines. The results of this analysis are close to those from the direct method. In this case the truncation of higher modes leads to erroneous results for the higher rotor speeds.

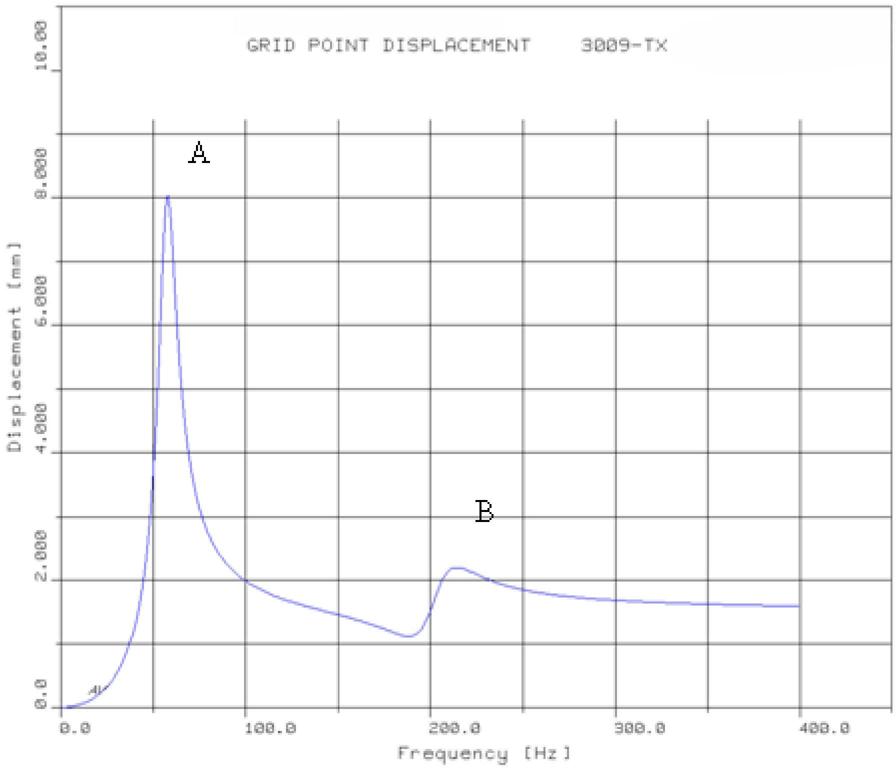


Fig. 214 Synchronous analysis. Magnitude of translation in the fixed system

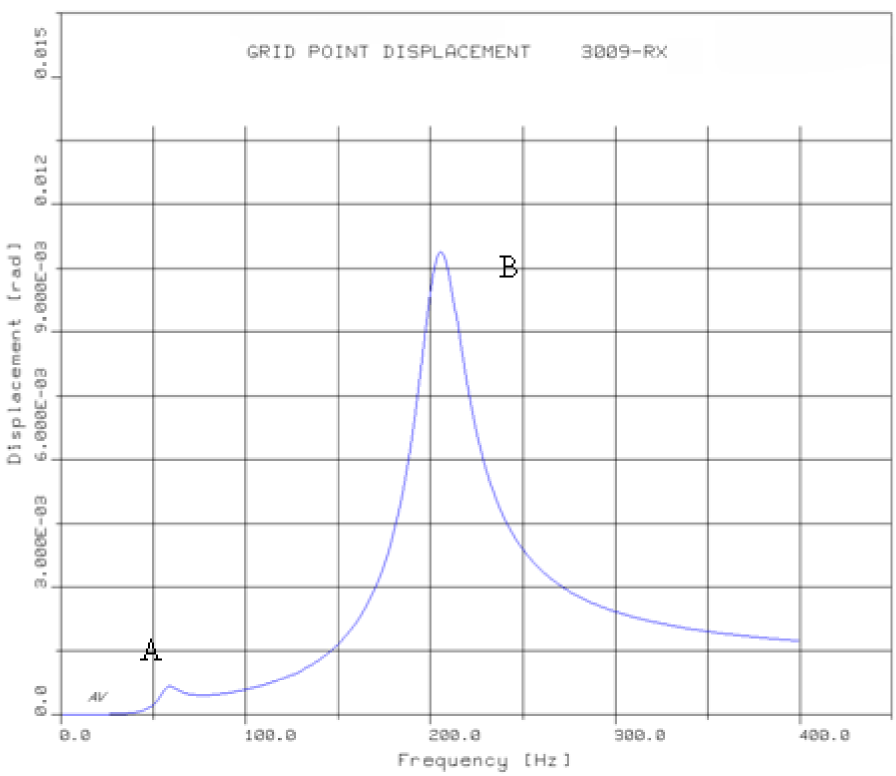


Fig. 215 Synchronous analysis. Magnitude of tilt in the fixed system

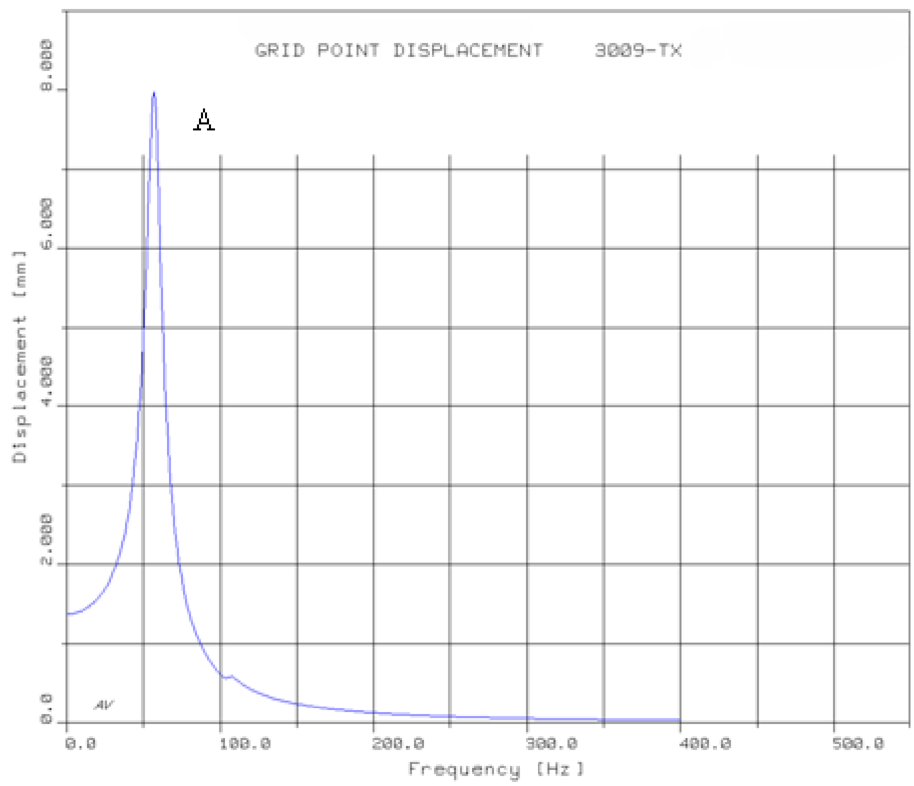


Fig. 216 Translation response of an asynchronous analysis for 57 Hz rotor speed.

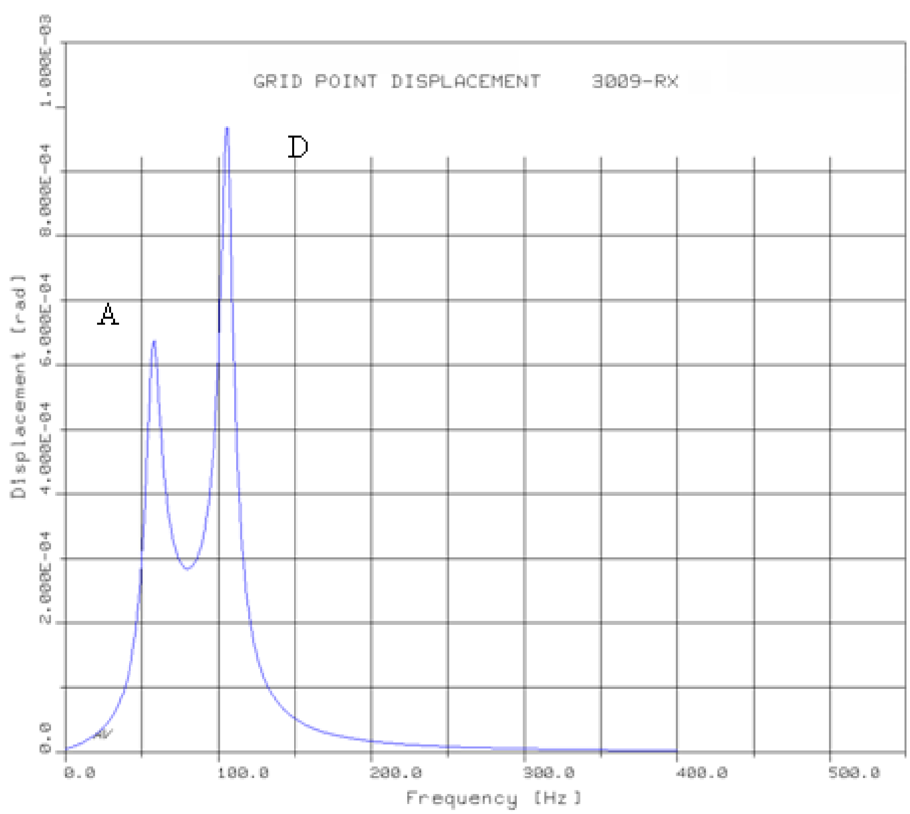


Fig. 217 Tilt response of an asynchronous analysis for 57 Hz rotor speed

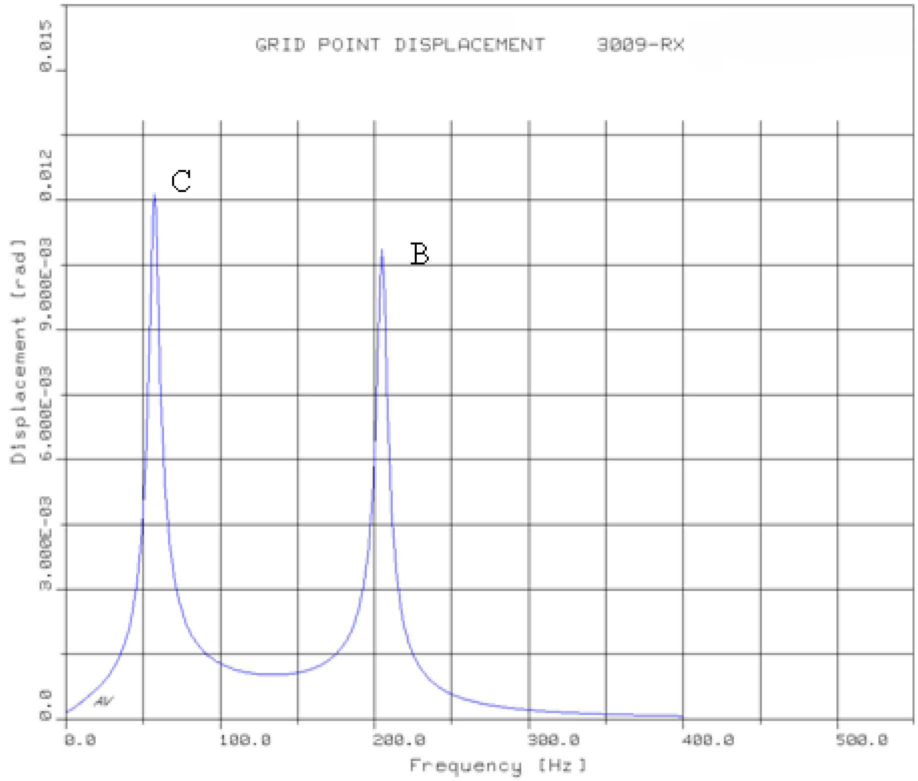


Fig. 218 Tilt response for an asynchronous analysis at 205 Hz rotor speed.

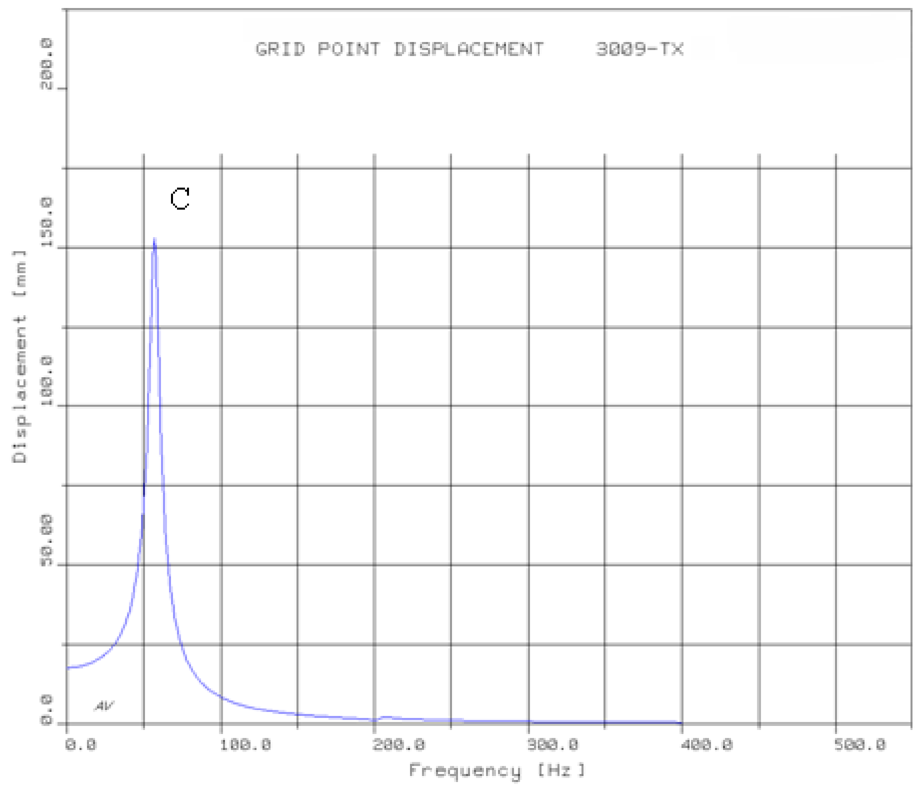


Fig. 219 Translation response for an asynchronous analysis at 205 Hz rotor speed.

