The Influence of Magnetic Forces on the Stability Behavior of Large Electrical Machines

Dr. H. Sprysl, Dr. H. Vögele, ABB Kraftwerke AG, CH-5242 Birr
Dr. G. Ebi, SENSOPLAN GmbH, D-79801 Hohentengen a.H.

1 Introduction

The unit size of hydro-generators has doubled in the last 20 years. In particular this increase has been made possible by the use of larger diameters and a higher level of utilization of the machines. At the same time the loading levels have also increased: Higher temperatures and centrifugal forces result in larger deformations of the laminated core and of the magnet wheel. When faults occur, such as short circuits or out-of-phase synchronization, there is an increase in the dynamic forces exerted on the rotor and stator. Consequently stringent requirements are placed on the design of hydro-generators.

This applies particularly for the circular form and centering of the rotor and stator, since there are high magnetic tensile forces in the small air gap. Small deviations in the concentricity generate a magnetic force, which further increases the magnitude of the deviations. Provided that the mechanical stiffness of the structure is able to exert a sufficiently large opposing force, then the deflections are harmless and the generator behaves in a stable fashion. The evaluation of this stability is an important criterion for the determination of the dimensions of large electrical machines.

The expressions “unbalanced magnetic pull” and “stability” will be explained. On this basis a procedure will be introduced, with which stability analyses of hydrogenerators can be easily performed using finite element methods. Finally, using the example of a hydrogenerator, it will be shown what design measures can be taken to optimally satisfy the requirements for the generator.

2 Unbalanced Magnetic Pull

A unbalanced magnetic pull (force) results when the air gap between the stator and the rotor is not concentric (Fig.1). The deviation from the ideal circular form can be described as follows:

$$\delta(\varphi) = \delta_m \cdot (1 + e \cdot \cos\varphi) \qquad (1)$$

Without taking account of magnetic saturation, the magnetic energy flux density is given as:

$$B(\varphi) = \frac{const}{\delta(\varphi)} \qquad (2)$$

And finally the force, which is proportional to the square of the energy flux density:

$$F(\varphi) = \frac{C}{(1 + e \cdot \cos\varphi)^2} \qquad (3)$$

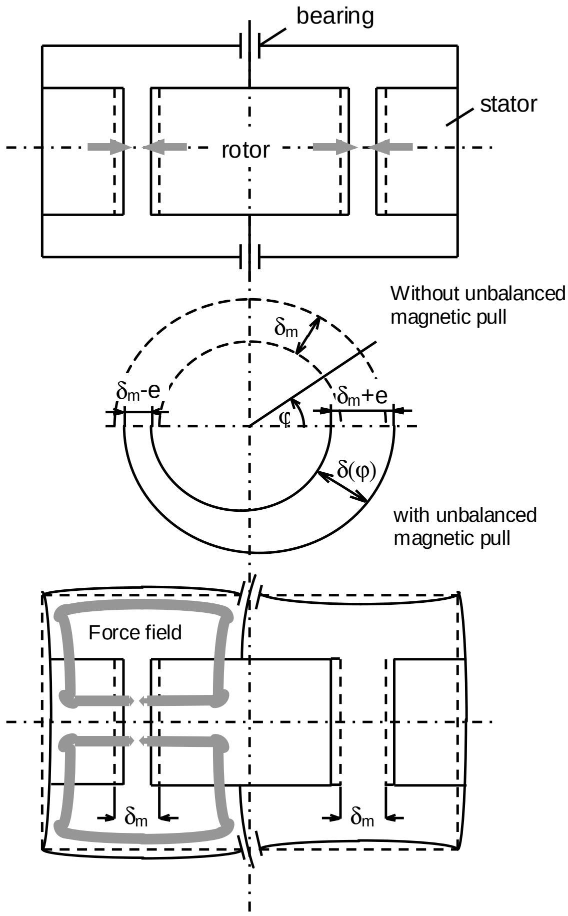


Fig.1: Eccentricity caused by unbalanced magnetic pull.

