USING MULTIBODY DYNAMICS FOR THE SIMULATION OF FLEXIBLE ROTOR BLADES – GETTING THE MECHANICAL COUPLING RIGHT?

J. Arnold
DLR – German Aerospace Center
Institute of Aeroelasticity, 37073 Göttingen, Germany

ABSTRACT

The process of structural dynamic modelling of the flexible rotor blade in a multibody system is demonstrated for the BO105 and ERATO rotors. Flexibility of the non-rotating blade is described in the multibody system SIMPACK with a modal approach which is gained from a real modes solution and additional geometric stiffness contributions for the rotating blade are considered by static load cases, both computed in a pre-processing step with the finite element software MSC.NASTRAN.

1. INTRODUCTION

A strong interest from industry to introduce generic software solutions next to available comprehensive helicopter analysis tools into the current design process is observed. These tools are not limited to pre-described rotor blade layouts or rotor hub systems and thus, allow the investigation of highly innovative concepts beyond standard solutions. For the description of the pure mechanical model of a rotor, a multibody system (MBS) might be a solution. Focus is on the simulation of the flexible rotor blade with MBS SIMPACK to allow a statement whether the correct mechanical representation of the elastic rotor blade including the coupling activated for the rotating structure, i.e. bending-torsion, tension-bending and tension-torsion, is possible with the available software.

The presented work is based on results gathered during the French-German SHANEL project [1] with the partners Eurocopter, Eurocopter Deutschland, ONERA and DLR. Two structural dynamic models comprising the conventional BO105 and the innovative blade layout of the ERATO rotor blade [2] have been investigated with MBS SIMPACK. According results not considering aerodynamics have been evaluated by the comparison of the processed BO105 fan diagrams against the comprehensive helicopter software CAMRAD II [3] and the modal parameters of the clamped ERATO rotor blade against experimental data from a vibration test. Focus of this work is on the comparison of simulation results for the flexible BO105 rotor blade.

2. FLEXIBLE ROTOR BLADE MODELS

2.1. Rotor blades modelled with FEM

Two finite element models are available for the BO105 rotor blade with 37 beam elements based on a CAMRAD II model and the ERATO rotor blade with 58 beams derived from HOST [4] input data. The rotor blade definitions comprise blade geometry, radial distributions for stiffness and mass, pitch joint position, blade pitch control stiffness, blade modal damping and lag damper data, if appropriate. Both models show the following features with MSC.NASTRAN:

- Beam elements are placed along the quarter chord line.
- Beam entries CBEAM / PBEAM define continous stiffness and mass distributions.
- Beam offsets relative to the quarter chord line define shear center, tension center and center of gravity.
- The pitch joint is modelled with a rotational spring element CELAS2 representing blade pitch control stiffness in the pitch degree-of-freedom and a rigid RBE2 element in the constraint degrees-of-freedom.
- The mass definition CONM2 is used to describe additional concentrated mass points.
- Rigidly connected nodes model trailing and leading edge positions.

Figure 1 and Figure 2 illustrate the used beam models of the BO105 and ERATO rotor blades with radial location of the pitch joint. In the present work, MSC.NASTRAN version 2007 has been used.

For BO105, the elastic axis is found in front of the straight quarter chord line due to stiffness concentration of the C-spar towards the leading edge. Simplifications comprise a single load path model, a single pitch joint modelled for the outboard pitch bearing position only, as well as the neglect of flap pendulum absorber and swashplate. The resulting mass including the distributed blade mass defined by the beam elements and the additional mass points is 51.16 kg.

