HELICOPTER GROUND RESONANCE ANALYSIS USING MULTIBODY DYNAMICS

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Abstract

Helicopter dynamic response is one of the most important criterions of the helicopter design. For an accurate prediction and calculation of the vibratory responses of the helicopter structure in different flight conditions, an exact modelling is essential. The accuracy of the results returns back to the accuracies of the model and calculation procedures.

The multibody dynamic simulation tool allows using modular models to build a complete detailed system. This could be used to model a complex helicopter. SIMPACK (**SI**mulation of **M**ultibody systems **PACK**age) as a multibody simulation tool has shown considerable ability to model and analyse linear and non-linear systems during different DLR-projects, however use of this tool for modelling the helicopter dynamics is something which is currently being investigated [1], [2]. This tool was originally developed by DLR (German Aerospace Center) and is now further developed and commercially distributed by SIMPACK AG.

In this work the multibody simulation of the ground resonance effect is investigated. For this purpose, the interface of SIMPACK with a FEM code for modelling the elastic rotor blades is considered. This interface creates a **S**tandatrd **I**nput **D**ata, **SID**, based on blade geometry and eigenvalue analysis. The SID-file is then used for the calculation of elastic deformation. SIMPACK provides also an interface with MATLAB, which allows performing a part of postprocessing within this program. Fig.1 shows the general view of the ground resonance analysis using multibody dynamics simulation.
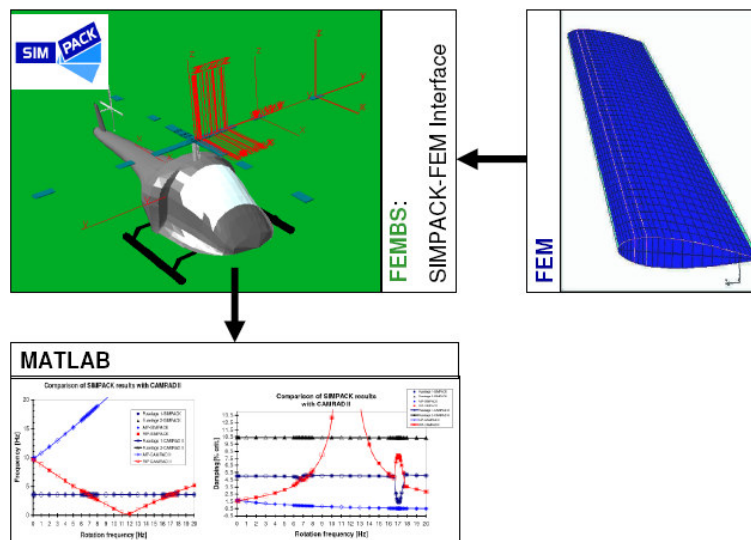
Fig.1 General view of the ground resonance analysis procedure

Helicopter ground resonance is a self exited dynamic instability, which may occur when helicopter is in contact with the ground. Elastic deformation and out of phase lagging motion of the rotor blades during rotor rotation lead to the oscillation of the rotor centre of gravity around the rotor rotation axis. Due to the interaction between rotor and fuselage, this oscillation excites the flexible fuselage and the rotor support and may cause a violent damage of the helicopter structure. This phenomenon is known as ground resonance. Modelling this phenomenon needs consideration of the following points: Linearisation, Geometric stiffening effect, Time-periodic system, Multiblade coordinates transformation. The work described within this paper is generally divided into three main parts. The first part is related to the "linearization and geometric stiffening

effect", which is a preliminary step for the dynamic stability analysis of a rotating system. Here, the simulation results are compared with analytical results. Creating the Fan-Diagram of an isolated rotor and comparison of the results with those produced with CAMRAD II belongs also to this part. The second part deals with the helicopter ground resonance analysis. Modelling the ground resonance starts with a simplified model (rigid blades with mechanically equivalent elastic lagging deformation and fuselage with two translational degrees of freedom) and is continued with more complex models comprising elastic blades and fuselage. Data used in this part belongs to DLR scaled (1:2.5) BO105 research rotor model (configuration K20) [3] and CAMRAD II model of full scale BO105 with modified skid gear provided by Eurocopter Germany. One of the features of the ground resonance model is its time-periodic characteristic, which leads to the invalidity of the direct usage of the classical eigenvalue method for the dynamic stability analysis. To perform the ground resonance analysis, the model is first linearized about the equilibrium state. Then the linear system matrix of the model is extracted and transformed from the rotating coordinates to the non-rotating multiblade coordinates. This coordinate transformation changes the system from a time-periodic to a time-invariant one. Finally, a classical eigenvalue analysis of the transformed system matrix results in the frequencies and damping of the system. To evaluate the simulation procedures, the SIMPACK results are compared with results produced with CAMRAD II model. To validate the model and calculation procedures in addition to a comparison with the CAMRAD II-results an analytical validation of the dynamic response of the system is also performed. The third part of the work deals with the parametric studies of the ground resonance effect and analysing the results and tendencies of the changes of the dynamic response.

1. INTRODUCTION

Testing of rotorcraft dynamic stability is a necessary step for the successful development of a new design. Testing a prototype at full scale or model scale is typically very expensive. Therefore usage of a simulation code for investigation of dynamic stability is a considerable alternative solution. Each simulation code has its own computational methods and modelling approaches, which results in different accuracies. Therefore usage of a simulation code for a special analysis demands pre-investigations and validations.

The multibody dynamic simulation tool allows using modular models to build a complete detailed system. This could be used to model a complex helicopter. SIMPACK (**SI**mulation of **M**ultibody systems **PACK**age) as a multibody simulation tool has shown considerable ability to model and analyse linear and non-linear systems during different DLR-projects, however use of this tool for modelling the helicopter dynamics is something which is currently being investigated [1], [2].

In general helicopter dynamic instability happens less frequently but ends more dramatically. A vibratory mode of a rotor can be stable over a wide range of operating conditions and with a small change in speed or torque may become violently unstable with a rapid growth in amplitude leading either to failure or to a limit cycle if nonlinearities limit the growth.

Having deep knowledge of the physics of the aeroelastic instability of a system makes it easier to find an effective technical solution to prevent it. To achieve this, parametric studies, using simulation code, for investigation of the effects of different system parameters on the instability will be beneficial.

Inside each simulation code first physical model is estimated with a mathematical model. This mathematical model is generally in form of a system of nonlinear differential equations:

$$(1) \quad \dot{x}_i = f_i\left(x_1, x_2, \ldots, x_n\right)$$

For stability analysis of this system, the response of the physical system after a small perturbation from its equilibrium state is studied. In general the behaviour of this nonlinear system could be approximated with the behaviour of a dynamically equivalent linear system.

Applying the Jacobian linearization method for equation (1) with consideration of the linearization point "LP" results the following linear equation:

$$(2) \quad \dot{X} = \begin{bmatrix} \dot{X}_1 \\ \dot{X}_2 \\ \vdots \\ \dot{X}_n \end{bmatrix} = \begin{bmatrix} \left.\frac{\partial f_1}{\partial x_1}\right|_{LP} & \left.\frac{\partial f_1}{\partial x_2}\right|_{LP} & \cdots & \left.\frac{\partial f_1}{\partial x_n}\right|_{LP} \\ \left.\frac{\partial f_2}{\partial x_1}\right|_{LP} & \left.\frac{\partial f_2}{\partial x_2}\right|_{LP} & \cdots & \left.\frac{\partial f_2}{\partial x_n}\right|_{LP} \\ \vdots & \vdots & \vdots & \vdots \\ \left.\frac{\partial f_n}{\partial x_1}\right|_{LP} & \left.\frac{\partial f_n}{\partial x_2}\right|_{LP} & \cdots & \left.\frac{\partial f_n}{\partial x_n}\right|_{LP} \end{bmatrix} \cdot \begin{bmatrix} X_1 \\ X_2 \\ \vdots \\ X_n \end{bmatrix}$$

This equation can be rewritten in the following form:

$$(3) \quad \dot{X} = [A] \cdot X$$

The matrix $[A]$ is called "system matrix".