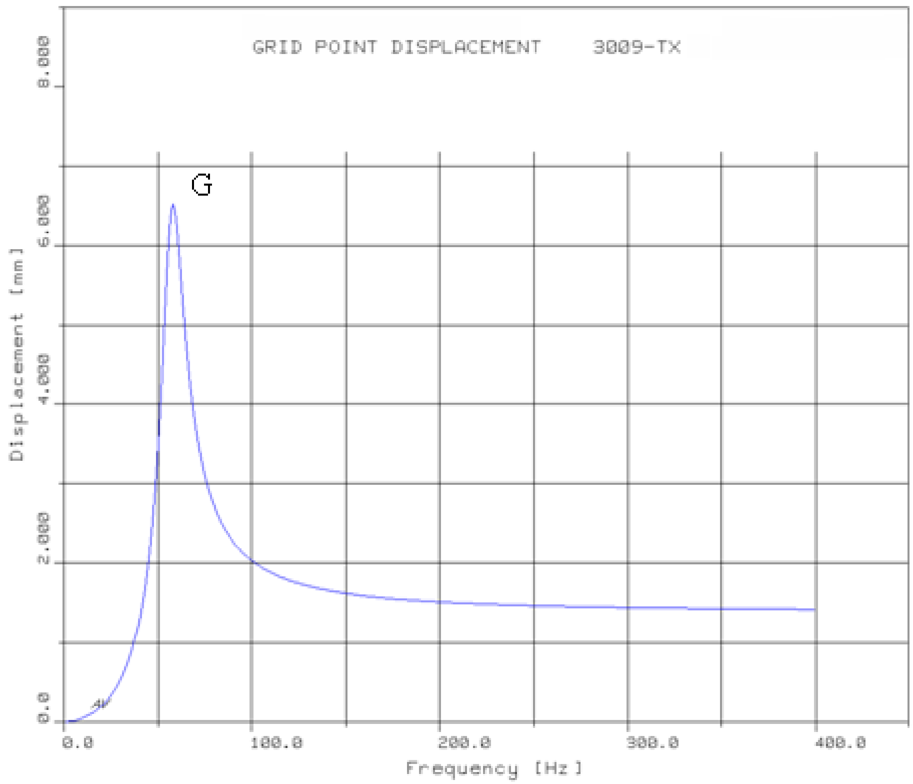


Fig. 220 Synchronous analysis, backward whirl response of translation

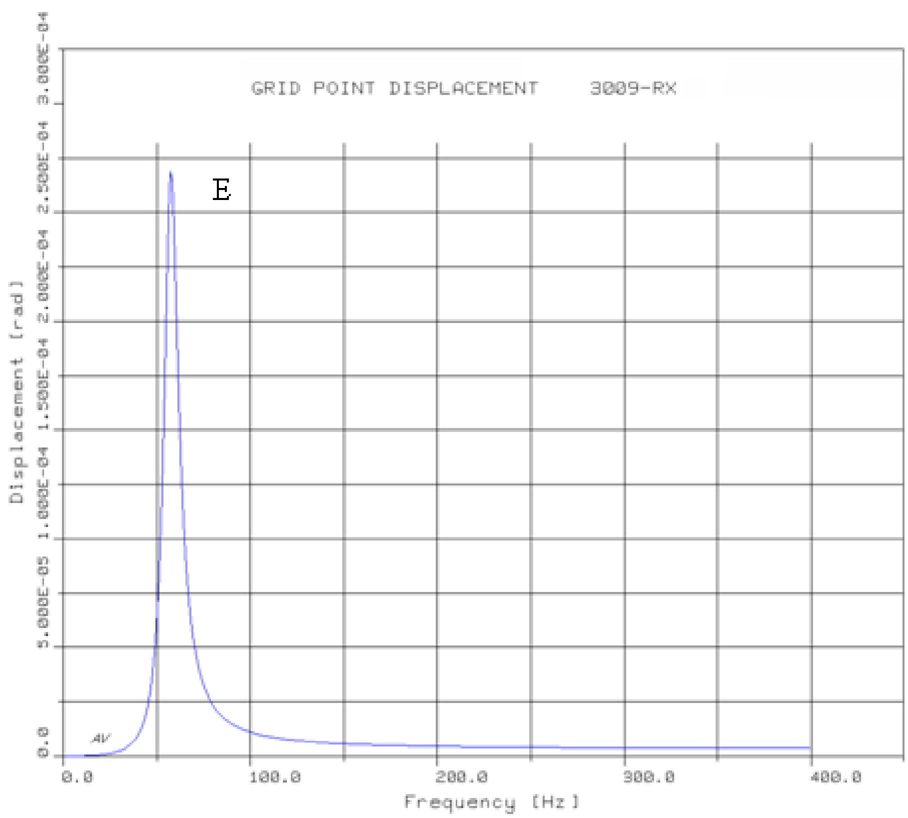


Fig. 221 Synchronous analysis, backward whirl response of tilt

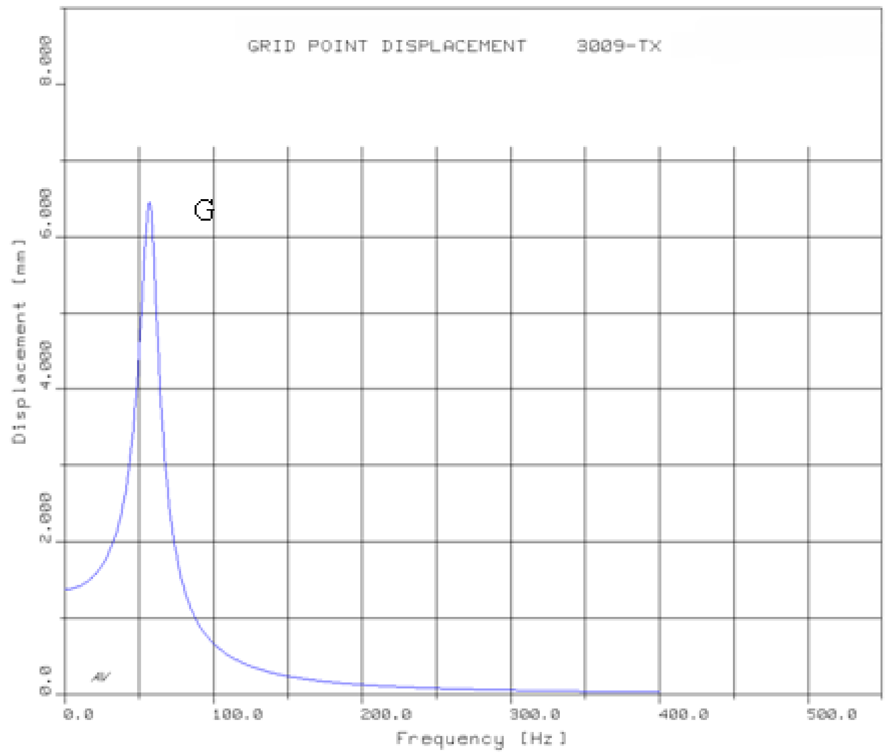


Fig. 222 Translation response of asynchronous analysis with backward excitation at 57 Hz rotor speed

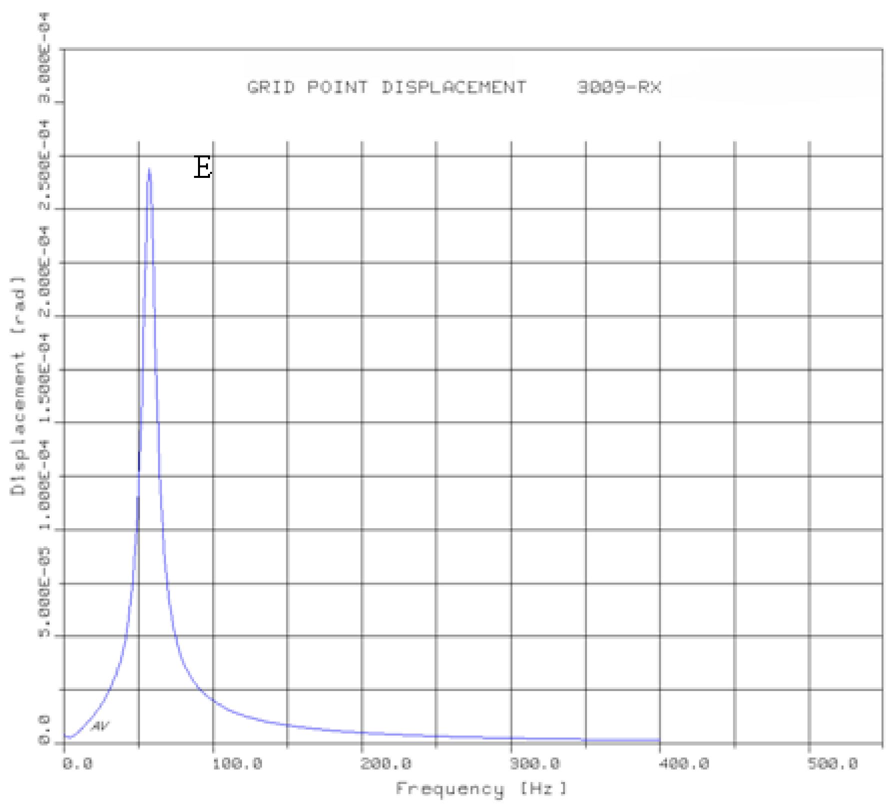


Fig. 223 Translation response of asynchronous analysis with backward excitation at 57 Hz rotor speed

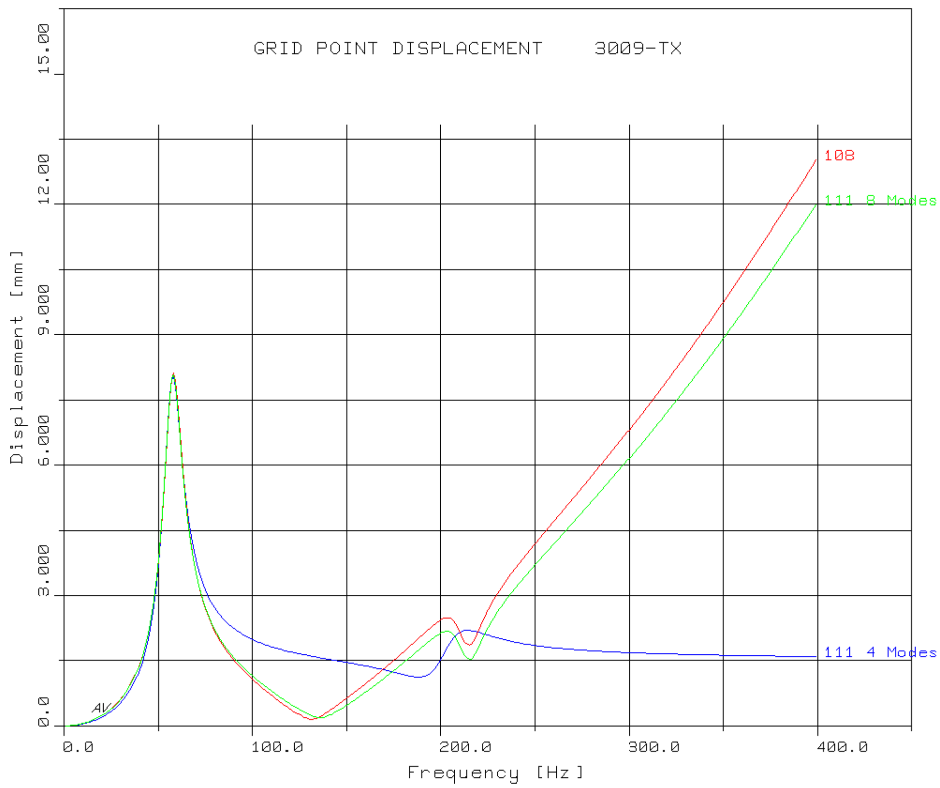


Fig. 224 Synchronous analysis, translation response for modal and direct solutions

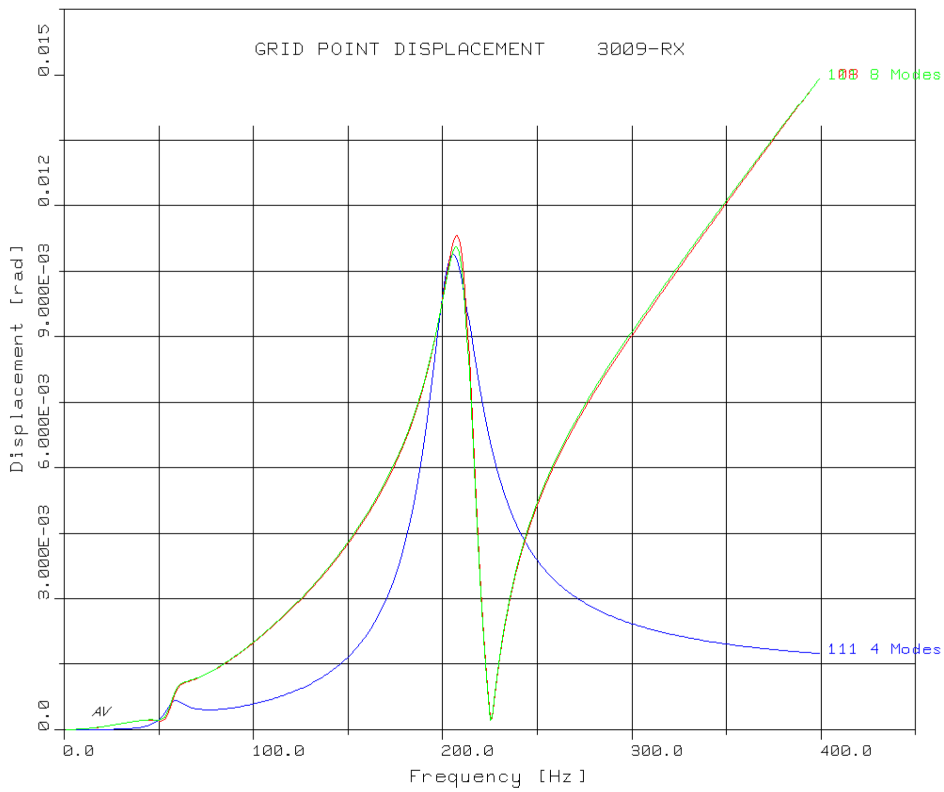


Fig. 225 Synchronous analysis, tilt response for modal and direct solutions

10.9.3.2 Rotating System

The results of a synchronous analysis with forward whirl excitation are shown in Fig. 226 and Fig. 227 for translation and tilt respectively. The results with forward excitation are depicted with blue symbols. The results of a synchronous analysis with backward excitation are depicted with red lines. The results are identical to those for forward excitation because at zero frequency there is no whirl motion at all. In this analysis EORDER = 0.0.

The backward whirl response is shown in Fig. 228 for translation and in Fig. 229 for the tilting motion. The excitation direction was forward. In the rotating system, the whirl motion is inverted. Here EORDER = 2.0.

The result of an asynchronous analysis at 57 Hz rotor speed is shown in Fig. 230. The amplitude of the translation mode at resonance point A is identical to the synchronous peak in Fig. 226. The response at 205 Hz rotor speed is shown in Fig. 231. The magnitude of peak B is equal to the peak in Fig. 227. Excitation in the forward sense, the response at 57 Hz rotor speed is shown in Fig. 232. The magnitude of peak A is the same as for the previous case. The magnitude of peak G is close to the peak found for the synchronous backward resonance in Fig. 228. The tilt response is shown in Fig. 233. Here the peak of the resonance E is shown. The value is close to that of the synchronous analysis in Fig. 229.

