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Test rig model development and validation for the diagnosis of rolling element bearings

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Abstract **– In the context of condition based maintenance, carrying out diagnosis and prognosis processes is a key. For that purpose the evaluation of the condition of a machine is necessary, for which the development of physical models is useful as the response of the modelled system can be obtained in different operating conditions. In this paper, an electromechanical model for a rotary machine is presented, making special emphasis on the modelling of rolling element bearings. Thus, the response to different damaged conditions is evaluated. The proposed model is validated by comparing the simulation results with experimental signals acquired by tests carried out at different operating conditions. This comparison shows a good agreement as differences less than 0.6 % for the model of the bearing and differences up to the 10 % for the modelling of the rest of the elements are obtained.**

I. INTRODUCTION

Machinery healthy management implies the appropriate decision making about maintenance actions based on diagnosis and prognosis information, available resources and operational demand. In order to make those decisions, the evaluation of the condition of machinery as well as the interpretation of the indicators obtained from this evaluation is required. At the time of performing this interpretation, generally a model is needed.

In this paper the spotlight has been put on rotary machinery. For this kind of machinery physical modelling is useful because its ability to describe the response of time-varying systems in which different operating and different faulty conditions must be taken into account.

When it comes to developing models for rotary machinery, models of rolling element bearings (REBs) are the most common ones due to the fact that they are key parts of this kind of systems. These models have been developed taking into account different levels of complexity and have been validated by means of different techniques, such as the comparison of the velocity amplitudes for an inner race defect [1], checking the rotational speed of the cage [2], the comparison between the analytical and simulated values of the static load distribution [3], or the use of time domain indicators [4]. Anyway, the most common strategy to validate a model consists in obtaining an agreement between the simulation results and the experimental results regarding the characteristic fault frequencies [5]. Models with higher complexity usually represent other components of a rotary system. For example, Zhang et al. [6] presented a rotor-ball bearing system model under elastohydrodynamic lubrication. A more complex model is presented by Sawalhi and Randall [7], who modelled the interactions between gears and bearings.

Whereas many of the cited papers present the modelling process of an only or few components, this paper has as aim the development and the validation of a model for a whole rotary machine, taking into account the different components in which it consists of, and putting an special emphasis on the dynamics of REBs. Both mechanical and electrical issues are considered in the modelling process, as well as the presence of damage in one of the REBs of the machine. A number of tests have been carried out and features from the measurement taken in these tests are extracted. The comparison between these features and those obtained from the simulations of the physics-based model gives a representative value of the accuracy of the model for the validation process.

II. ROTARY MACHINE TO BE MODELLED

A commercial test rig manufactured by SpectraQuest Inc. called Gearbox Prognostics Simulator is used as the rotary machine to be modelled, which is shown in Fig. 1. It consists of an induction motor that drives an axis. Then, there is a gearbox which is going to be monitored, specifically one bearing of the intermediate shaft. It is a ball bearing REXNORD ER16K. After the monitored gearbox there is another gearbox and a motor that applies some load. The control is the responsible of assuring that the speed of the drive motor (DM) and the load applied by the load motor (LM) are the desired ones.

A number of sensors are placed in different points of the machine. They provide the following measures with sampling speeds of 10240 Hz and 50000 Hz: the current of the DM and the LM, the voltage of the LM, the speed and the angular position of the DM's rotor and the vibration of the surroundings of the monitored bearing.

As the aim of the modelling comes from the need of getting knowledge of the state of a system, different damaged scenarios have been considered in this study. For that purpose, damages of six different sizes have been seeded to the outer ring of six bearings, each bearing having an only damage. Damage has been produced by a drilling process using the following damage diameters: 0.4, 0.6, 0.8, 1, 1.5 and 2 mm. An extra bearing in healthy state has been considered for comparison.

Fig. 1. Test rig used for the experiments

III. MODEL OF THE ROTARY MACHINE

With the aim of developing a physics-based model of the rotary machine described in the previous section a multi-body approach has been employed. The models of the different elements of the rotary machine have been firstly developed and after that they have been joined together. Both mechanical and electrical issues are considered in the modelling process. Modelica® object-oriented modelling language has been used in this process due to its ability for the modelling and simulation of multi-domain physical systems.

A. Mechanical model

In the test rig model a detailed model for REBs developed by Leturiondo et al. [8] is used. Following a multi-body approach, it takes advantage of the reusability of models to cover a wide range of bearing configurations, as well as to generalize the dimensioning of the bearing, the application of the operating conditions and the definition of different damaged conditions.

