Virtual Commissioning of Large Machines with COMSOL

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Abstract: Despite well-advanced functions built-into the COMSOL engine, there is the need to integrate dynamics properties of rotating components both measured and pre-computed. An example is given on how to integrate the fluid-film characteristics derived with the specialized software ALP3T and take full advantage of COMSOL post-processing and graphics to present the results to customers using the Cloud.

COMSOL simulations of rotor-bearing systems are benchmarked for steady-state operations, transients and beyond. Stability thresholds are determined by solving the complex eigenvalues. However, for post-stability behavior non-linear fluid-film interactions need to be reconstructed from the pre-computed look-up tables.

Final demonstration of the concept of virtual commissioning is given through two industrial applications involving motor-compressor (MC) and turbo-generator (TG). In both cases, the drivelines are representing multi-supported rotor-bearing systems. Thus, there is a strong demand during installation or commissioning to account for alignment changes and adjustments of the couplings and supports stiffness's.

Keywords: Rotating machinery, rotor-dynamics, critical speeds, instability, journal bearings.

1. Introduction

ABB offers a comprehensive range of reliable, highly efficient motors and generators. All products are supported by service and expertise to save energy and improve customers' processes over the total life cycle. One of the ideas behind the professional installation and commissioning service is to get it right from the very start. Thus, the concept of virtual commissioning of large machines and drive-lines is attractive both to OEMs and to the customers. COMSOL Server provides a platform for creating and deploying applications and to back-up commissioning engineers by the expertise of the design team, making the installation process much faster and smoother. By distributing the models and computational capability to the entire team it is possible to adjust remotely the specific parameters or variables by running a dedicated simulation package with instant feed-back from the field. The approach transforms the workflow and becomes accepted and commonly appreciated by the engineering teams and business managers. But, to gain trust the service has to guarantee reliable, well-proven solutions.

Rotating machinery dynamics is an interesting combination of multi-physics, utilizing first principles of all mechanical engineering fundamental areas, solid dynamics, fluid-dynamics, electro-magnetism and heat transfer as well as the control. It seems to be quite natural to test multi-physics software, such as COMSOL, its limits and potentials in the rotor-dynamics applications.

The challenge is then taken to benchmark the COMSOL rotor-dynamics solutions, compare the outcomes against other commercial simulations of rotating components, fluid-film interaction and heat transfer.

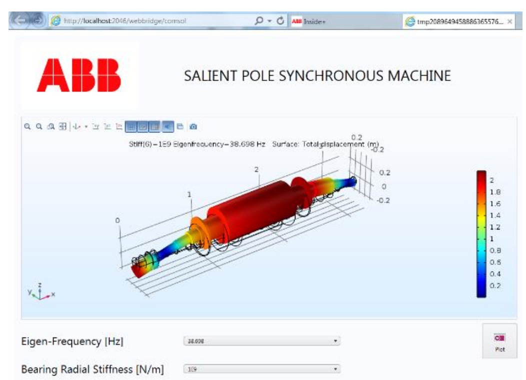


Figure 1. WEB-Based application assessing the critical speeds in terms of the bearing stiffness.

Figure 1 demonstrates an example of web-deployed application for determining the critical speeds of the rotor in terms of bearing stiffness.

2. Rotating Components

At the start, the focus was set on modelling fundamental rotating components such as shafts, couplings, bearings and supports. It is the tradition in rotor-dynamics to test a flexural rotor with so called Jeffcott rotor, consisting of a point mass attached to a massless shaft. The method provides clear understanding of dynamics, behavior of the components as well as the assessment of properties and characteristics before assembling a complex rotating structure.

2.1 Shaft

Several authors compared accuracy of COMSOL shaft modelling against ANSYS and ABACUS. It can be concluded that the satisfactory accuracy is reached within the frequency range of several multiples of nominal speed providing the use of sufficient number of elements (i.e. proper mesh).

Table 1 compares the critical frequencies of a uniform rotor subjected to different boundary conditions. The reference values, expressed through the coefficient-A [1], were derived analytically and normalized by the beam geometry and material properties. The boundary conditions were defined in terms of the fixed constrains or predefined displacements.