Asynchronous analyses at 205 Hz rotor speed are shown for translation and tilt in Fig. 234 and Fig. 235 respectively. Here the resonances B, C, F and H are shown and can be compared to the Campbell diagram in Fig. 212.

The synchronous results for analysis in the rotating and fixed system are shown in Fig. 236 and Fig. 237 for translation and tilt respectively. The results are identical.

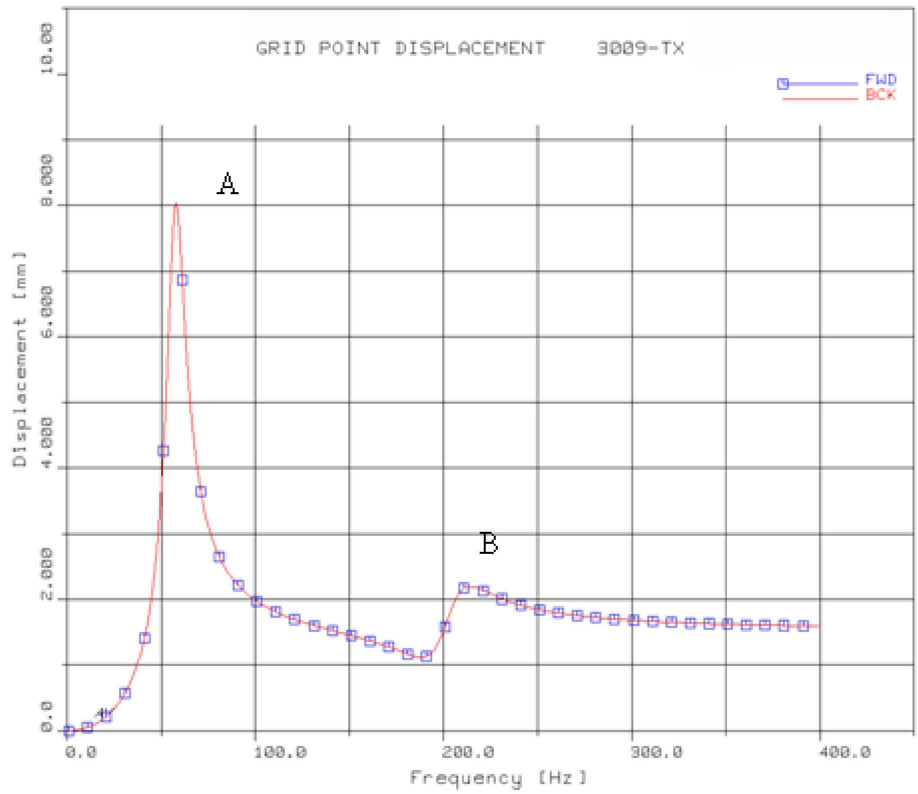


Fig. 226 Synchronous analysis, Magnitude of translation. Forward (symbols) and backward excitation

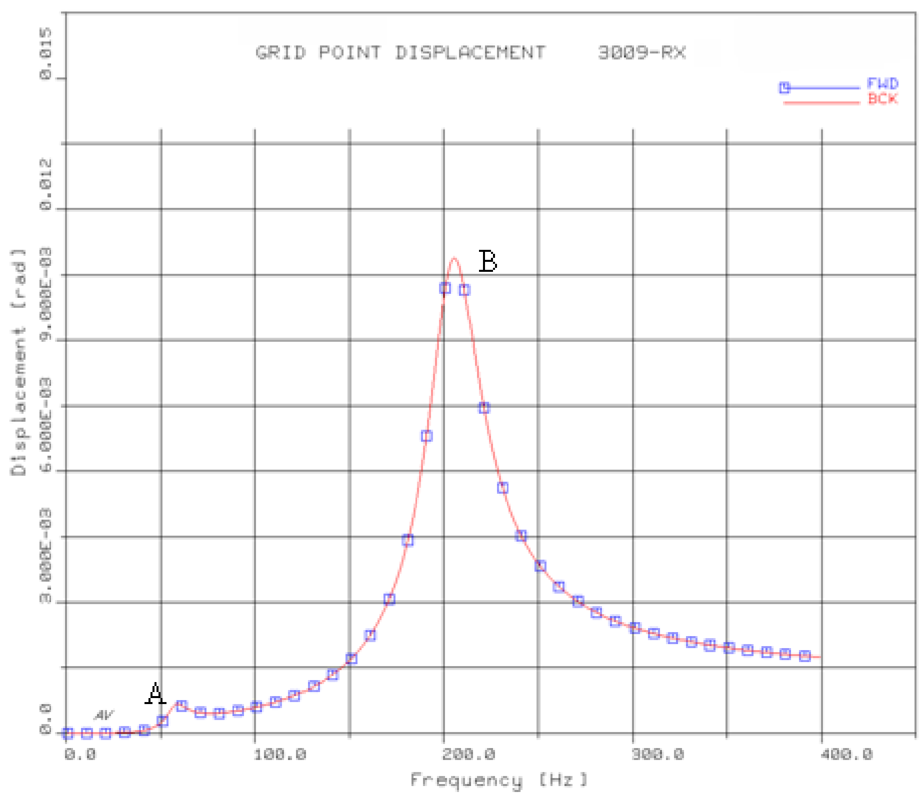


Fig. 227 Synchronous analysis, Magnitude of tilt angle. Forward (symbols) and backward excitation

