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Dynamic analysis of a spindle-bearing system based on finite element method

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Abstract

Background: Most important component of the machine tool is the spindle-bearing system. Dynamic behavior of the spindle-bearing system directly affect to the machining productivity and quality of the work pieces. Therefore at the stage of design and development, it is necessary and very important to know its spindle dynamic behaviors for avoiding forced vibration due to resonance.

Methods: The finite element method (FEM) has been adopted for obtaining spindle dynamic behaviors. The FEM model for the spindle-bearing systems is similiar to those developed in rotor-dynamic. The main purpose of this study, an ANSYS Parametric Design Language (APDL) program based on finite element method has been developed for obtaining full analysis of rotordynamic in order to investigate the spindle-bearing dynamic behaviors. The programs efficiently performed the full analysis by determining the Campbell diagrams, critical speeds, and response of unbalance due to mass unbalance at the cutting tool. In this paper, evaluation of the results and how the program was verified with beam finite element (FE) model are presented. For this purpose, a certain mechanical and geometrical properties of a grinding machine tools spindle-bearing system was modeled and its analysis of rotordynamic was done by ANSYS (APDL) program.

Results: The results show that the accuracy of the model and the solution technique has been demonstrated by comparison and very good agreement has been obtained.

Keywords: ANSYS parametric design language (APDL), Campbell diagram, Critical speeds, Unbalance response, Spindle-bearing system.

INTRODUCTION

Most important component of machine tool system is the spindle-bearing system. Dynamic characteristic of the spindle-bearing system directly affect to the machining productivity and quality of the product. In the design and development stages, then it is very important and necessary to know the dynamic behaviors of spindle for avoiding resonance of critical speeds due to machining operations. To obtain dynamic analysis of the spindle-bearing system analytically in the first design stage, the finite element method (FEM) has been frequently adopted for modeling of spindle-bearing dynamic behaviors. Basically, the FEM model and dynamic analysis for the spindle-bearing systems of a machine tools is similiar to those developed in rotordynamic. However, the spindle shafts used in machine tools usually have smaller shaft diameters and bearings, and possess disk-like in turbomachinery components. Thus in this paper, we attempt to review some researches relating to the field of rotordynamics.

Lin et al. [1] in their paper stated that the most popular approch for modeling the spindle dynamic behavior is the finite element model (FEM), because of its capability to manage complex geometry and boundary condition and the calculation approaches save time consuming while solving the finite element system equation. Most of the ”FE model” papers, such as that written by Nelson and McVaugh, following the FE model that was developed in the rotor-dynamics field. Lin [2] developed a genetic algorithm (GA) optimization approach to search the optimal location of bearings on the motorized spindle shaft. The goal is to maximize its first-mode natural frequency (FMNF). In order to achieve the results, a spindle-bearing system dynamic model is formulated using finite element method (FEM) that was developed in the rotordynamics. Nelson and McVaugh [3], and Nelson [4] applied the theory of Timoshenko's beam to build matrix of systems in order to analyze the rotor systems dynamics including the influences of gyroscopic moments, rotational inertia, axial load, and shear deformation. Zorzi and Nelson [5] presented the influences of damping on the rotating systems dynamics.

Cao and Altintas [6] proposed common procedure of finite element method for modeling the spindle-bearing system. The spindle shield cover and spindle shaft are modeled as Timoshenko beam element which involving the effects of gyroscopic moments and centrifugal force. The stiffness matrix of the bearing, the contact angle, preload and deflection of spindle and spindle housing (shield cover) are all coupled by using the model of finite element of the spindle assembly. Erturk et al. [7] proposed method of analytical about modification of structural and involving receptance coupling for modeling the assemblies of cutting-tool, tool-holder, and spindle-bearing. These component assemblies are considered as multi-segment model of Timoshenko's beam and Euler-Bernoulli's beam, and they compared its results with those formulation method. They found that the accuracy of Euler-Bernoulli model may give the low accuracy results at higher frequencies.

Chateled et al. [8] presented a modeling approach of analysis modal method to calculate the rotor system dynamics (the frequencies of natural and mode shapes) of turbo-machinery. Two methods were compared using the model approaches based on finite element model of three-dimensional and one-dimensional. The results show that low accuracy was obtained when the dynamic behavior of system was formulated using one-dimensional beam. Coupling vibrations of the blade-bending and shaft-torsion in a multi-disk rotor system was studied analytically by Chiu and Chen [9]. Natural frequencies and mode shapes of the system for one to three-disk cases have been investigated. The found that natural frequencies of the system varied with number of blades.

Whalley and Abdul-Ameer [10], by using simple harmonic response methods, they used contoured shaft profiles in order to obtain the natural frequencies and critical speeds of rotor shaft systems.

Taplak and Parlak [11] used the Dynrot program based on finite element method for performing rotordynamics analysis of gas turbine with certain mechanical and geometrical properties. This program was used to obtain Campbell diagram, determining critical speeds, and investigating unbalance response of the rotor due to mass unbalance of compressor. Jalali et al. [12] investigated full analysis of rotordynamics on high-speed-turbine-impeller system using finite element model of ANSYS program and one-dimensional beam, and experiment of modal testing. as performed by [11], the diagram of Campbell, critical frequencies, and unbalance response analysis due to mass unbalance were obtained using the two models of ANSYS and one-dimensional finite element. Based on the comparison results of theoretical and experiment show that the good accuracy of the behavior of such systems can be achieved using two of these models. Villa et al. [13] have been investigated the behavior of non linear flexible unbalanced rotor which supported by roller bearings. In this case, the method of harmonic balance was used to find a periodic response of this non-linear system. In frequency form, stability of the system was identified based on a perturbation applied method. They stated that the method of harmonic balance with the AFT application can be used to obtain harmonic solutions.

Bai et al. [14] examined the dynamic behaviors of hydroturbine main shaft by using program named ANSYS. They developed ANSYS Parametric Design Language to generate the geometry model of 3D, analysis of modal, and obtaining critical speed at the spin speed. By using ANSYS Parametric Design Language, critical speed determination and analysis of imbalance response for a multi segmen rotor have been presented by Gurudatt et al. [15] and Jagannath [16]. They showed the advantage of using this method is that with typing one of input script as example shaft diameter, rotor segmen length, loads experienced by the rotor, command of ”ANTYPE,MODAL”, and ”HARMIC” command, all of these commands can be read as variable input and execution command of the program. These scripts are executed by ANSYS Programming Design Language. Results of these programming language are validated with results of theoretical and measurement, which are good agreement of the acceptable limits.

In this study, an ANSYS Parametric Design Language (APDL) program based on finite element method has been developed for obtaining full analysis of rotor-dynamic in order to investigate the behaviour of spindle-bearing dynamic. The program was used to perform analysis by determining the Campbell diagrams, critical speeds, and unbalance response due to mass unbalance at the cutting tool. In this paper, the program was verified with beam finite element (FE) model and evaluation of the results were compared. For this purpose, certain mechanical and geometrical properties of a grinding machine tools spindle-bearing system were modelled by using the element of BEAM188.

METHODS AND MATERIALS

The equations of motion

The model typical for analyzing a Leadwell STD V-30 spindle-bearing system of grinding machine tool is shown in Fig. 1. The spindle is designed to operate at up 8,000 rev/min with a 5.6 kW motor connected to the shaft with a pulley-belt system. In this model, cutting tool, tool-holder, spindle shaft, and bearings were included. All components of cutting tool, bearings, spindle shaft, and tool-holder assemblies are considered as elements of discrete disk and bearings, and the multi-segment beams with elasticity and distributed mass. Based on procedure of the finite element discretization in many literatures [17-19], the detail of equations will not be derived here and only the general equation of motions are shown.