If the rotor and stator are concentric, i.e. e = 0, then the magnetic forces are in equilibrium. As soon as the concentricity is lost, e.g. through deformation or manufacturing tolerances, then there is an unbalanced magnetic pull of magnitude:

$$F_{magn} = \int_{Umfang} F(\varphi) \cdot d\varphi = C \cdot \int_0^{2\pi} \frac{\cos\varphi \cdot d\varphi}{(1 + e \cdot \cos\varphi)^2} \qquad (4)$$

F_{magn} is exerted in a radial direction (Fig.2).

A magnetic stiffness can be defined on the basis of equation (4):

$$c_{magn} = \frac{F_{magn}}{e} \qquad (5)$$

This permits an analogy to be made with mechanical spring elements.

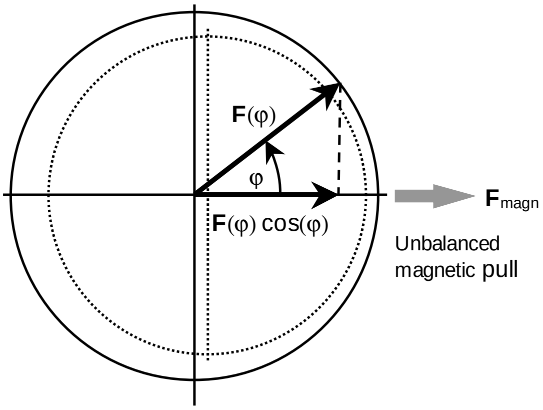


Fig.2: Eccentricity determines the direction of the unbalanced magnetic pull.

3 Stability

The stability problem can be illustrated most easily using a substitute model: A beam is firmly fixed at one end and a block of mild steel is attached to the free end. A bar magnet is located immediately below. If the beam is deflected by the amount u, then it responds with a mechanical reaction force:

$$F_{mech} = c_{mech} \cdot u \qquad (6)$$

At the same time the magnetic force

$$F_{magn} = c_{magn} \cdot u \qquad (7)$$

is generated and attempts to increase the deflection. However, as long as the mechanical reaction force is greater, i.e.

$$c_{mech} \geq c_{magn} \qquad (8)$$

then the beam can return to its original position. Under this condition the state of the beam is stable. However, as soon as the magnetic force is dominant, i.e. $c_{mech} < c_{magn}$

then the end of the beam is attracted to the bar magnet and adheres to it. The behavior of the beam is unstable. A simple estimate of the stability of a generator or a motor can be made by ignoring the influence of the stator laminated core and only considering the stator frame. As a first approximation the rotor can be considered rigid. If the frame is subjected to periodic deformation u(ϕ) (Fig.3), then it becomes oval. In this case the ring-shaped frame produces a reaction force (per radian):

