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USING MULTIBODY DYNAMICS FOR THE SIMULATION OF FLEXIBLE ROTOR BLADES – MODELLING OF AN INNOVATIVE BLADE LAYOUT BASED ON BEAM APPROACH

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ABSTRACT

The process of structural dynamic modelling of the flexible rotor blade in a multibody system is demonstrated for the ERATO rotor. 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] and follow-on activities with the partners Eurocopter, Eurocopter Deutschland, ONERA and DLR. Three structural dynamic models of beam type comprising the conventional BO105 and 7AD rotor blades with straight reference axes as well as the innovative blade layout of the ERATO rotor blade [2], [3] have been investigated with MBS SIMPACK. According simulation results not considering aerodynamics have been evaluated by the comparison of processed BO105 fan diagrams [4] against the comprehensive helicopter software CAMRAD II [5]. Lessons learned from the simulation of the BO105 rotor blade with the multibody system SIMPACK comprise the following aspects:

- Comparison of fan diagrams to CAMRAD II is encouraging.
- Rotating aspects of the flexible blade (gyroscopic terms, geometric stiffening) seem to be represented well.
- Major elastic couplings of the blade are defined in SID-format, but the flexible representation is limited to the finite element beam definition itself (i.e. MSC.NASTRAN 2007: missing mechanical coupling tension-torsion, beam offsets not allowed for derivation of geometric stiffness).
- Meanwhile, beam offsets are possible for the derivation of geometric stiffness contributions with MSC.NASTRAN 2008.

Focus of this work is on the comparison of simulation results gained for the NASTRAN finite element model with experimental modal data of the clamped ERATO blade from a vibration test and the generation of fan diagrams using multibody system SIMPACK.

2. FLEXIBLE ROTOR BLADE MODELS

2.1. Rotor blades modelled with FEM

Three finite element models are available for the BO105 rotor blade with 37 beam elements based on a CAMRAD II model, the 7AD rotor blade with 52 beam elements and the ERATO rotor blade with 58 beam elements, both derived from HOST [6] input data. The rotor blade definitions comprise blade geometry, radial distributions for stiffness and mass, blade modal damping, blade hinge positions with lag damper data, and pitch joint position with blade pitch control stiffness, if appropriate. All models show the following features with MSC.NASTRAN:

- Beam elements are placed along the quarter chord line representing the blade reference line.

- Beam entries CBEAM / PBEAM define continuous stiffness and mass distributions.
- Beam offsets relative to the quarter chord line define shear center, tension center and center of gravity.
- The blade hinge for lead-lag motion is modelled with a rotational spring element CELAS2 and damper element CDAMP2, whilst stiffness and damping is neglected for the flap hinge. A rigid RBE2 element constrains all remaining degrees-of-freedom. Here, the pitch hinge with control stiffness is not taken into account (7AD / ERATO rotor with blade hinges).
- 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 (BO105 hingeless rotor).
- The mass definition CONM2 is used to describe additional concentrated mass points.
- Rigidly connected nodes model trailing and leading edge positions.

The blade models utilize two congruent beam lines to define the stiffness and mass distribution, but different beam offsets as depicted in Figure 1. Beam offsets for shear center, tension center and center of gravity are major drivers for the blade dynamics. For example, a conventional rotor blade with straight reference axis shows, more or less, elastic modes in terms of pure flap, lag and torsion for the beam offsets neglected, whilst elastic modes with mixed contributions from flap, lag and torsion are obtained if beam offsets are taken into account.