The shaft element

The spindle shaft segments are considered as a timoshenko's beam and having a constant geometrical cross section. The finite element used has two nodal points and having four (4º) degrees of freedom which are the two direction of rotations and translations, respectively at nodal point of the each element. Each shaft element has a translational mass matrix (M_e^T), a rotational mass matrix (M_e^R), a gyroscopic matrix (G_e), a stiffness matrix (K_e), and a force vector (F_e). The motion equation in a global coodinate, for one element which rotates at a constant operating speed **Ω** can be expressed as

$$\left(M_e^T + M_e^R\right)\ddot{q}_e - \Omega G_e \dot{q}_e + K_e q_e = F_e \tag{1}$$

Where q_e is the nodal displacement vector, containing the eight degrees-of-freedom for one of shaft element (two rotations and two translations in each node). By combining the individual matrices of each shaft element, one can obtain the global matrices that represent the whole shaft, thus resulting to the following equation of motion:

$$\left(M_G^T + M_G^R\right)\ddot{q} - \Omega G_G \dot{q} + K_G q = F_G \tag{2}$$

Where (M_G^T) is a matrix of global translational mass, (M_G^R) is a matrix of global rotational mass, (K_G) is a matrix of global stiffness, (G_G) is a matrix of global gyroscopic, (F_G) is a global force vector matrix acting on the shaft element, and (q) is the displacement vector containing all $4(n_e + 1)$ degrees of freedom of the shaft elements that represent the physical shaft (n_e is the number of shaft elements).

The disk element

Mass element is considered as a rigid disk. The rigid disk is placed at a certain nodal point of finite element. Here, q_d is matrix of the nodal displacement vector of the center mass of disk. For assuming a constant operating speed **Ω** then the equation can be expressed as

$$\left(M_d^T + M_d^R\right)\ddot{q}_d - \Omega G_d \dot{q}_d = F_d \tag{3}$$

Where (M_d^T) is translational disk matrix, (M_d^R) is rotational disk matrix, (G_d) is gyroscopic disk matrix.

The bearing element

Coefficients of stiffness and damping represent dynamic characteristics of the bearings. The motion equation for dynamic characteristics of the bearings on shaft element can be written as

$$C_{br}\dot{q}_{br} + K_{br} q_{br} = F_{br} \tag{4}$$

Where (K_{br}) and (C_{br}) are called matrices the bearing stiffness and the damping matrices, and (F_{br}) is called a bearing force matrix.

Equation of system and analysis of eigenvalue

Based on the element equations (2), (3), and (4) then a certain global element equation can be established and other global equations also can be generated for the other elements. These elements are constructed to form the general equation, which represents behaviour of the whole system. Then here, the motion equation of the damped system for coordinate of global is expressed as

$$M_G \ddot{q} + C_G \dot{q} + K_G q = F_G \tag{5}$$

Where $C_G = C_{br} - \Omega G_e - \Omega G_d$, $K_G = K_e + K_{br}$

In order to obtain the natural frequency of system, then eigenvalue must be solved and expressed by Eq. (5), the system equation can be set as state variable vector.

$$A_s \dot{x} + B_s x = 0 \tag{6}$$

Where the matrices of $\mathbf{A_s}$, $\mathbf{B_s}$, and displacement x consist of element matrices given as

$$A_s = \begin{bmatrix} M_G & C_G \\ 0 & I \end{bmatrix}, \quad B_s = \begin{bmatrix} 0 & K_G \\ -I & 0 \end{bmatrix}, \quad x = \begin{vmatrix} \dot{q} \\ q \end{vmatrix}$$

For assuming harmonic solution $x = x_0 e^{\lambda t}$ of equation (6), the solution of an eigen-value problem is

$$(A\lambda + B)x_0 = 0 \tag{7}$$

Refer to the equation (7), to obtain the matrix solution then the determinant of this matrix equation equal to zero

$$\left|\lambda I + C_a\right| = 0 \tag{8}$$

Where, $C_a = A_s^{-1}B_s$ and λ is an eigen-value. The eigen-values are usually as the complex number and conjugate roots.

$$\lambda_k = \alpha_k \pm i\omega_k \tag{9}$$

Where α_k and ω_k are the stability factor of growth and the k^{th} mode of damped frequencies, respectively.

Unbalance response

The forces of mass unbalance (F) which is shown in equation (5) can be expressed as:

$$F = F_u \Omega^2 e^{i\Omega t} \tag{10}$$

The response of unbalance mass is considered to be as the form

$$P = P_u e^{i\Omega t} \tag{11}$$

Substituting equation (10) and (11) into (5), the equation can be expressed as

$$(K - \Omega^2 M + i\Omega C) p_u = F_u \Omega^2 \tag{12}$$

By solving equation (12), the response of steady state can be obtained.

ANSYS parametric design language (APDL)

In this study, A practical APDL macro scripting language has been developed to generate all the required results, containing amplitude plots and frequencies plots at all the nodes of the model, with minimal effort of the user. The algorithm incorporated in the macro is as

1) Setup the model. Impose boundary conditions and apply excitation force.
2) Performing the analysis of modal for obtaining the natural frequencies and the critical speeds. Set solution using ”ANTYPE,MODAL” command. Retrieve mode frequency and critical speed frequency using ‘*GET’ command and store in using ‘*VFILL’ command.
3) Perform harmonic analysis for obtaining unbalance response and provides validation for the frequency found by modal analysis through harmonic analysis. Set solution using ”HARMIC” command and set the range of excitation frequencies to increment from 0 to maximum operating speed in number of step (using ” NSUBST” command).
4) Solve for unbalance response. Plot results to get unbalance response at nodal point ‘n’.
5) Increment parameter ‘n’ by 1. If n > 18 (since the spindle-bearing system model here contains 18 nodal points), if ok then go to next step. Otherwise, go back to step 3.
6) End of program.

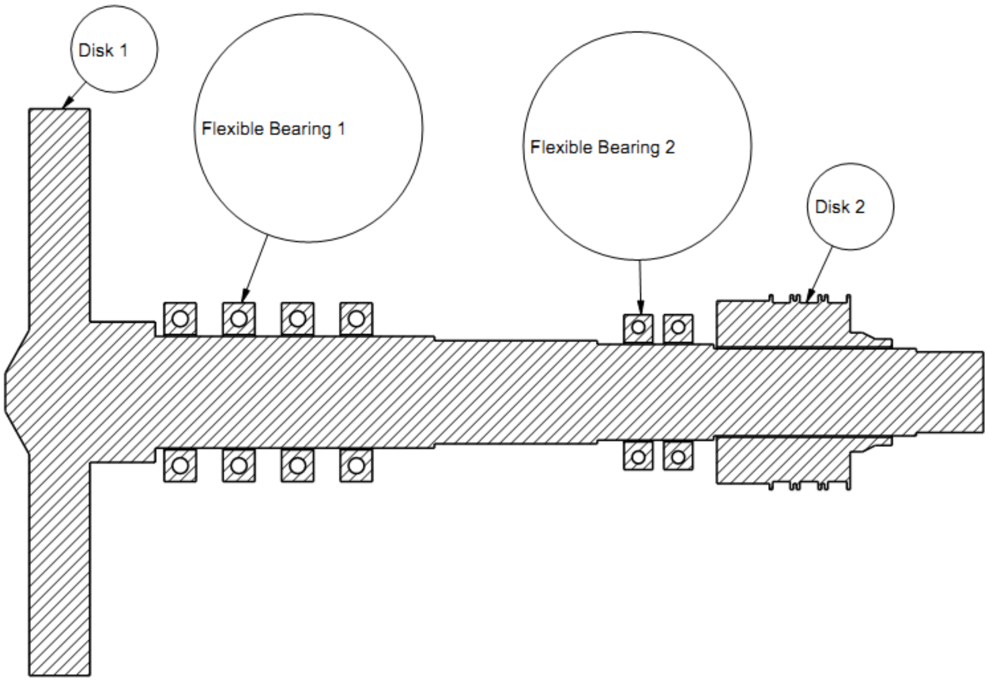

Figure 1: Cross section geometry of spindle-bearing system.

THE MODEL OF FINITE ELEMENT

Finite element model of beam

Table 1 shows the mechanical properties and geometric of the element. In this study, the spindle shaft is modelled into 17 elements of beam and points of node at the end of each element. Two-mass the cutting tool and pulley-belt components can be considered as 1 and 3 elements of rigid disk, respectively. These elements of rigid disk are located at the nodal points number 1, 14, 15 and 16. The parameters of the mass element are tabulated in Table 2. In addition the two set of bearings, located at the nodal points 5 and 12. These bearings are modeled as symmetric isotropic bearings and stiff elastic constrains. Table 3 show model of bearing elements. A

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schematic of spindle's finite element model is shown in Fig. 2.