For the stability analysis of the linear equation (3) following methods can be utilized:

- If the system matrix is constant in time, then eigenvalue analysis leads to stability/instability prediction
- If the system matrix is time dependent and periodic, then applying the Floquet theory or an approximation method (for example usage of multiblade coordinates transformation with time average approximation) is common.

1.1. Helicopter Ground Resonance

The theory of helicopter ground resonance was originally developed by Coleman and Feingold [4]. This phenomenon is a self-excited dynamic instability caused by the interaction of the lagging motion of the rotor blades with other modes of the motion of the helicopter. Elastic deformation and out of phase lagging motion of the rotor blade during rotor rotation causes the oscillation of the rotor centre of gravity around the rotor rotation axis. Due to the interaction between rotor and fuselage, this oscillation excites the flexible fuselage and rotor support and may cause a violent damage of the helicopter structure. This dynamic instability happens when helicopter landing gear or skid gear is in contact with the ground. Therefore this instability is called ground resonance.

Some of the characteristic parameters of this instability are:

- Blade lag frequency
- Blade lag damping
- Frequencies of the structure supporting the rotor
- Daming of the structure supporting the rotor

To perform ground resonance analysis, degrees of freedom of the modelled helicopter are reduced. An Elimination of a DOF is based on its effect on the ground resonance. For a classical ground resonance model just the lagging of the rotor blades and longitudinal and lateral in-plane motion of the fuselage are considered. For this model aerodynamic forces have little influence on the instability effect in compare to structural and inertial forces, therefore aerodynamic forces can be neglected [5]. With consideration of the flapping motion of the rotor blades aerodynamic forces can be neglected.

1.2. Multiblade Coordinates

Generally the equation of motion of a rotor is derived in the rotating frame. In this frame each rotor blade is considered separately but mostly the rotor responds as a whole to an excitation. Changing the coordinate system from rotating to the non-rotating allows to analyse the rotor as a whole system. In the non-rotating frame the so called "multiblade coordinates" are defined. Transformation of the differential equations of motion from rotating frame to non-rotating frame has the following advantages:

- It simplifies the analysis and helps to understand the behaviour of the rotating system, especially, when they are in interaction with non-rotating parts.
- In the case of time-periodic system, this transformation reduces the number of time-periodic elements in the differential equations.
- For a time-periodic system, a combination of multiblade transformation and time average approximation leads to a system of differential equations with constant coefficients.

Consider a rotor with N equally spaced blades. The azimuth of the ith blade is given by:

$$(4) \quad \psi_i = \psi_1 + (N-1)\frac{2\pi}{N}$$

ψ_1 is the azimuth of the first blade. The arbitrary degree of freedom of the rotor blade "β" is then expressed in the multiblade coordinates as follows [6]:

$$(5) \quad \beta_i = \beta_0 + \sum_{1}^{n}\left(\beta_{nc}\cos n\psi_i + \beta_{ns}\sin n\psi_i\right) + \beta_d(-1)^i$$

with $n=1$ to $\dfrac{N-2}{2}$ when N even

or $n=1$ to $\dfrac{N-1}{2}$ when N odd

β_d appears only when N is even.

The multiblade coordinates from (5) are then calculated by following equations:

$$(6) \quad \beta_0 = \frac{1}{N}\sum_{i=1}^{N}\left(\beta_i\right) \qquad \text{collective coordinate}$$

$$(7) \quad \beta_d = \frac{1}{N}\sum_{i=1}^{N}\left(\beta_i \cdot (-1)^i\right) \qquad \text{differential coordinate}$$

$$(8) \quad \beta_{nc} = \frac{2}{N}\sum_{i=1}^{N}\left(\beta_i \cos n\psi_i\right) \quad \text{cyclic coordinate}$$

$$(9) \quad \beta_{ns} = \frac{2}{N}\sum_{i=1}^{N}\left(\beta_i \sin n\psi_i\right) \quad \text{cyclic coordinate}$$

These coordinates defines the new degrees of freedom, which describe the motion of the rotor as a whole in a non-rotating frame. Fig. 2 shows as an example the mode shapes of a four bladed rotor in multiblade coordinates considering the lag degree of freedom.

ξ_0 collective mode

ξ_d differential mode

ξ_{1c} cyclic mode
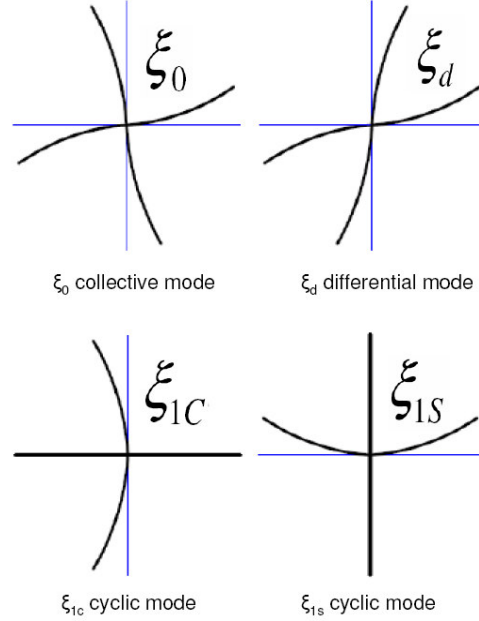
ξ_{1s} cyclic mode

FIG. 2   Multiblade modes of lagging rotor blade

There are similarities between the multiblade coordinates and coefficients of the Fourier series but they are not the same. If the motion in the rotating frame represented by a Fourier series, then the coefficients of series are constant in time but infinite in number and if by Multiblade coordinates then the coefficients or not constant in time and the number of coefficients depends to the number of rotor blades.

1.3. Simulation tool SIMPACK

SIMPACK (**SI**mulation of **M**ultibody systems **PACK**age) software package is used to simulate, analyse and design all types of mechanical system. It can analyse the vibrational behaviour of multibody systems and allows predicting and describing the motion of a complex machine or mechanisms [7]. This tool was originally developed by DLR (German Aerospace Center) and is now further developed and commercially distributed by SIMPACK AG. The software has a comprehensive range of modelling and calculation features. Therefore it is applied within industry, university and research institutions. There are different modelling elements, which are used to simulate a complex system. Data related to these elements can be entered in to SIMPACK via the graphical user interface. Most important modelling elements and options are:

- Reference Frames
- Bodies

- Joints, Constraints
- Force elements
- Sensors, Control elements
- Substructures, Substitutions variables

SIMPACK creates first of all the equations of motion for a modelled mechanical system and then solves it with different mathematical procedures. A wide range of analyses features are available to analyse dynamic systems. These are:

- Static Analysis
- Kinematics Analysis
- Non-linear Dynamic Analysis
- Linear System Analysis
- Symbolic Code Generation
- Eigenvalue Analysis

2. GROUND RESONANCE MODELLING APPROACHES

There are different modelling approaches of the ground resonance effect. The differences of these approaches returns back to the accuracies of the modelling and to the methods of analysis. The structure of a helicopter is in general divided in to two main groups. The first group includes rotor and the second group fuselage. For each group there are different modelling methods. For reduction of the complexity of the problem, the degrees of freedom of each group can be reduced to the ground resonance dominant degrees of freedom.