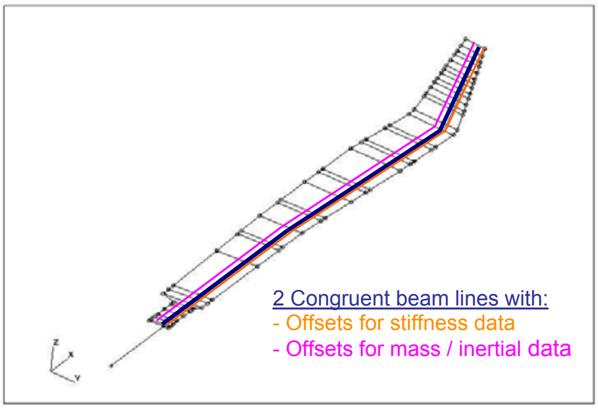

Figure 1: Beam definition using offsets

Figure 2 to Figure 4 illustrate the used beam models of the BO105, 7AD and ERATO rotor blades with radial locations of the pitch joint or the blade hinges. In the present work, MSC.NASTRAN version 2008 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.

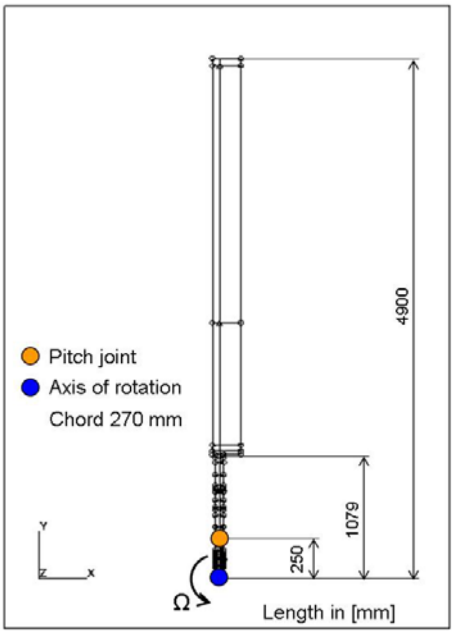

Figure 2: BO105 finite element model

The input data of the 7AD rotor blade shows an elastic axis which coincides with the quarter chord line. The quarter chord line remains straight up to a radial station of approximately 95 % from where it follows the swept-back blade tip. Model assumptions comprise the same radius of 75 mm for both blade hinges and an estimated radial distribution for tension stiffness. The distributed blade mass is 3.18 kg and the blade attachment is found at the radial station of 275 mm.

The acoustically optimized ERATO rotor is a collaborative design of ONERA and DLR resulting in a non-conventional blade shape with a swept beam axis. Like the 7AD rotor, ERATO was used in several test campaigns in the S1 Modane wind tunnel and the DNW-LLF [2]. Lessons learned during the refinement of the finite element models for the BO105 and 7AD rotor blades with straight blade axes have been transferred to the beam model of the ERATO rotor.

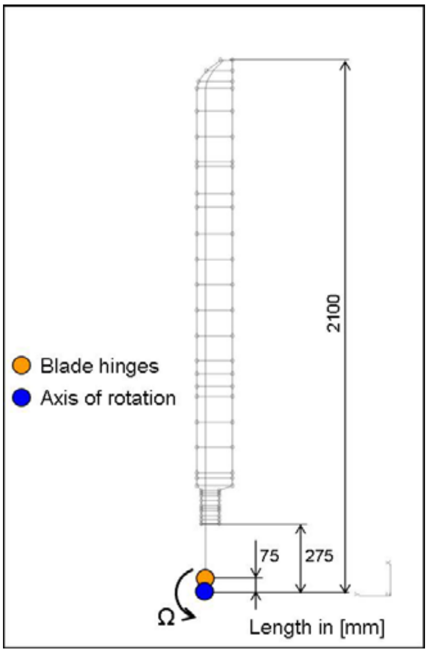

Figure 3: 7AD finite element model

Also the input data of the ERATO rotor blade shows an elastic axis which coincides with the quarter chord line. The quarter chord line remains straight up to a radial station of approximately 45 % changing into a slight forward sweep which is followed by a clear back sweep towards the blade tip. Again, the model assumptions comprise the same radius for the blade hinges and an estimated radial distribution for tension stiffness. The distributed blade mass is 3.55 kg and the blade attachment is found at the radial station of 275 mm.