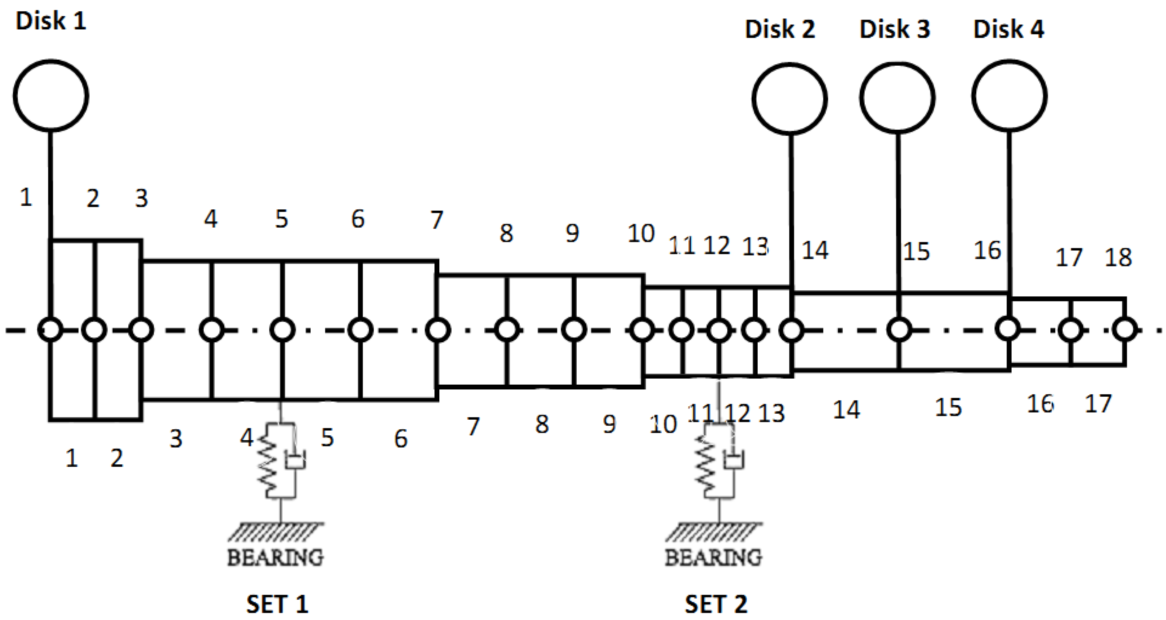

Figure 2: Discretization models of spindle-bearing system.

Table 1 : Mechanical and geometrical properties of shaft element.

Element number	1	2	3	4	5	6	7	8	9
Diameter *Do* (mm)	88	88	70	70	70	70	64.5	64.5	64.5
Length *L* (mm)	20.5	20.5	43.75	43.75	43.75	43.75	34	34	34
ρ (kg/m^3)	7800	7800	7800	7800	7800	7800	7800	7800	7800
E (GPa)	210	210	210	210	210	210	210	210	210

Element number	10	11	12	13	14	15	16	17	18
Diameter *Do* (mm)	60	60	60	60	54.5	54.5	50.4	50.4	-
Length *L* (mm)	18.25	18.25	18.25	18.25	63.5	63.5	21	21	-
ρ (kg/m^3)	7800	7800	7800	7800	7800	7800	7800	7800	-
E (GPa)	210	210	210	210	210	210	210	210	-

Table 2 : Element model of disk.

Mass number	1	2	3	4
Nodal number	1	14	15	16
m (kg)	15.866	2.415	2.415	2.415
J_p (kg.m^2)	0.486	0.005	0.005	0.005
J_t (kg.m^2)	0.247	0.003	0.003	0.003

Table 3: Element model of bearing.

Bearing number	1	2
Nodal number	5	12
K_{yy} (N/m)	1.911×10^8	2.476×10^8
C_{yy} (N.m/s)	191.10×10^2	247.60×10^2
K_{zz} (kg.m^2)	1.911×10^8	2.476×10^8
C_{zz} (N.m/s)	191.10×10^2	247.60×10^2

Geometric modeling and finite element modeling using APDL

As can be seen from Fig. 3, a three dimensional geometric model of the spindle shaft is established using the APDL (ANSYS Parametric Design Language) program. The spindle shaft is considered as the elements of BEAM188 with an internal node and the function of quadratic shape to increase the element accuracy. The characteristic of BEAM188 has two nodal points and having twelve degree of freedom at each element, the motions are translation in the *x*, *y* and *z* axis direction and rotation about *x*, *y* and *z* axis. The element of MASS21 is used for modeling of disk element (mass of rigid disk) and element of COMBIN14 is used for modeling the symmetry bearings. The nodal points, elements, material properties, real constants, boundary conditions and other physical system-defining features that constitute the model have been created by using APDL commands such as *RO*, *PEX*, *PGXY*, *MP*, *ET*, *MAT*, *K*, *N*, *LSTR*, *R*, *RMORE*, *LATT*, *LESIZE* and *E*.

The effect of shear cannot be ignored in the spindle shaft. The constraints are applied to the element motions of displacement in the *x* axis direction and rotation about *x* axis, thus the spindle shaft would not experience any displacements of translation and twist motion about the *x* axis direction. Parameters for the material and element properties of this spindle shaft model are the same as in beam finite element model.

By QRDAMP method, a modal analysis on spindle shaft system was performed to obtain speeds of the critical whirl and value of Campbell frequencies. In a stationary frame, “Coriolis Effect” can be included in the rotating structure by activating the CORIOLIS command. The intersection between natural frequencies and spin speeds (speeds of the critical whirl) for gradient 1x (synchro line) is determined. Harmonic analysis also performed with SYNCHRO command to determine amplitude response values.

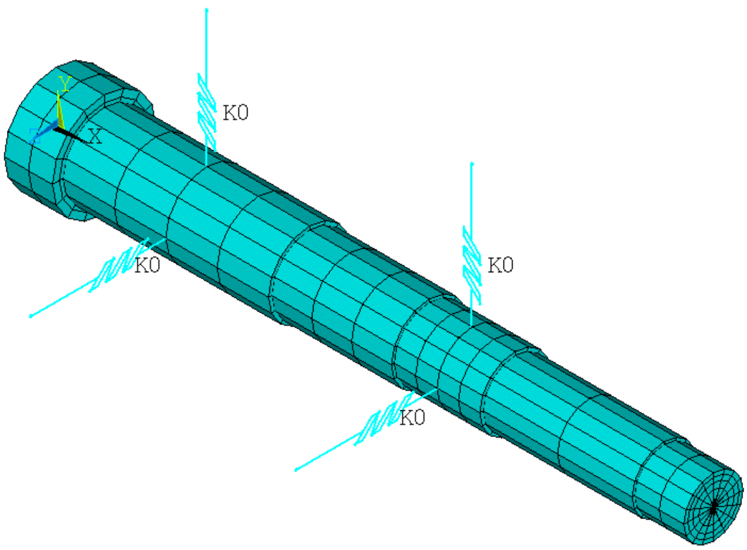

Figure 3: 3D finite element view of spindle shaft generated using APDL program.