2.1. Modelling of the Rotor Blades

Two methods for modelling a rotor blade are used. In the first method the rotor blade is modelled as rigid body. The elasticity and structural damping of the blade are modelled with equivalent spring and damper. For a hingeless rotor besides the equivalent spring and damper an equivalent lagging hinge is also considered. The calculations of the equivalent hinge, spring and damping coefficients are done under assumption of the equivalency of the first lagging harmonic of both hingeless and equivalent rotor. Fig.3 shows a general form of a rotor blade created with the first method.

In the second method the blade is modelled as an elastic body using directly a FEM code or SIMPACK beam elements (SIMBEAM). In both cases an eigenvalue analysis of the structure should be first performed. Usage of these results in SIMPACK interface with FEM code (FEMBS) allows creating a standard input data file (SID), which is used by SIMPACK to calculate the elastic deformation of the structure.

2.1.1. Frequency Analysis of the Rotor Blade

Generally for the stability analysis of a nonlinear system in frequency domain, its linearized form is considered. Therefore for blades created with the first or second method the linearization method of SIMPACK should be evaluated.
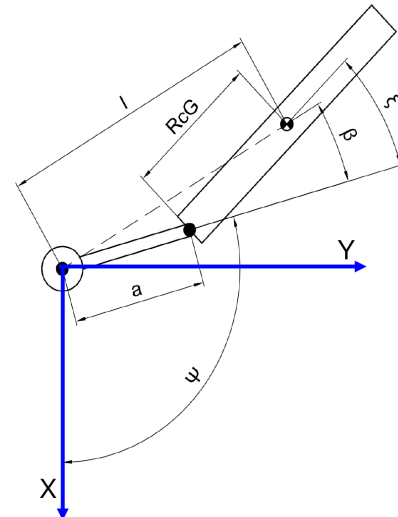

FIG. 3   Rigid blade with equivalent elasticity

Using the first method of the blade modelling, a blade of a windmill-powered plant, shown on Fig.4 was created. "I" given on Fig.4 defines the blade mass moment of inertia at the rotor centre and about the "Z" axis and "K" gives the rotational spring stiffness. This model was linearized for different rotational velocities and azimuth angles.
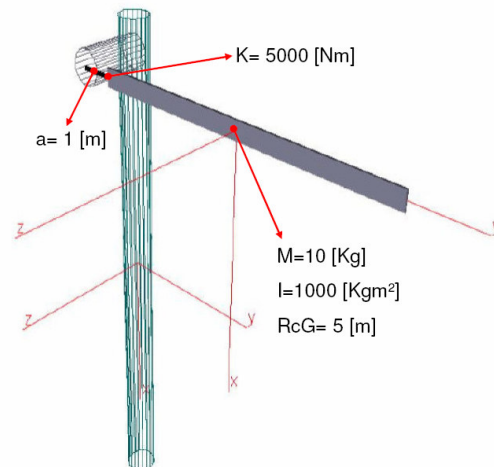

FIG. 4   SIMPACK windmill-powered plant model

Lead-Lag frequencies [rad/s]					
	$\Omega=5$ [rad/s]			$\Omega=20$ [rad/s]		
$\phi=\Omega t$ [deg]	Theoretical	SIMPACK	MSC.ADAMS	Theoretical	SIMPACK	MSC.ADAMS
0	4.2258	4.2258	4.2258	8.4515	8.4515	8.4515
30	4.1421	4.1421	4.1421	8.41	8.41	8.41
60	4.0798	4.0798	4.0798	8.3795	8.3795	8.3795
90	4.0567	4.0567	4.0567	8.3683	8.3683	8.3683

Table 1   Comparison of Eigenfrequencies: Analytical, SIMPACK and MSC.ADAMS

The lead-lag frequencies obtained after the linearization were compared with analytical results and MSC.ADAMS results published on paper [8]. Table 1 gives and compares these frequencies for four different azimuth angle and two rotor rotational velocities.

For frequency analysis of a rotor blade created with the second method of the modelling, an elastic rotating blade with 32 elements, created with SIMBEAM option of SIMPACK, was analyzed. Fig.5 shows this model. To

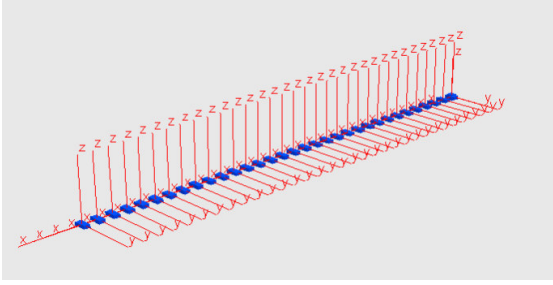evaluate this model the same beam was modelled using MSC.NASTRAN.

FIG. 5   Elastic beam created with SIMBEAM

Eigenfrequencies of these rotating beams were compared together for different rotational velocities. Fig.6 shows as an example the eigenfrequencies of different eigenforms of these two beams for rotational frequency 6 Hz and compares them together. This comparison shows, that the elastic beam created directly with SIMPACK considers the rotational effects (centrifugal effect, which changes the stiffness of the blade and gyroscopic effect, which leads to the coupling of DOFs) as it is considered for NASTRAN model.
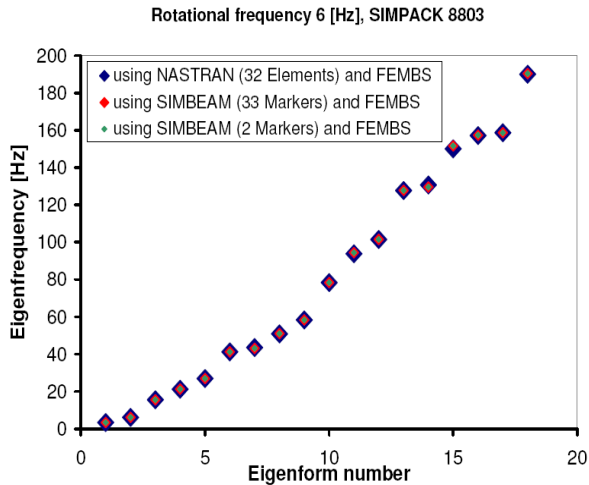
FIG. 6   Eigenfrequencies of different Eigenforms of the rotating beam shown on Fig.5

2.1.2.   Frequency Analysis of an Isolated Rotor

With blades created with the first method of the blade modelling, an isolated rotor was created. The data related to this rotor belongs to the scaled BO105 rotor [3] and will be given later for description of the ground resonance model. For this isolated rotor a created sensor measures the location of the rotor centre of gravity. The blades are first disturbed out of phase in lag direction and then time integration is performed. Fig.7 shows the result of the time integration of the first rotor blade.
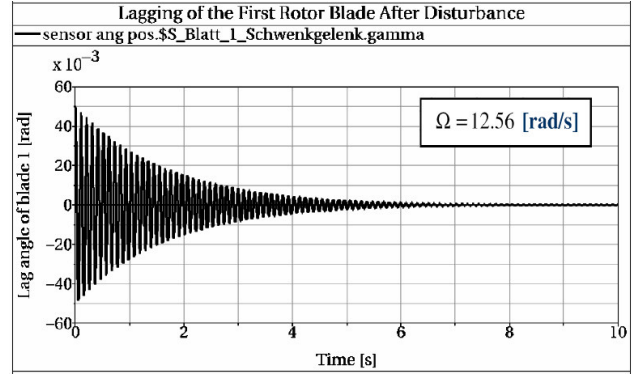