Regarding the rest of mechanical components, the LossyGear model and the Inertia model implemented in the library of Mechanics of Modelica® are used for the modelling of the gearboxes and the axes, respectively. The former does not only take into account the speed rate and the consequently torque rate, but also the efficiency of the transmission, whereas the latter provides the inertia of the axis. As there are other mechanical elements that imply torque losses, such as the bearings or the couplings, a model for estimating these torque losses is constructed as a polynomial surface function of the operating conditions.

B. Electrical model

For the modelling of the electrical motor, the AIM_SquirrelCage model of the Machines library of Modelica® is used, which represents the behaviour of an asynchronous induction machine with squirrel cage [9]. It is based on the dynamic equivalent circuit of the induction motor, whose equation relating voltages v and currents i as well as the equation relating flux linkages λ and currents i and the one relating the torque with the rest of variables are given by Rashid [10]. It should be highlighted that these equations can be simplified as the rotor voltages in the q and d axes v_{qr} and v_{dr} are both equal to 0.

Regarding the control of the speed of the machine, there are many solutions in the literature [11]. In this work, the strategy proposed by Bojoi et al. [12] has been adapted, obtaining a structure with three proportional-integral (PI) controllers. The first PI controller is the one responsible of comparing and controlling the rotor speed ω_r to its reference value ω_{ref}, whereas the other two do the same task for the flux in the rotor frame λ_r and the current of the stator in the q axis i_{sq}. The dynamics of the control are described as:

$$i_{sq_{ref}} = k_{p\omega} \cdot e_{\omega} + \int_0^t k_{i\omega} \cdot e_{\omega} \cdot dt \tag{4}$$

$$v_{sq} = k_{pi} \cdot e_{i_{sq}} + \int_0^t k_{ii} \cdot e_{i_{sq}} \cdot dt \tag{5}$$

$$v_{sd} = k_{p\lambda} \cdot e_{\lambda} + \int_0^t k_{i\lambda} \cdot e_{\lambda} \cdot dt \tag{6}$$

where $k_{p\omega}$, k_{pi} and $k_{p\lambda}$ are the gains of the proportional controls, $k_{i\omega}$, k_{ii} and $k_{i\lambda}$ are the gains of the integral controls; and e_{ω}, e_{isq} and e_{λ} are the tracking errors for the speed, the current and the flux, respectively.

IV. MODEL VALIDATION

The validation process is done in a range of operating conditions. Thus, the tests were carried out at constant speeds ranging between 250 and 1500 rpm. As the first stage of the monitored gearbox has a transmission ratio of 5:2, the speed at which the monitored bearing is running goes from 100 to 600 rpm. Radial loads from 0 to 5 kN were applied to the intermediate shaft, i.e. the monitored bearing supports a radial load between 0 and 2.5 kN. Finally, the LM applies constant torque loads that defined as the 30 % or the 60 % of the torque applied by the DM.

The simulations for the model have been carried out by using the software Dymola® with a tolerance of 10^{-4} and a time step of 10^{-3} s using the Dassl integration method.

A. Parameter tuning

In order to estimate the energy losses due to the presence of gearboxes that consist of spur gears, the model presented by Petrescu et al. [13] is used.

Besides the gearboxes, electrical issues, the bearings, and the couplings also imply torque losses. The collected data have been used to find a polynomial surface of degree 4 as a function of the applied speed and the applied torque that represent these torque losses. Similarly, another polynomial surface has been fitted to obtain a constant value for the applied torque for each torque level. This function has one degree lower than the previous one, as this degree is enough for accurately defining the behaviour of the applied torque in different conditions.

When it comes to estimating the parameters of the motor, a recursive approach has been adopted. Thus, the resistances R and the inductances L have been iteratively changed in order to obtain values of the current i_q as closest to the experimental values. This strategy leads to the following values of the parameters: R_s = 0.03 Ω, R_r = 1.012 Ω, L_m = 0.0312 H, L_s = 0.0312 H and L_r = 0.0312 H, where subscripts s, r and m stand for stator, rotor and magnetizing, respectively.

B. Validation

The experimental and simulated data have been processed and some features have been selected to validate the model. Regarding the motor current experimental data, the three phases have been filtered by means of a low pass finite impulse response filter taking 500 Hz as passband-edge frequency. After that, the simulated and the experimental i_{sq} currents are obtained and the root mean square error (RMSE) of the i_{sq} current is computed in order to analyse the accuracy of the model with respect to the experimental values. Similarly, the experimental signal of the torque of the DM is also low pass filtered by the use of the same filter and the RMSE is calculated.

The values of the RMSE of the current and the torque are presented in Table 1 and Table 2, respectively, for different operating conditions. As it can be seen, the model provides differences up to 10 %, which are considered accurate enough for the followed purposes.