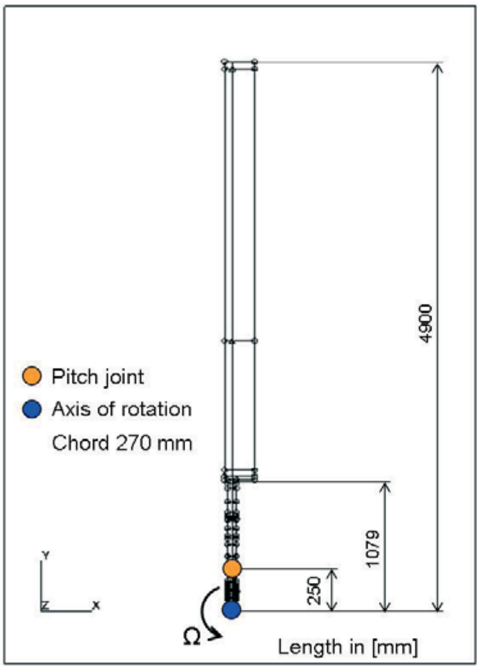

Figure 1: BO105 finite element model

The acoustically optimized ERATO rotor is a collaborative design of ONERA and DLR resulting in a non-conventional blade shape.

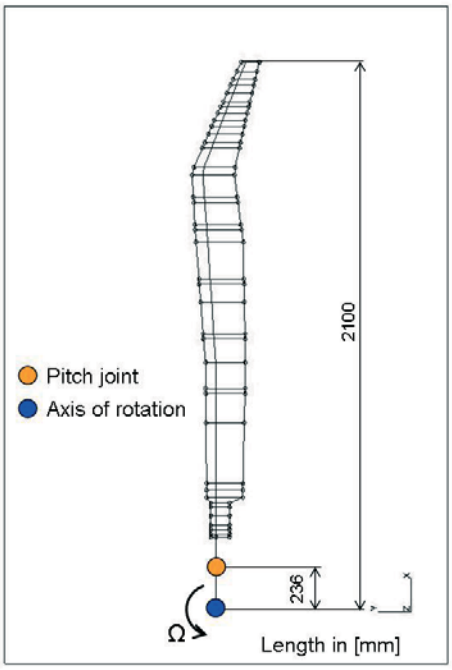

Figure 2: ERATO finite element model

2.2. Rotor blades modelled with MBS

The multibody system SIMPACK [5] is used to simulate the mechanical model of the flexible rotor blades including large rigid body motions and small deformations of the elastic structure. The development of the simulation package was initiated by DLR and later out-sourced to INTEC [6] for further development and commercial distribution. It provides all non-linear inertial coupling terms and allows the setup of elastic simulation models.

Flexibility of the non-rotating blade is described in the multibody system SIMPACK with a modal approach [5]-[8] for available finite element models. Several features allow the introduction of elastic rotor blades:

(1) Implementation of complete elastic model as one elastic body and additional geometric stiffness terms via standard FEMBS interface

(2) Implementation of elastic model with connected elastic substructures via standard FEMBS interface

(3) Application of the intrinsic elastic beam model SIMBEAM

(4) Application of the Rotor Blade Generator based on SIMBEAM in new version 8900

Feature (1) allows the straight-forward use of finite element models of industrial model size. Currently, the FEMBS interface [6] supports ABAQUS, ADINA, ANSYS, IDEAS, MSC.NASTRAN, NX.NASTRAN and PERMAS. Feature (2) might be advantageous to add further non-linear characteristics of the multibody joints that interconnect elastic substructures. Features (3) and (4) provide a solution, if a finite element code is not available or is not supported by FEMBS. In the present work, FEMBS version 8.705b and SIMPACK version 8.803 has been used.

Feature (1) which is based on the implementation of the complete elastic model together with additional geometric stiffness using the FEMBS interface is chosen for the presented work on the BO105 and ERATO rotor blade. Two FEM solutions from a preprocessing step with the finite element code MSC.NASTRAN are required in FEMBS in order to process the blade:

- A modal solution provides the modal elastic model with natural frequencies and mode shapes for the non-rotating blade at Ω=0 Hz (SOL103).

- A static solution provides the geometric stiffness terms for the description in the relevant degrees-of-freedom of the rotating blade for Ω>0 Hz (SOL101, RFORCE centrifugal forces due to angular velocity).