FIG. 7   Oscillation of the rotor blade after a small perturbation

FFT analysis of this result, shown on Fig.8, gives as expected the lag frequency of the blade, which depends to rotor rotational velocity.
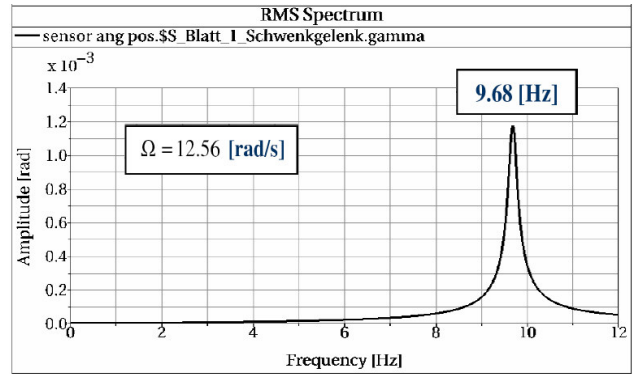

FIG. 8   Frequency spectrum of the blade oscillation shown on Fig.7

Because of the out of phase oscillation of the rotor blades, the rotor centre of gravity oscillates too. The Frequencies of this oscillation can be obtained with FFT analysis of the movement of the rotor centre of gravity, which is captured by a sensor. Fig.9 shows these frequencies. It can be seen that, between the frequencies of the rotor centre of gravity, rotor rotational frequency and blade lagging frequency the following relation exists [9]:

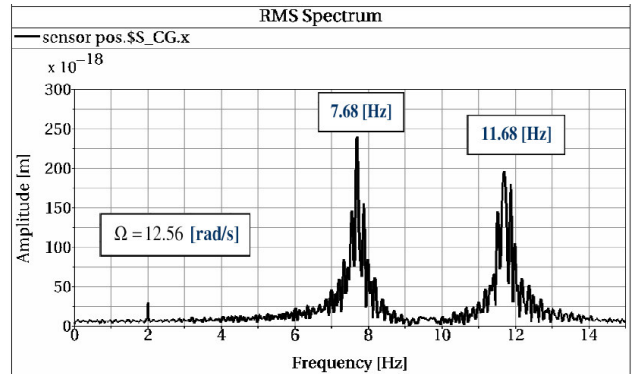$$(10) \quad f_{CG} = f_{Blade,Lagging} \pm f_{Rotor,\Omega}$$

FIG. 9   Frequency spectrum of the oscillation of the rotor centre of gravity.

Performing of this frequency analysis for different rotor rotational velocities (rotational frequencies) and plotting all

the frequencies results in the lines shown on Fig.10. A Similar frequency diagram will be seen later in ground resonance analysis of a whole helicopter.
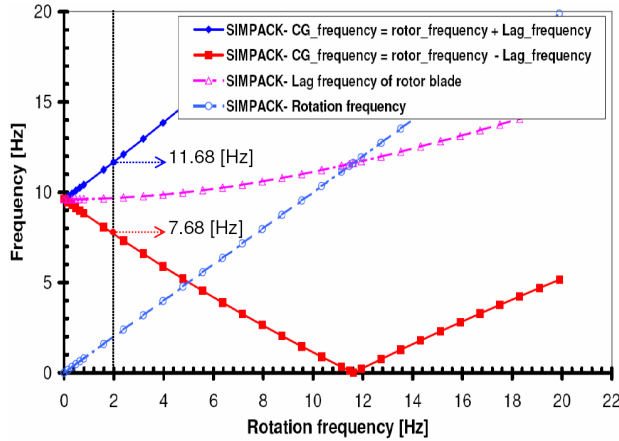

FIG. 10   Frequencies of the rotor centre of gravity and blade lag mode

As a further analysis for this isolated rotor, Fig.11, 12 and 13 show graphically the planer displacement of the centre of gravity of this rotor for three different rotational velocities.   Comparison of these figures shows the irregularity of the patterns of these displacements.   For this isolated rotor the multiblade coordinates were defined as outputs. These Outputs were used to calculate the position of the rotor centre of gravity with the following equations [9], see Fig.3:

$$(11)\quad X_{CG} = \frac{\xi_{1C}(a+R_{CG})}{2}$$

$$(12)\quad Y_{CG} = \frac{\xi_{1S}(a+R_{CG})}{2}$$

The results obtained from these equations were plotted on Fig.11, 12 and 13 and compared with the results obtained directly from the sensor measurements. A Conclusion in respect to equations 11 and 12 and Figures 11 to 13 would be that, instead of frequencies of the oscillation of the rotor centre of gravity the frequencies of the multiblade coordinates can be analysed.
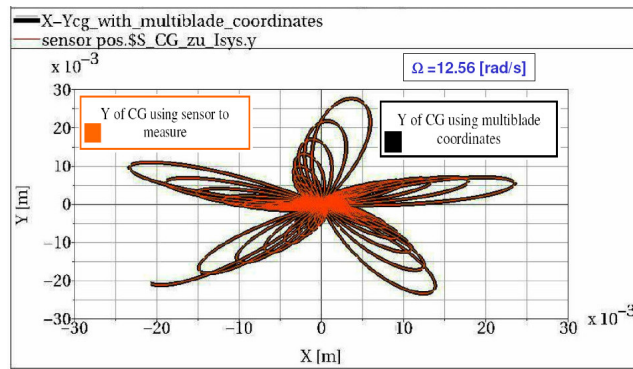

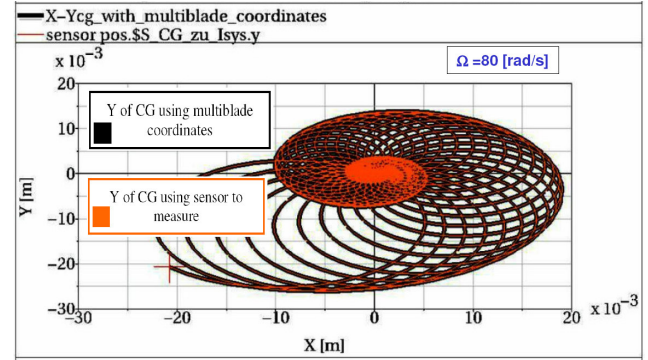FIG. 11   Displacement of the rotor centre of gravity

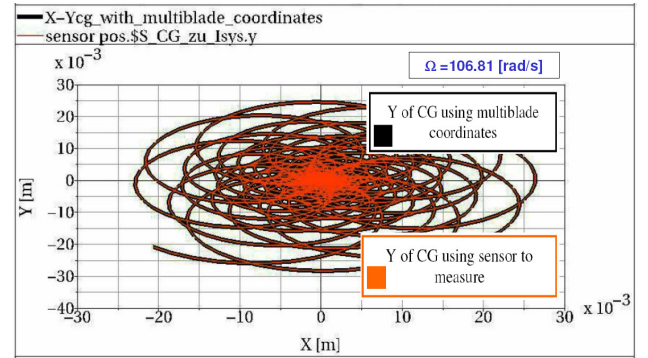FIG. 12   Displacement of the rotor centre of gravity

FIG. 13   Displacement of the rotor centre of gravity

As a next step, using the SIMBEAM option of SIMPACK the elastic blade of BO105 helicopter was modelled (structure data was provided by Eurocopter). After modelling the blade, an isolated rotor as shown on Fig.14 was created.
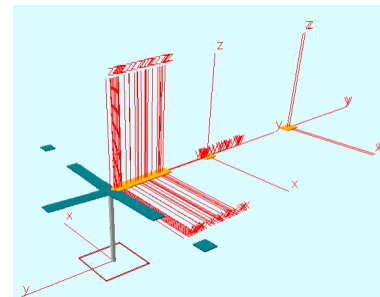

FIG. 14   Isolated BO105 rotor

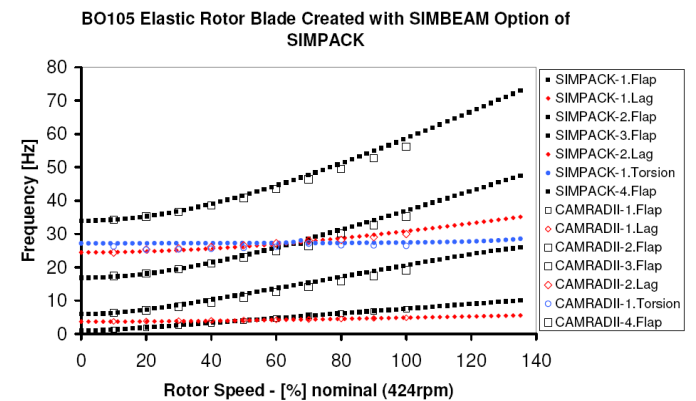For this rotor frequency diagram (Fig. 15) was created.