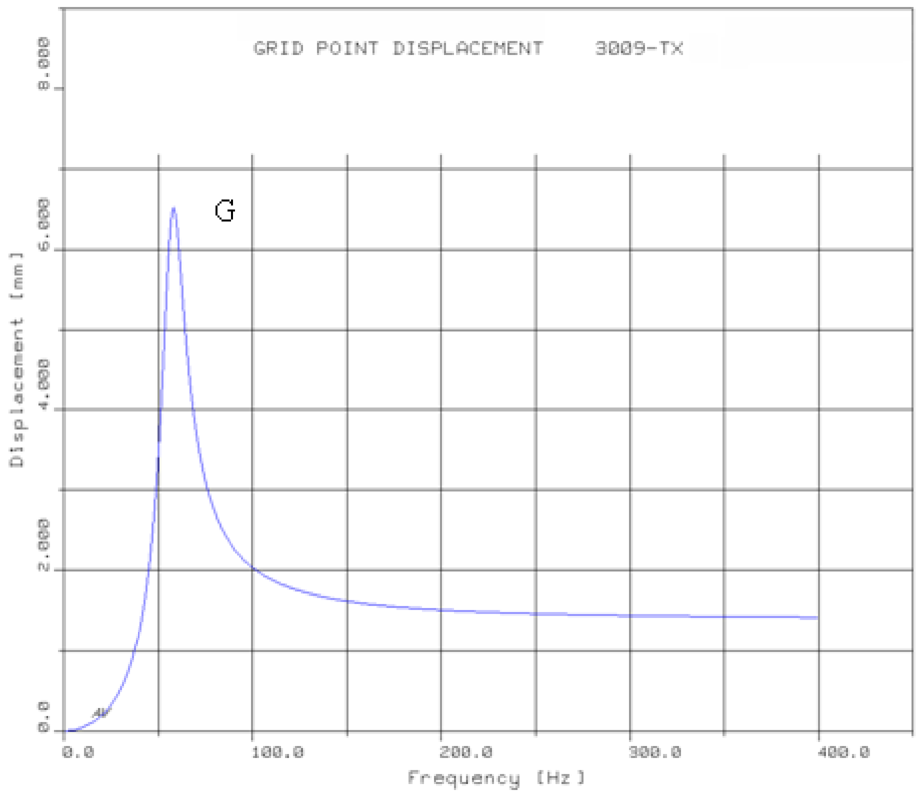


Fig. 228 Backwards whirl resonance for translation, EORDER=2

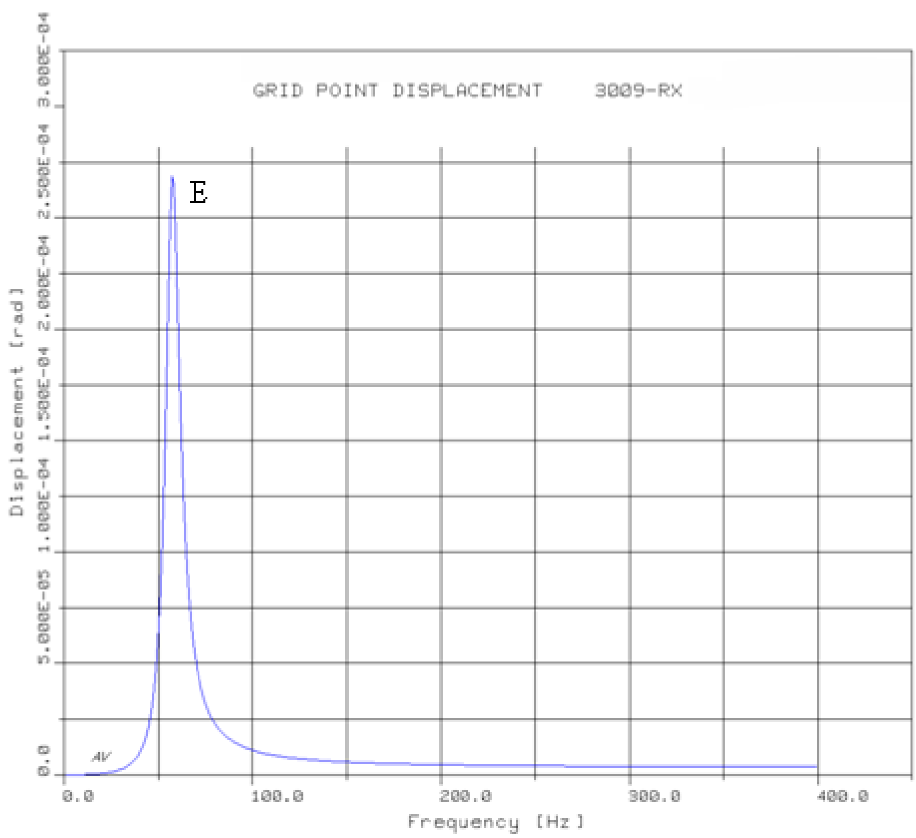


Fig. 229 Backwards whirl resonance for translation, EORDER=2

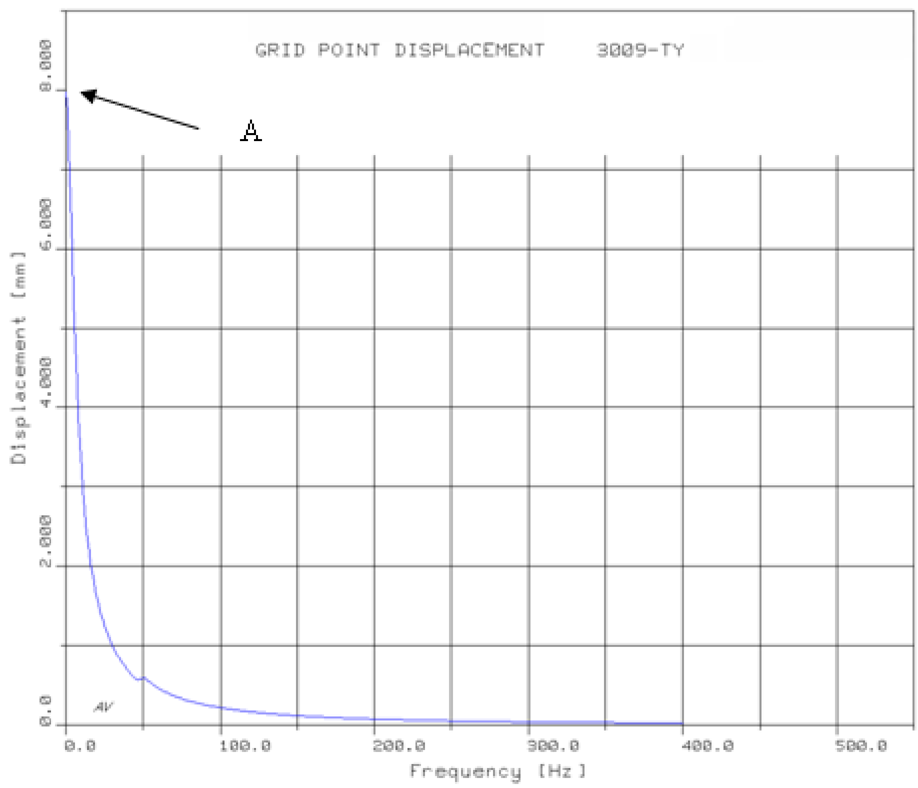


Fig. 230 Translation response for backward excitation at rotor speed 57 Hz

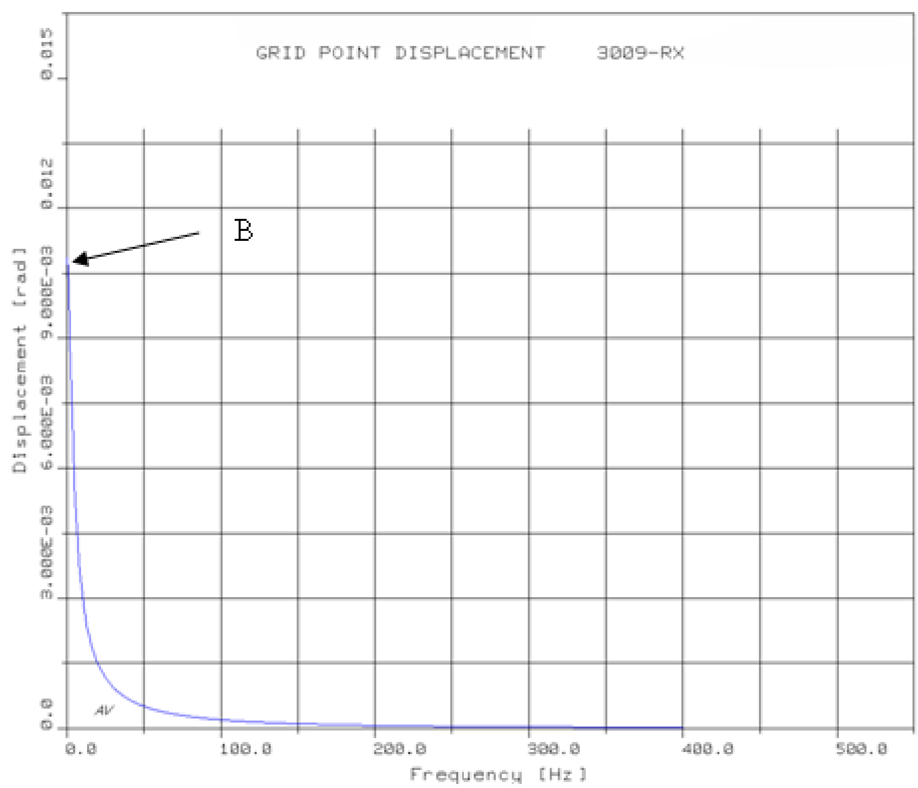


Fig. 231 Tilt response for backward excitation at rotor speed 205 Hz

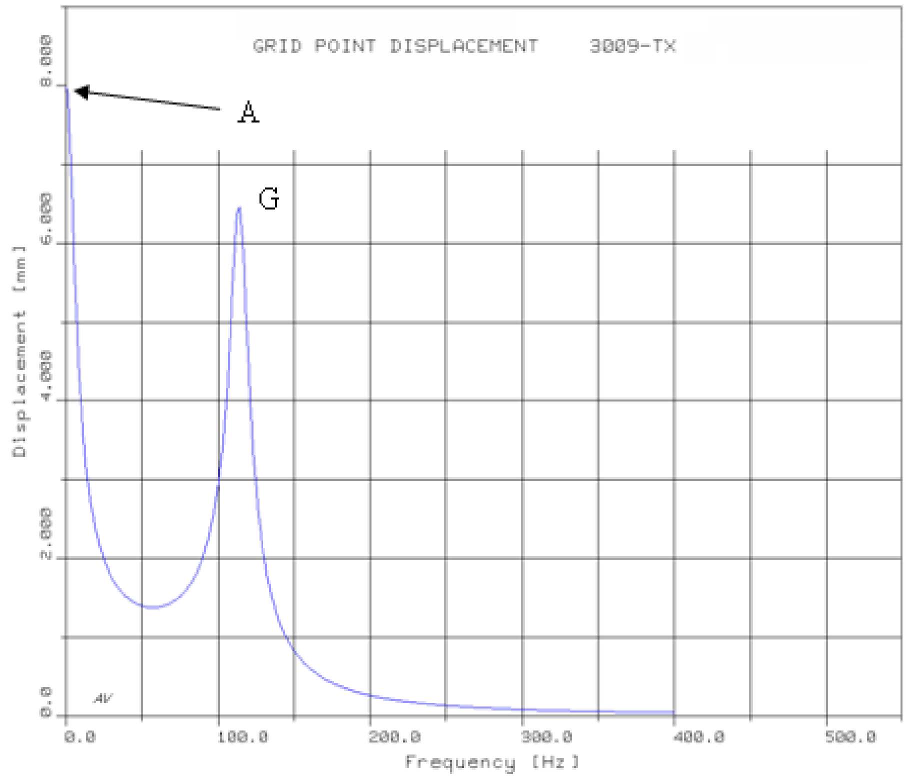


Fig. 232 Asynchronous analysis at 57 Hz rotor speed, forward whirl excitation, translation response

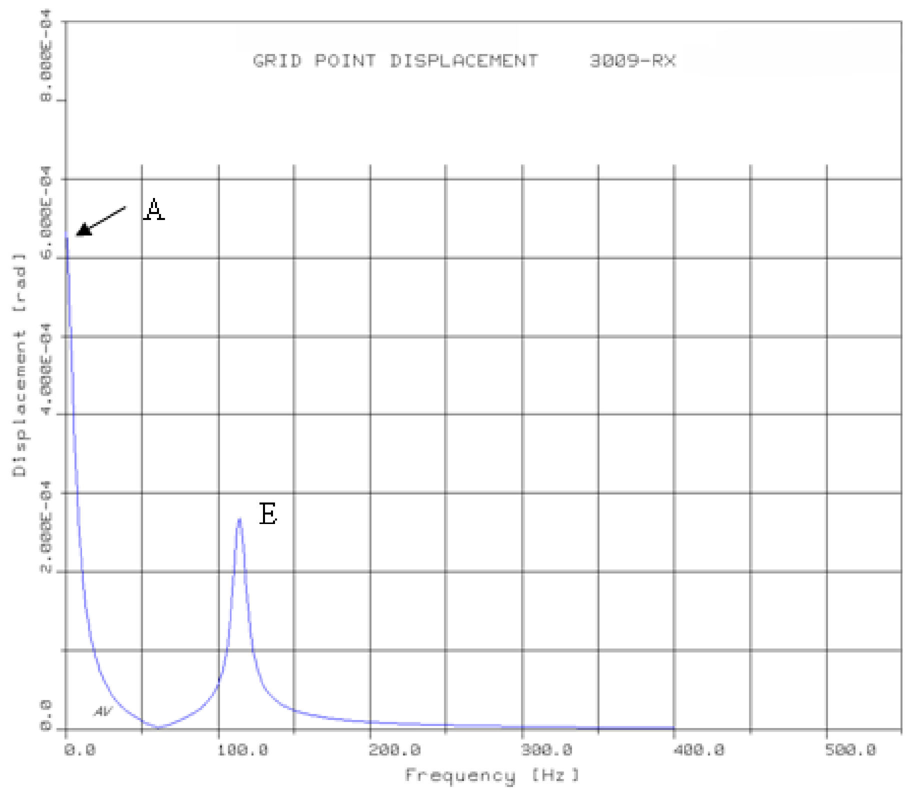


Fig. 233 Asynchronous analysis at 57 Hz rotor speed, forward whirl excitation, tilt response

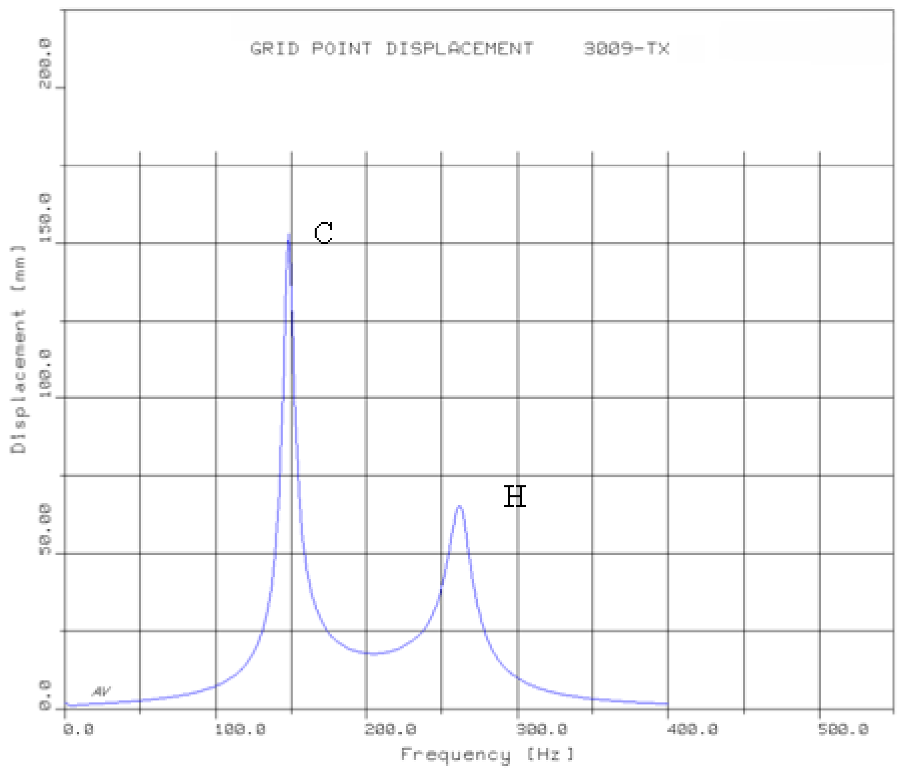


Fig. 234 Asynchronous analysis at 205 Hz rotor speed, forward whirl excitation, translation response

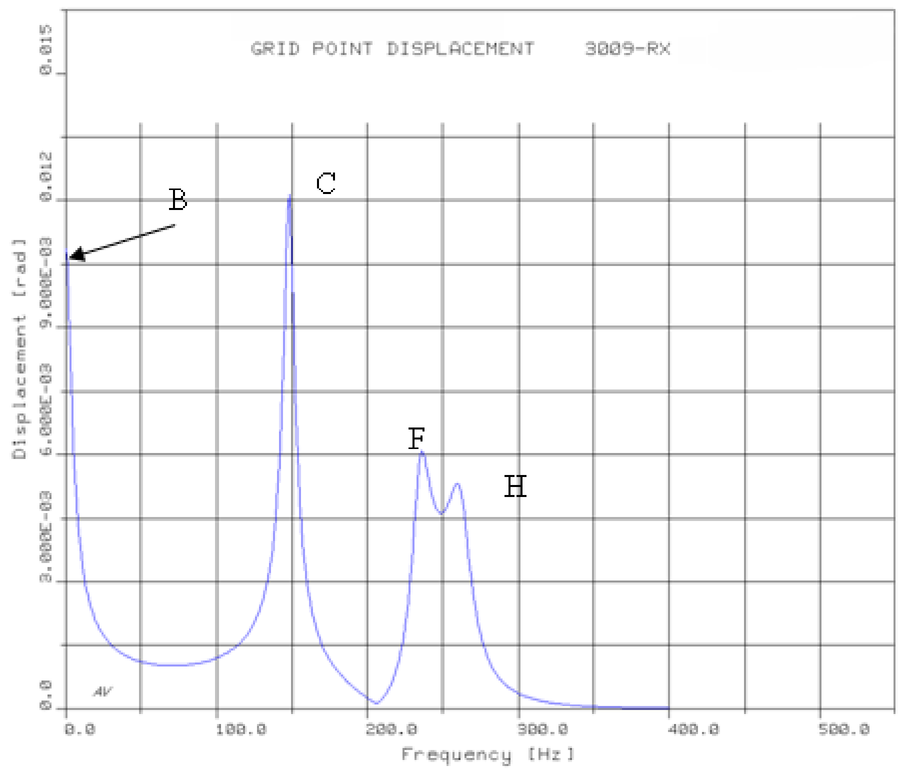


Fig. 235 Asynchronous analysis at 205 Hz rotor speed, forward whirl excitation, Tilt response

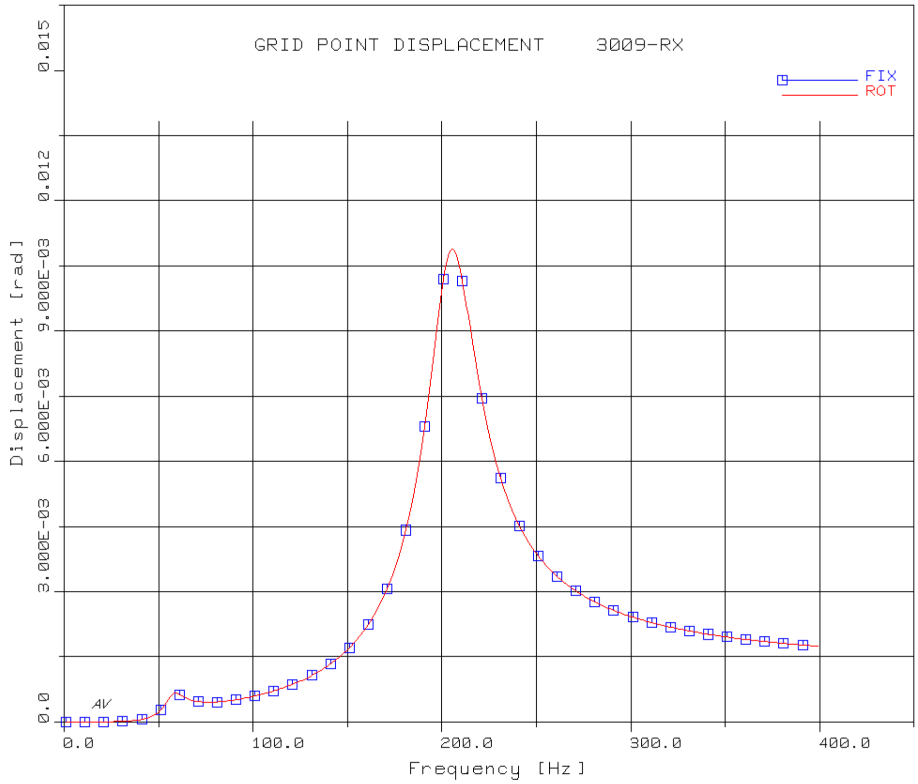


Fig. 236 Synchronous analysis, translation amplitudes are identical for both analysis systems

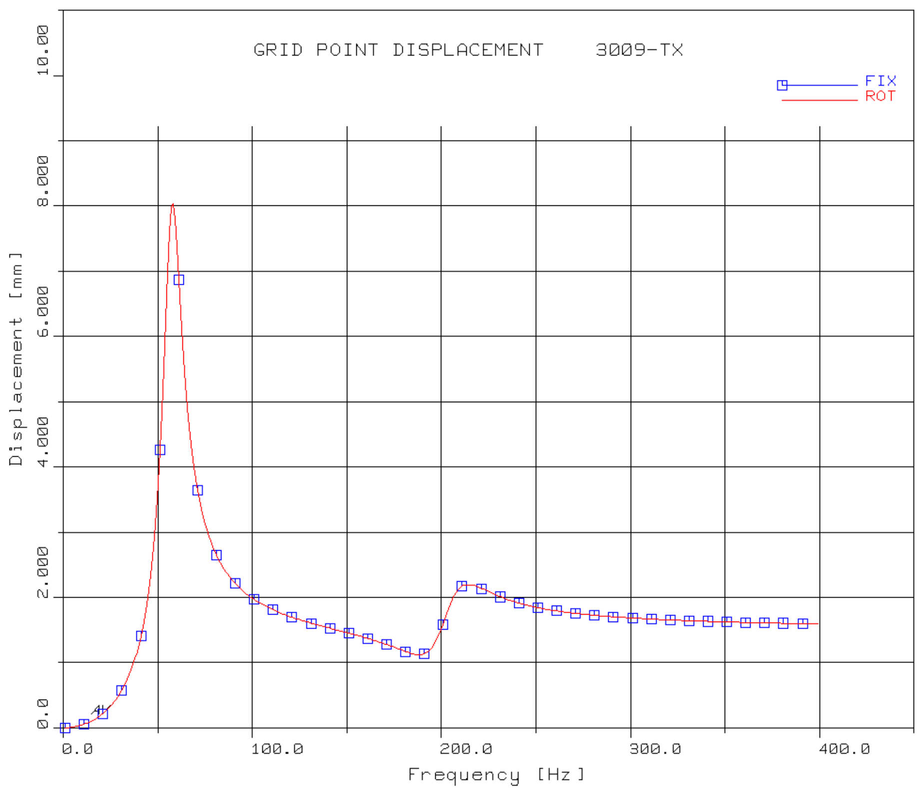


Fig. 237 Synchronous analysis, tilt amplitudes are identical for both analysis systems

10.9.4 Transient Analysis

Transient analysis is done using a sweeping time function with variable rotor speed in the synchronous case and for a fixed rotor speed in the asynchronous case. It is also possible to do analyses with fixed excitation frequency and variable rotor speed. Because the rotor is passing through the resonance points, the structure does not have the time to come into steady resonance. Therefore, the resonance peaks found in transient analysis will normally be lower than the steady-state solution found in the frequency response analysis.