FEMBS reads the model geometry, mass and stiffness matrices, natural frequencies, mode shapes, geometric stiffening terms and the used load case entries for the derivation of the geometric stiffening terms from the pre-processed finite element data. Then, FEMBS translates this input into a common flexibility description which is based on the Standard Input Data (SID) format as described in [7]. The SID-file with the flexible substructure is added in SIMPACK with marker locations which are similar to the node locations in the finite element model, changing a rigid into an elastic body. Regarding aspects of component modal synthesis and substructuring, any hybrid model consisting of rigid and flexible bodies can be built. Rigid bodies can be made flexible by the introduction of a modal elastic model as described above or spring stiffness can be applied to a joint between two bodies from the MBS element library. SIMPACK adds the additional equations for the modal degrees-of-freedom to the set of differential algebraic equations (DAE) and solves the resulting set of equations on a time step basis using an O(N) formalism with the diagonal matrix structure.

2.3. Comparison to measured vibration data

Verification of the simulation results obtained with multibody dynamics comprise the modal properties in terms of eigenfrequencies and mode shapes for the non-rotating ERATO blade against the finite element results and available GVT results for the clamped blade. Results are not included in this work and will be presented at a later stage.

3. FLEXIBLE ROTOR BLADE MODELS IN THE ROTATING FRAME

3.1. BO105 fan diagrams processed with MBS

The processing of the fan diagram for the rotating flexible blade with SIMPACK is demonstrated for the isolated BO105 rotor blade. In general, the diagram illustrates the dependency of the mode related natural frequencies to the rotor speed due to the consideration of geometric and gyroscopic terms. Also, the mode shapes change due to rotational effects and are reassembled in the multibody system with the natural modes of vibration as shape functions, related participation factors and phase angles. The practical relevance of the fan diagram, which is also known as Campbell diagram, is given for rotor speeds larger than 20% of the nominal rotor speed and up to natural frequencies which correspond to 10/rev.

With a nominal speed of 424 RPM for the BO105 main rotor, the approximate frequency of 7.07 Hz equal to an angular velocity of 44.4 rad/s is obtained for one rotor revolution equivalent to 1/rev. Hence, the BO105 fan diagram is plotted for natural frequencies up to 70 Hz based on 11 modal solutions which are calculated in steps of 5 rad/s up to a maximum angular velocity of 50 rad/s. All blade models have 20 modal degrees-of-freedom and use the rheonom motion joint type within SIMPACK to set the rotor speed around the axis of rotation. By means of varying the angular velocity and subsequent modal analysis, the natural frequencies are processed and the modes are tracked through shape visualization.

Further, a blade setting with a precone of 2.5° and a collective pitch of 3.8° is taken into account for all fan diagrams. Cone angle and collective pitch are adjusted with a zero degree-of-freedom joint relative to the axis of rotation. The blade setting influences flap and lag frequencies and resulting mode shapes in the rotating frame. Three different cases of the BO105 rotor are considered within the NASTRAN beam model and translated with the FEMBS interface to investigate the influence of beam offset definitions comprising shear center, tension center and center of gravity as well as the introduction of the discrete spring stiffness of the pitch joint at the radial station of 250 mm. They are listed in Table 1.

Rotor Blade	Beam Offsets	Pitch Joint
Case 1	no	no
Case 2	yes	no
Case 3	yes	yes

Table 1: MBS models of the BO105 rotor blade

The reference CAMRAD II results plotted for comparison in the following fan diagrams consider all beam offset definitions and the pitch joint with blade pitch control stiffness. Case 3 investigated with MBS has the same features. Seven modes comprising four flap bending (F1-F4), two lag bending (L1-L2) and one torsion mode (T1) are available.

3.2. Rotor blade without beam offsets

The fan diagram of Case 1 for the rotor blade not including beam offsets and pitch joint is depicted in Figure 3. Since the beam offsets are neglected, congruent beam lines for the mass and stiffness distribution are obtained. Shear center, tension center and center of gravity are the same. Mode order and natural frequencies for the non-rotating blade and the blade rotating with nominal rotor speed are found in Table 2. It can be seen that the mode order is changed with increasing rotor speed for flap bending F1 and lag bending L1 as well as for torsion mode T1 and flap bending F3. Frequency increases the most with all flap bending modes, also with lag bending mode L2. The natural frequency of the torsion mode remains nearly the same, whilst the first lag bending mode L1 gives a slightly decreasing value with SIMPACK in the rotating frame.