Fixed Free	3.51	21.3	57.0	105.1
	3.52	22.0	61.7	121.0
Fixed Fixed	21.6	56.2	103.5	159.6
	22.4	61.7	121.0	200.0
Fixed Hinged	15.1	46.8	92.2	147.7
	15.4	50.0	104.0	178.0
Free Free	21.8	57.6	107.0	-
	22.4	61.7	121.0	200.0

Table 1. Comparison COMSOL normalized frequencies against the reference A-coefficients [1] shown in the high-lighted field

2.2 Couplings

Couplings are used to join the rotors and reduce the sensitivity to any eccentricities and angular misalignments of the centerlines. Three general types of couplings are traditionally distinguished:

- Rigid coupling: often a spline connection locked tight on the shaft diameters;
- Flexible coupling: a splined joint with crowned teeth and crowned tip diameter allowing for large axial motion and modest amount of angular misalignment;
- Distance piece with a joint on each side to accommodate wide range of misalignments.

Coupling data provided by OEMS often includes:

- Torsional stiffness;
- Masses on each end with the center of gravity;
- Bending stiffness in case of transmission of large power;
- Damping ratios in terms of quality factors in case of a rubber coupling.

A benchmark test was performed for four rigid masses (disks) coupled by the springs together. The equations of motion were derived analytically to form the mass and stiffness matrices as shown in Figure 2.

$$M = \begin{bmatrix} m & 0 & 0 & 0 \\ 0 & m & 0 & 0 \\ 0 & 0 & m & 0 \\ 0 & 0 & 0 & m \end{bmatrix}$$

$$K = \begin{bmatrix} 2*k & -k & 0 & 0 \\ -k & 3*k & -2*k & 0 \\ 0 & -2*k & 3*k & -k \\ 0 & 0 & -k & k \end{bmatrix}$$

Figure 2. Mass and stiffness matrices of the multi-mass system flexibly coupled

Analytical MATLAB© vibrating modes and corresponding eigenvalues have matched perfectly the COMSOL simulations as shown in Table 2.

	Frequency [Hz]			
COMSOL	1.4748	4.4261	6.2872	9.3065
MATLAB	1.4750	4.4266	6.2879	9.3076

Table 2. Comparison of resonating frequencies for m=0.1541[kg], k=100 [N/m]

2.3 Rolling Element Bearings

In general rolling element bearing appear to be speed dependent. Due to centrifugal forces and gyroscopic effects of the elements, the contact angles changes are generating non-linear relations between forces and displacements.
Jeffcott rotor supported by non-linear bearings was used to benchmark a rotor unbalance response. A small (0.01%) third order stiffness term was combined with the bearing linear stiffness

characteristics. Figure 3 shows the parametric frequency sweep around the critical speed.

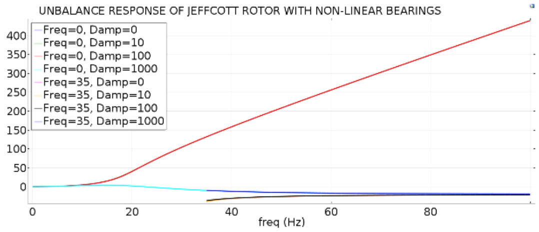


Figure 3. Unbalance response of Jeffcott rotor with non-linear bearings.

It was interesting to note that by starting the calculations of the unbalance responses for rotational speed bellow and above the (linear) resonance two different solutions were found.

This, however complies with the theoretical solutions indicating existence of two steady state oscillations with the possibility of amplitudes jump between them.

Additional parametric sweep of damping property showed that the solutions were converging to each other with sufficient amount of dissipation (modal damping) introduced.

In most of industrial cases, it is sufficient, just to rely on the force-displacement relations linearized around equilibrium position. Dynamic behavior is then described by the 5x5 speed dependent stiffness matrix. Then, if justified, the additional 5x5 speed-related damping matrix can be introduced to reflect e.g. the interaction of squeeze dampers or shims submerged in oil.

Further references to rolling element bearings are given through the website: www.mesys.ch. The MYSYS software has a comprehensive Dbase covering wide range of different bearing types, geometry and boundary conditions. It is possible to export bearings properties through the XML files.