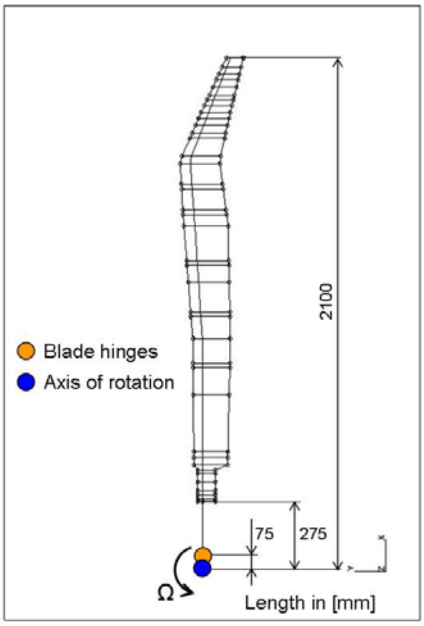

Figure 4: ERATO finite element model

2.2. Rotor blades modelled with MBS

The multibody system SIMPACK [7] 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, later out-sourced to INTEC for further development and commercial distribution, being recently renamed as SIMPACK AG [8]. 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 [7]-[10] 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

Feature (1) allows the straight-forward use of finite element models of industrial model size. Currently, the FEMBS interface [8] 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 (2) and (3) 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 ERATO rotor blade. Two FEM solutions from a pre-processing 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 $\Omega>0$ 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 [10] allowing the consideration of all major elastic couplings, if provided by the finite element model. 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 sub-structuring, 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.

Modal data comprising natural frequencies and mode shapes might be extracted for the rotating blade. Here, a resulting mode shape is described in terms of the shapes from the non-rotating blade with related modal contribution factors and phase angles.

2.3. Comparison to measured vibration data

Available ground vibration test (GVT) results for the clamped blade comprise modal properties in terms of natural frequencies and mode shapes for the non-rotating ERATO blade. The GVT setup is depicted in Figure 5. A rowing hammer technique allowing a minimum number of three accelerometers was applied to excite of the rotor blade.

The comparison of simulation results gained for the NASTRAN finite element model with experimental modal data of the clamped ERATO blade takes the measured frequencies into account. The obtained natural frequencies are given in Table 1 in terms of a relative frequency deviation with the GVT results as reference values. Here, the analytical values of Case 1 do not consider beam offsets for the shear center, tension center and center of gravity, whilst Case 2 does.

Mode	f_{GVT} [-]	$\Delta f_{Case\ 1}$ [%]	$\Delta f_{Case\ 2}$ [%]
1 / F1	1.00	-2.4	-2.2
2 / L1	1.00	+1.9	+2.2
3 / F2	1.00	-1.0	-1.0
4 / F3	1.00	-4.0	-4.4
5 / T1	1.00	+7.0	+7.8
6 / F4	1.00	-5.3	-6.4
7 / L2	1.00	+6.4	+7.0
8 / F5	1.00	+2.6	+2.9
9 / F6	1.00	-4.2	-5.5
10 / T2	1.00	+8.1	+6.7
(F = flap mode / L = lag mode / T = torsion mode)			

Table 1: Natural frequencies of the clamped blade

The resulting frequencies from the finite element model without beam offsets are found slightly smaller (except F5) for flap, slightly larger for lag and approximately 8 % larger for both torsion modes in comparison to the experimental vibration data. Frequencies obtained for the finite element model with beam offsets show the same characteristics, partially with a small increase in frequency deviation. In general, the improvement of the initial beam model is possible by the adjustment of global stiffness parameters like modulus of elasticity and shear or direct re-modelling of beam properties.

Analytical mode shapes of the blade have not been compared to the experimental shapes from the vibration test yet.