In this case, the values of length and diameter are utilized to generate model, the generated model from the script is shown in Fig. 4.

```
/TITLE, DAMPED MODE SHAPE OF SPINDLE USING SYMMETRIC BEARINGS
/COM,   REF: "The Dynamics of Spindle-Bearing for Grinding Machine Tool'
/COM,   MEPPO-BPPT - 3 January 2011.
/COM
/PREP7
*DIM,SPIN,,4                ! SPIN VELOCITY (RPM)
RO = 7800                   ! MATERIAL #1 : STEEL
PEX = 2.1E+11
PGXY = 8.268E+10            ! WITH SHEAR
ET,1,BEAM188,,,2            ! ELEMENT TYPE #1 : SHAFT
NBDIAM = 17                 ! SHAFT SECTION PROPERTIES
*DIM,DIAM,ARRAY,NBDIAM
*DO,I,1,NBDIAM
   SECTYPE,I,BEAM,CSOLID
   SECDATA,DIAM(I)/2,16,4
*ENDDO
ET,2,MASS21                 ! ELEMENT TYPE #2 : DISK
ET,3,COMBIN14               ! ELEMENT TYPE #3 : BEARINGS
KEYOPT,3,2,2                ! Y DIRECTION
ET,4,COMBIN14
KEYOPT,4,2,3                ! Z DIRECTION
R,30,1.911E+8,19110         ! BEARINGS
N,18,560E-3
BRG = 0.15                  ! BEARING "LENGTH" FOR VISUALISATION
TYPE,1                      ! CREATE SHAFT ELEMENTS
TYPE,2                      ! CREATE DISK ELEMENTS
TYPE,3                      ! CREATE BEARING ELEMENTS
FINI
/SOLU
D,ALL,UX                    ! NO TRACTION & NO TORSION
D,ALL,ROTX
/com, Activate Coriolis command and pick the QRDAMP eigensolver

antype, modal                    ! Perform Modal analysis
modopt, qrdamp, 6,,,on           ! Use QRDAMP solver to extract 6 complex m
mxpand, 6                        ! Expand all the modes
coriolis, on,,, on               ! Last field specifies stationary referen(
```

Figure 4: APDL macro scripting language.

RESULTS AND DISCUSSION

Results

Modal analysis of spindle system

The spindle-bearing system for modal analysis was solved using two solution methods. The pseudo-modal method is used in beam FE model and the QRDAMP method is used in APDL as the solution methods to solve the equation of motion system. Figs. 5-7 show the mode shapes of the natural frequencies of the spindle at rest (**Ω = 0)** obtained by APDL program. As can be seen from Figs. 5-7 that for the 1st and 2nd, 3rd and 4th, 5th and 6th of the natural frequencies, respectively have the same mode shapes. This characteristic is affected by symmetric bearings. The spindle natural frequencies which obtained using beam FE model and the FE model that constructed in APDL, and also the percentage difference of the results are tabulated in Table 4. The table shows that natural frequencies of the spindle generated using beam model and APDL model are in good agreement and the maximum difference is still below about 2 %.

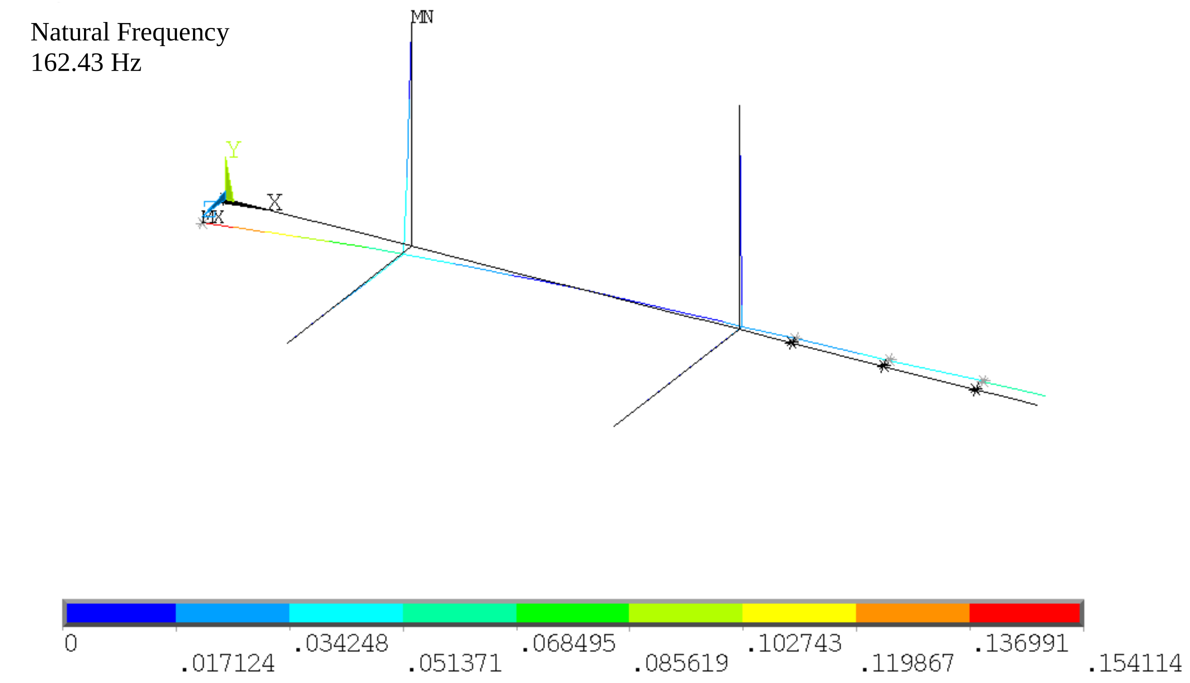

Figure 5: Mode shape 1st and 2nd (Ansys APDL model).

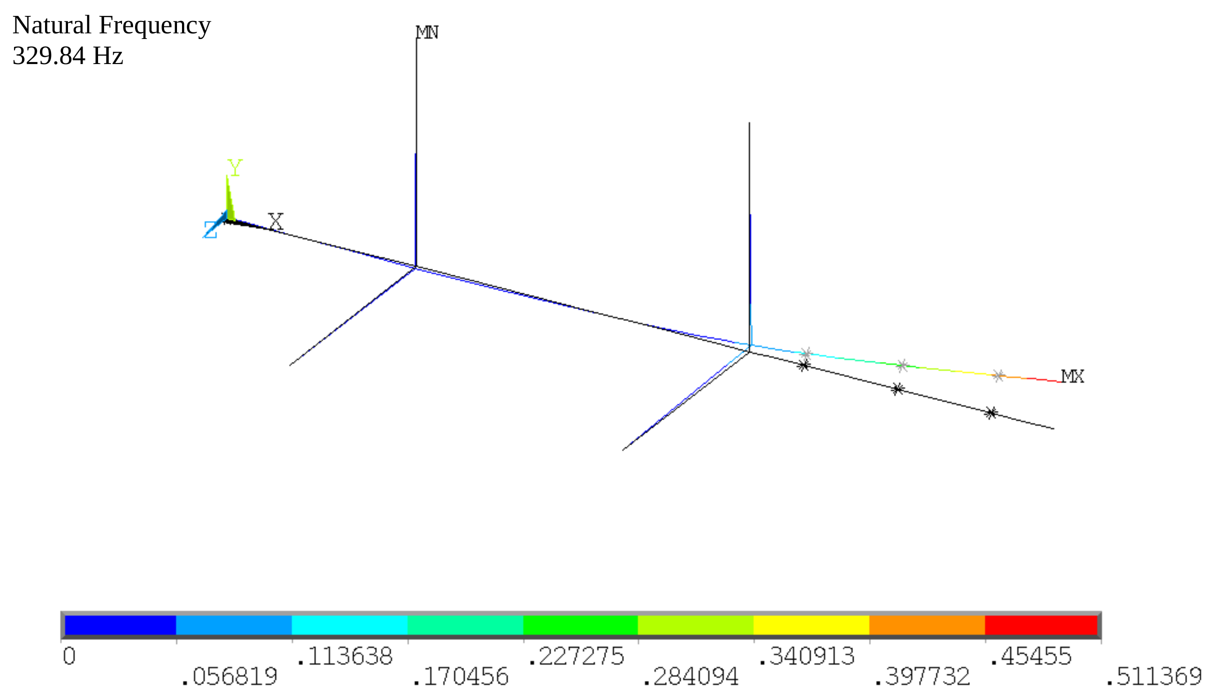

Figure 6: Mode shape 3rd and 4th (Ansys APDL model).

