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VIBRATION CONTROL OF A LARGE TURBOGENERATOR BY ELECTROMAGNETIC DAMPERS

CHAN HEW WAI C.
Electricité de France. Direction des Etudes et Recherches
1 av. du Général de Gaulle
92141 Clamart - FRANCE

Abstract

The passing through critical speeds of rotating machines with long shafts may often lead to vibratory problems. In such cases, one solution would be to add damping on the shaft line during running through the critical speeds in order to reduce vibrations. These additional dampings can result from active electromagnetic devices. The Research and Development Department at Electricité de France has demonstrated the feasability of such an attempt. For this particular case, a test bench has been developed and a 10,000 daN load capacity electromagnetic damper was studied. After the promising results obtained, we have undertaken new works in order to improve the load capacity and the reliability of the electromagnetic damper. Our ultimate aim is to prove that this solution may be applied to a large 900 Mwe turbogenerator in operation.

1. Introduction

Today, Electricité de France (EDF) operates an important number of machines for electricity production. These machines with long shafts often operate above several critical speeds. These machines were designed using two different technologies. The first consists on supporting each rotor by two hydrodynamic bearings and in the second, each rotor is supported by only one bearing. In this case, the shafts may sometimes show high levels of vibrations during the running through certain critical speeds due to little damping offered by fluid-film bearings.
The Research and Development Department at EDF has been interested in this problem for many years and has proposed an interesting solution which consists on adding one or many electromagnetic dampers for active vibration control on the machines under consideration.
Several papers have reported the use of electromagnetic forces to dampen vibrations, but only in laboratory or for small-size machines.
Nikolajsen et al. [1] were the pioneers who have demonstrated the effectiveness of a controlled magnetic damper to reduce vibration in a long transmission shaft.
Burrows and Sahinkoya [2,3] used an electromagnetic damper to a laboratory rotor supported by fluid film bearings.
Recently, Allaire et al. [4] have developed and tested a magnetic device used as a bearing or a damper for vibration reduction in a multimass flexible rotor.

The first works have lead us in designing and constructing a prototype of a large size damper. So, an experimental bench to test the electromagnetic damper was developed. In that time, its load capacity was defined to 10,000 daN. Tests conducted have shown that the electromagnetic damper met the requirements of design specifications.
The main results were reported at the 1987 ASME Conference [5] and at the first International Symposium on Magnetic Bearings in Zurich [6].
Today, taking these promising results obtained into account , we deal with the industrialization phase, including some major changes.
Indeed, in normal cases, the load capacity of 10,000 daN is sufficient to significantly reduce vibration during running through the most difficult critical speeds on the aimed machine. However, the requirements with respect to machine operating conditions and reliability are such at EDF that it is better to increase the damper load capacity to 30,000 daN to take into account exceptional unbalances. This excess load capacity of three times is required for the worst case unbalance distribution estimated on the existing machine. But, the dimensional constraints are unchanged. So, the design of the future magnetic damper components was guided by three major concerns : load capacity, size and reliability.
The purpose of this paper is to report on new studies which have been undertaken in order to improve the damper load capacity and the complete device reliability.

2. Magnets Design

The first damper tested is described in Fig.1. It included 6 poles with 4 magnetic axis. It only covered the upper 270° of the rotor. It was an "incomplete" damper.

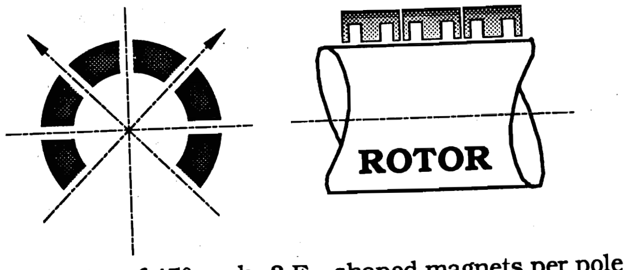


Fig.1 First magnetic damper working principle.