$$F_{mech} = c_{Ring} \cdot u = \frac{9 \cdot E \cdot I}{R^3} \cdot u \tag{9}$$

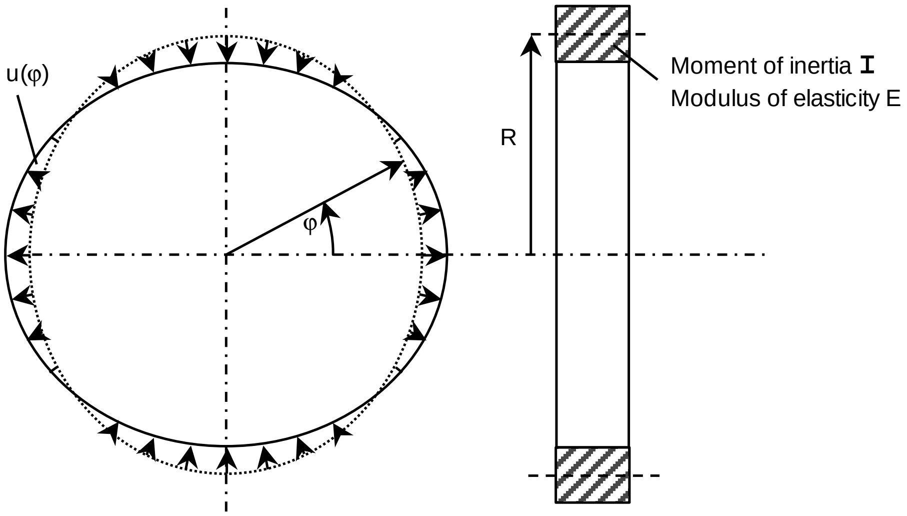


Fig. 3: Oval shape of the air gap

At the same time an unfavorable distribution of magnetic forces is generated in the air gap. In the narrowed zone the magnetic forces increase and in the widened gap they decrease. Nevertheless the resulting force is zero. However the condition constitutes an unsteady equilibrium. Disturbances, which are always present as a result of static deformation or dimensional deviations, immediately cause eccentricity and thereby generate a resulting magnetic force with the stiffness c_{magn} (per radian). The quotient of the mechanical and magnetic stiffness provides a measure of the safety of the machine against instability:

$$S = \frac{c_{Ring}}{c_{magn}} \tag{10}$$

If S < 1, then the stator collapses and adheres to the rotor. This case would result in complete destruction of the equipment with huge economic damages. Hence the evaluation of the stability constitutes an important criterion for the design of large electrical machines.

4 Modal Analysis as Stability Criterion

Of course the simple estimate derived above is not sufficient to perform a reliable and precise evaluation of the stability of the generator. Since only the dimensions of the stator frame are considered in the calculation of C_{Ring}, the designer cannot determine the effect of the fixing of the core laminations in the stator frame or of the frame itself on the stability. The investigation of the mechanical behavior of the entire structure requires the use of the finite element method. The challenge then is to formulate an analysis procedure on the basis of FEM, with which the stability calculation is as efficient as possible and which correctly takes account of the influence of the magnetic pull.

The unbalanced magnetic pull varies within certain limits, in proportion to the variation in the eccentricity. In this respect it behaves like a mechanical spring, but with the difference that the spring constant is negative. Indeed the magnetic pull increases when the separation is reduced and becomes smaller when the gap is increased. Hence the effect of the magnetic pull can be very simply incorporated in the FE model through the introduction of spring elements with negative spring constant. The value of the spring constant is obtained from equation (5). The springs are attached to the inner diameter of the stator in a star configuration and have a common node at the center of the stator (Fig.4). Since F_{magn} acts in a radial direction, the spring constant of the individual springs is given by:

$$k_i = -\frac{c_{magn}}{0{,}5 \cdot N} \qquad (11)$$

where N is the number of springs around the circumference. It must be stated explicitly here that not all FE programs accept elements with negative spring constant. This is often caused by the fact that before the calculation of the total stiffness matrix they check the sign of the element stiffness values and discontinue if it is negative.

Since both the stator and the unbalanced magnetic pull can be represented with the help of linear-elastic structural elements, the stability analysis can be reduced to a modal analysis. The oval shape represents the lowest mode of the bending vibrations of the stator: The four-node vibration. The corresponding natural frequency ν_e provides an appropriate measure for the evaluation of the stability. It is reduced when the magnetic force increases. When the stability limit is reached the four-node vibration disappears, i.e. ν_e becomes zero. The elastic reaction forces are then no longer sufficient to compensate the magnetic pull.

First the connection between stability and modal analysis should be illustrated again using the example of the ring. The FE programs ANSYS® and MADYN will be used for the calculation. ANSYS® is a universal program, which has access to a wide range of structure elements, whereas MADYN is designed for beam structures.