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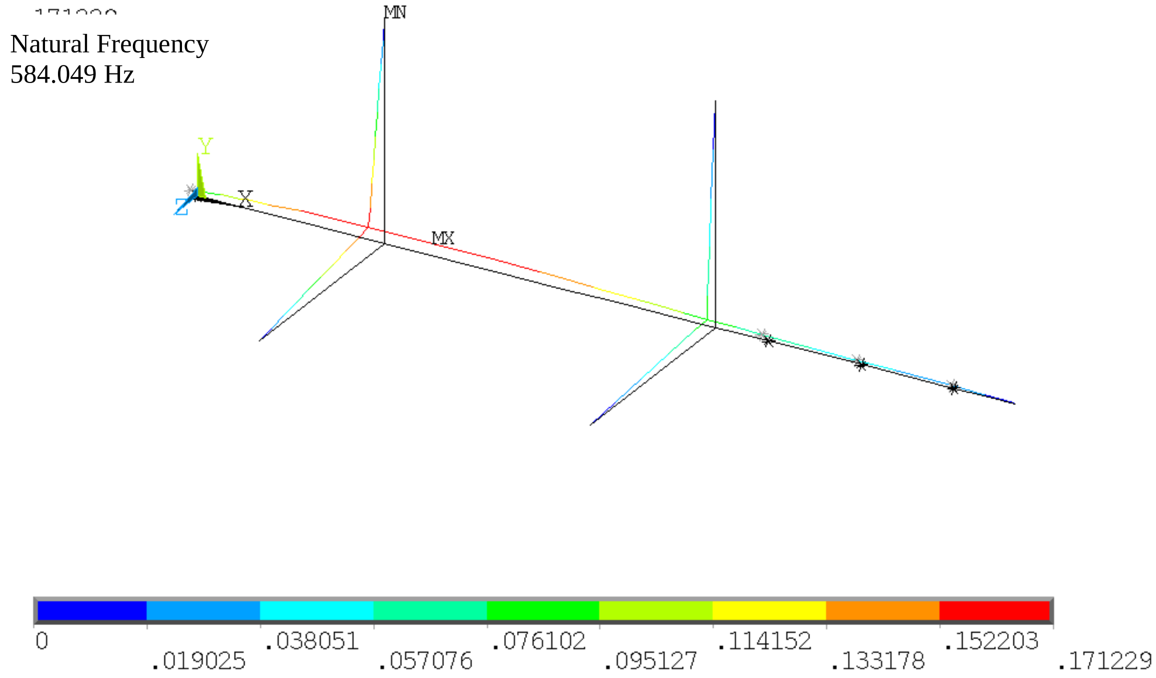

Figure 7: Mode shape 5th and 6th (Ansys APDL model).

Table 4 : Comparison of damped natural frequencies at rest.

Spin speed (rpm)	Beam model (Hz)	APDL model (Hz)	Difference (%)
Mode 1 (BW)	163.645	162.430	0.74
Mode 2 (FW)	163.645	162.430	0.74
Mode 3 (BW)	333.428	329.840	1.08
Mode 4 (FW)	333.428	329.840	1.08
Mode 5 (BW)	595.965	584.049	1.99
Mode 6 (FW)	595.965	584.049	1.99

Campbell diagram and critical speed of spindle-bearing system

The natural frequencies as a function of speed of rotation if the spindle-bearing system under rotating condition ($\mathbf{\Omega \neq 0}$). In this study, determining the Campbell diagrams and obtaining the critical speeds are very useful to evaluate the spindle-bearing system dynamics. Analyzes of the numerical simulation was performed with considering the speed ranges from 0 to 27000 rpm. Figs. 8 and 9 show the Campbell diagrams obtained by beam FE model and the FE model constructed in APDL. The damped natural frequencies of the spindle system (at operating speed 8000 rpm) obtained by beam FE model and the APDL FE model are tabulated in Table 5. The percentage difference of results between beam model and APDL predictions are also shown in Table 5. Table shows that the damped natural frequencies of spindle obtained using beam and APDL finite element model are also in good agreement and the maximum difference is about 2 %.

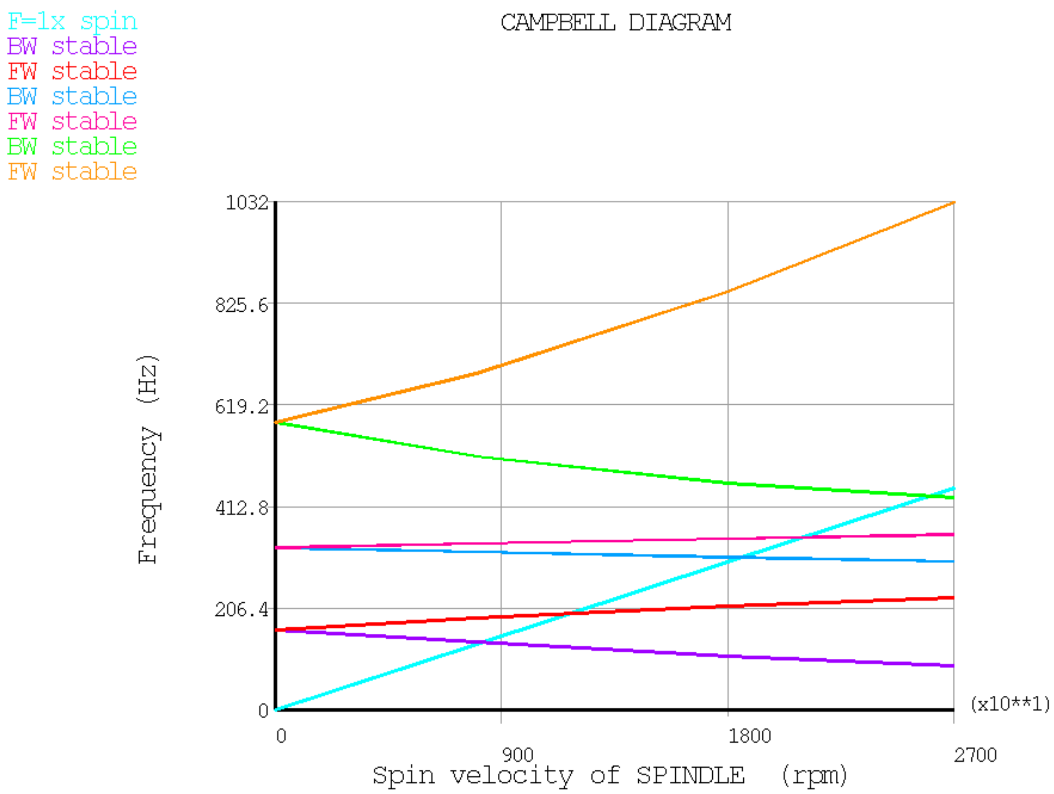

Figure 8: Campbell diagram from APDL model.

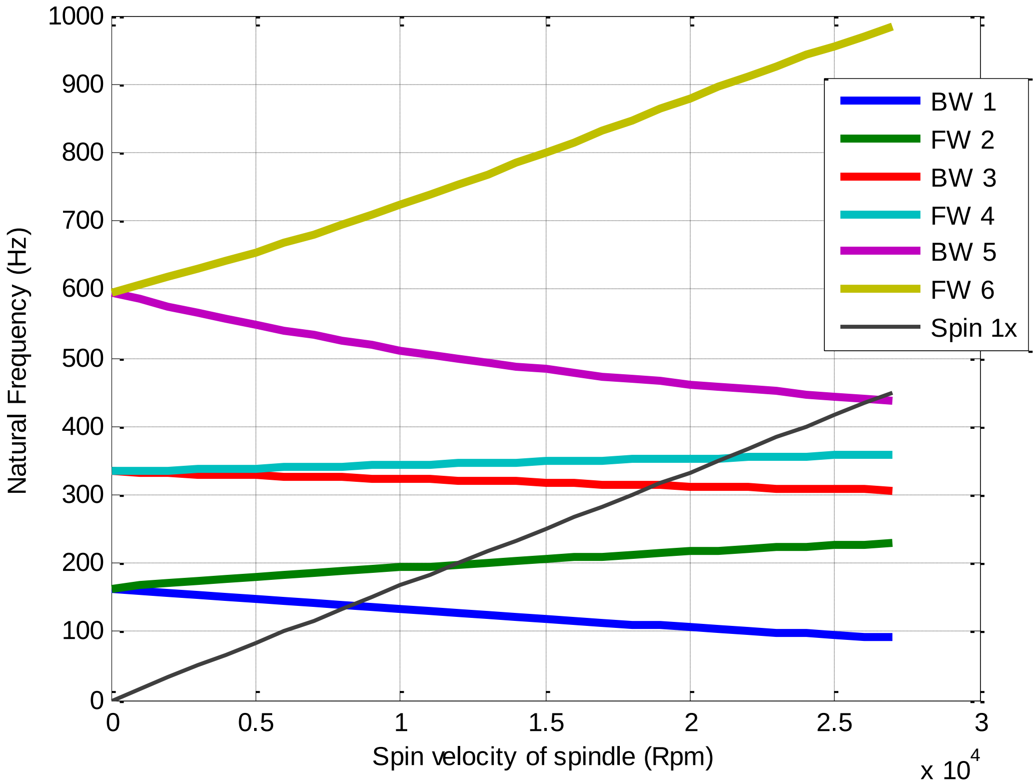

Figure 9: Campbell diagram from beam finite element (FE) model.