Each 45° pole contained 3 independant E-shaped magnets. Each of the 18 power amplifiers which drived electromagnet operated on 160 V DC with a maximum current of 50 A. Thus, each pole was to have provide 7,000 daN in order to develop 10,000 daN rotating force. The nominal air gap was 2 mm. The magnet sheets used were in silicon iron.
The tests conducted showed the effectiveness of such a device; even if this " incomplete " damper did not work in the best conditions.
But, in order to take exceptional unbalances into account, it is necessary to increase the damper load capacity to 30,000 daN and the air gap to 3 mm to avoid any possible contact. These represent the new design constraints.
So, the first phase of the study consisted in determining by numerical calculations using Finite Elements Method (FEM) the load capacity limit of the former electromagnets.
Firstly, these preliminary computations have shown that with the present E-shaped electromagnets with silicon iron sheets, it is impossible to reach the desired rotating force of 30,000 daN.
Secondly, they also demonstrated that the use of an "incomplete" damper led to the same result.
So, further calculations with various material sheets, but with the same shape, have been made. Figure 2 shows the first magnetization curves of the candidate materials.

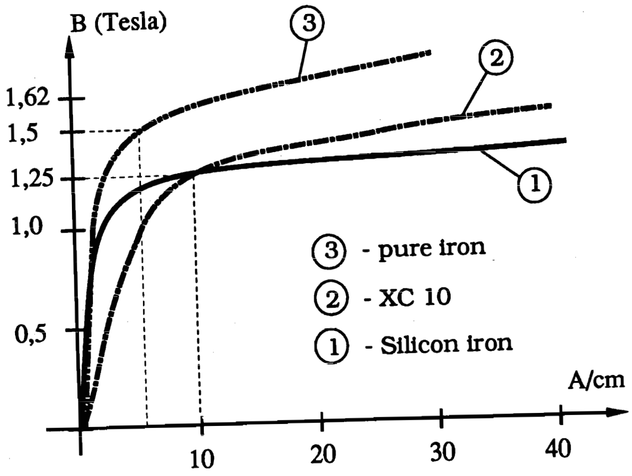


Fig.2 First magnetization curves for different materials.

The main results obtained with the E-shaped magnets are indicated in table 3 according to the nature of sheets used.

Nature of Sheets	One Magnet Load Capacity (daN)	Amplitude of the Rotating Force (daN)
Silicon Iron Presently used	2,225	13,350
XC 10	2,805	16,830
Pure Iron	3,972	23,830

Table 3. Theoretical rotating force amplitude for a 360° damper with E-shaped magnets according to different sheets.

Furthermore, these computations reveal that E-shaped magnets does not represent the optimum. Indeed, Fig.4 shows one induction calculation with this magnet design.

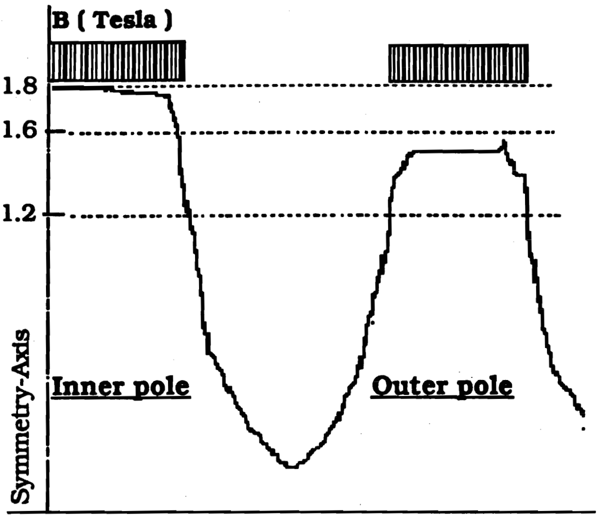


Fig.4 One flux density calculation with E-shaped magnet.

It can be seen that the outer pole induction is clearly smaller than it is under the inner pole. All other calculations made with this magnet shape give similar results.
Tests in order to equalize the two inductions have been carried out. A 80% surface reduction of the outer pole has been made and an unequal coil turns distribution tested (the outer pole had twice as much turns as the inner). In all cases, results have been disappointing for the equality of the two inductions have never been obtained.
Thus, further calculations with U-shaped magnets have been realized. These enable changes due to magnet shape to be evaluated, the other parameters being unchanged.