Figure 5: GVT setup for clamped ERATO blade

3. FLEXIBLE ROTOR BLADE MODELS IN THE ROTATING FRAME

3.1. ERATO fan diagrams processed with MBS

The processing of the fan diagram for the rotating flexible ERATO blade is performed with MBS SIMPACK. In general, the diagram illustrates the dependency of the mode related natural frequencies of an isolated rotor 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. All rotor harmonics are related to the rotor speed (1/rev).

The fan diagram is evaluated for 10 modal solutions which are calculated in steps of 10 % of the nominal rotor speed up to a maximum rotor speed of 120 %. 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.

Three different cases of the ERATO 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 blade hinges for lead-lag and flap motion at the radial station of 75 mm. They are listed in Table 2.

Rotor Blade	Beam Offsets	Blade Hinges
Case 1	no	no
Case 2	yes	no
Case 3	Yes	yes

Table 2: MBS models of the ERATO rotor blade

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 and 3, 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. 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.

3.2. Rotor blade without beam offsets

Case 1-straight (see Figure 6 and Table 3): Rotor blade without beam offsets and without hinges (mode animation shows well separated shapes in terms of flap, lag and torsion); mode 7 and 8 change order; T1 and T2 nearly constant because of “artificially” straight blade axis

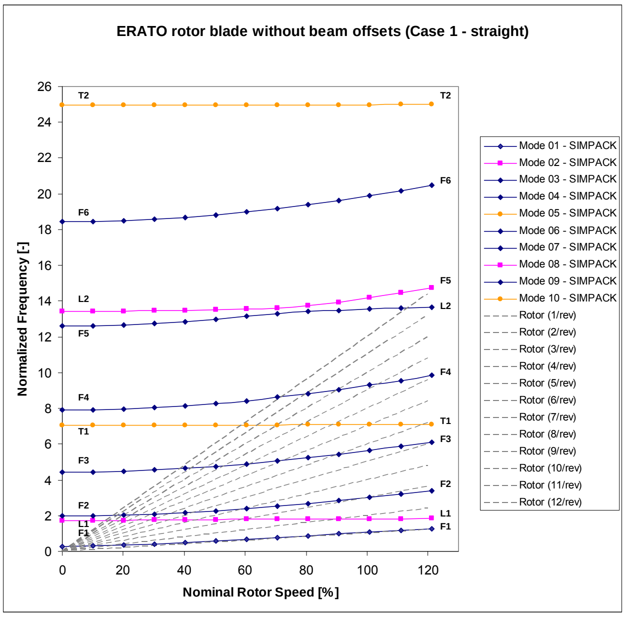

Figure 6: Fan diagram ERATO, Case 1-straight

Mode No.	0% Rotor Speed	100% Rotor Speed
1	F1 / 0.29	F1 / 1.07
2	L1 / 1.71	L1 / 1.82
3	F2 / 1.99	F2 / 3.02
4	F3 / 4.43	F3 / 5.65
5	T1 / 7.07	T1 / 7.08
6	F4 / 7.90	F4 / 9.30
7	F5 / 12.62	L2 / 13.55
8	L2 / 13.44	F5 / 14.18
9	F6 / 18.43	F6 / 19.89
10	T2 / 24.96	T2 / 24.98
(F = flap mode / L = lag mode / T = torsion mode)		

Table 3: MBS mode shapes and natural frequencies for the ERATO blade model, Case 1-straight

Case 1 (see Figure 7 and Table 4): Rotor blade without beam offsets and without hinges (mode animation shows shapes in terms of flap, lag with torsion contributions); mode 7 and 8 change order; T1 and T2 not constant and increasing in frequency