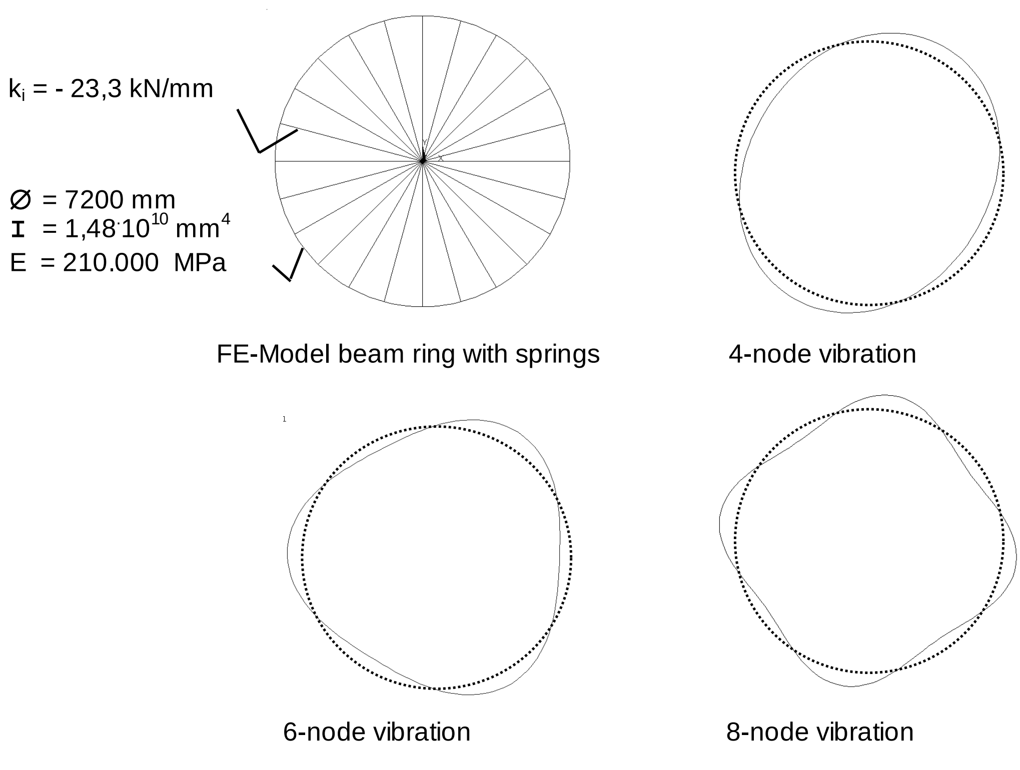


Fig.4: Modes of the ring bending vibrations

Both programs offer the advantage that they automatically suppress the rigid body modes, so that no unnecessary limitations of the degrees of freedom need be made in the determination of the structure natural frequencies. The modal analysis is carried out in ANSYS® using the Householder method; in MADYN the simultaneous vector criterion was employed. The calculation model and the modes of the four, six and eight-node vibrations are shown in Fig.4. The corresponding natural frequencies are given in Table 1.

mode	$c_{magn} = 0$			c_{magn} = 280 kN/mm	
	analytic	MADYN	ANSYS®	MADYN	ANSYS®
4-node / Hz	31,6	31,2	31,7	28,7	29,3
6-node / Hz	89,3	86,2	89,3	85,2	88,2
8-node / Hz	171,2	160,6	172,9	160,1	170,0

Table 1: Natural frequencies of the ring bending vibrations in [Hz].

The results of the closed solution are listed for the simple loading case without magnetic pull:

$$\nu_i = \frac{i\cdot(i^2-1)}{2\cdot\pi\cdot R^2\cdot\sqrt{i^2+1}}\cdot\sqrt{\frac{E\cdot I}{m}} \qquad (12)$$

i is the number of vibration periods around the circumference (i = 2 for the four-node vibration), m is the mass of the ring per unit length of the circumference.