Nature of Sheets	One Magnet Load Capacity (daN)	Amplitude of the Rotating Force (daN)
XC 10	6,070	36,422
Pure Iron	6,198	37,188

Table 5. Theoretical rotating force amplitude for a 360° damper with U-shaped magnets according to different sheets.

Table 5 summarizes the main results obtained with U-shaped magnets with different sheets.
In fact, in the same outside dimensions, U-shaped magnets give a 20% active surface increase: less coils are used.

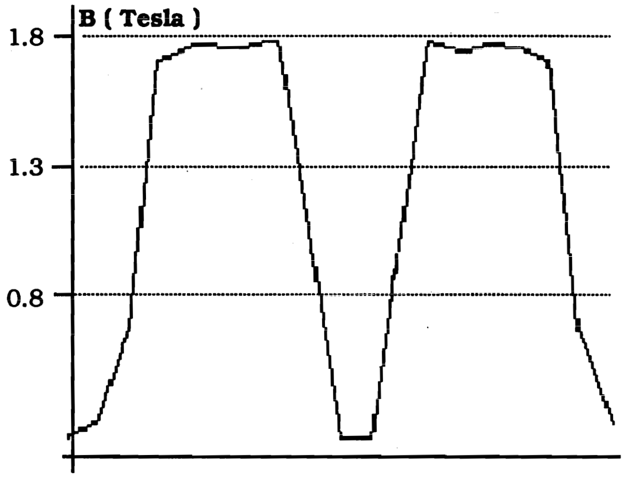


Fig. 6 Flux density calculation with U-shaped magnet.

Fig.6 shows the good flux density symmetry within the two poles. As an electromagnet magnetic force is proportional to the square of the flux density, this explains the significant gain obtained with this magnet shape.

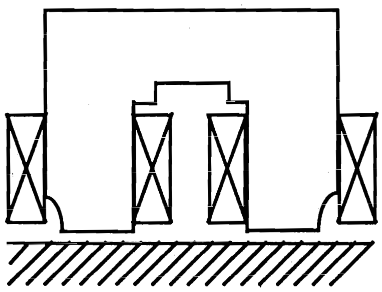


Fig.7 The adopted U-shaped magnet.

This, along with the development of a new power amplifier of 30 KVA, indicates that it is theorecally possible to have access to the value of 30,000 daN on condition to have a full damper which covers the 360° of the rotor. Tests on real scale will confirm these results.

3. Power amplifier

With the former damper, power amplifiers of 8 KVA were used and were sufficient in order to develop 10,000 daN rotating force at 17 Hz with a 2 mm nominal air gap.

As the new goal is the 30,000 daN rotating force obtention with 3 mm air gap, the development of a power amplifier of 30 KVA has been undertaken.

The specifications submitted to the Société de Mécanique Magnétique (S.2.M), the damper manufacturer, contained the following stipulations :

- Reactive power : 30 KVA,
- Load inductance : 200 mH.
- Phase displacement : 5° at 17 Hz with 200 mH maximum.
- The flux feedback control is required.

The amplifier is designed to provide the reactive power of an electromagnet to the future damper of 30,000 daN in order to obtain a rotating force as F = Fo + Fo sin (wt). The maximum force Fmax= 2Fo must correspond to the maximum current value. For that, the amplifier must be able to supply a 60 A 17 Hz current without phase displacement of the flux feedback control.

4. First tests

Before the manufacturing of the second damper model, it was interesting to verify the calculations results and the new 30 KVA amplifier performances. So, a real scale model of the U-shaped magnet was constructed.

The new amplifier was developed and tested with this representative inductive load. The main results are displayed in Figs.8,9 and 10.

Fig.8 represents the induction and the electromagnet force as a function of the current. The attained force with a 60 A current is about 6,400 daN. It means that the damper rotating force using these magnets will be 38,400 daN (6,400x3x2). The air gap is 3 mm.