Foundation springs available in COMSOL are suitable to model the rolling element bearings attached directly to the ground. However, if this is not the case, as met on rotors connected to flexible housing or to another rotor e.g. shaft in shaft arrangements, very different approach is required. COMSOL is actually offering a nice solution (discussed later on) of defining the reaction forces at the interfaces in terms of dependent variables.

2.4 Fluid-Film Interaction

Large process machines often rely on fluid-film bearings to support the rotating or reciprocating elements. Although seals, process-fluids or electro-magnetic fields coexist, the oil-film bearings provide the majority of the stiffness and damping for moving assembly.

Bearings oil-film characteristics are available through several publications or can be computed with the specialized programs. One of them is the software package ALP3T developed within the jointed FVV and FVA project. It handles lubrication with cavitation, double-phase flow and thermal effects. It accounts for turbulence occurring at high speed, affecting heat conduction, power loss and stability reserve [2].

The algorithm has proven itself in number of industrial applications and for that reason it was integrated into the applications to demonstrate the concept of virtual commissioning.

Unfortunately, ALP3T software is not open to the public, it requires the license [3]. There are alternatives, one of them is the JSME’s Dbase covering more than 50 bearing types [4].

Anisotropic stiffness and damping characteristics of fluid-film represent the bearing behavior at steady-state equilibrium established under a static load such as gravity. The journal is however pushed away from its equilibrium by dynamic forces, such as rotor unbalance causing it to whirl on an elliptical orbit. To justify the use of linearized bearing properties the orbit size needs to be relatively small in comparison to the bearing radial clearance. The linearized coefficients are then used to predict the stability thresholds or the bifurcation point beyond which the journal orbits nearly filling the entire bearing clearance and forming the limit cycles [5], [6].

Stability analysis of a Jeffcott rotor supported by the fluid film bearings was used again for next benchmarking. Figure 4 shows the stiffness and damping characteristics of 4-lobe bearing.

Stability threshold derived with MATLAB© has matched the rotational speed for which the COMSOL model anticipated the vanishing of real part of the eigenvalue.

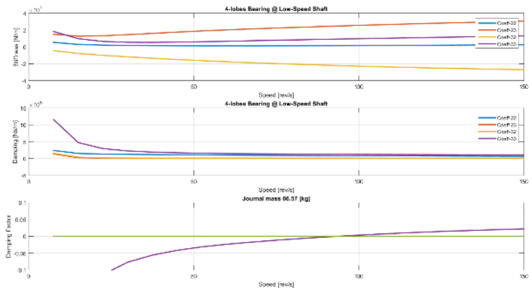


Figure 4. Journal bearing characteristics and stability map of Jeffcott rotor in terms of rotational speed.

Predicted stability threshold was actually validated experimentally. Figure 5 shows a spectrogram of the journal displacement during the run-up and cast-down (note a gear ratio of 1:4). Oil whirl phenomenon occurred at the rotational speed of 390 [rev/s] measured on high speed shaft (HSS) which corresponded to the speed of 98 [rev/s] on low speed shaft (LSS).

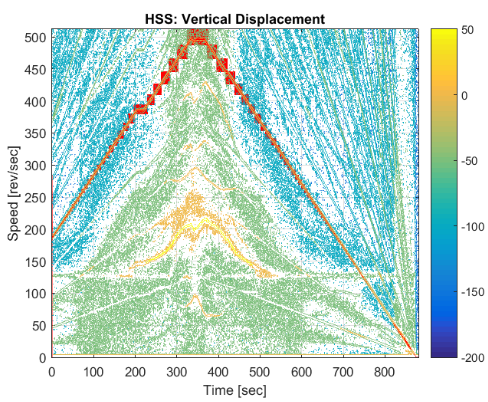


Figure 5. Experimental spectrogram of the journal displacements during the run-up and cast-down

3. Virtual Commissioning

COMSOL provides unique feature of a set of built-in modules (library) of partial differential equations (PDE) to be either arbitrarily connected to each other or to user defined one. Couplings are formulated and linked via flexible mathematical models by defining at the interfaces:

- Neumann conditions: specifying the surroundings effects and interactions. Those are often expressed in terms of forces, flux or current;

- Dirichlet conditions: specifying the constraints resulting from the subsystems interaction at boundaries.