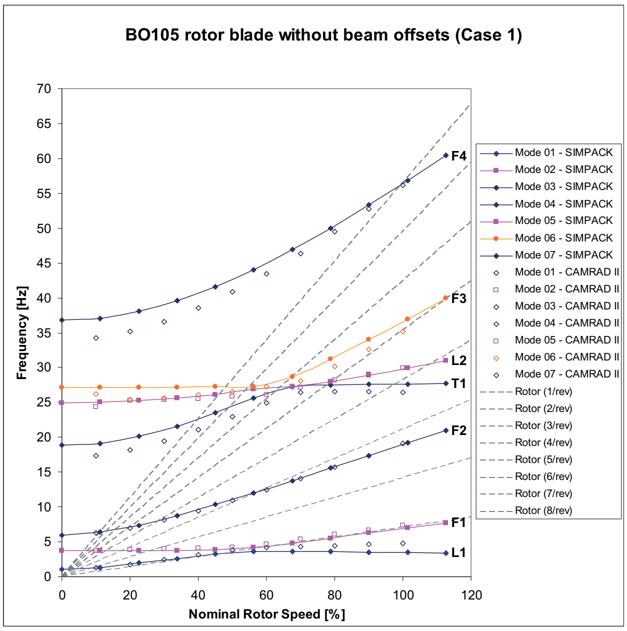

Figure 3: Fan diagram of BO105 rotor blade, Case 1

For a better comparison to Case 2 and 3, also the concentrated mass points with a small mass offset are included to obtain the same mass of 51.16 kg with Case 1 to 3. The mass points comprise 7.27 kg in total with 3.97 kg being placed without offset on the quarter chord line and 3.30 kg being placed with an offset of 15.5 mm in chord direction towards the leading edge at a radial station of 808 mm. Since this concentrated mass and offset are both small, the resulting center of gravity is still very close to shear and tension center. Hence, the mode animation shows well separated shapes in terms of flap, lag and torsion.

Mode No.	0% Rotor Speed	100% Rotor Speed
1	F1 / 0.99 Hz	L1 / 3.45 Hz
2	L1 / 3.77 Hz	F1 / 6.87 Hz
3	F2 / 5.99 Hz	F2 / 18.97 Hz
4	F3 / 18.83 Hz	T1 / 27.72 Hz
5	L2 / 24.95 Hz	L2 / 29.83 Hz
6	T1 / 27.12 Hz	F3 / 36.62 Hz
7	F4 / 36.76 Hz	F4 / 56.37 Hz
(F = flap mode / L = lag mode / T = torsion mode)		

Table 2: MBS mode shapes and natural frequencies for the BO105 blade model, Case 1

3.3. Rotor blade with beam offsets

The resulting fan diagram of Case 2 for the rotor blade including beam offset definitions, but still neglecting the pitch joint, is plotted in Figure 4. The offsets are the same as those used with the reference CAMRAD II model. Since all beam offsets are taken into account, shear center, tension center and center of gravity are different. Mode order and natural frequencies for the non-rotating blade and the blade rotating with nominal rotor speed are found in Table 3. In comparison to the blade model without offsets investigated as Case 1, the natural frequencies found for the non-rotating and rotating blade with higher flap bending modes F3 and F4 decrease by around 2%. All other natural frequencies remain more or less the same and a similar dependence for the flap, lag and torsion modes to rotor speed is obtained. Again, the mode order is changed with increasing rotor speed for flap bending F1 and lag bending L1 as well as for torsion mode T1 and flap bending mode F3.