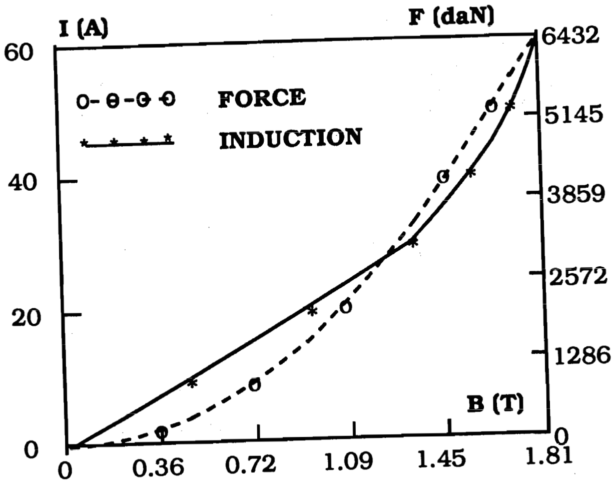
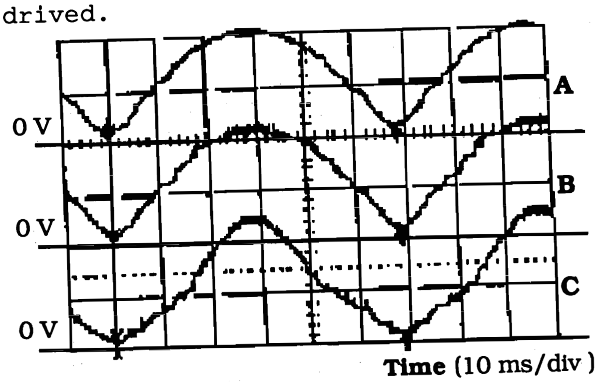


Fig.8 Force and flux density curves of the tested U-shaped magnet as a function of the current. Air gap is 3 mm.

The electromagnet force is proportional to the square of the flux measured by the flux measurement coil. For this reason, the reference force signal first passes through a linearizer. The linearizer output becomes the flux feedback control reference. The electromagnet is therefore in force drived.



A - Flux Reference (5 V / div)
B - Flux (5 V / div)
C - Current (30 A / div)

Fig.9 Amplifier results at 17 Hz during force modulation tests.

The amplifier results during force modulation tests are indicated in Figs.9 and 10.

It appears in Fig.9 that the amplifier modulates with the real load a force at 17 Hz without significant difference in phase between the flux reference and the flux itself. Note the difference between flux and current due to magnetic iron saturation flux density.

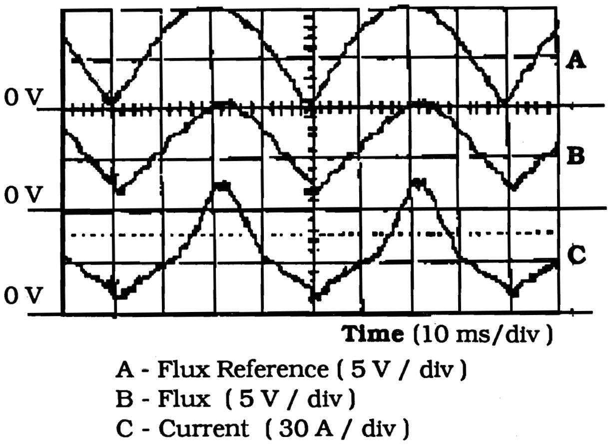


Fig. 10 Amplifier results at 25 Hz during force modulation tests.

The results at 25 Hz are presented in Fig.10. The amplifier shows voltage saturation and the current increases with its maximum slope. So, a phase displacement in flux appears.

Thus, we can say that the experimental results of the U-shaped magnet drivén by the new 30 KVA amplifier are completely in accordance with theoretical predictions.

5. Digital control

Reliability is our major concern. In order to improve the complete device reliability, a digital control has been studied and developed. The digital control is based on a DSP card with a Texas Instruments microprocessor (TMS320C25).

This card, VMEbus compatible, includes 64 Kbytes dual port RAM for exchanges with the control processor (Motorola 68000), I/O auxiliary ports for information transferts with the A/D and D/A converters, a DMA controller for I/O data transferts.

Sensor signal acquisition card is composed by a synchronous demodulator, analog anti-aliasing filter, sample and hold circuit and a 12 bits A/D converter. The amplifiers are driven by 13 bits D/A converters (see Fig.11).