10.9.4.1 Fixed system

The structure is now excited by a sine function in the x-direction and a cosine function in the y-direction. The first parts of the functions are shown in Fig. 238 and the last parts in Fig. 239 where the integration time steps are shown as symbols. The frequency is varying linearly from 0 to 400 Hz in 4 seconds as shown in Fig. 240.

Asynchronous analysis is done with increasing frequency with time and keeping the rotor speed constant. This is a vertical line in the Campbell diagram. The results of an asynchronous analysis at 205 Hz rotor speed is shown in Fig. 241. The response is harmonic. The magnitude of the oscillation amplitude is shown in Fig. 242. The peak value is around 0.009 and is lower than the value found from the frequency response analysis of 0.0109. The reason is that the structure is not vibrating in steady-state during the sweep. The analysis was repeated with a lower sweep rate of 400 Hz in 8 seconds. The peak amplitude increased. The functions for the two sweep rates are compared to the result of the frequency response analysis in Fig. 243. The peak for the higher sweep rate is closer to the steady-state value of SOL 111 shown as a green line. Also the frequency of the peak decreases slightly to the steady-state value with slower sweep rate. The response to the fast excitation shows dynamic “ripples” after the resonance peak. The response of the translation motion is shown in Fig. 244 and the magnitude in Fig. 245. Also in this case, there is an influence of sweep rate as shown in Fig. 246.

Asynchronous analysis can also be done by keeping the excitation frequency constant and varying the rotor speed. This corresponds to a horizontal line in the Campbell diagram denoted by Z in Fig. 210. The results for a fast sweep (0 to 400 Hz in 4 seconds) are shown in Fig. 247 and the magnitude in Fig. 248. With a slowly increasing rotor speed from 195 to 215 Hz in 4 seconds the amplitude is increased as shown in Fig. 249.

A comparison of the magnitudes calculated for the synchronous case with SOL 112 and SOL 111 are shown in Fig. 250 for the translation response. The magnitudes of the peaks are close, even for the fast sweep rate. The response of the second peak (tilt) is very weak for the transient analysis. A similar plot of the tilt response is shown in Fig. 251. Here the influence of sweep rate is great and the peak is higher for the steady-state case of SOL 111.

The results of an analysis with SOL 109 are shown in Fig. 252 and Fig. 253 for translation and tilt, respectively. Also here, there is a truncation effect like the one found for the frequency response shown in Fig. 224 and Fig. 225. The modal solutions with eight real modes are in reasonable agreement with the direct method.

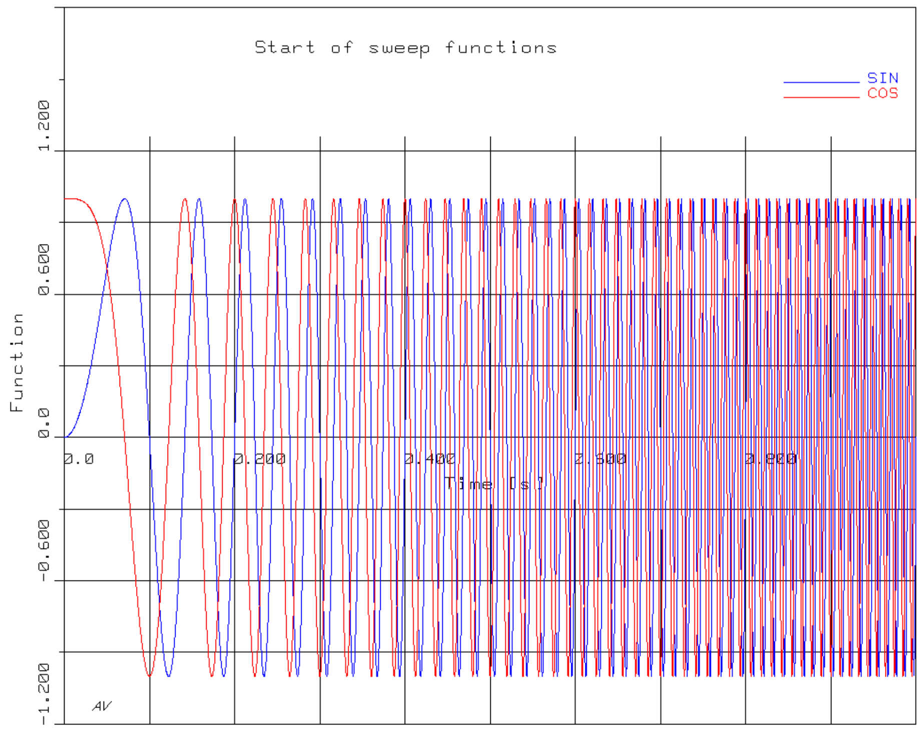


Fig. 238 First part of excitation function

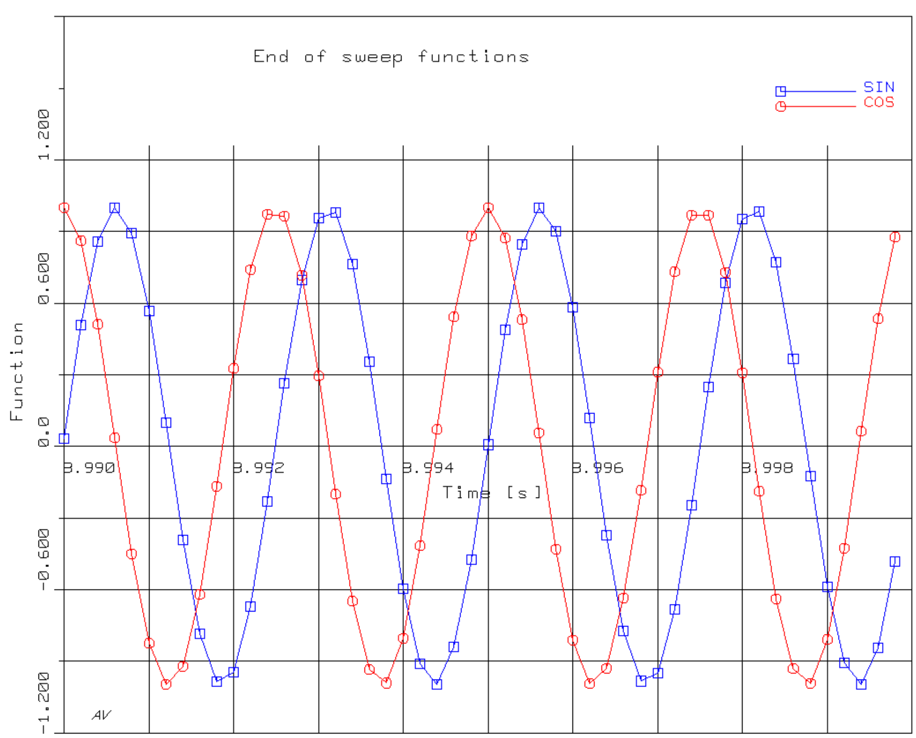


Fig. 239 Last part of excitation function with symbols for the integration points

