The Electromagnetic Damper – Towards a First Large-Scale Industrial Application

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Summary :
Studies concerning the magnetic bearing were begun at the Research and Development Department when the first industrial applications on large machines appeared. Our work concerned, in particular, the modelling of this device and the construction of models used in performing validation tests. Furthermore these studies involved the search for new applications. Thus a large-sized electromagnetic damper was constructed and tested on a second test bench to demonstrate that this solution may be applied to a large 900 MWe turbo-generator in operation.

Since advances in power amplifiers indicated the possibility of supporting large sized rotors, the Research and Development Department at Electricité de France became actively interested in the magnetic bearing. From 1979 studies have enabled the potential of this new technology to be evaluated. In consideration of all that is at stake and the serious constraints concerning the availability and reliability of machines, considerable efforts are being made in the areas of modelling, behaviour prediction and reliability. Validation tests on large-scale models complete this work.

The project included three phases :

1) The development of software enabling the dynamic behaviour of the active magnetic bearing to be predicted,
2) The construction of test benches designed to verify the computing programmes and to measure bearing characteristics.
3) Application tests.

The requirements with respect to machine operating conditions and reliability are such at E.D.F. that it is impossible to put a new machine into operation unless its operating characteristics, under all possible abnormal conditions, are known, including exceptional transient currents (broken turbine blades, earthquakes, etc...). It is therefore indispensable to be able to simulate the behaviour of the machines under these conditions. Computing programmes which the E.D.F. has had in its possesion for many years now make this analysis possible. They were adapted with regard to the particular type of support provided by the magnetic bearing. The models used are capable of handling the following types of problems :

. linear (small displacements) at steady state,
. linear transient currents,
. non-linear transient currents,
. stability.

The above concern machines with several types of bearing.

FIRST CONCLUSIVE TEST

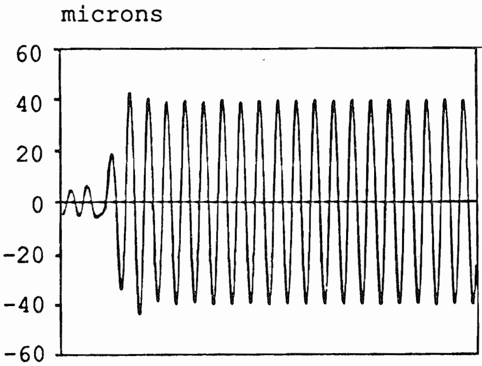
Along with theoretical studies and simulation calculations, tests involving real equipment were conducted, either to evaluate the characteristics of the active magnetic bearings (AMB), or with the intention of direct applications to machines in operation.
To this end, the Electricité de France constructed a test bench around a 4 t supporting capacity bearing. Its purpose was :

. to verify the computing programmes
. to measure bearing characteristics
. to enable reliability testing
. to test the emergency devices

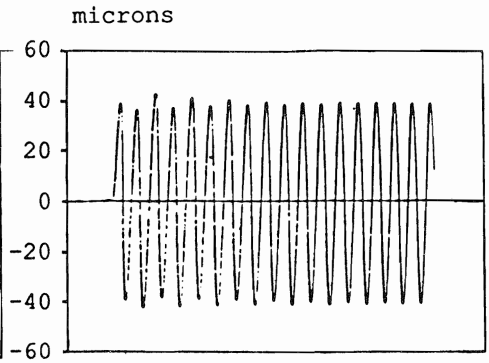
The test bench consists of a 4 m long 2 t rotor, which is supported by two oil-film bearings (end bearings). The magnetic bearing, with a load capacity of 4 t, manufactured by the French company S2M (Société de Mécanique Magnétique) is located in the middle. This arrangement is best adapted to the first two points of the study. The first tests conducted in 1984 are very encouraging since there is an accurate correspondance between the measured and computed characteristics (see table). The dynamic coefficients of the magnetic bearing were determined by finding the first two natural frequencies of the rotor. For the second natural frequency, the magnetic bearing is located on a vibration node . This enables the dynamic coefficients of fluid bearings to be evaluated. This, along with the knowledge of the first natural frequency, enables the dynamic coefficients of the magnetic bearing to be estimated.