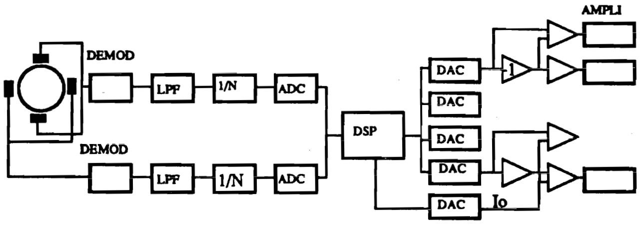


Fig.11 Block diagram of the digital control.

Displacement probes measure rotor motions. These primary signals pass through a run-out suppression block. A large bandwith phase correction network composed mainly by a filtered derivation function ensures a 90° phase lead in the range of 4 to 20 Hz. In this way, we create a damping force due to a velocity feedback.
In order to obtain a high damping coefficient without damaging stability, a tracking filter is used. Such a filter (synchronized with the rotating speed) is in fact a pure numerical integrator which bandwith is adjusted to have an adequate response of the machine. Its function is to reduce the output signal to zero. So, its gain is nearly "infinite". That means the damping value is very high and constant whatever the frequency.
The sine and cosine signals which synchronize the tracking filter are issued from a sin/cos table stored on the DSP card. A key phaser, along with a Phase Locked Loop (PLL), enables the right sinus and cosinus values to be obtained. Finally, a coefficients weighting circuit guides the right signal to its corresponding amplifier (see Fig. 12).

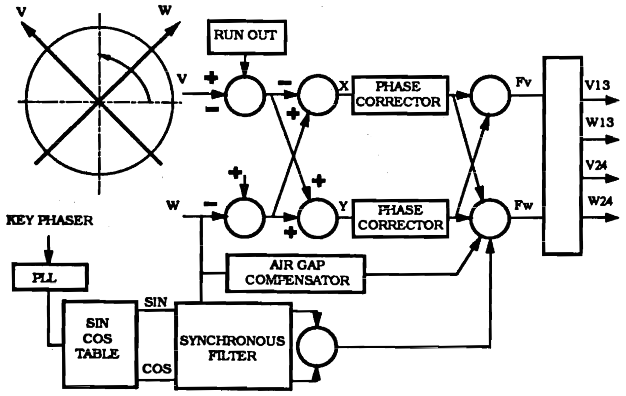


Fig.12 Digital control system.

This digital control scheme has been realized and tested with the former damper of 10,000 daN.

6. Main results

Typical results when the digital control operates are presented in Figs.13, 14, 15 and 16. The flexible rotor used must pass a bending critical speed at about 16.4 Hz, up to a nominal speed of 20 Hz.

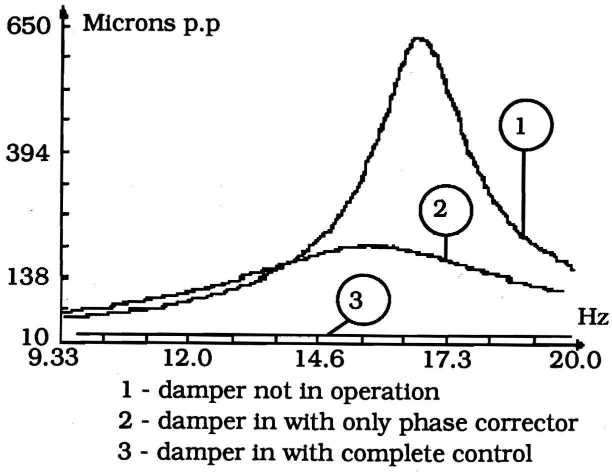


Fig.13 Vertical rotor vibrations according to different damping coefficients. (unbalance=0,10 m.kg)