Concerning the vibration analysis, the data acquired by the accelerometers is low pass filtered taking 150 Hz as passband-edge frequency. It is known that a bearing with a fault in the outer ring entails the appearance of the ball pass frequency of outer ring (BPFO) and its harmonics in the frequency spectrum [14]. Table 3 shows the values of this characteristic frequency and its harmonics obtained from the simulation results, the experiments and the theoretical equation [14]. The differences between the values presented in Table 3 are less than 0.6 %.

The effect of the damage size is analysed by some features calculated near the characteristic frequencies and its harmonics [15]. The features used in this work for each of the zones are the following: the averaged root mean square (RMS), which is a dimensionless value computed as the value of the RMS of the signal in one zone of interest divided by the RMS of the whole filtered signal; and the sum of peak amplitudes, which is the sum of the amplitudes of the peaks found in one zone of interest.

Table 1. RMSE in the estimation of the current

speed / torque	30 [%]	60 [%]
250 rpm	2.59 A (10 %)	2.08 A (6.3 %)
500 rpm	1.7 A (6.9 %)	2.73 A (8.5 %)
1000 rpm	1.89 A (7.4 %)	2 A (5.9 %)
1500 rpm	1.23 A (4.9 %)	2.56 A (7.8 %)

Table 2. RMSE in the estimation of the torque

speed / torque	30 [%]	60 [%]
250 rpm	0.74 N·m (4.2 %)	1.59 N·m (4.8 %)
500 rpm	1.89 N·m (10.9 %)	1.64 N·m (5.3 %)
1000 rpm	0.96 N·m (4.9 %)	1.08 N·m (3.2 %)
1500 rpm	1.85 N·m (9.9 %)	2.16 N·m (6.6 %)

Table 3. Comparison between the BPFO values

Value	Theoretical	Simulation	Experimental
BPFO [Hz]	23.82	23.68	23.75
2xBPFO [Hz]	47.63	47.36	47.5
3xBPFO [Hz]	71.45	71.29	71.25

Fig. 2 shows the sum of peak amplitudes computed for different bearing conditions near the BPFO (note that the healthy case is denoted by H). It can be seen that in general the simulation results are within the limits provided by the experimental results and that they follow an increasing pattern as the size of the damage gets higher. This situation also occurs for the features analysed near the first harmonic, as shown in Fig. 3.

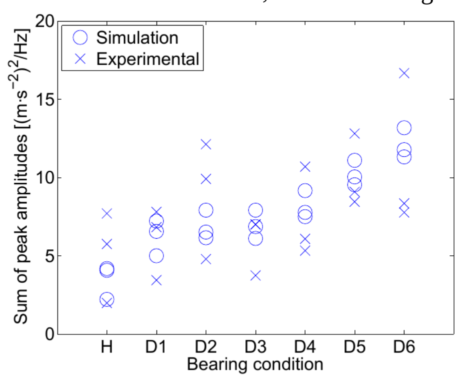

Fig. 2. Sum of peak amplitudes in the zone of BPFO

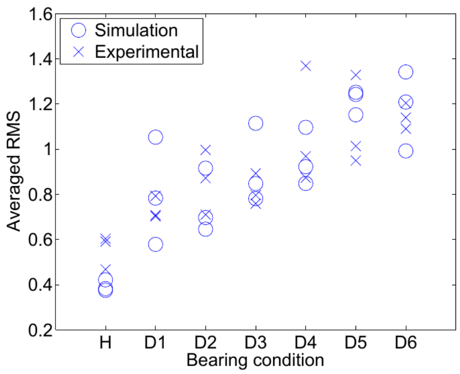

Fig. 3. Averaged RMS for the first harmonic of BPFO

V. CONCLUSIONS

A model for a rotary machine is developed in this paper from an electromechanical point of view by the use of Modelica® object-oriented modelling language. A detailed model for rolling element bearings is used to analyse the dynamics of one of the bearings mounted in the monitored gearbox of the test rig considering it to be in healthy condition or in different damaged states.

A number of tests in different operating conditions have been carried out, as well as in different conditions of the bearings. The model of the electromechanical elements of the test rig is validated as differences lower than 10 % are obtained. In particular, for the case of the bearings, the validation of the model is done by comparing the response of the experimental results and the simulation results in the frequency-domain, obtaining differences lower than 0.6 %. Moreover, the degradation process can be determined by the use of some indicators related to the characteristic fault frequency and its harmonics. This leads to the ability to do the diagnosis process with simulated data obtained from the developed model.

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