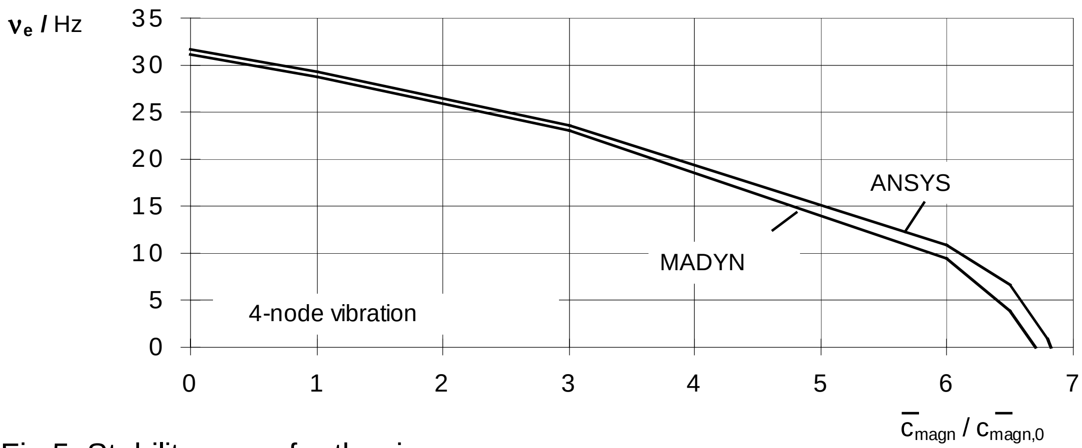


Fig.5: Stability curve for the ring.

Fig.5 shows the stability curve: The natural frequency of the four-node vibration is illustrated as a function of the resulting magnetic pull. The stability limit, $\nu_e = 0$, is achieved for MADYN when c_{magn} has increased to 6.7 times its initial value $c_{magn,0}$. The value in ANSYS® lies 0.1 higher. The safety calculation according to equation (10) provides a value of 6.7. These results show very good agreement and confirm that the modal analysis constitutes a suitable stability criterion for the evaluation of electrical machines.

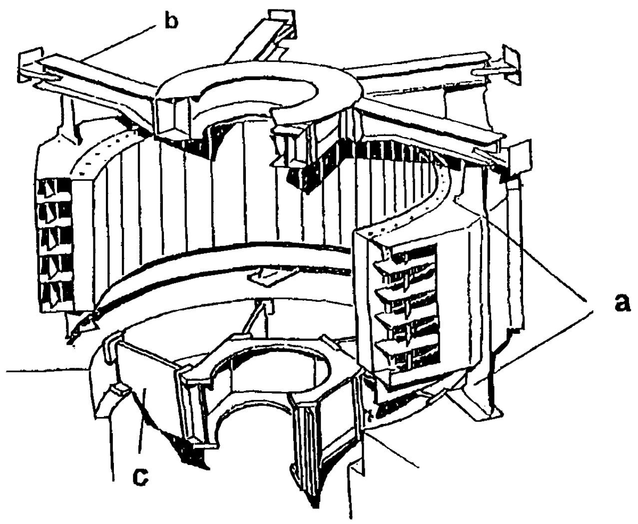
5 Design Requirements for Hydrogenerators

In addition to stability, two further important criteria for the mechanical design of hydrogenerators are the strength and particularly the dynamic behavior during fault conditions. This leads to the following three design requirements:

A) Undesired movements, such as oval deformation or eccentric deflection, should be suppressed.
B) Concentric deformations, which result from thermal effects, should be permitted.
C) The stiffness in the circumferential direction, should be adjustable in order to fix the stator torsional natural frequency.

The first requirement provides the machine with a high stiffness. The result of the second requirement is that the radial forces during warming remain small, since the thermal expansion is not restrained. Finally the third requirement serves to minimize the fatigue loading in the case of out-of-phase synchronization, and hence to protect the foundation.