FIG. 15   Cambell diagram of BO105 rotor

On Fig.15 frequencies of the different eigenforms calculated with SIMPACK are compared with results produced with CAMRADII. This comparison shows a good agreement of the both SIMPACK and CAMRADII models. Because the results of ground resonance analysis produced with SIMPACK will be compared with CAMRADII results, therefore it is necessary both models to be dynamically equivalent. For classical ground resonance analysis the first lag frequency of the blade is most important and is often considered as only model degree of freedom of the blade, therefore a modification (calibration) is necessary to match these lag frequencies with those obtained from CAMRADII. This ensures the dynamically equivalence of both models. Fig.16 shows the first lag frequencies of the rotor blade after and before modification and compares them with CAMRADII results.
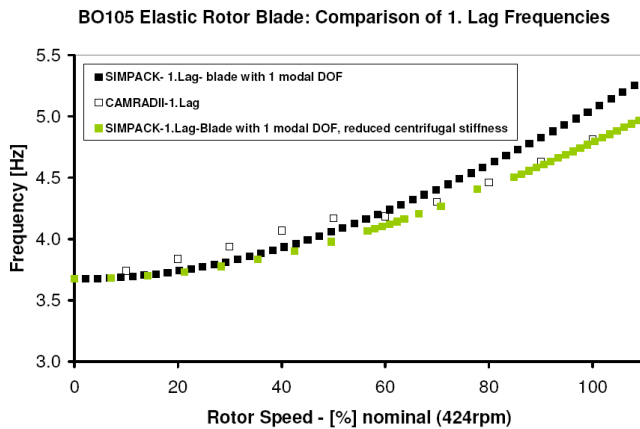

FIG. 16   Lag frequencies of the BO105 rotor blade

For the modification of the lag frequencies a reduction in centrifugal stiffness was considered.

2.2.   Modelling Approaches of the Fuselage

For modelling the helicopter fuselage three methods can be applied. Fig.17 shows schematically these methods. In the first method the fuselage is modelled directly using SIMPACK modelling elements (mass, spring, damper). For the definition of the spring or damping coefficients, we consider this fact that, the fuselage model and real fuselage should be dynamically equivalent. The calculations of these values are normally based on the results of the experimental ground vibration test or modal values. These equivalent values of the fuselage are normally determined experimentally by ground vibration test. The second method applies the SIMPACK interface with FEM code. The fuselage is first modelled using a FEM code and then the information related to this elastic model is sent to SIMPACK. The SIMPACK interface with FEM code (FEMBS) creates a standard input data file (SID), which is used to define the structure and elasticity of a body. All matrices inside this file are in modal coordinates. In the third method one can edit or create directly a SID file using the modal values (modal mass matrix, modal stiffness matrix, modal damping matrix and etc.) of the fuselage. Modal values can be obtained for example after a ground vibration test.
For ground resonance calculation, it is sufficient to evaluate fuselage frequencies not higher than the rotor frequency minus lagging frequency. The Fuselage natural frequency on its landing gear varies depending on its percent airborne condition. Because of this a helicopter may be stable when fully on the ground but go in to ground resonance in a partially airborne condition. It is assumed that the helicopter on its landing gear can be represented by effective parameters applied at the fuselage model.
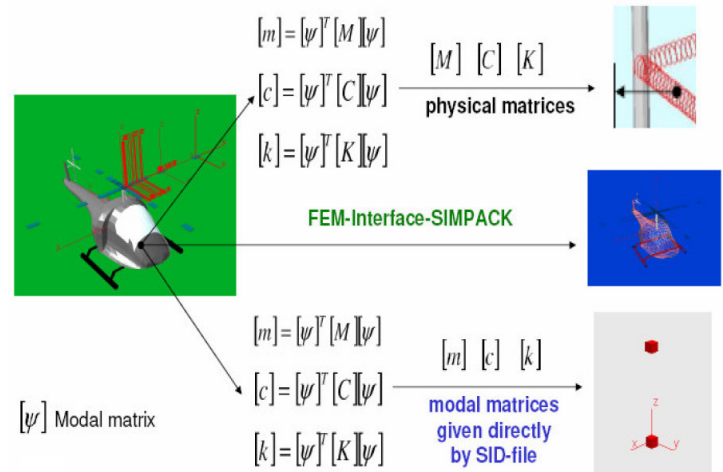

FIG. 17   Modelling options of the Fuselage

3.   GROUND RESONANCE ANALYSIS (G.R.A)

In this part two types of ground resonance model are considered. In the first modelling type, blades are rigid and their elasticity are defined with equivalent springs (first method of the blade modelling) and in the second model the blades are elastic and are created by finite element method.   To perform a ground resonance analysis after creating the model the model is first linearized. For this linear system then   a multiblade transformation is performed to convert the rotor coordinates from a rotating frame in to a fixed frame. This transformation averages out the periodic terms in the motion equations over time, which leads to a system of equations with constant coefficients. Finally an eigenvalue analysis leads to the frequencies and damping of the system. This procedure is repeated for different rotor rotational velocities and at the end the obtained frequencies and damping are plotted together.

3.1.   G.R.A: Rigid Blade with Equivalent Elasticity

With the data given on the next page, a whole helicopter model was created to perform the ground resonance analysis. Fig.18 shows this model. In this model the fuselage has two translational degrees of freedom and rotor blade has only lag degree of freedom. Blades are jointed to the lag hinge with rotational springs and dampers. The data related to the rotor are based on the scaled research rotor "BO105, DNW, configuration K20 [3]. This SIMPACK blade model is in fact a mathematically equivalent model of the hingless scaled BO105 rotor blade. This hingless rotor blade is modelled with an equivalent lagging hinge and equivalent spring and damper to model the stiffness and structural damping of the hingless blade.
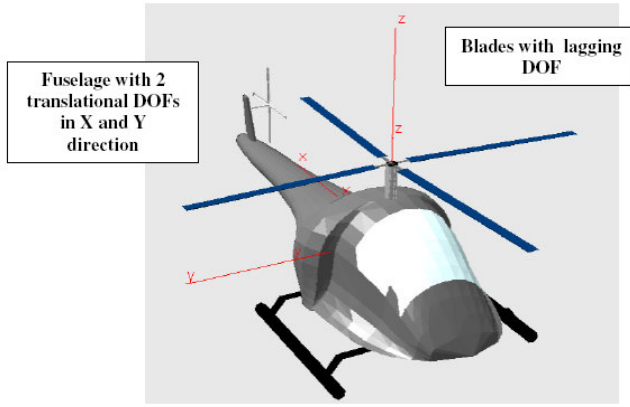
FIG. 18   SIMPACK ground resonance model