	Measurements		Computations	
Control loop gain	Magnetic bearing stiffness K (N/m)	Magnetic bearing damping C (N.s/m)	Magnetic bearing stiffness K (N/m)	Magnetic bearing damping C (N.s/m)
0,25	$2,25 \cdot 10^7$	$9,09 \cdot 10^4$	$2,15 \cdot 10^7$	$8,39 \cdot 10^4$
0,5	$4,59 \cdot 10^7$	$1,62 \cdot 10^5$	$4,50 \cdot 10^7$	$1,64 \cdot 10^5$
0,7	$6,23 \cdot 10^7$	$2,22 \cdot 10^5$	$6,51 \cdot 10^7$	$2,29 \cdot 10^5$
1,0	$9,17 \cdot 10^7$	$3,20 \cdot 10^5$	$9,81 \cdot 10^7$	$3,24 \cdot 10^5$

Table. Determination of the dynamic coefficients of the magnetic bearing as a function of the control loop gain. This table shows a good concordance between the computed values and measurements on the test bench with respect to magnetic bearing stiffness and damping.



a) Measurement



b) Simulation

Fig.1. Comparison between simulation and measurement on the test bench during the detachment of a rotating explosive bolt (transient unbalanced mass). Overvoltage levels are very similar in both cases. However, numeric computations lead to greater amplitudes, especially in the actual transient phase.

Furthermore, the response to a disturbance, (the detachment of an explosive bolt during rotation) has proven to be in conformance with predictions (see figure 1). These results, obtained for small displacements (linear) still need to be validated with respect to large displacements, and especially in the vicinity of saturation areas (magnetic fields greater than 1.3 Tesla).

SUPPORTING OR CORRECTING BEARING? STEPS TOWARDS A FIRST LARGE-SCALE APPLICATION

Another application of the AMB consists in using it not as a support, but as a corrector or damper. Indeed, traditional bearings generally offer little damping and exceeding of critical speeds may lead to vibratory problems. In these cases, one solution would be to insert an AMB in the shaft line which serves as a damper facilitating the crossing of these critical speeds. In this instance the bearing does not have to support the rotor. Its only function is to develop a force in constant opposition to that created by eventual unbalances. This type of application is already being considered by Electricité de France on high-power turbogenerators in operation. The idea is not new. Several papers have described the use of electromagnetic forces to dampen vibrations, but only for small-size machines.

Nikolajsen and Holmes (1) have demonstrated the effectiveness of a controlled electromagnet in reducing vibration in a transmission shaft.

Allaire and Humphris (2) have developed magnetic bearings which were tested on a flexible rotor as both supporting bearings and dampers. Our approach is original for three main reasons :

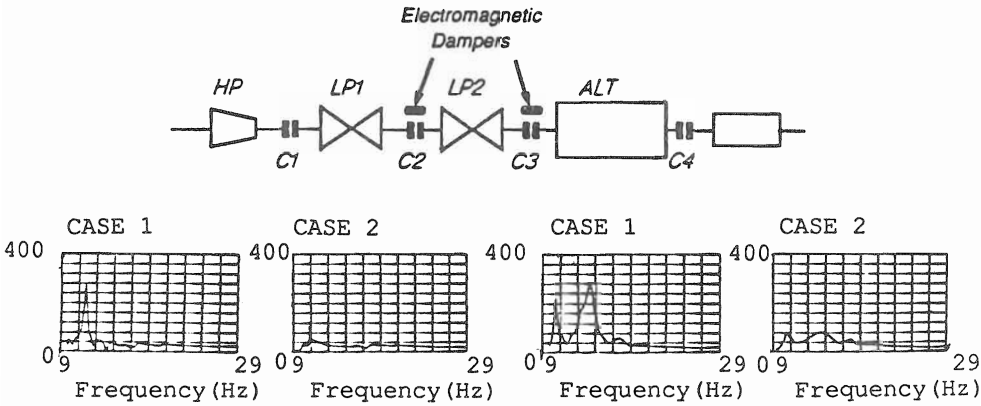
1) Our purpose is to show that this method may be applied to a 900 Mwe turbogenerator in operation.
2) The electromagnetic damper only uses the upper 270° of the rotor to facilitate mounting and dismounting at the site.
3) The rotor is massive and evaluation of rotor heating is an important point of our study.