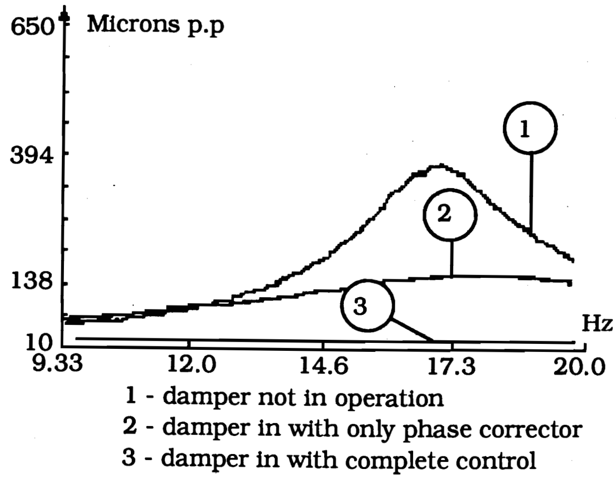


Fig.14 Horizontal rotor vibrations according to different damping coefficients. (unbalance=0,10 m.kg)

Figures 13 and 14 indicate rotor vertical and horizontal vibrations with a residual unbalance mass of approximately 0,10 m.kg during deceleration for different damping values provided by the electromagnetic damper.

Firstly, it can be seen that the rotor's critical speed does not shift when the damper is operating, and secondly, that vibrations are well damped when it is running through the critical speed.

Therefore the electromagnetic damper well fulfills its damper function (stiffness is practically equal to zero). This first damper was designed to provide a rotating force of 10,000 daN at 17 Hz. To verify these figures, tests were conducted on the test bench by applying unbalance masses up to 8,6 m.kg which represent a rotating force of about 10,000 daN at 17 Hz.

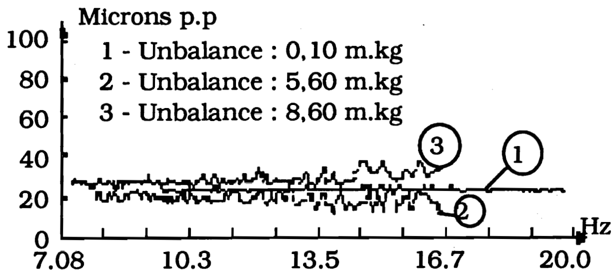


Fig.15 Vertical rotor vibrations with the complete digital control according to different unbalances.

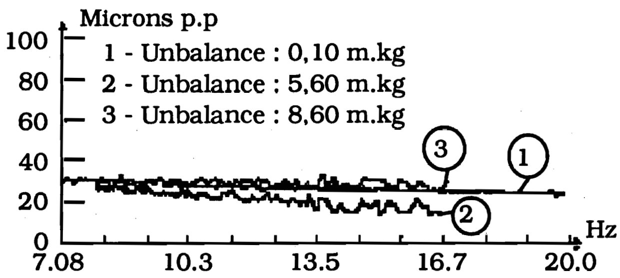


Fig.16 Horizontal rotor vibrations with the complete digital control according to different unbalances.

Figures 15 and 16 show these tests results. The rotor vibrations are practically constant(30 microns peak to peak) whatever the unbalance masses. These residual vibrations correspond to inevitable errors issued from the run-out suppression block. That demonstrates the great efficiency of the damper and shows the high damping coefficient it can deliver. These very interesting results have already been observed with the former analog controlled damper. The main change is the fact that the digital control gives better results with a very high degree of stability.

7. Conclusion

The results observed during different tests on the U-shaped magnet driven by the new 30 KVA amplifier allow us to say that the 30,000 daN rotating force will be obtained without any major problem.
The new amplifier in question gives complete satisfaction.
The digital control used is consistent with the expectations. Tests conducted on it show that it is possible to effectively dampen the rotor vibrations with a high damping coefficient without damaging the stability.
The digital control will certainly be implemented on the industrial prototype. Indeed, it will improve the reliability and will make easier the damper setup, specially on site. Also, it allows, if necessary, the implementation of special functions, not easily available in analog circuits, in order to optimize the rotating force modulus. Here, the expected gain is about 20 %.
These works have represented the components feasability studies (magnet, amplifier, digital control scheme). The next step consists on realizing and testing the complete second damper model with these previous components. These tests are necessary in order to define the design specifications of the industrial device.
Our ultimate goal is to design and realize an industrial prototype damper which should be mounted on site to an existing 900 Mwe turbogenerator set in operation.

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