The data related to the model shown on Fig.18 are :
Rotor data
0.35 [m]   Distance between the lagging hinge and the axis of rotation.
1.39 [Kg]   Blade mass
1.66 [m]   Blade length
1.281[Kgm²]   Blade mass moment of inertia about lagging hinge
81.6 [Nm/grad]   Lagging stiffness
1.5 [% crit.]   Damping ratio

Fuselage data
160000 [N/m]   Spring stiffness in Y direction
77.519 [Ns/m]   Damping coefficients in Y direction
160700 [N/m]   Spring stiffness in X direction
65   [Ns/m]   Damping coefficients in X direction
3.51   [Hz]   Frequency of eigenform in Y direction for $\Omega = 0$
3.52   [Hz]   Frequency of eifenform in X direction for $\Omega = 0$
0.54   [% crit.] Damping ratio in Ydirection for $\Omega = 0$
0.45   [% crit.] Damping ratio in X direction for $\Omega = 0$
322   [Kg]   Fuselage mass

To perform the stability analysis the rotor rotational velocity is changed stepwise. For each value of the rotational velocity the system is linearized about its equilibrium state. Then the linear system matrix of the model is extracted and saved in MATLAB m-file format. A MATLAB program, implemented by writer, is used to perform the coordinates transformation. This program transforms the linear system matrix "$[A]$" from rotating coordinates to non-rotating coordinates. The new matrix is called "$[T]$". Fig. 19 shows the linear differential equation of the system in rotating and non-rotating coordinates and illustrates the definition of the system matrices A and T. After performing the transformation, the eigenvalues of the transformed linear system matrix $[T]$ are determined and then with respect to the eigenforms are sorted. Finally the frequencies and damping are calculated and evaluated.
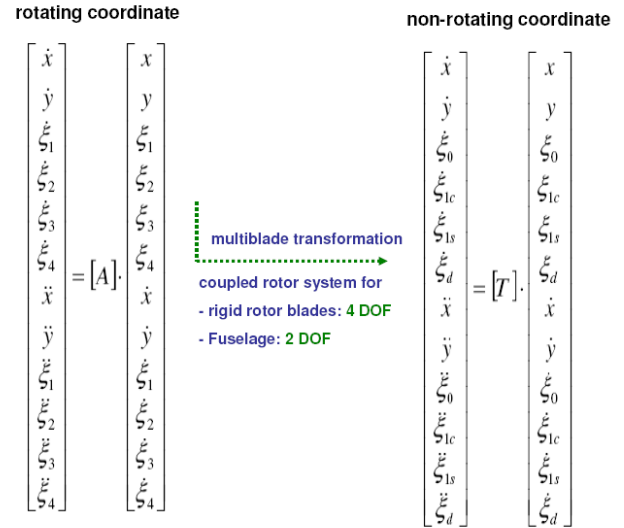

FIG. 19   Multiblade coordinates transformation

A CAMRAD II model of the described ground resonance model was created to validate the performed procedures. From the data used for SIMPACK model, the data needed for the CAMRAD II model were calculated. This is necessary due to the differences between the modelling approaches of CAMRADII [10] and SIMPACK. Fig.20 shows schematically the CAMRAD II rotor blade.
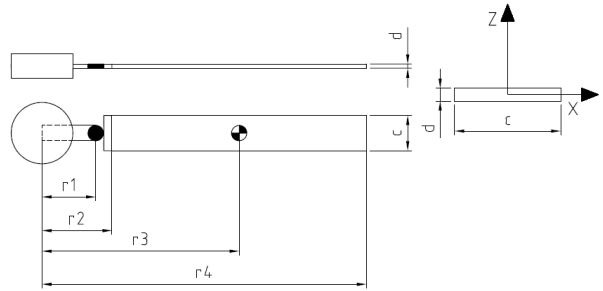

FIG. 20   CAMRADII rotor blade model

From the mass moment of inertia of the SIMPACK model and the rotor length one can calculate the rotor chord length (see, Fig.20 and Equ.13) as follows:

$$(13) \quad \begin{aligned} & b = r4 - r2 = 1.658[m] \\ & I_{CG} = \frac{m(b^2 + c^2)}{12} = \frac{1.3896(1.658^2 + c^2)}{12} = 0.3245 \\ & \Rightarrow c = 0.2308[m] \end{aligned}$$

Then density of the blade is calculated from its geometry and mass:

$$(14) \quad \rho = \frac{M_{blade}}{b \cdot c \cdot d} = \frac{1.3896}{1.658 \cdot c \cdot 0.12c} = 131.1[Kg/m^3]$$

Equations 15 and 16 give the values of the two other parameters needed in the blade input file of CAMRAD II.

$$(15) \quad I_{THETA} = \int (x^2 + z^2)\rho \cdot dA = 0.00377359[kgm]$$

$$(16) \quad I_{POLAR} = \int \left(x^2 - z^2\right)\rho \cdot dA = 0.00366645[Kgm]$$

Fig. 21 shows graphically the frequencies and damping of the both SIMPACK and COMRAD II models in multiblade coordinates (first model). Comparison of the results shows the agreement of the both models and validates the implemented MATLAB program for multiblade coordinates transformation and also approved the considered procedure of ground resonance analysis.

According to the Fig.21 the ground resonance of the model happens by rotational frequency ~17 [Hz]. This instability happens when the regressing frequency of the rotor interfaces the fuselage frequency. However this happens in two different points. Investigating the damping at both intersection points shows that just in one intersection point the damping of the fuselage dominant eigenform reduces and leads to instability. The instability of this ground resonance model depends to the following parameters:

natural frequencies of the helicopter and the blades, the distance between the rotor hub and the lead-lag hinge, the moment of inertia of the blade about the lead-lag hinge, the rotor speed, the mass of blade, and the effective mass of the fuselage.

To see the effect of each of these parameters on the stability, one can perform parameter studies using the parameter variation option of SIMACK "ParVariation".
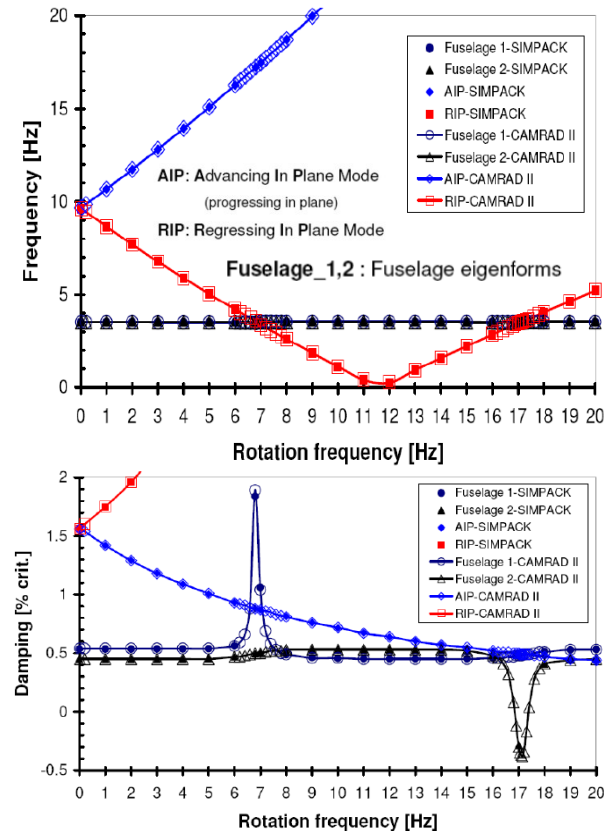

FIG. 21   Frequency and damping of the eigenvalues of the transformed system matrix, first model