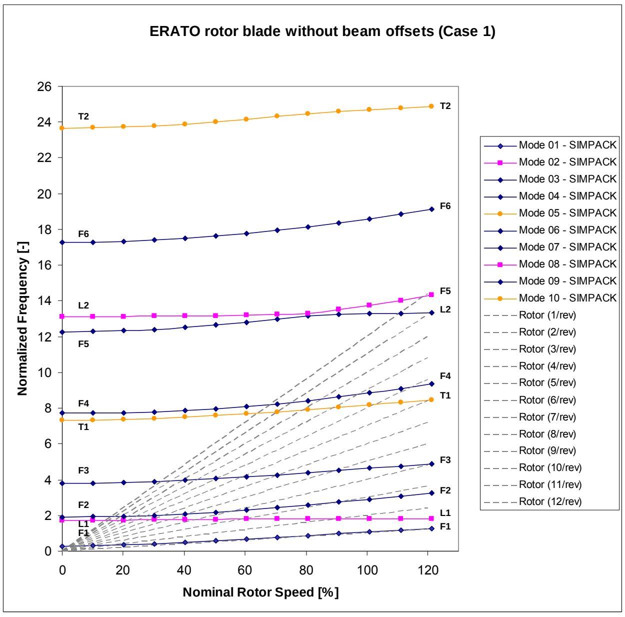

Figure 7: Fan diagram ERATO, Case 1

Mode No.	0% Rotor Speed	100% Rotor Speed
1	F1 / 0.28	F1 / 1.07
2	L1 / 1.72	L1 / 1.81
3	F2 / 1.91	F2 / 2.90
4	F3 / 3.80	F3 / 4.64
5	T1 / 7.31	T1 / 8.16
6	F4 / 7.71	F4 / 8.85
7	F5 / 12.27	L2 / 13.27
8	L2 / 13.12	F5 / 13.76
9	F6 / 17.27	F6 / 18.60
10	T2 / 23.66	T2 / 24.71
(F = flap mode / L = lag mode / T = torsion mode)		

Table 4: MBS mode shapes and natural frequencies for the ERATO blade model, Case 1

3.3. Rotor blade with beam offsets

Case 2 (see Figure 8 and Table 5): Rotor blade with beam offsets and without hinges (especially higher modes show bending and torsion combined, mass effect is dominant); mode 7 and 8 change order; T1 and T2 not constant and increasing in frequency

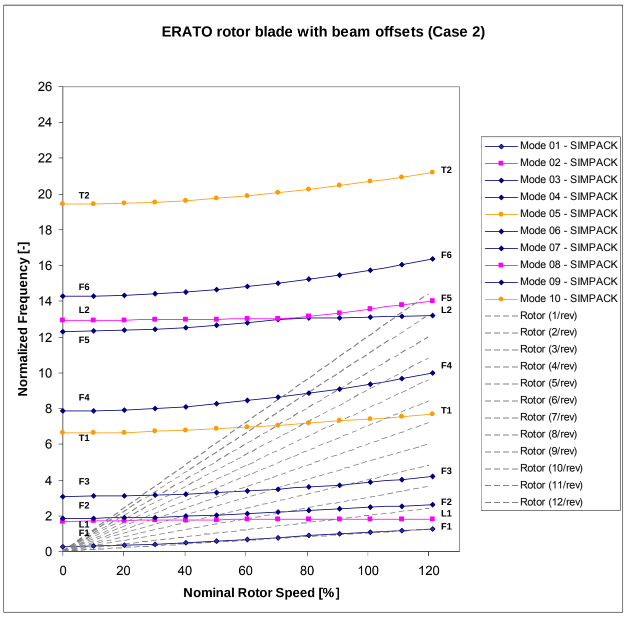

Figure 8: Fan diagram ERATO, Case 2

Mode No.	0% Rotor Speed	100% Rotor Speed
1	F1 / 0.29	F1 / 1.08
2	L1 / 1.69	L1 / 1.82
3	F2 / 1.87	F2 / 2.47
4	F3 / 3.10	F3 / 3.88
5	T1 / 6.63	T1 / 7.43
6	F4 / 7.85	F4 / 9.38
7	F5 / 12.32	L2 / 13.12
8	L2 / 12.94	F5 / 13.56
9	F6 / 14.27	F6 / 15.74
10	T2 / 19.43	T2 / 20.69
(F = flap mode / L = lag mode / T = torsion mode)		