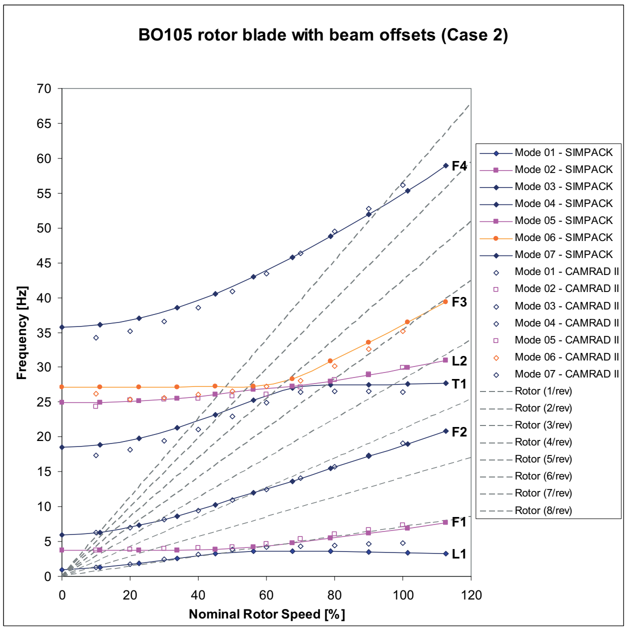

Figure 4: Fan diagram of BO105 rotor blade, Case 2

In general, rotating systems show a coupling between flap, lag, torsion and tension motion, since shear center, tension center and center of gravity have relative offsets to each other as well as to the resulting centrifugal force vector. Elastic couplings activated by the blade rotation comprise bending-torsion, tension-bending and tension-torsion. For Case 2, the mode animation within SIMPACK shows highly coupled mode shapes in terms of bending and torsion. Higher flap and lag bending modes have remarkable torsional contributions, especially found with the second lag mode L2. The influence of mass effect is dominant. This corresponds to general experience regarding the following two aspects: First, frequencies are less sensitive to these model changes than the mode shapes. Second, the higher the mode, the more intense the coupling between flap, lag and torsion in the mode shape is observed.

Mode No.	0% Rotor Speed	100% Rotor Speed
1	F1 / 0.98 Hz	L1 / 3.43 Hz
2	L1 / 3.76 Hz	F1 / 6.84 Hz
3	F2 / 5.93 Hz	F2 / 18.82 Hz
4	F3 / 18.52 Hz	T1 / 27.70 Hz
5	L2 / 24.89 Hz	L2 / 29.77 Hz
6	T1 / 27.10 Hz	F3 / 36.09 Hz
7	F4 / 35.78 Hz	F4 / 54.96 Hz
(F = flap mode / L = lag mode / T = torsion mode)		

Table 3: MBS mode shapes and natural frequencies for the BO105 blade model, Case 2

3.4. Rotor blade with pitch joint

The fan diagram of Case 3 for the rotor blade including beam offset definitions and the pitch joint is shown in Figure 5. The offsets are those from Case 2 and used with the reference CAMRAD II model. The rotational spring element representing the blade pitch control stiffness is adjusted to reach the first torsion mode T1 of approximately 3.7/rev equal to 26.4 Hz at 100% of nominal rotor speed, as found with the CAMRAD II model. Since all beam offsets are taken into account, shear center, tension center and center of gravity are different. Mode order and natural frequencies for the non-rotating blade and the blade rotating with nominal rotor speed are found in Table 4.

Mode No.	0% Rotor Speed	100% Rotor Speed
1	F1 / 0.98 Hz	L1 / 3.43 Hz
2	L1 / 3.76 Hz	F1 / 6.84 Hz
3	F2 / 5.93 Hz	F2 / 18.82 Hz
4	F3 / 18.52 Hz	T1 / 26.32 Hz
5	L2 / 24.89 Hz	L2 / 29.78 Hz
6	T1 / 25.85 Hz	F3 / 36.09 Hz
7	F4 / 35.77 Hz	F4 / 54.92 Hz
(F = flap mode / L = lag mode / T = torsion mode)		

Table 4: MBS mode shapes and natural frequencies for the BO105 blade model, Case 3

Of course, the modelled pitch joint and related control stiffness changes the natural frequency obtained for torsion mode T1 which fits the reference value at 100% rotor speed perfectly. On the other hand, the additional stiffness does not effect the obtained frequencies of the bending modes and their natural frequencies are identical for all flap and lag bending modes to those from Case 2. Mode animation within SIMPACK shows similar couplings and the mode order is changed again for increasing rotor speed with flap bending mode F1 and lag bending L1 as well as for torsion mode T1 and flap bending F3.