By the first ground resonance model the frequencies of the fuselage dominant eigenforms had nearly the same values. For the next model (second model) these frequencies were separated from each other (by model

modifications) to prevent any coupling effect. The data used for this model are:

Rotor data
Same data used for the first model

Fuselage data

31135	[N/m]	Spring stiffness in Y direction
143	[Ns/m]	Damping coefficients in Y direction
15645000	[N/m]	Spring stiffness in X direction
6500	[Ns/m]	Damping coefficients in X direction
3.48	[Hz]	Frequency of eigenform in Y direction for $\Omega = 0$
79.17	[Hz]	Frequency of eigenform in X direction for $\Omega = 0$
5	[% crit.]	Damping ratio in Y direction for $\Omega = 0$
10.39	[% crit.]	Damping ratio in X direction for $\Omega = 0$
59.09	[Kg]	Fuselage mass

With these data two models were created using SIMPACK and CAMRADII. For both models the ground resonance analyses were performed. Fig. 22 shows the frequencies and damping of both SIMPACK and CAMRADII models in multiblade coordinates. Comparison of the results of the both simulation tools approved again the procedure defined for ground resonance analysis with SIMPACK.
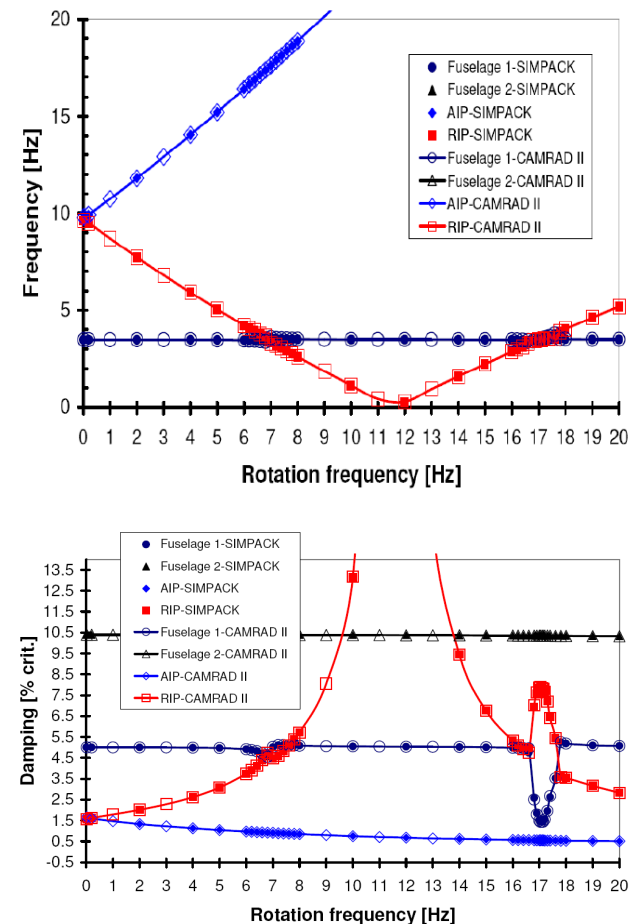

FIG. 22   Frequency and damping of the eigenvalues of the transformed system matrix, second model

Analysing the damping on Fig.22 shows that at the point of instability the energy needed for increasing the amplitude of the fuselage oscillation comes from the rotor. Increase or decrease of the damping can be better

understood, when the exchange of the energy between the fuselage and rotor is analysed.

Fig.23 shows the results of ground resonance analysis of two further considerable test cases. In one case the structural damping of the fuselage is zero while these values for blades are non zero and in the second case the structural damping for both blades and fuselage are zero. The upper diagram of Fig.23 shows the frequencies for both test cases. Diagram in the middle shows the damping for the first case and lower diagram shows the damping for the second case.

stability analysis using multiblade transformation fuselage with 2/1DOF

stability analysis using multiblade transformation fuselage with 1DOF

stability analysis using multiblade transformation fuselage with 1DOF, structural damping of fuselage and blade=0
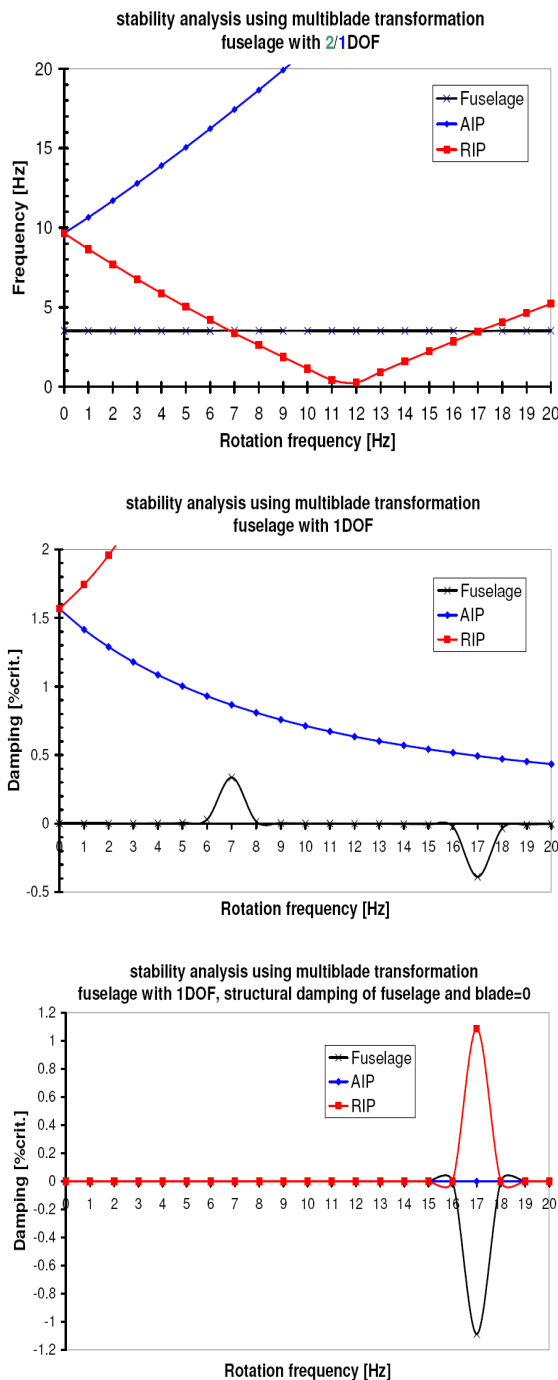

FIG. 23   Ground resonance analysis, test cases

The change of the damping in the neighbourhood of the frequencies intersection point causes the changes in dynamic behaviour. It can be also seen on lower diagram

of Fig.23 that, for a rotor with constant rotational velocity the sum of the energy exchange between the rotor and fuselage at the ground resonance point will be zero.

3.2.   G.R.A: BO105 Blade Created with FEM

It was described, that the hingless rotor blade of BO105 was created using SIMPACK and calibrated to be dynamically equivalent with its similar CAMRADII model. Considering the modal values used by CAMRADII model to model the BO105 fuselage, an dynamically equivalent fuselage with mass, spring and damper was inside SIMPACK created. The dynamic equivalency means here the equivalency of the eigenforms and eigenvalues of the both models under the assumption of the equality of the masses and mass distributions.