In the search for a design, which best satisfies the requirements, ABB developed oblique elements. This includes also oblique arms and leaf springs. The forbidden movements generate tensile and compressive forces in the inclined arms. In this loading direction the arms are rigid and suppress the deformations. On the other hand the permitted thermal strains result in a bending load on the oblique arms. However the low level of rigidity does not permit the generation of significant stresses. Oblique leaf springs have been introduced in the stators in order to achieve a good adjustment of the torsional natural frequencies. These are plates which extend in the axial direction and which all adopt the same angle of inclination to the radial direction (Fig.6). The design can be optimized for all requirements by variation of the length, width, thickness and angle of the plates. The oblique leaf springs are drawn through as plates from the foundation through the entire stator to the upper radial support. This design is so simple that it has also been successful for small machines.



a = Connection stator / foundation using oblique columns
b = Upper bearing bracket with oblique arms
c = Lower thrust bracket with oblique arms

Fig.6: Oblique elements for large electrical machines

The design concept of ABB is conceived in such a way that the different loads can be specifically reduced with the help of the FE model, depending on the power level and component size. Fig.7 shows the FE model of an oblique leaf stator for a vertical machine. The stability investigation gave an natural frequency of 39.4 Hz for the four-node vibration. The corresponding mode of vibration is also illustrated in Fig.7. This high value clearly indicates that the stability can be safely ensured through the use of the oblique elements.

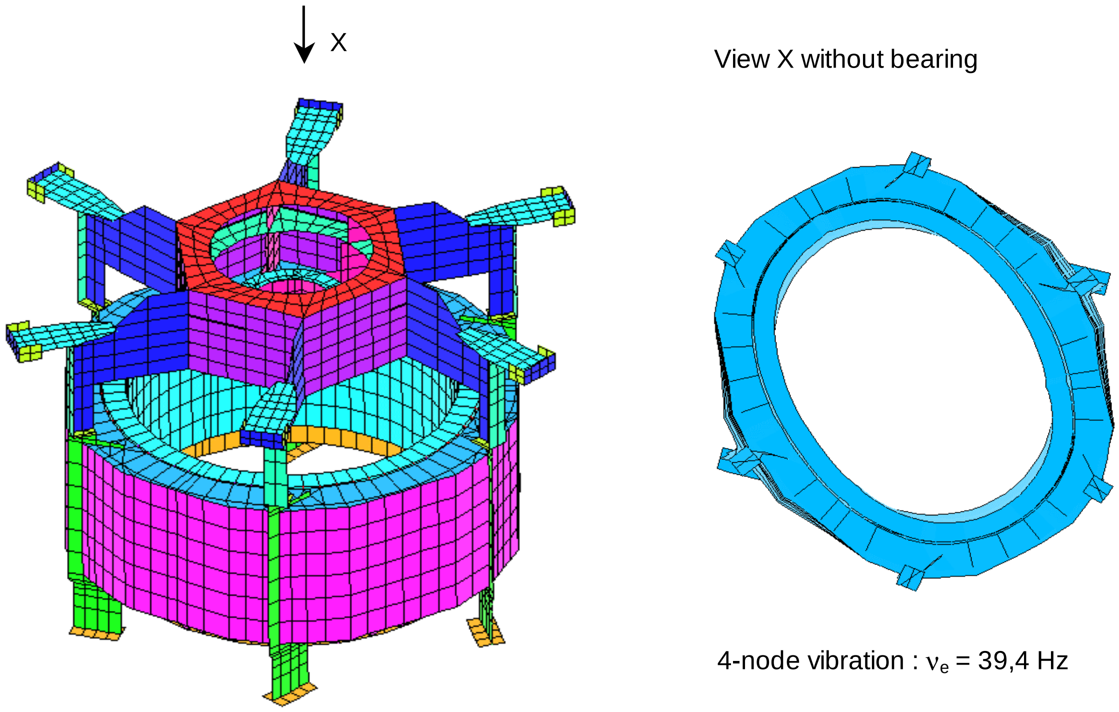


Fig.7: FE model of oblique element stator and mode of the four-node vibration.