In order to demonstrate the feasability of such an undertaking, a test bench has been developed. This test bench takes into account the dimensional restrictions of the actual shaft line with respect to the planned installation point of the electromagnetic damper (coupling plates).

DIMENSIONING OF A LARGE SIZE ELECTROMAGNETIC DAMPER

The first phase of the study consisted in determining (by digital simulation on shaft line models) possible installation locations and the performance of the electromagnetic dampers so as to provide the required impedance to the shaft lines.

These preliminary computations have shown that on the machine under consideration, a device which only provides damping offers the optimal solution with respect to vibration reduction at all critical speeds. Furthermore, they reveal that a rotating force of approximately 10 tons would be sufficient to significantly reduce vibration when passing the most difficult critical speeds. Figure 2 shows the peak vibrations obtained by digital simulation in the horizontal direction at coupling 2 and in the vertical direction with respect to coupling 3 during a deceleration. With magnetic dampers, the horizontal amplitude is divided by 7 and the vertical amplitudes by 3.5 during the crossing of critical speeds. In this simulation, the damping of each damper is 8.10^6 N.s/m.

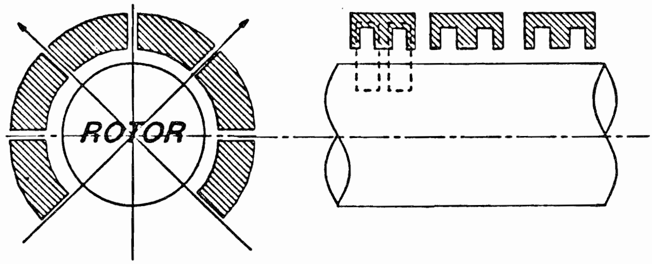


a) Horizontal vibration at coupling 2 b) Vertical vibration at coupling 3

Fig.2. Digital simulation (finite element method) of the effect of 2 electromagnetic dampers installed on couplings 2 and 3 during deceleration.
Case 1 : Initial
Case 2 : With 2 magnetic dampers each producing 8.10^6 N.s/m.

The second phase therefore consisted in developing and constructing a large size damper for controlling the vibrations of an existing shaft line.
The damper which was proposed, following all of the above considerations, is described in figure 3.
It includes 6 poles with 4 magnetic axes. It only covers the upper 270° of the rotor to facilitate installation on the actual machine. This particular characteristic means that it must operate in the vicinity of an average static force so as to create the necessary rotating force to dampen vibrations. Each 45° pole contains 3 independant electromagnets. Each of the 18

power amplifiers which drive each electromagnet operates on 160 V dc with a maximum current of 50 A. In our application, flux circulation is axial in order to reduce losses in the rotor.



6 poles of 45° each   Axial flux circulation

Fig.3. Electromagnetic damper working principle.

The rotating force is optimised via a compensation method which consists in supplementing to the detector signals of the various poles with part of the signals that the missing poles would have detected if they existed. Use of the latter improves the modulus of the rotating force approximately 43 % (3). Thus, each pole must provide 7,000 daN in order to develop 10,000 daN of rotating force.

The stability and dynamic characteristics of an active magnetic damper (or bearing) are determined by the electronic control system. In our application, the various filters used provide a phase of approximately 90° in the 10 to 20 Hz frequency range. The damping function is assured in this way. To obtain a high damping coefficient without reducing stability, a synchronous peak gain filter is used. This type of filter (synchronised with the rotation speed) significantly increases the loop gain only at the rotation speed without modifying the phase. The overall control system transfer function which represents damper impedance and the resulting damping curves are represented as a function of frequency for two different gains (see fig.4). To minimize rotor heating due to eddy currents, the static force is automatically adjusted by the control system according to the amplitude of the vibrations.

TEST BENCH

The rotor is 5 meters long and its total weight is approximately 7 tons. The middle of the rotor represents the coupling of a LP

rotor in operation and therefore has the same dimensions. The rotor is supported by two circular hydrodynamic bearings. It is driven by a dc motor and may attain a maximum speed of 1,200 RPM. The frame consists of stiff metal beams. Its total weight is approximately 70 tons. It stands upon isolating devices so as to avoid transmitting its vibrations to the building.
Since the rotor is massive, the knowledge of rotor temperature is essential in order to establish the operating limits of the damper. These temperatures are measured via 12 sensors distributed along the surface and at different depths (see fig. 5). These signals are transmitted via a remote sensor and plotted in real time on a paper recorder.