Table 5: MBS mode shapes and natural frequencies for the ERATO blade model, Case 2

3.4. Rotor blade with beam offsets and hinges

Case 3 (see Figure 9 and Table 6): Rotor blade with beam offsets and with lag / flap hinges (modes show bending and torsion combined, mass effect is dominant); modes order remains unchanged; T1 and T2 not constant and increasing in frequency

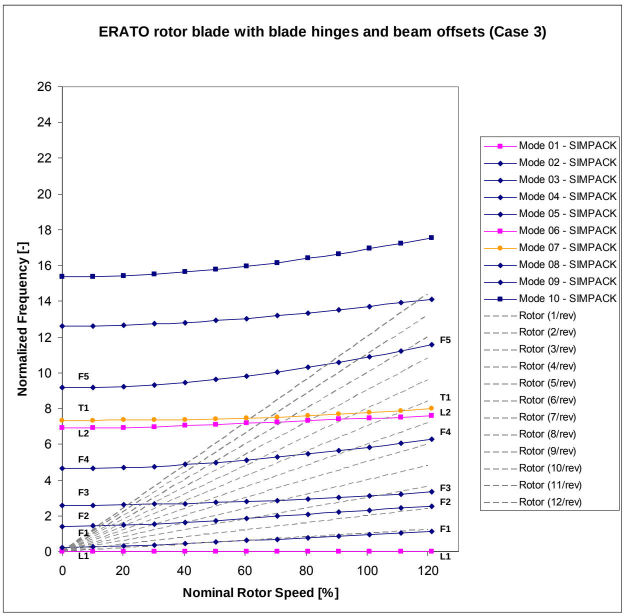

Figure 9: Fan diagram ERATO, Case 3

Mode No.	0% Rotor Speed	100% Rotor Speed
1	L1 / 0.00	L1 / 0.00
2	F1 / 0.24	F1 / 0.95
3	F2 / 1.42	F2 / 2.32
4	F3 / 2.59	F3 / 3.12
5	F4 / 4.65	F4 / 5.85
6	L2 / 6.91	L2 / 7.46
7	T1 / 7.34	T1 / 7.77
8	F5 / 9.17	F5 / 10.90
9	- / 12.62	- / 13.71
10	- / 15.36	- / 16.93
(F = flap mode / L = lag mode / T = torsion mode)		

Table 6: MBS mode shapes and natural frequencies for the ERATO blade model, Case 3

4. CONCLUSIONS AND OUTLOOK

A multibody approach with MBS SIMPACK has been used to model several isolated rotor blades. Beam models are available for the BO105, 7AD and ERATO blades for investigation, even though the focus is on ERATO with swept blade reference line.

The structural dynamic model of the ERATO rotor blade was translated from the comprehensive rotorcraft code HOST into a NASTRAN beam model using standard finite element beam definitions. Resulting natural frequencies are compared with experimental modal data of the clamped ERATO blade from a vibration test showing slightly smaller values for flap, slightly larger frequencies for lag and also larger ones for both torsion modes. Probably, the improvement of the initial beam model is possible by the adjustment of global stiffness parameters like modulus of elasticity and shear or selected beam property entries. The multibody approach is then used to model the isolated rotor blade for rotating condition based on the modal data extracted from the NASTRAN model. A study comprising three different cases taking beam offsets and blade hinges into account shows the influence of shear center, tension center and center of gravity on the resulting fan diagrams of the ERATO blade.

After the refinement of the analysis model, a comparison of the fan diagram taking the beam offsets and blade hinges into account might be possible against published results from ONERA [3]. Also the analytical mode shapes of the clamped blade will be correlated with the available vibration test results.

5. ACKNOWLEDGEMENTS

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