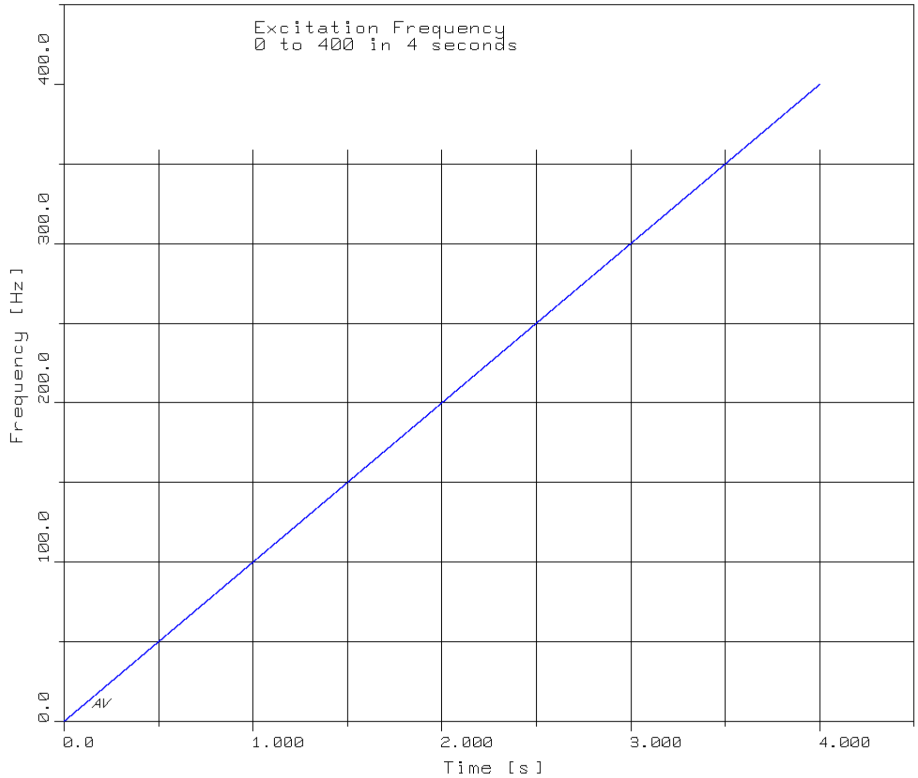


Fig. 240 Sweep frequency as function of time

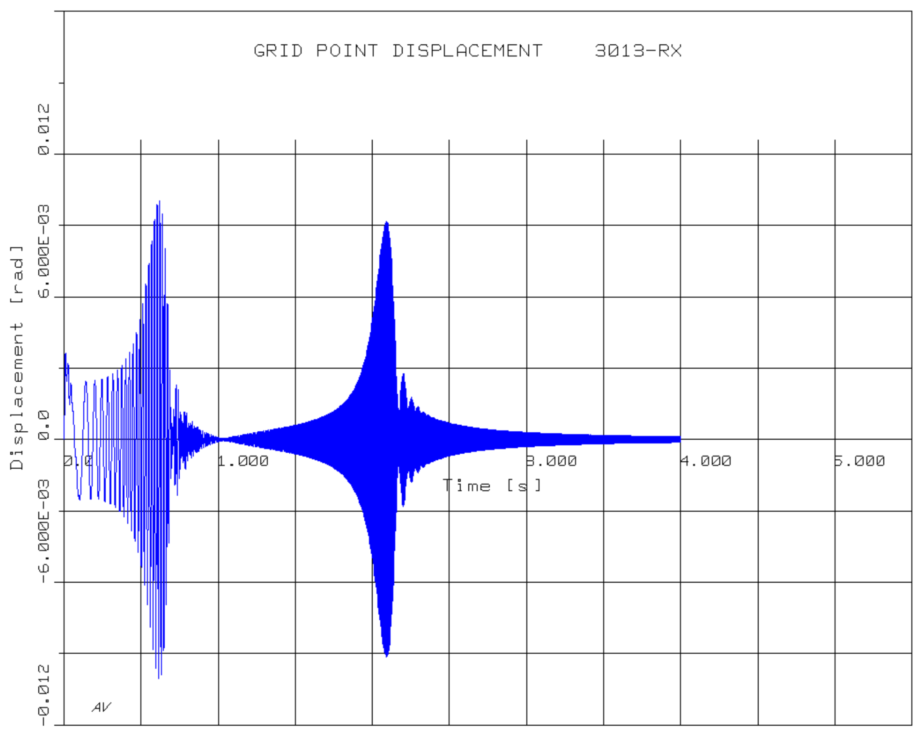


Fig. 241 Asynchronous analysis at 205 Hz rotor speed

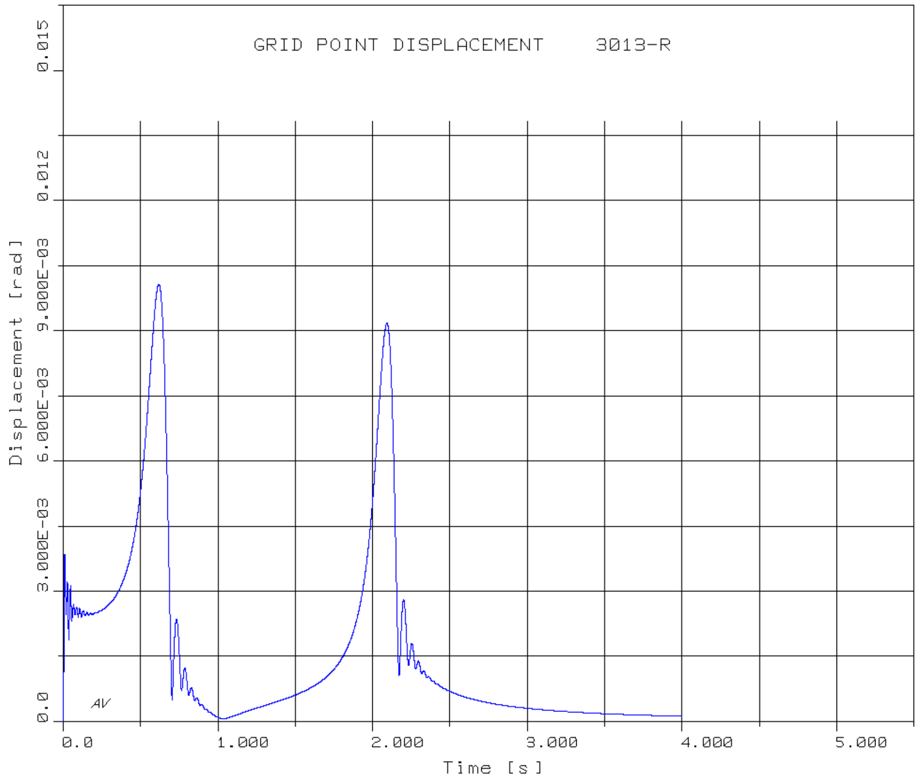


Fig. 242 Magnitude of tilt angle for asynchronous analysis at 205 Hz rotor speed

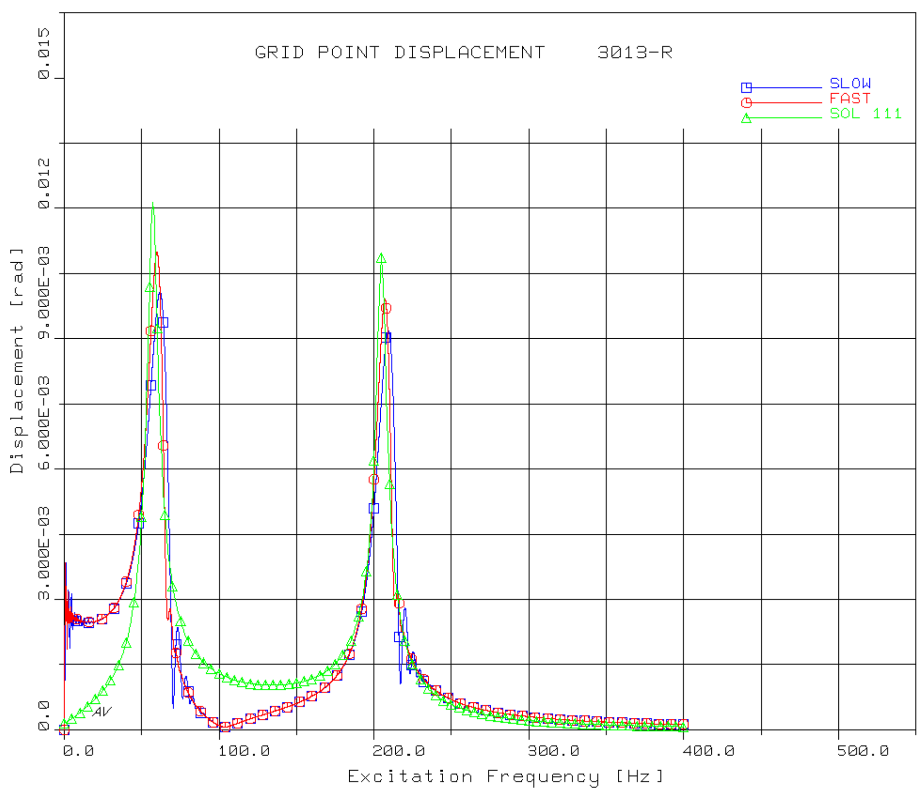


Fig. 243 Slow and fast sweep compared to frequency response analysis

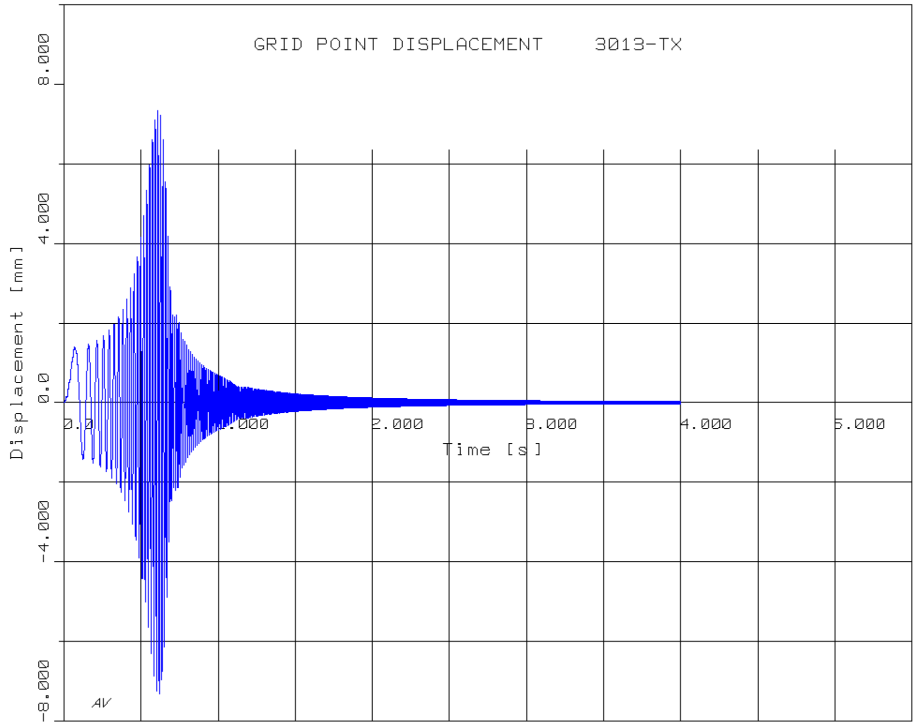


Fig. 244 Translation response for asynchronous analysis at 57 Hz rotor speed

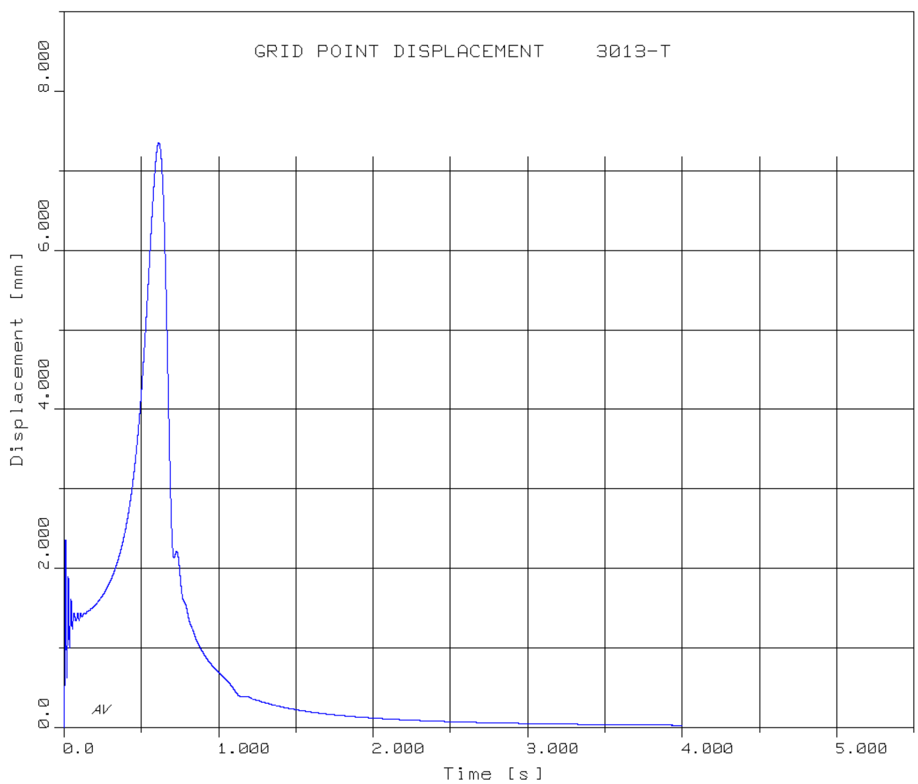


Fig. 245 Magnitude of translation for asynchronous analysis at 205 Hz rotor speed

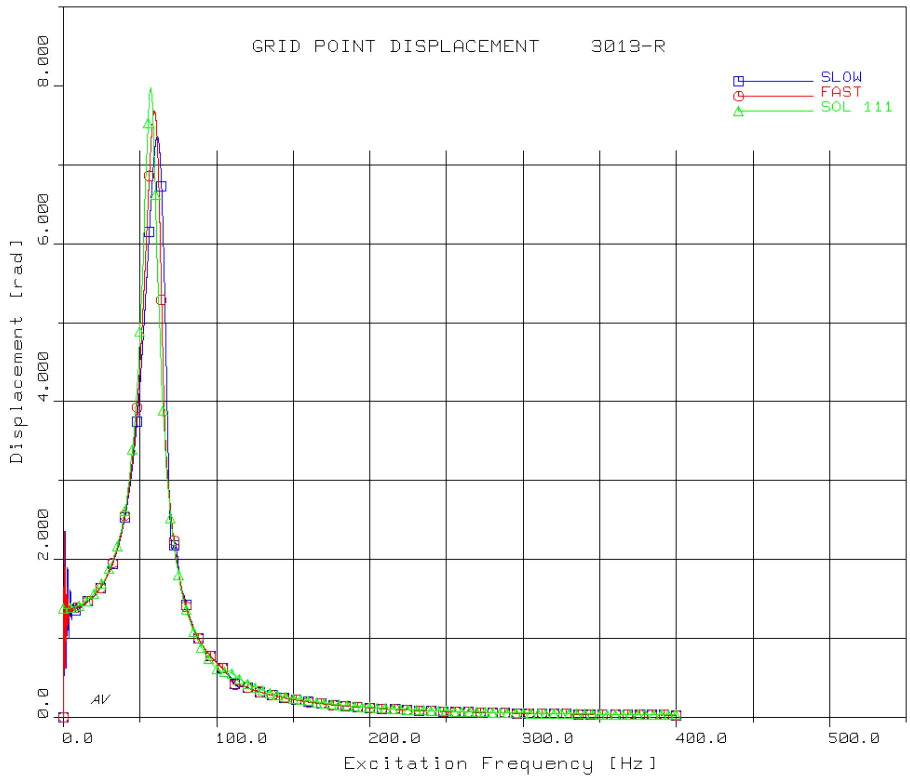


Fig. 246 Slow and fast sweep compared to frequency response analysis

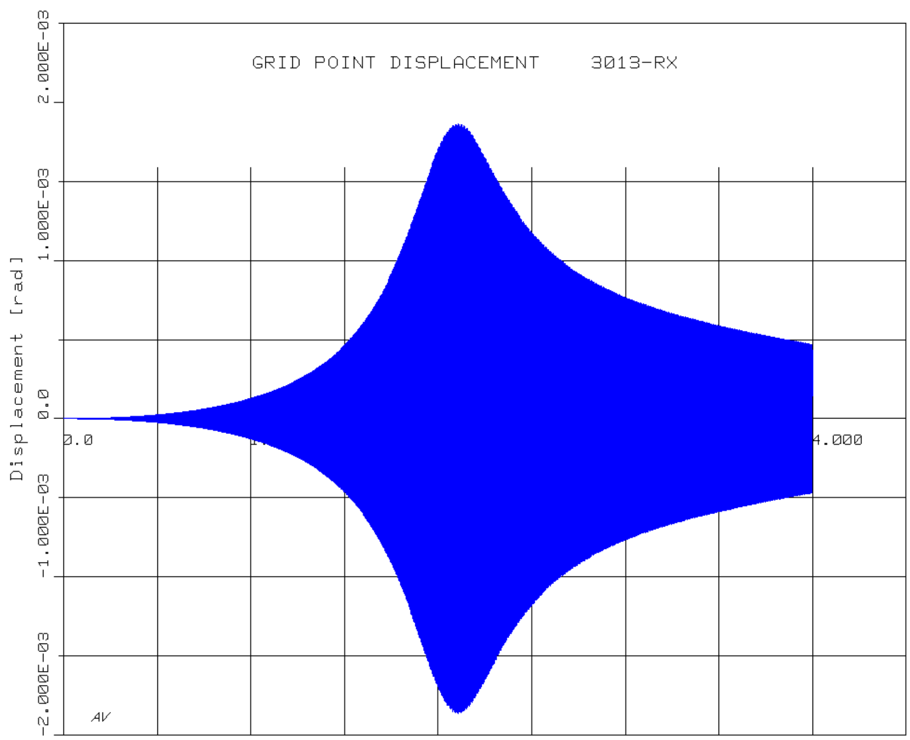


Fig. 247 Asynchronous analysis with excitation at 205 Hz and linearly increasing rotor speed

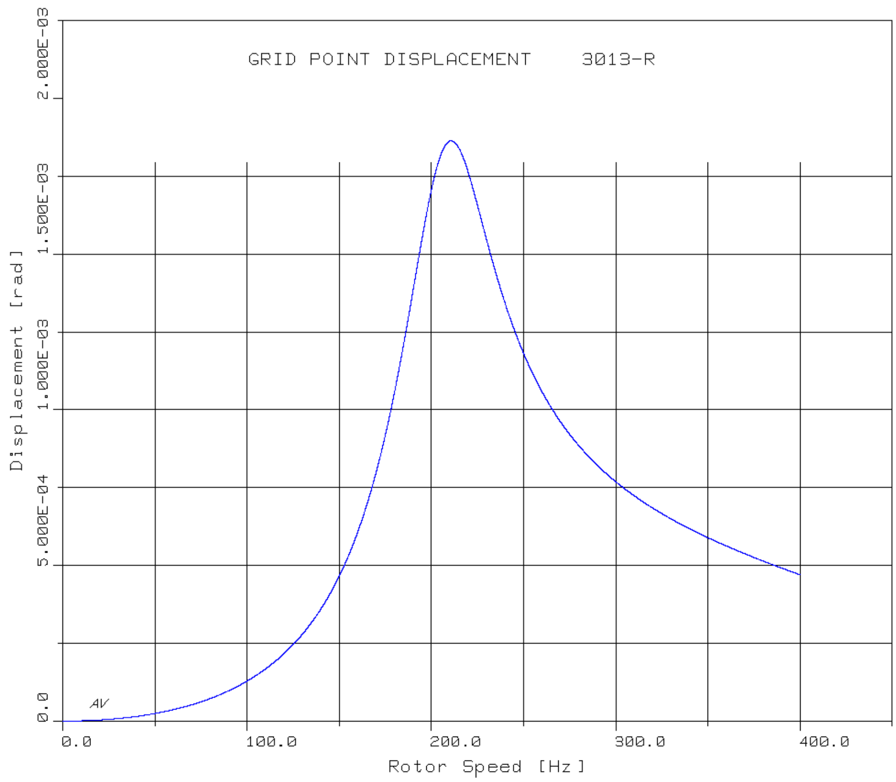


Fig. 248 Magnitude of tilt angle with excitation at 205 Hz and linearly increasing rotor speed

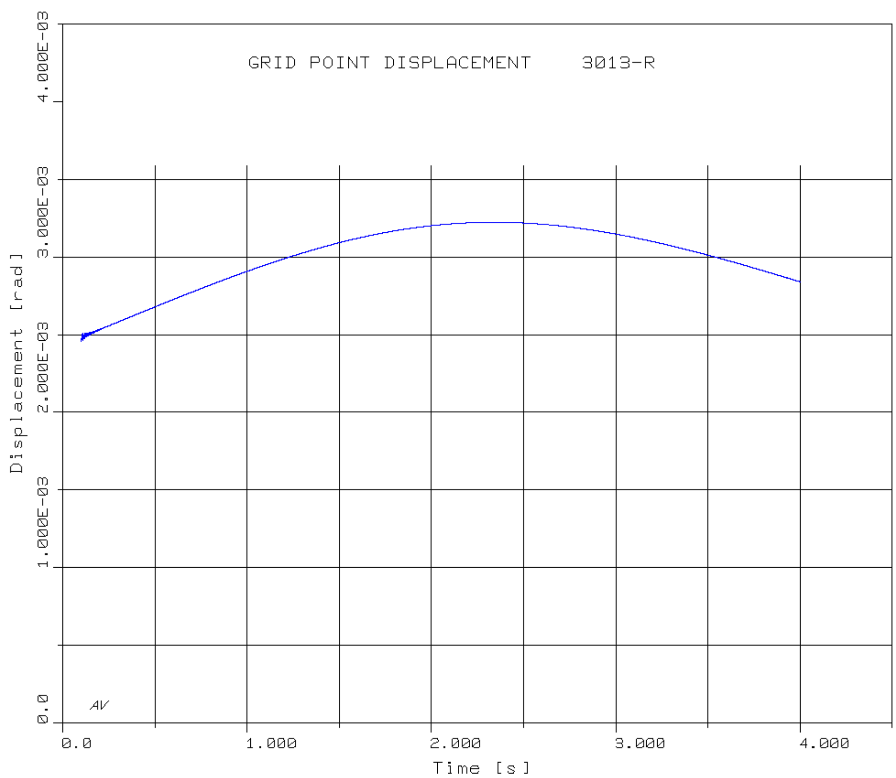


Fig. 249 Magnitude of tilt motion for constant excitation at 205 Hz and slowly increasing rotor speed from 195 to 215 Hz in 4 seconds