Both types are related as every flux condition results in some unique values of the dependent variables, and every constraint requires unique flux to enforce the expected values.

The following examples are demonstrating the use of boundary conditions and discussing some benefits of user defined couplings.

3.1 Machine

3-D finite element model of a salient pole synchronous rotor is shown in Figure 6. The shaft is supported by two oil-film bearings positioned at the high-lighted planes (blue areas).

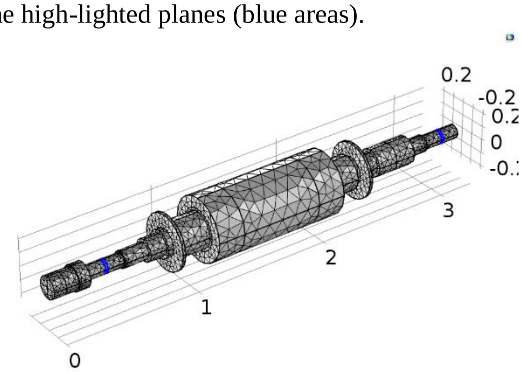


Figure 6. Rotor of synchronous motor with marked rigid connectors

Rotor-bearing interactions are expressed via the Neumann boundary conditions, set in terms of the force reactions acting along the rigid connectors.

Each of the hydrodynamic forces depends on the relative displacements and velocities predefined at the respective node. Those are accessible through the dependent variables:

- comp1.solid.uvw_rig1
- comp1.solid.uvw_rig2

Velocities in the planes need to be differentiated first at the respective planes:

- d(comp1.solid.uvw_rig1,TIME)
- d(comp1.solid.uvw_rig2,TIME)

Naturally, if the constraints are given in the form of predefined displacements or velocities, the Dirichlet conditions should be applied.

COMSOL allows for using not just fixed values but also for the expressions. Additionally software offers a set of pre-defined connectors to simplify model description, when applicable.

Journal bearing stiffness and damping properties, as it was explained earlier, are anisotropic and speed-dependent. Their characteristics are modelled then through the look-up table. Using the MATLAB code the stiffness and damping matrices were imported to define the piecewise functions, all being synchronized by the rotation speed, also given in a piecewise-function form.

By using the Eigen-frequency study with the sweep over the rotational speed the complex modal shapes, resonating frequencies and modal damping were calculated. Those are presented on the Campbell diagrams. Figure 7 shows the critical frequencies in terms of the rotational speed index.

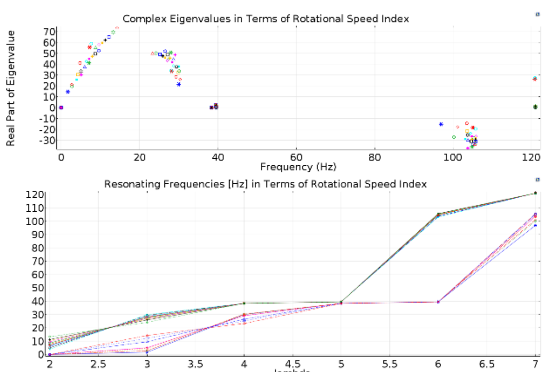


Figure 7. Complex Eigenvalues in terms of rotor speed index.

3.2 Motor-Compressor (MC)

Next, a synchronous electric motor was coupled to the compressor unit. Both units were mounted on two separate, concreate blocks, each being supported by set of flexible springs. The resulting assembly model is shown in Figure 8.

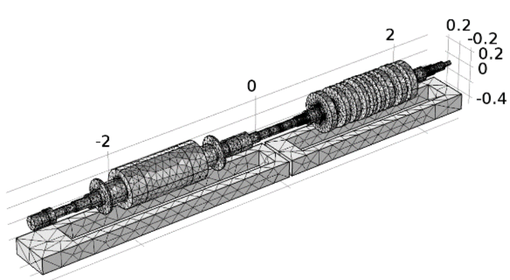


Figure 8. Motor-Compressor unit on flexible foundation

Solving the eigenvalue problem both resonating modes and corresponding frequencies were found to indicate if any structure modifications are required. The parameter sweep analysis was used to maximize the safety margins and separate the rotation speed and its multiples from the resonances. Figure 9, 10 and 11 are showing bending and torsional modes of the MC unit, respectively.