8215.225 rpm
BW (Backward Whirl)

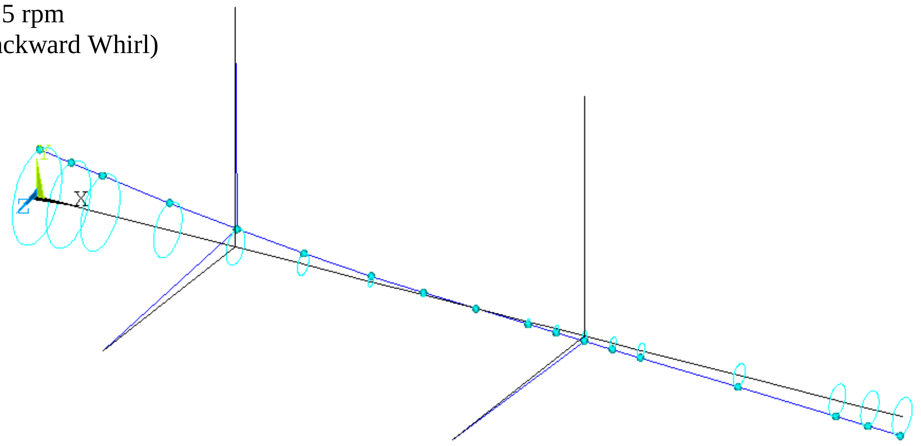

Figure 10: Mode shape relating to 1st critical speed (BW).

11751.541 rpm
FW (Forward Whirl)

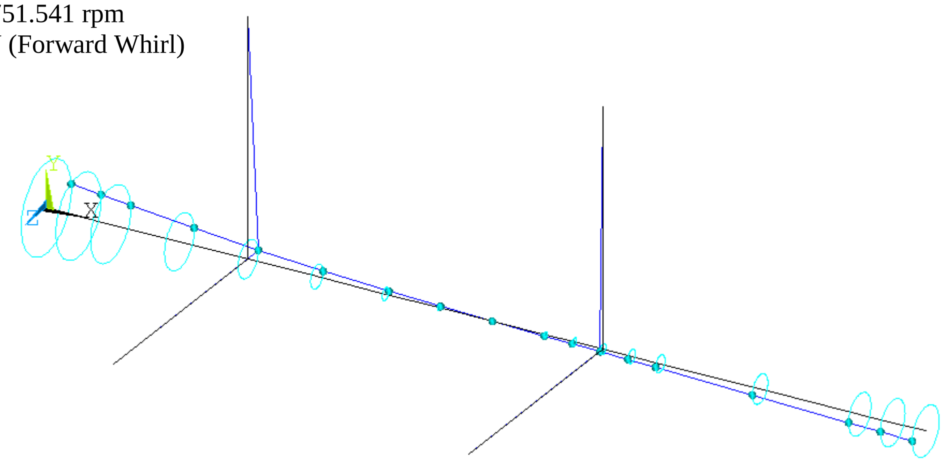

Figure 11: Mode shape relating to 2nd critical speed (FW).

18610.909 rpm
BW (Backward Whirl)

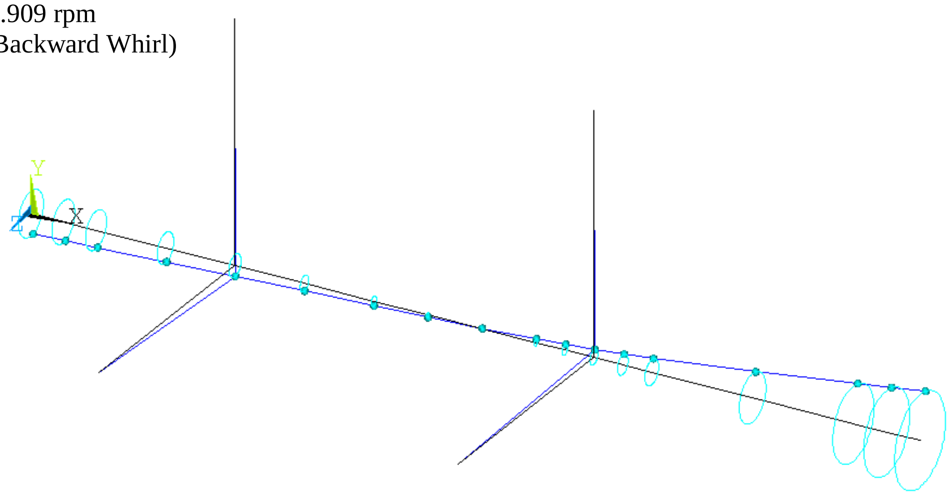

Figure 12: Mode shape relating to 3rd critical speed (FW).

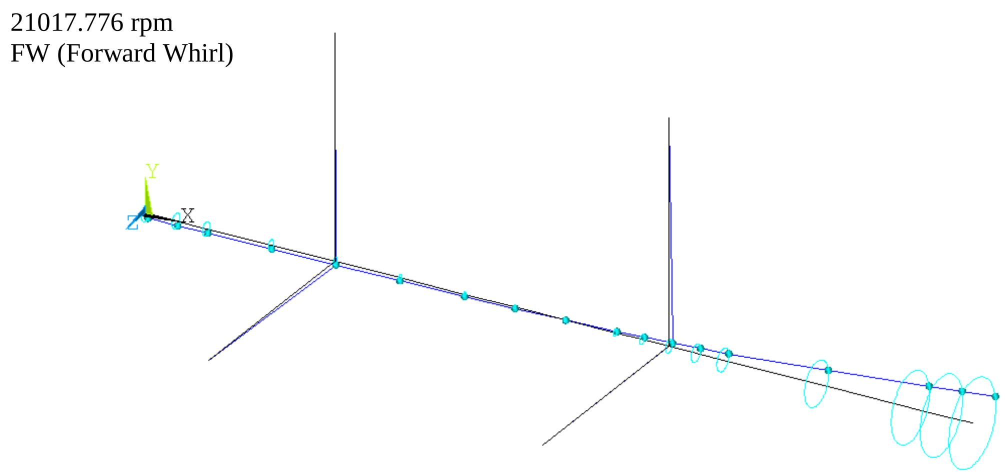

Figure 13: Mode shape relating to 4th critical speed (FW).

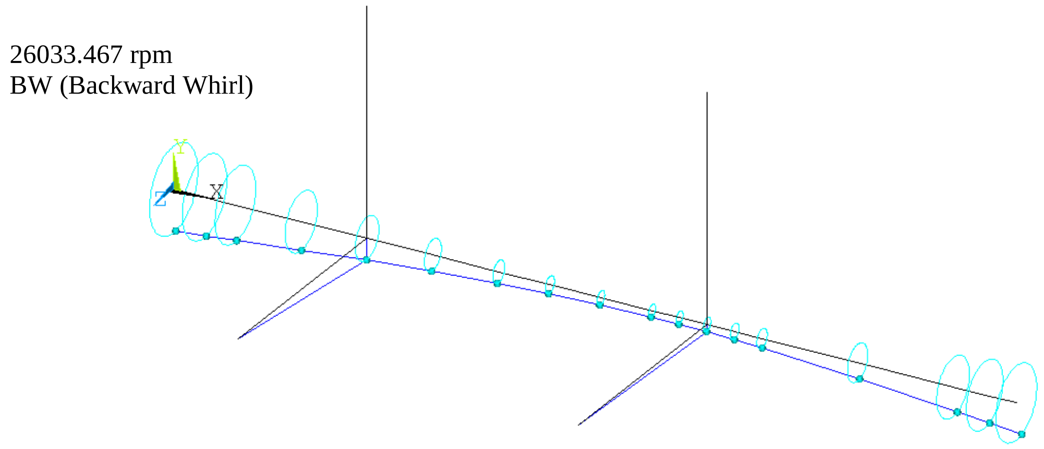

Figure 14: Mode shape relating to 5th critical speed (FW).