MODELLING

It is not always possible to determine all of a machine's parameters through experiments. Modelling is one way of achieving this. Thus the rotor and frame were modeled so as to :

1) Verify that damping of the electromagnetic damper conforms to the theoretical value predicted.
2) Predict test bench behaviour for different values of the exciter energy (unbalanced mass) and the damping supplied by the electromagnetic damper.

The rotor was modeled using 100 continous beam elements under plane deflection. The rotor model was validated using the first two natural free-free frequencies experimentally obtained by modal analysis as shown in the table below :

Rotor free-free Frequency	Computed	Measured	Difference
1st natural frequency	43,43 Hz	43,55 Hz	0,3 %
2nd natural frequency	99,57 Hz	99,61 Hz	0,04 %

The model of the frame consists of continuous beams which are interconnected and which have the same main dimensions and mass as the real frame.

MAIN RESULTS

The main results are shown in figures 6 to 8. They make possible the comparison between the values measured on the test bench and

the values computed with the model.
Figure 6 indicates rotor vibration with a residual unbalanced mass of approximately 0.07 m.kg during deceleration for different damping values provided by the electromagnetic damper.
Firstly, we observe that the rotor's critical speed, which is located at approximately 17 Hz, does not shift when the damper is operating, and secondly, that vibrations when this speed is exceeded are well damped. Therefore the electromagnetic damper well fulfills its damper function (stiffness is practically zero). The curves computed make it possible to estimate the damping values and to readjust rotating models. The distance between the fluid bearing axes was 4,5 m.
The electromagnetic damper was designed to provide a rotating force of 10 tons at 15 Hz and maximum damping of 8.10^6 N.s/m.
To verify these figures, the distance between the fluid bearing axes was reduced to 3 m, and tests were conducted on the test bench by applying larger and larger unbalanced masses up to 11 m.kg which represent a force of approximately 10 tons at 15 Hz.
Figure 7 shows that the results obtained on the test bench correspond well with those first calculated.
The various measurements made on the test bench indicate that the increase in temperature at the hottest point of the rotor (across from the unbalanced mass) is approximately 2° C per ton and per minute at 15 Hz during the first moments.
Figure 8 represents the results of a long duration test. It indicates that the slope of the graph decreases as rotor thermal equilibrium is approached. An estimation of the power loss was made using a relatively simplified calculation (bidimensional calculation - localised source of heat...).

CONCLUSIONS

The results obtained from the various series of tests show that the main objectives aimed for were attained.
The device does behave as a genuine damper since the rotor's critical speed is indeed damped and not shifted.
We have observed a good concordance between measurements and calculations. Differences arise primarily from measurement uncertainties (sensitivity of the sensors to type of material and to temperature, attenuation in the filters...) and from the simplified modelling of the test bench frame. It should be noted

that differences are especially significant below 10 Hz and that above this level theoretical and measured values are very close. Thus we can say that the theoretical damping curves of the device are completely in accordance with real operation, especially within the range of 10 hz to 20 Hz which is the most important.Tests with large unbalanced masses have shown that the damper is well capable of providing a force of 10 tons at 15 Hz. Temperature sensors in the rotor also made it possible to have a good idea of the change in temperature at different points of the rotor according to the unbalanced mass applied and according to time. However, the total power loss in the rotor could not be calculated with accuracy, since the existing thermal calculation codes do not enable exact simulation of the actual phenomenon (due to its three-dimensional and transient nature and non-sinusoidal source). The simplified calculation made still provides a good indication of the order of magnitude of rotor power loss.

In conclusion, the tests conducted show that the electromagnetic damper in question meets the requirements of design specifications even if certain improvements may still be made. In particular, the installation of a control system gain limiter would enable saturation of the amplifiers to be avoided. The stability of the unit may be improved so as to increase damper effectiveness.

Power loss in the rotor is not neglegible, even if it is lower than the value predicted by the manufacturer. It could be significantly lowered by the development of a full damper.