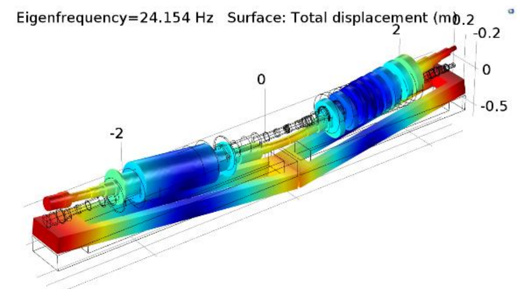


Figure 9.First bending mode @ 24.2 Hz

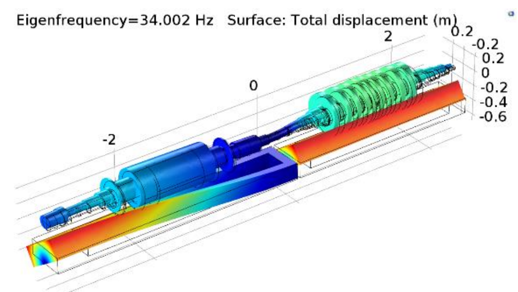


Figure 10. First foundation torsional mode @ 34.0 Hz

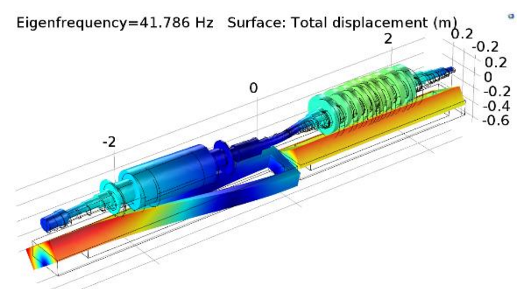


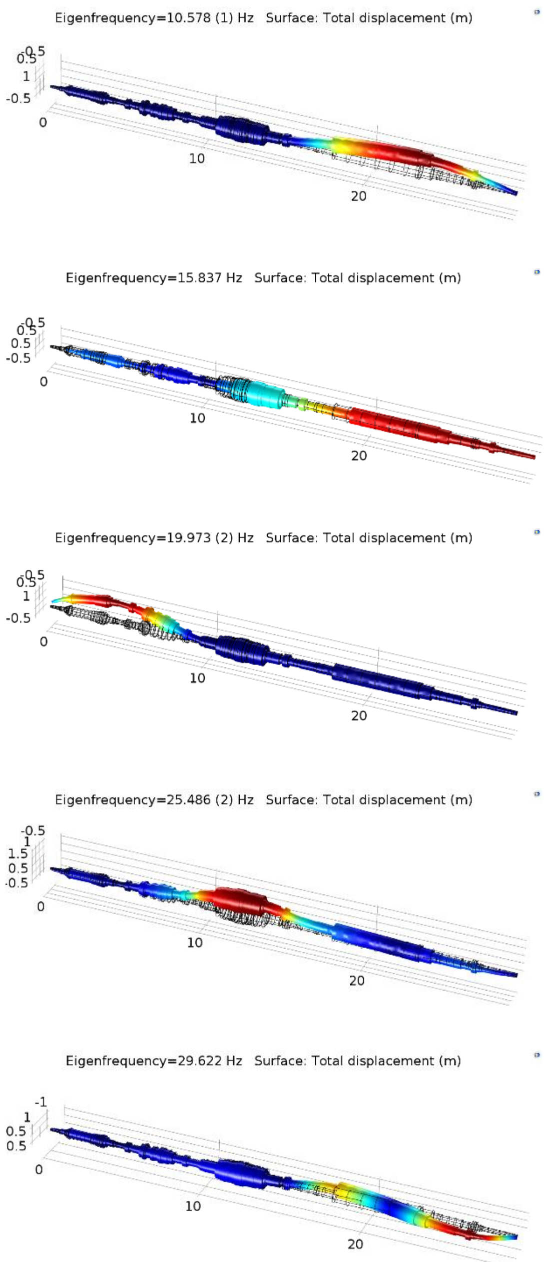
Figure 11. Second torsional mode @ 41.7 Hz

Using the server, it was very convenient to view the modes in the web-browser. This kind of tool was seamlessly generated by developing the model in the Application Builder and publishing it on the COMSOL server.