Table 6 shows the value of critical speeds are obtained from both method of beam FE and APDL FE models. As can be seen from Table 6, the critical speeds of the spindle are also in good agreement and the maximum difference is also still below about 2 %. Figs. 10-14 show the mode shapes of the critical speeds obtained by FE model constructed in APDL. There are three backward whirls (BW) and two forward whirls (FW) modes are considered. We can find 1st, 2nd, 3rd, 4th, and 5th critical speeds from Figs. 8 and 9. These critical speeds are obtained by the frequency modes which intersecting spin speed line.

Table 5 : Damped natural frequencies at operating speed.

Spin speed (at 8000 rpm)	Beam model (rpm)	APDL model (rpm)	Difference (%)
Mode 1 (BW)	138.6285	137.518	0.80
Mode 2 (FW)	187.6727	186.467	0.64
Mode 3 (BW)	324.9887	321.266	1.15
Mode 4 (FW)	341.6226	338.133	1.02
Mode 5 (BW)	525.2697	514.517	2.05
Mode 6 (FW)	694.2787	683.315	1.58

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Table 6 : Comparison of critical speed mode shape.

Critical speed	Beam model (rpm)	APDL model (rpm)	Difference (%)
Critical speed 1 (BW)	8268.6618	8215.225	0.65
Critical speed 2 (FW)	11889.0000	11751.541	1.16
Critical speed 3 (BW)	18833.0000	18610.909	1.18
Critical speed 4 (FW)	21233.0000	21017.776	1.01
Critical speed 5 (BW)	26404.0000	26033.467	1.40

Unbalance response analysis

Analysis of unbalance response is carried out for investigating the behaviors of spindle-bearing dynamic which provides validation for the frequency found by modal analysis through harmonic analysis.

Analysis of unbalance response is carried out for investigating the behaviors of spindle-bearing dynamic which provides validation for the frequency found by modal analysis through harmonic analysis. An unbalance of 9.981×10-5 kg.m for center mass of the cutting tool is considered in APDL model. The nodal solutions of unbalance responses have been obtained using the APDL FE model which are tabulated in Table 7. The percentage difference between both model are in good agreement and the maximum difference is about 4 % also shown in Table 7. As can be seen from the table, the maximum amplitudes occur near at the second and fourth forward critical speeds points which were calculated in the previous section. It's mean that if the system has damping, the system will resonance when it approaches at these critical points. It is shown at the Table 8, the maximum amplitudes are not coincide 195.9 Hz and 350.3 Hz which were calculated previously. Figs 15 and 16 show the comparison unbalance responses between disk 1 and bearing set 1, and bearing set 1 and bearing set 2, respectively. Figs 17 and 18 show the mode shapes at two first critical speeds.

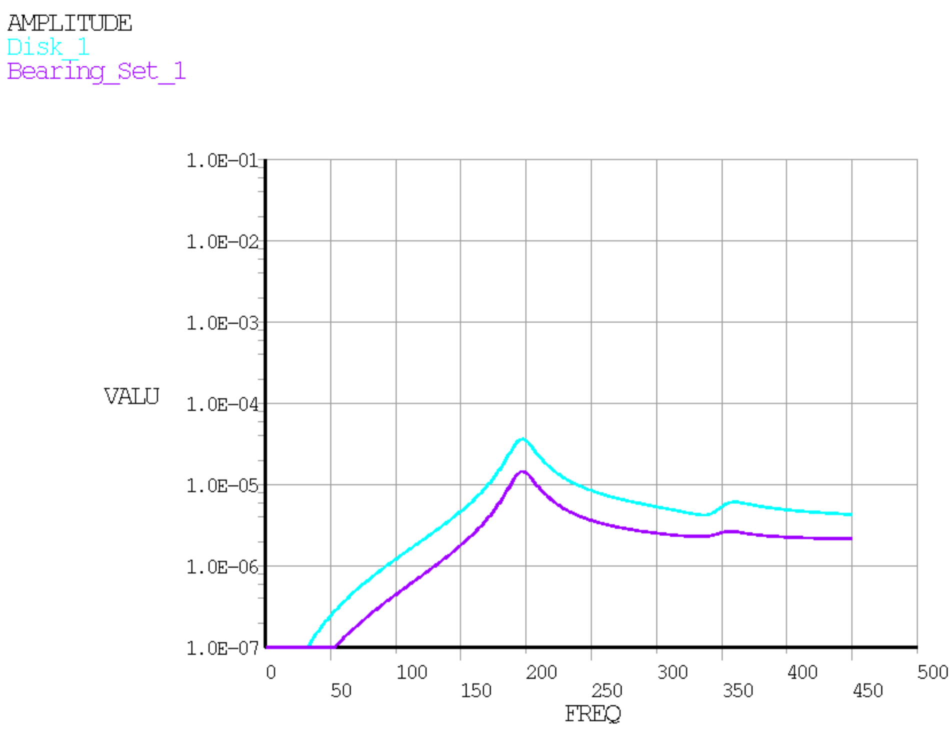

Figure 15: Unbalance response at disk 1 and bearing set 1.

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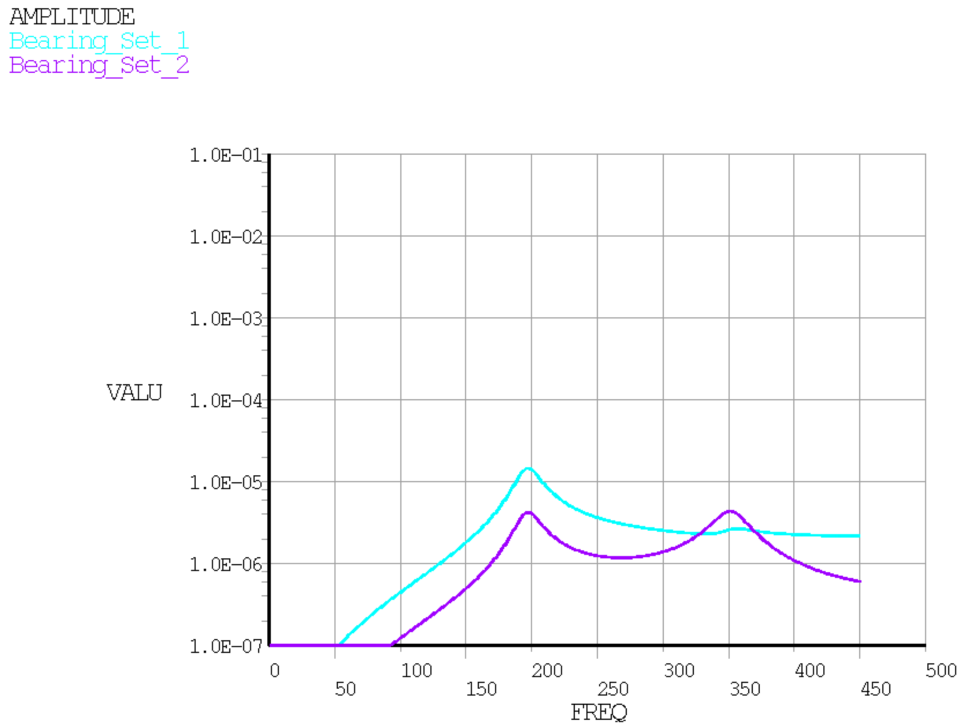

Figure 16: Unbalance response at bearing set 1 and bearing set 2.