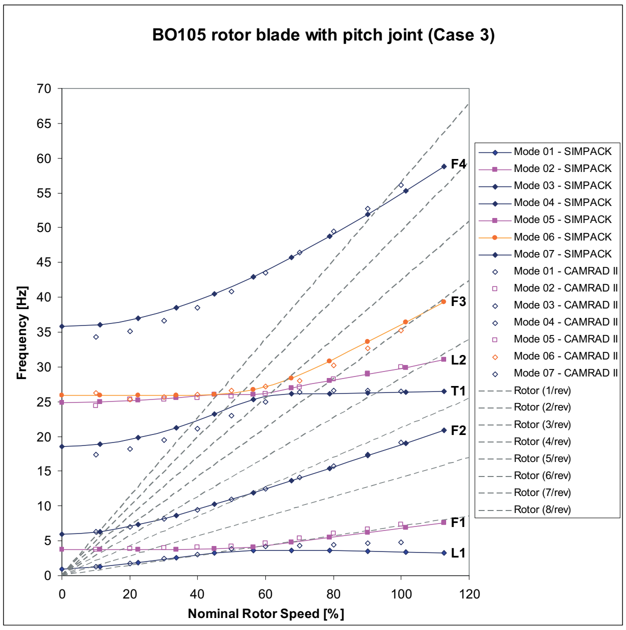

Figure 5: Fan diagram of BO105 rotor blade, Case 3

An additional fan diagram for Case 3 with mode tracking applied to the SIMPACK results is depicted in Figure 6. The used colour code is blue for flap bending modes, pink for lag bending modes and the torsion mode is plotted in orange colour.

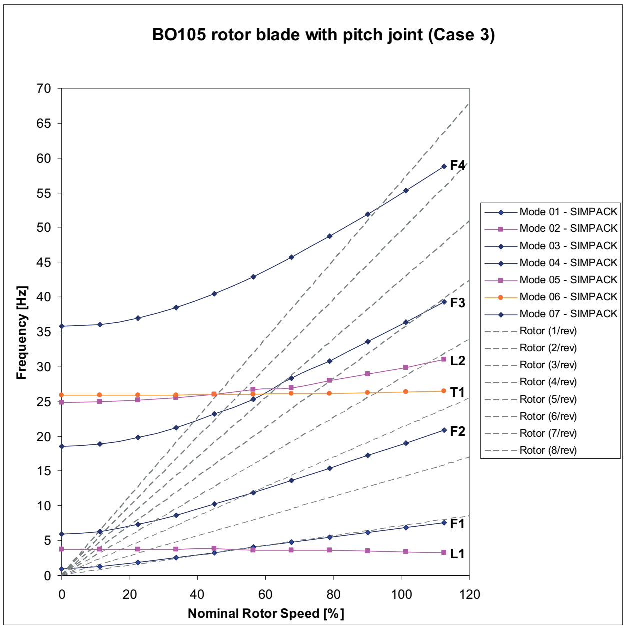

Figure 6: Mode tracked fan diagram of BO105 rotor blade, Case 3

3.5. Comparison of MBS results to CAMRAD II

Finally, the frequency results gained with MBS SIMPACK for Case 1 to 3 are compared to the CAMRAD II reference values at 100% nominal rotor speed. Table 5 specifies the relative frequency deviations for the investigated modes. According reference values from CAMRAD II for the non-rotating rotor blade are not available.

Mode	$\Delta f_{Case\ 1}$ [%]	$\Delta f_{Case\ 2}$ [%]	$\Delta f_{Case\ 3}$ [%]
1 / L1	-28.3	-28.8	-28.8
2 / F1	-5.8	-6.2	-6.2
3 / F2	-0.7	-1.5	-1.5
4 / T1	+4.9	+4.8	-0.4
5 / L2	-0.4	-0.6	-0.6
6 / F3	+4.1	+2.6	+2.6
7 / F4	+0.4	-2.1	-2.2
(F = flap mode / L = lag mode / T = torsion mode)			

Table 5: Relative frequency deviations at nominal rotor speed between SIMPACK and CAMRAD II

From a frequency point of view, all investigated BO105 blade models show in common a good agreement for modes F2, L2 and F4, an acceptable match with F1, whilst for the lag frequency L1 a bad agreement to CAMRAD II results is given. Further, Case 1 without beam offset definitions gives acceptable frequencies for torsion mode T1 and flap mode F3. The other two models with beam offsets included show both a good match for flap frequency F3. The result of Case 2 in terms of the frequency deviation to the CAMRAD II reference value for torsion is acceptable, but Case 3 including the pitch joint and related blade pitch control stiffness matches torsion T1 very well.