3.2 Turbo-Generator (TG)

As the final benchmark the operational deflection shape (ODS) analysis of the turbo-generator (TG) was performed. The system consisted of HP, IP and LP turbines coupled with a generator and exciter.

The rotor was supported by six journal bearings pre-loaded by the gravity and pre-defined turbines alignment.



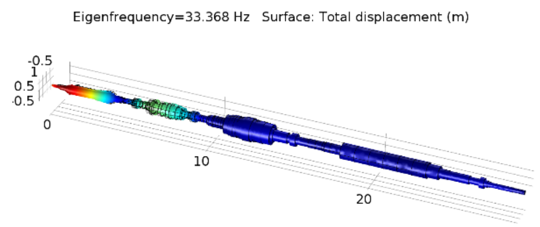


Figure 12. Turbo-Generator resonating modes

Due to the rigid couplings between the turbine stages the system was considered statically in-determined. Any change in one of the bearing has immediately impact on the loads in other bearings.

Re-distribution of loads i.e. the bearing reactions was handled by the stationary module in COMSOL. After updating the Sommerfeld numbers, new equilibria positions and respective stiffness and damping coefficients were found to re-solve the complex eigenvalue problem.

A rotor balancing and support modifications are often tasks met during the commissioning. It is advantageous to know the effects of modifications and the sensitivity of each of the correction planes.

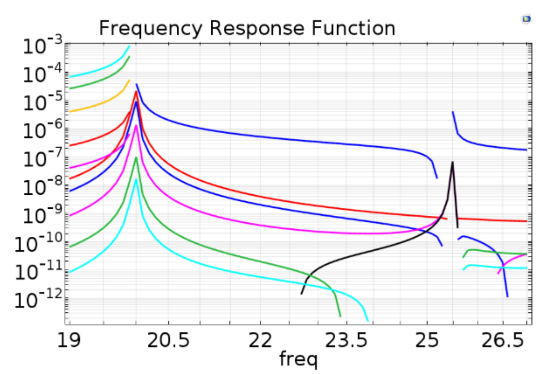


Figure 13. Turbo-Generator vector of the FRF matrix

Figure 13 shows the frequency response functions (FRF) derived through the Frequency Domain Study. The characteristics were produced by applying the force at the predefined correction planes and measuring the response at the observation planes.

Similar approach was also used to estimate the unbalance response. However, the difference was that instead of a fixed force vector, the force vector proportional to the angular velocity in square was applied. Figure 14 shows the operational deflection shape (ODS) due to unbalance introduced on the HP turbine.

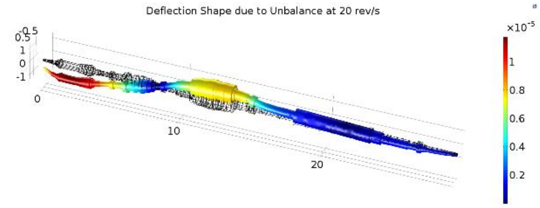


Figure 14. Turbo-Generator operational deflection shape due to unbalance at 20 rev/s

4. Conclusions and recommendations

- Application Builder combined with the COMSOL Server appears to be the right way of sharing the models and experience across the product development chain;

- Despite the presence of CFD module and functions allowing for modelling oil-film, it is still recommended to integrate the bearings characteristics, either measured or derived by third-party algorithms [2] [7] or apply oil-film force interpolation [6]. The reason is not the lack of computational power but the time needed to develop the solutions complying with the industrial standards.

- Modal reduced order, modal and time dependent model are crucial in rotor-dynamics to size-down the equations and separate non-linear parts, foundation interactions, etc. There are still few benchmarks to go as the initial tests showed some divergences.

- AC/DC, thermo-dynamics, acoustics, etc. modules are very tempting to be combined in large-machine simulations. A dedicated multi-physics package for rotor-dynamics would be more than welcome.

5. References

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