Fig.24 shows the ground resonance model of BO105 created inside SIMPACK. This model has 4 modal DOFs for the blades (one for each blade) and 2 translational DOFs (longitudinal and lateral) for the Fuselage.
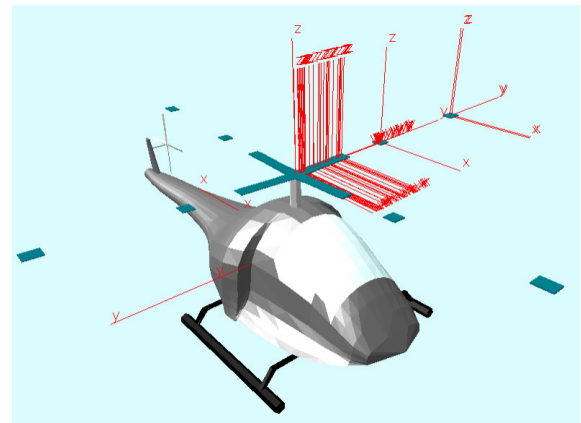

FIG. 24   SIMPACK BO105 ground resonance model

The procedure of the ground resonance analysis for this model is identical to the procedure performed for previous model.

Fig. 25 shows the results of this analysis. The upper diagram shows the frequencies and the lower diagram the damping. On this Figure the results are compared with CAMRADII results. It can be seen that, there is a small differences between the CAMRADII and SIMPACK results. This difference is firstly due to the difference between SIMPACK and CAMRADII fuselage models and secondly due to the rotor models. Although the SIMPACK blades were calibrated with CAMRADII blades, still the eigenfrequencies of the blades do not match 100% to each other and this results in the different ground resonance points (offset between the damping picks on Fig.25). It was shown in previous part, if both rotors/Fuselages are exactly similar, then the ground resonance results of both tools will be the identical.

an Interesting effect seen on this model is that, in the neighbourhood of the frequencies intersection point, damping of the fuselage dominant mode increases (fuselage gives energy) and damping of the rotor dominant modes decreases (rotor receives energy). It should be mentioned that, for coupled fuselage and rotor system, fuselage modes refer to fuselage dominant eigenmodes and rotor modes refer to rotor dominant eigenmodes. For a fuselage dominant eigenmode the amplitude of the fuselage oscillation is larger than the amplitude of the blade oscillation.

Comparison of the different ground resonance simulation results shows that, at the ground resonance point the eigenmdoe with reduced damping would be the one with smaller damping before frequencies intersection point. This fact can be seen for example by comparison of Fig.22 and 25.
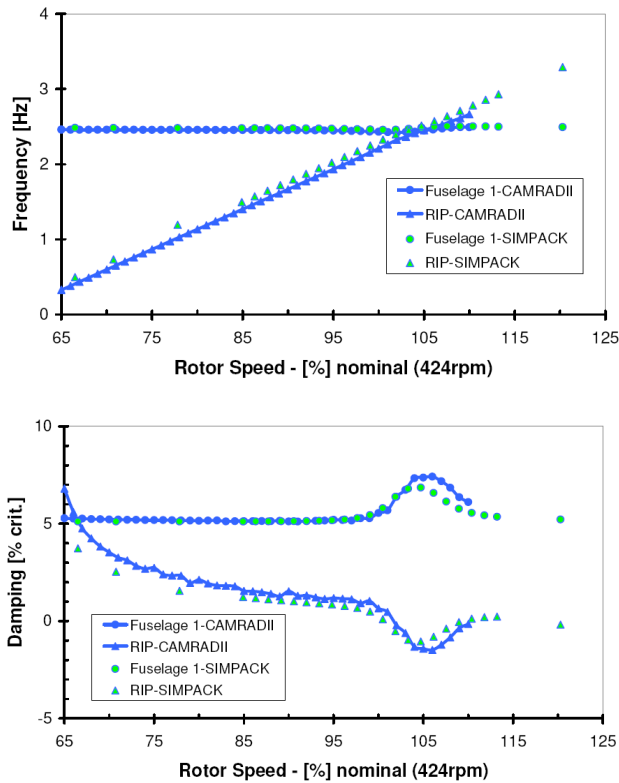

FIG. 25   Result of BO105 ground resonance analysis: Frequency and Damping

Different parametric studies were performed for the created ground resonance models and different results were produced. Fig.26 shows as an example one of these results. It shows the effect of the increase or decrease of the blade structure damping on the ground resonance effect.
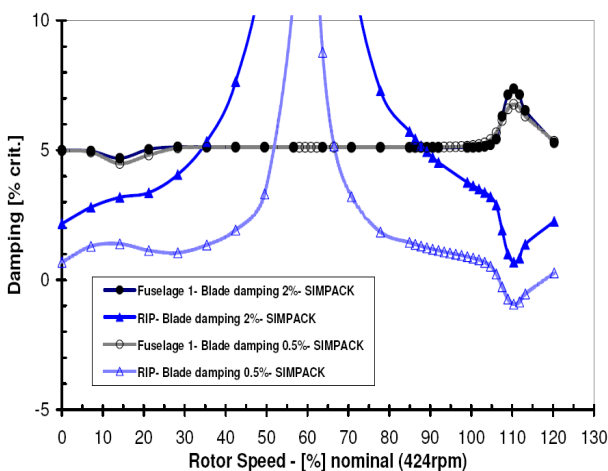

FIG. 26   SIMPACK ground resonance analysis: parametric studies

4.   CONCLUSION

During this work, SIMPACK was investigated as a multibody dynamics tool for ground resonance analysis. This investigation is divided in to two main parts: first part deals with the modelling approaches and the second part deals with the methods of the calculation and analysis.

In the first part some modelling options and features of SIMPACk for creating a helicopter-model with elastic blades were evaluated. Some of these options are:

- Modelling of the elastic blade and fuselage
  a-rigid blade with equivalent elasticity
  b-elastic blade created using FEM
  c-rigid fuselage with equivalent elasticity
- Parametric modelling and studies
  SIMPACK allows defining parameters inside the model and performing different analysis with parameter variation. This option can be used for example to produce the cambell diagram. In this case the rotor rotational velocity is defined as a model parameter and eigenvalue analysis is performed for different values of this parameter.
- Expressions (user defined functions)
  Using the expressions, the multiblade coordinates can be defined inside SIMPACK and can be requested as output. This allows analysing the multiblade coordinates in time domain.

Related to the methods of the calculation, the following points were analysed:

- Linearization of a nonlinear system
  Linearization option of SIMPACK gives valid results for a rotating system. These results are used for ground resonance analysis.
- Rotational effects(geometrical stiffening, gyroscopic effect and etc.)
  For a rotating system SIMPACK considers the forces resulting from rotation.
- Multiblade coordinates transformation
  After linearization of a non-linear system, SIMPACK creates the linear system matrices. These matrices can be saved in MATLAB m-file format. An implemented MATLAB program is then used to perform the multiblade coordinates transformation. This coordinates transformation was validated before its usage for ground resonance analysis.
- SIMPACK post-processing options (for example coordinate transformation, FFT analysis)
  In most cases SIMPACK can be used directly as a post-processor for the evaluation of the simulation results. SIMPACK can perform the Fast Fourier Transformation (FFT) for the results saved as an output or can filter the results with user defined or selected filter.

For the evaluation of the simulation results, SIMPACK results were compared with analytical results (in case of linearization) and results of the other simulation tool (in case of ground resonance analysis the results were compared with the results of CAMRADII).

In this work two main different ground resonance models were created. In the first main model the blade was rigid and its elasticity was modelled with an equivalent spring and in the second model the blade was modelled directly as an elastic part using FEM.

After performance of the ground resonance analysis for

each model the results were compared with the results of its similar/equivalent CAMRADII model. Comparison of the results shows the validity of the ground resonance modelling approaches and calculation methods defined within this paper and shows SIMPACK as a convenient multibody simulation tool for the helicopter ground resonance analysis. However, for the usage of this tool for a complex and exact helicopter dynamics analysis still new methods should be developed.

ACKNOWLEDGMENT

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