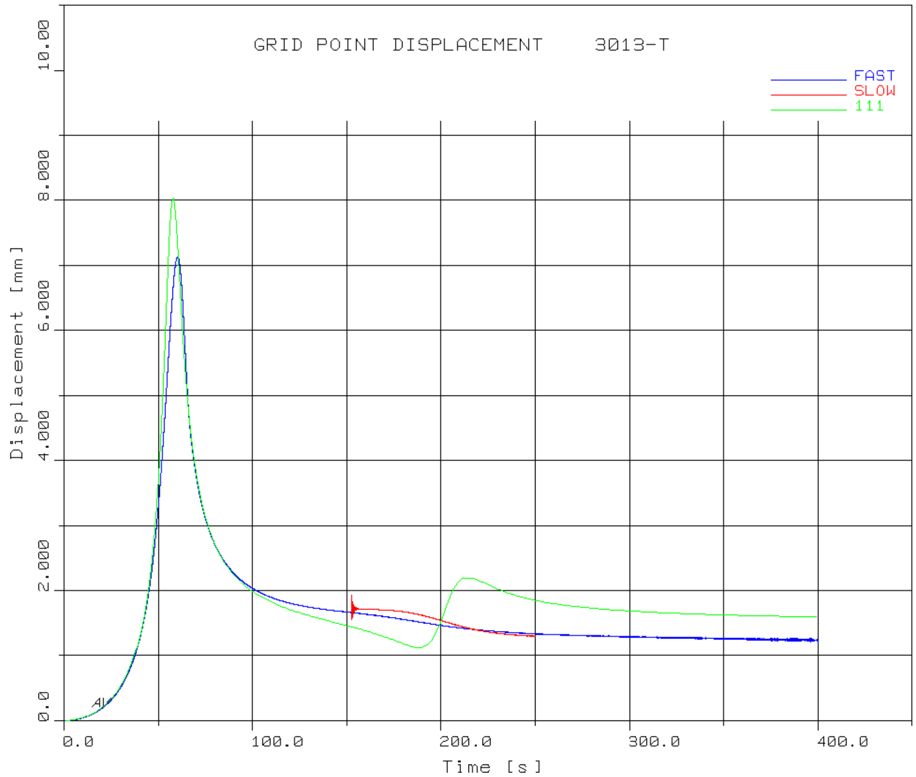


Fig. 250 Fast and slow transient response of translation compared to SOL 111

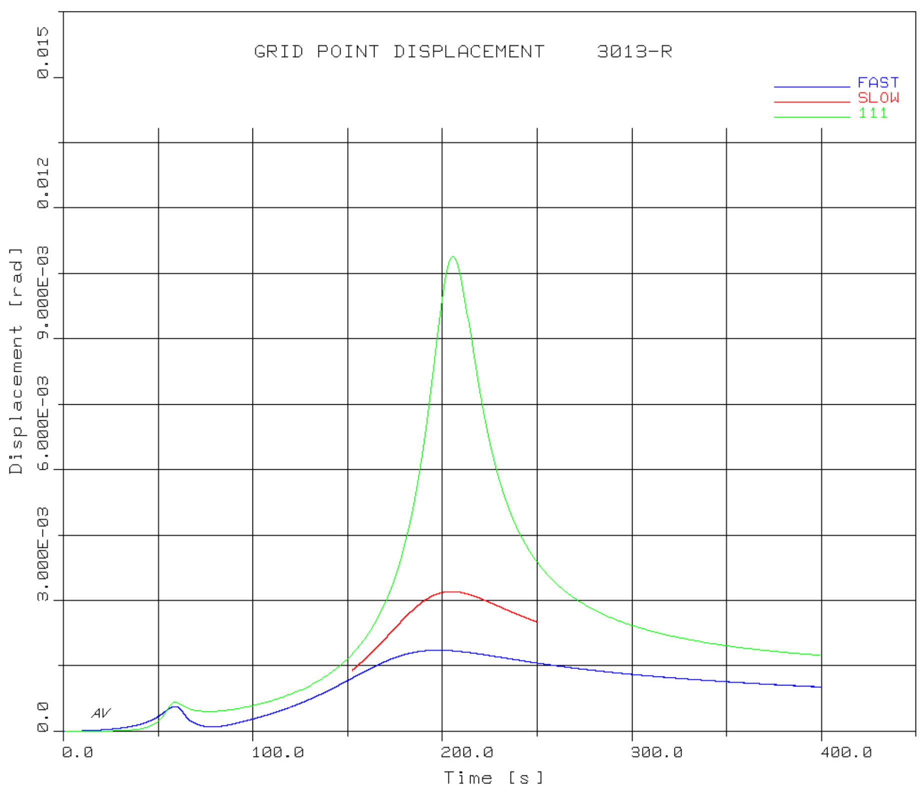


Fig. 251 Fast and slow sweep, synchronous analysis, tilt motion. SOL 112 compared to SOL 111

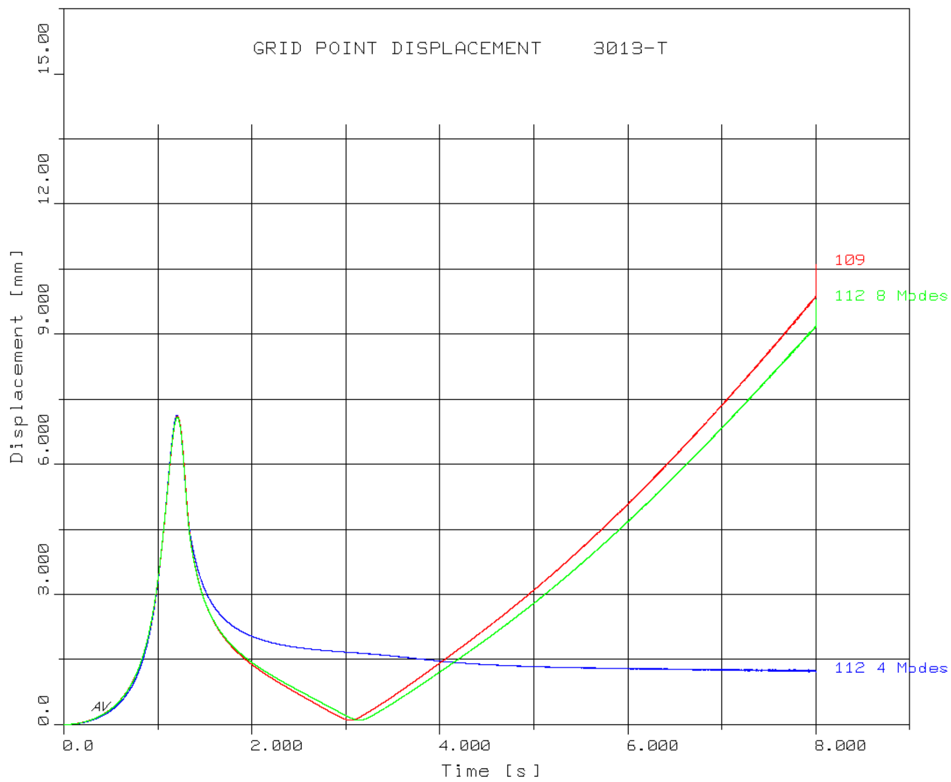


Fig. 252 Synchronous analysis, translation response for modal and direct solutions

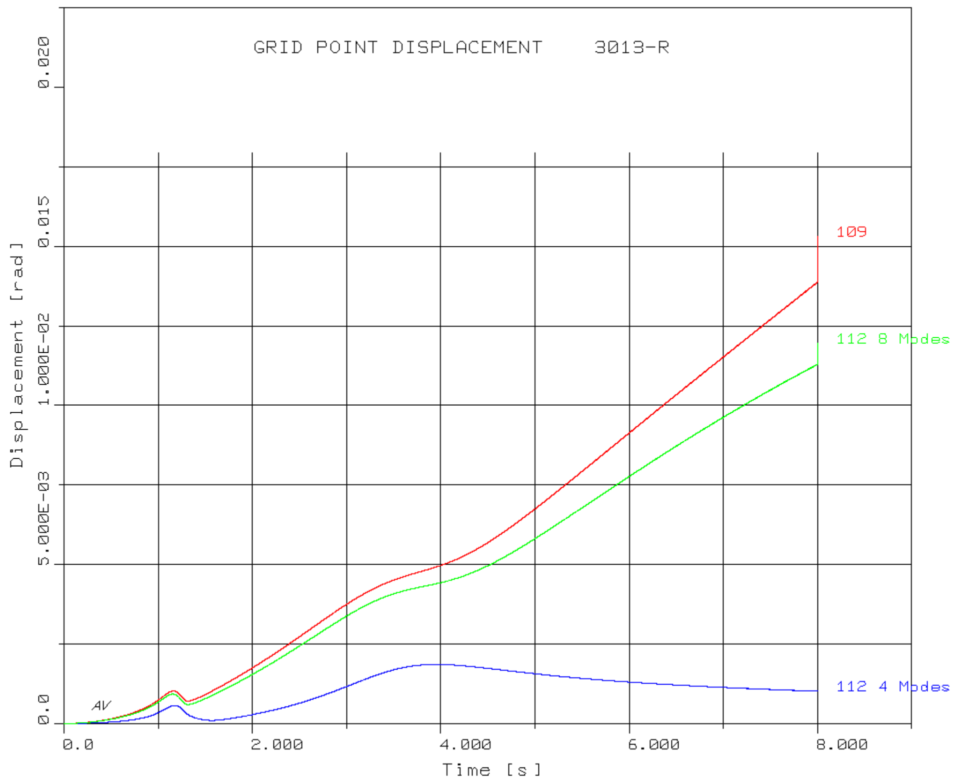


Fig. 253 Synchronous analysis, tilt response for modal and direct solutions

10.9.4.2 Rotating System

The transient analyses were repeated for analysis in the rotating system. The results of an asynchronous analysis at 205 Hz rotor speed are shown in Fig. 254 and Fig. 255 for translation and tilt motion, respectively. Fast (0-400 Hz in 4 seconds) and slow (0-400 Hz in 8 seconds) sweep results agree well with the peaks of the frequency response analysis with SOL 111. The results of a synchronous analysis are shown for translation and tilt motions in Fig. 256 and Fig. 257, respectively. The peaks A and B are in agreement with the previously found values. The results for fast and slow sweeps are compared to the frequency response amplitudes in Fig. 258 and Fig. 259. For the slow sweep the agreement is good. The peaks of the fast sweep are lower and occur at a higher frequency due to the lag of the steady-state oscillation. In the transient analysis there may be convergence problems in the time integration. For the synchronous analysis, a larger time step could solve the problem. For the backwards whirl analysis, the integration diverged after the resonance peak. The resonance of the backward whirl to the 2P sweep is shown in Fig. 260 for slow and fast sweep compared to the steady-state solution from SOL 111. The results get closer to the steady-state solution for slow the sweep rate. The peaks of the tilt modes are shown in Fig. 261. Here the peaks from the transient analysis are larger than that from the frequency response analysis. The different results for this case may be due to the fact that the backward resonances for the tilt and translation motion occur almost at the same rotor speed of 57 Hz.

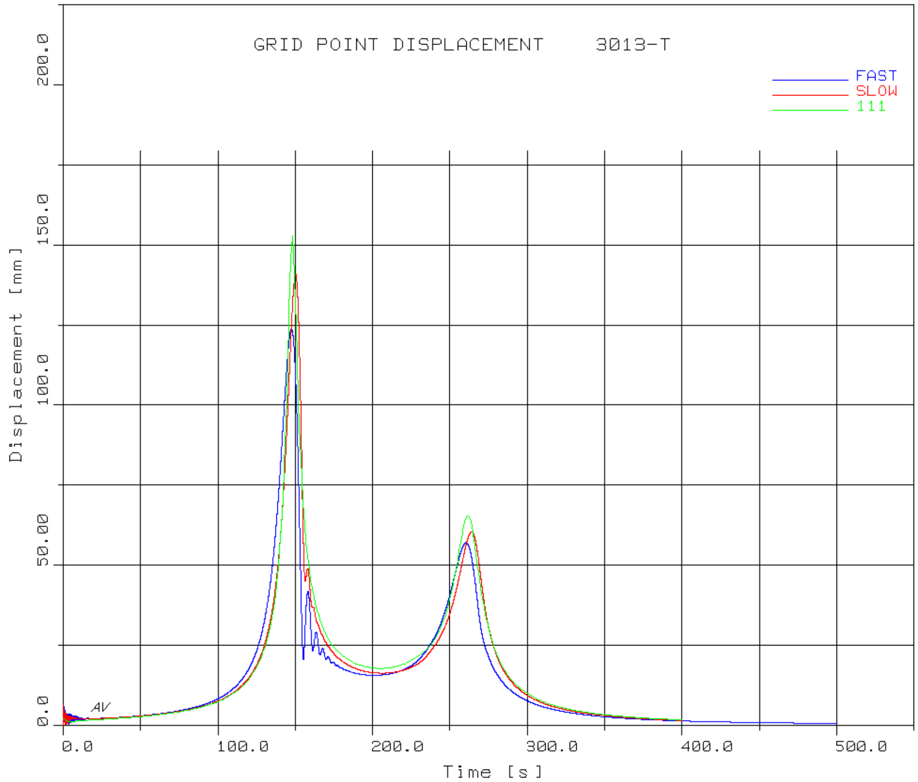


Fig. 254 Frequency response compared to transient response for asynchronous analysis at 205 Hz

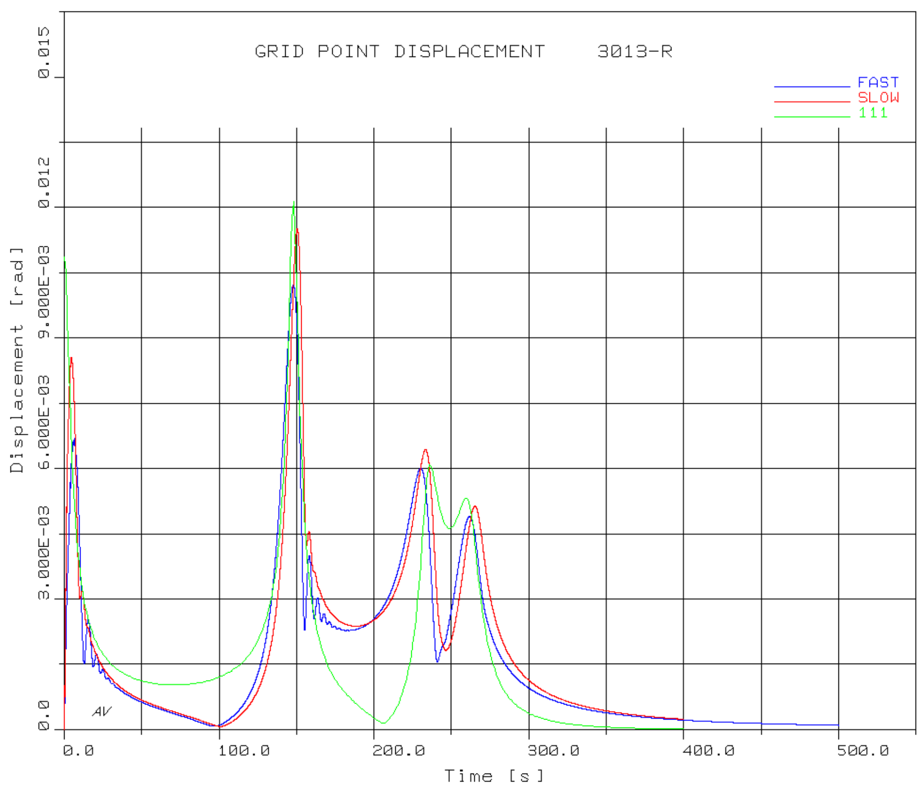


Fig. 255 Frequency response compared to transient response for asynchronous analysis at 205 Hz