4. CONCLUSIONS AND OUTLOOK

A multibody approach with MBS SIMPACK has been used to model an isolated rotor blade. Beam models are available for BO105 and ERATO rotor for investigation, even though the focus is on BO105 with straight blade reference line to gain initial experience. The objective of the study presented in this work is to find an answer related to the question: Getting the mechanical coupling right for the flexible rotor blade with multibody dynamics?

The comparison of processed BO105 fan diagrams to the comprehensive helicopter tool CAMRAD II is encouraging. In general, the generic multibody system SIMPACK seems to be capable to simulate a rotating flexible rotor blade with respect to the consideration of geometric stiffness and gyroscopic terms. Elastic couplings being of interest for a rotating flexible rotor blade comprise bending-torsion, tension-bending and tension-torsion. Here, the coupling bending-torsion is driven by mass effects in the plane of the blade cross section due to the offset between shear center and the center of gravity, whilst tension-bending and tension-torsion are activated by centrifugal force. Nevertheless, the flexible representation of the rotor blade is always limited to the structural definition within the finite element code itself. For example, the used beam model based on CBEAM and PBEAM entries in MSC.NASTRAN cannot reproduce the mechanical coupling tension-torsion and further, the beam offset definitions are not allowed with the computation of differential stiffness for the derivation of the geometric stiffness terms and need to be neglected in the related static solution. The effect of tension-torsion coupling is usually small for a conventional rotor blade, but could be included within the flexible description of SIMPACK, if provided.

Future steps to improve input data from finite element codes for MBS SIMPACK focus two aspects:

- Use of MSC.NASTRAN non-linear solution (SOL106)
- Use of non-linear MSC.MARC beam element definition (not supported in actual FEMBS version, but in former version 8.5)

The comparison of natural frequencies with fan diagrams is a first step only to qualify the multibody system SIMPACK for the simulation of an elastic rotor blade model. BO105 mode shapes for nominal rotor speed are not compared to CAMRAD II reference results yet, but all information comprising the basic mode shapes from the non-rotating blade and their modal contributions and phase angles for the rotating blade is available.

5. ACKNOWLEDGEMENTS

The author thanks all partners from the French-German SHANEL consortium for their great support. Special gratitude is addressed to Oliver Dieterich from Eurocopter Deutschland GmbH, Didier Petot and Khiem van Truong, both ONERA for the provision of BO105 and ERATO related main rotor data and very fruitful discussions.

6. REFERENCES

[1] Costes M, Raddatz J, Borie S, Sudre L, d'Alascio A, Embacher M, Spiegel P. Advanced Rotorcraft Aeromechanic Studies in the French-German SHANEL Project. Proc 35th European Rotorcraft Forum, Hamburg, 2009.

[2] Truong VK. Dynamics studies of the ERATO blade, based on Finite Element Analysis. Proc 31st European Rotorcraft Forum, Florence, 2005.

[3] Johnson W. Technology drivers in the development of CAMRAD II. Proc AHS Aeromechanics Specialists Conference, San Francisco, 1994.
[4] Benoit B, Dequin AM, Kampa K, von Grünhagen W, Basset PM, Gimonet B. HOST, a General Helicopter Simulation Tool for Germany and France. Proc 56th AHS Annual Forum, Virginia Beach, 2000.
[5] Lugner P, Arnold M, Vaculin O (Eds). Vehicle System Dynamics (Special issue in memory Professor Willi Kortüm). Vol. 41, No. 5, 2004.
[6] www.simpack.de
[7] Schwertassek R, Wallrapp O. Dynamik flexibler Mehrkörpersysteme. Vieweg, 1999.
[8] Wallrapp O. Standardization of Flexible Body Modeling in Multibody System Codes, Part I: Definition of Standard Input Data. Mechanics of Structures and Machines, 22(3), pp. 283-304, 1994.