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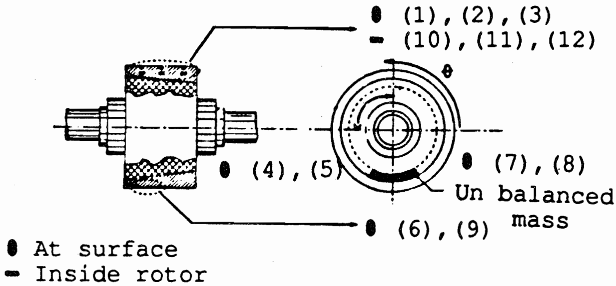


Fig.5. Placement of temperature sensors.

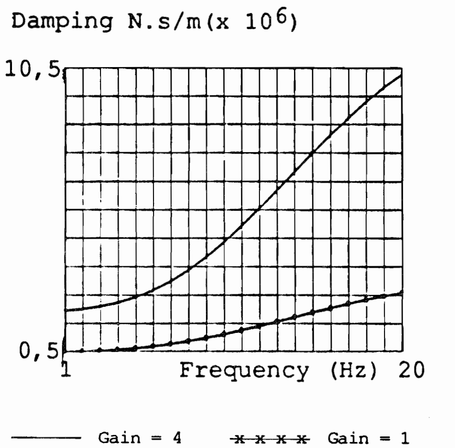
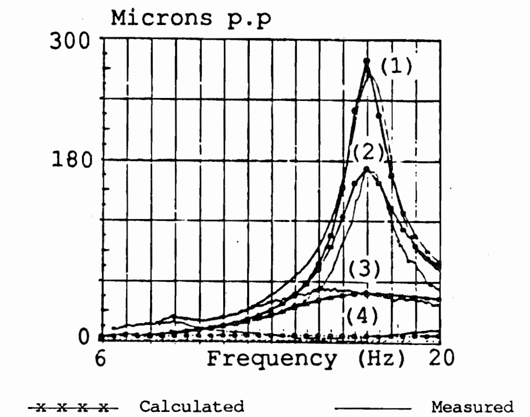


Fig.4. Damper damping curves as a function of frequency for 2 different gains.



1)Cd=0 2)Cd=310^4 Ns/m 3)Cd=210^5 Ns/m 4)Cd=210^6 Ns/m

Fig.6. Vertical rotor vibrations with a residual unbalanced mass of 0,07 m.kg for different damping values Cd.

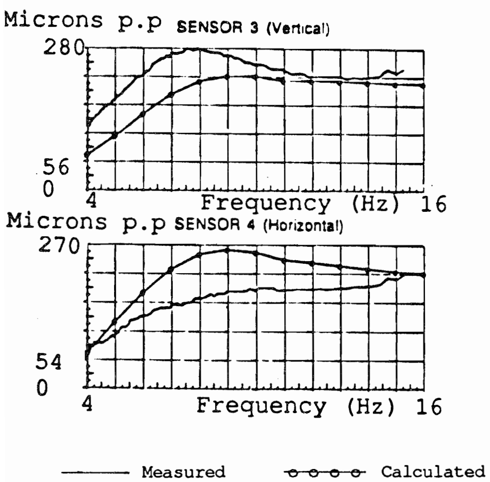


Fig.7. Rotor vibration with an unbalanced mass of 11,22 m.kg and damping of $7{,}610^6$ Ns/m. Distance between bearing axes is 3 m.

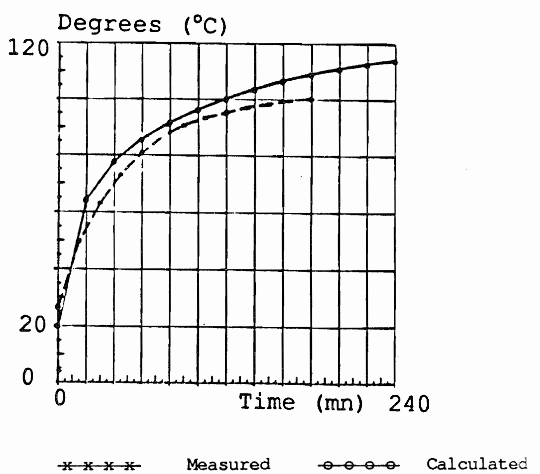


Fig.8. Change in rotor temperature as a function of time for an unbalanced mass of 1,8 m.kg. Power loss is estimated at 5 kW at 15 Hz.