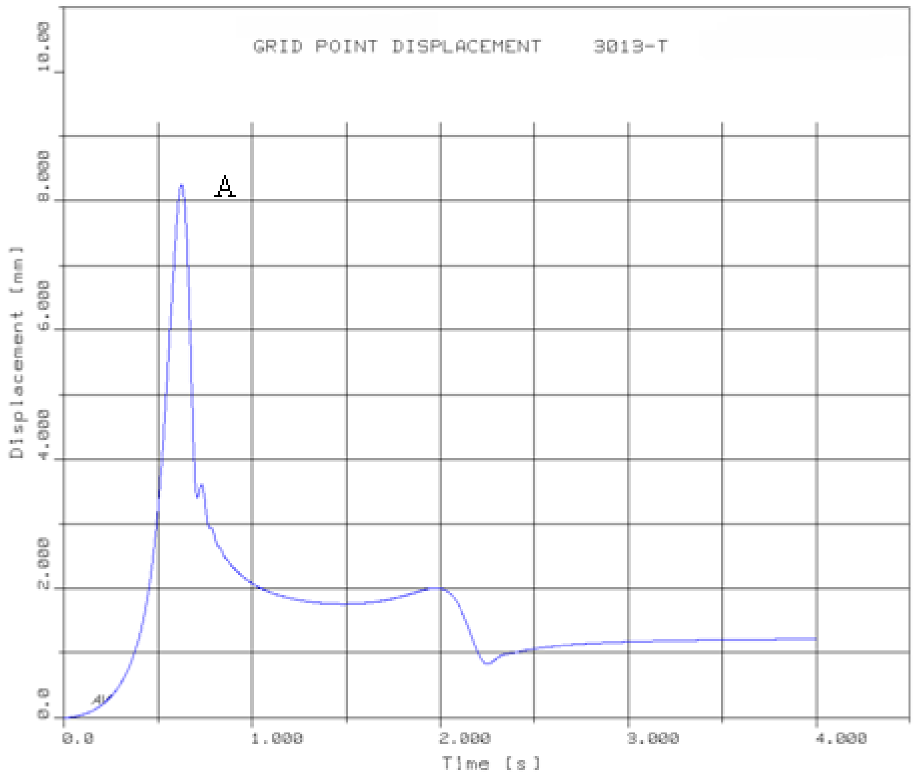


Fig. 256 Translation of synchronous analysis

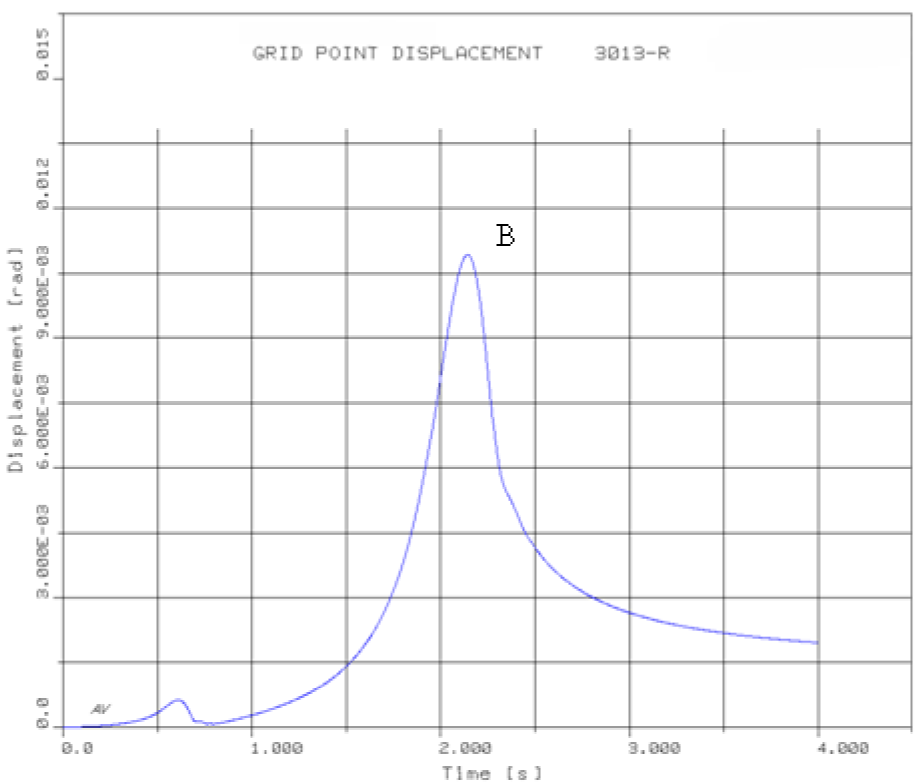


Fig. 257 Rotation of synchronous analysis

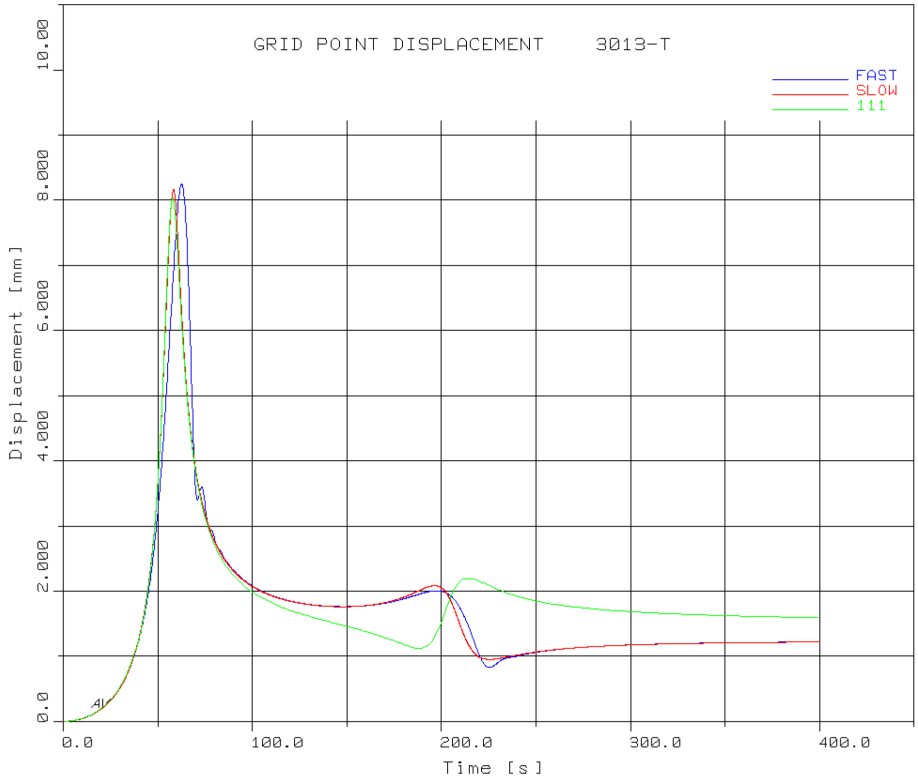


Fig. 258 Fast and slow transient response of translation compared to SOL 111. Synchronous analysis

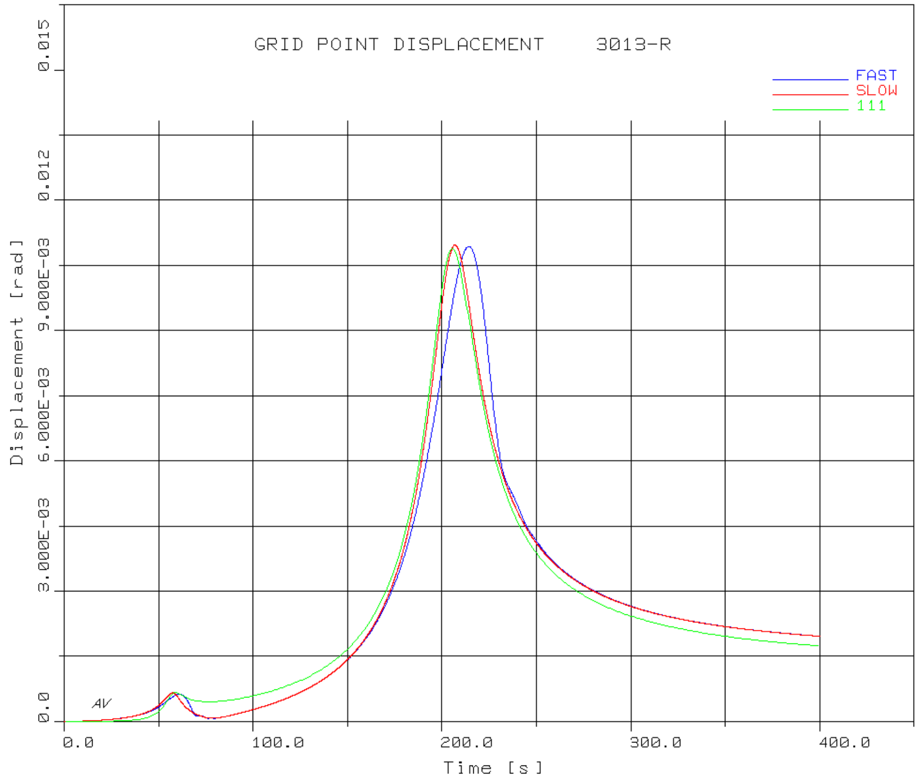


Fig. 259 Fast and slow transient response of tilt compared to SOL 111. Synchronous analysis

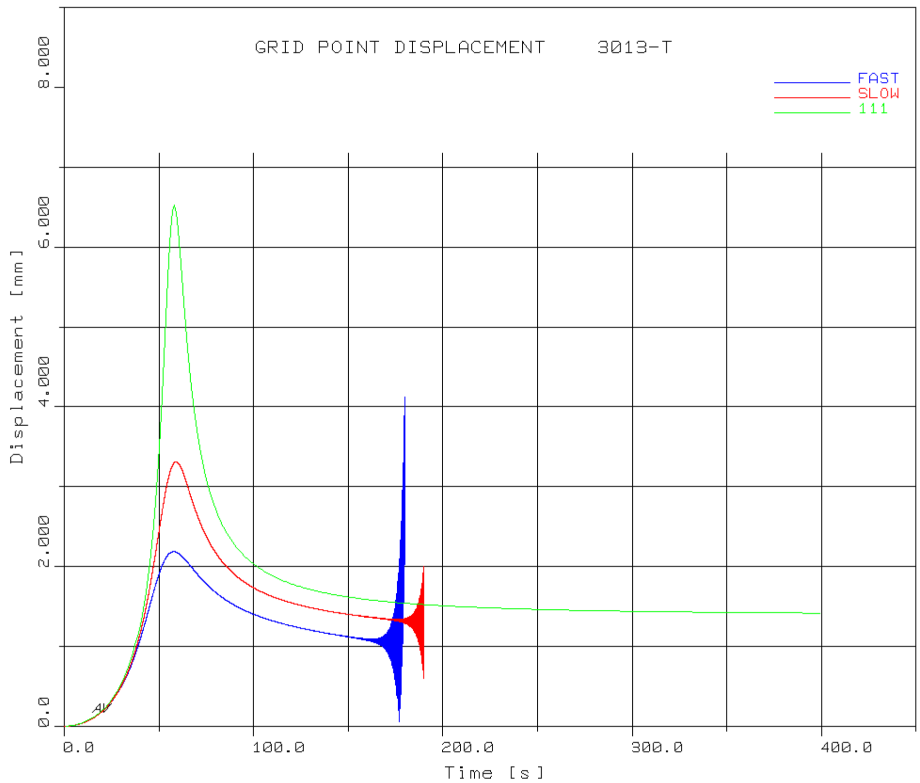


Fig. 260 Backwards whirl of translation

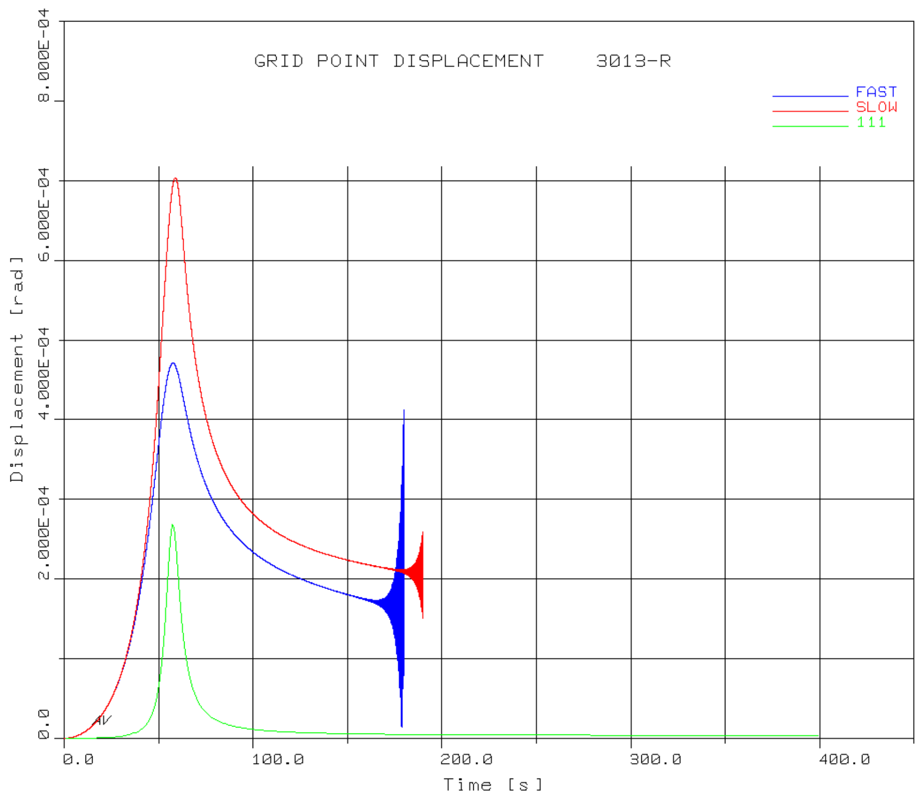


Fig. 261 Backwards whirl of tilt motion

CHAPTER

11

References

11 References

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