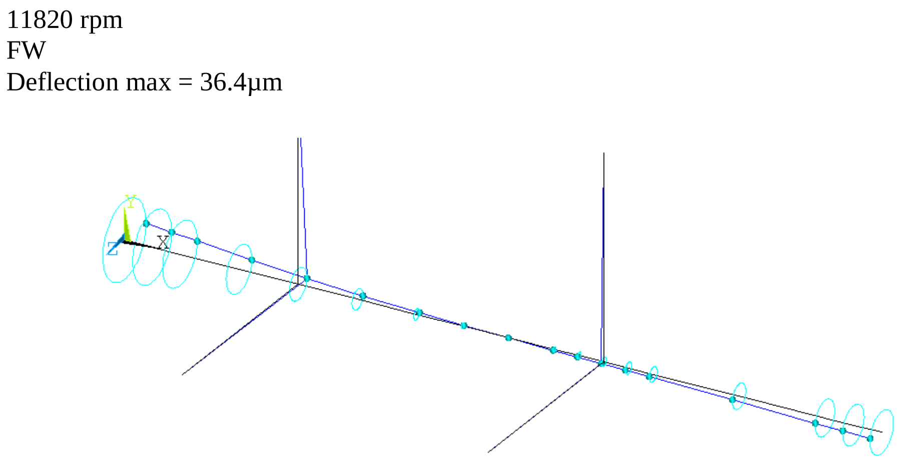

Figure 17: Mode shape of critical speed for 1st forward whirl.

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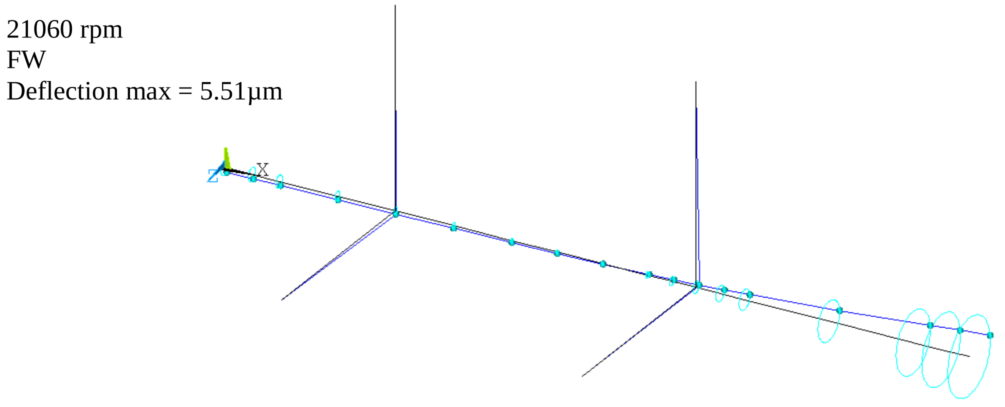

Figure 18: Mode shape of critical speed for 2nd forward whirl.

Based on this unbalance response analysis, it is easy to understand that the critical speed of the spindle is the speed corresponding to the intersection of the natural frequency (Hz) equal to spindle spin's (rpm) line with only the forward whirl mode. Fig 19 shows the unbalance response of the spindle-bearing system that was evaluated at nodal points 1,5, and 12, which stated disk 1 (cutting tool), bearing set 1, and bearing set 2, respectively. The maximum amplitude of these respons can be seen at the Table 7.

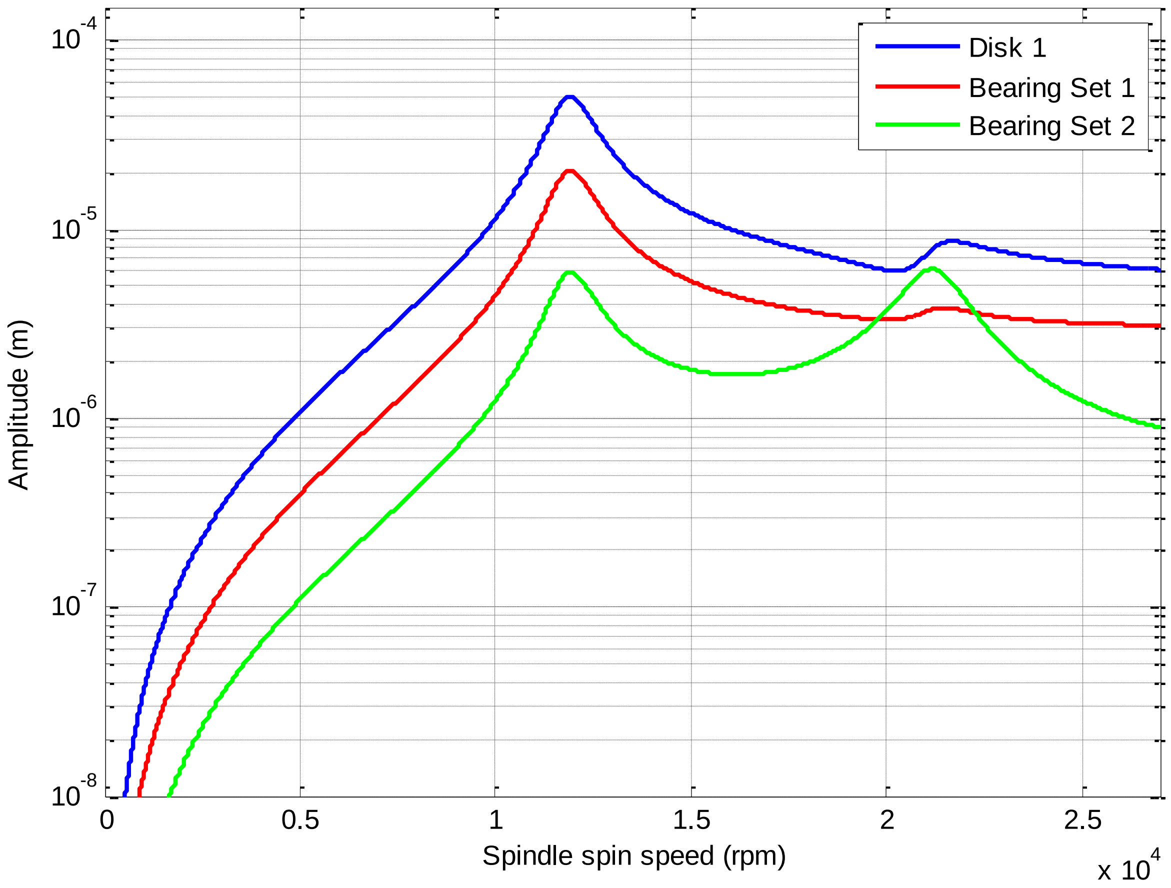

Figure 19: Unbalance response of the spindle-bearing system (generated by APDL model).

Table 7 : Comparison of nodal solution unbalance response.

Node	Beam model (Hz)	Maximum Amplitude (m)	APDL model (Hz)	Maximum Amplitude (m)	Difference (%)
Node #1 (Disk 1)	198.62	0.372×10^{-4}	197	0.364×10^{-4}	2.15
Node #5 (Bearing set 1)	198.65	0.150×10^{-4}	197	0.146×10^{-4}	2.67
Node #1 (Disk 1)	353.82	5.754×10^{-6}	351	5.510×10^{-6}	4.24
Node #12 (Bearing set 2)	353.25	4.561×10^{-6}	351	4.370×10^{-6}	4.19

Table 8 : Comparison of critical speed analysis.

Critical speed	Modal Analysis (Hz)	Harmonic Analysis (Hz)	Difference (%)
1st Critical speed (FW)	195.9	197	0.56
2nd Critical speed (FW)	350.3	351	0.19

CONCLUSIONS

A program named ANSYS Parametric Design Language (APDL) has been developed to make dynamic analysis and evaluation of the results. A grinding spindle-bearing system with certain geometrical and mechanical properties was modeled as beam finite element (FE) model and APDL model technique. Full rotor-dynamic analysis for investigating the modal and harmonic analysis has been carried out in order to evaluate the dynamic behaviors of spindle-bearing systems. The results show that accuracy of the model and solution technique are in a good agreement, and a way to reduce time and effort involved in performing modal analysis and multiple iterations of harmonic analysis to obtain unbalance response of the considered system. Thus, ANSYS Parametric Design Language (APDL) can be used by a spindle designer as tools, which increasing product quality, reducing cost, and time consuming in the design and development stages.

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Authors Contributions

The author (KJ) conducted the design model and FE simulations, collected all data and wrote the whole part of the paper. All authors read and approved the final manuscript.

Competing Interests

The authors declare that they have no competing interests.

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