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LARGE CONVERTER-FED ADJUSTABLE SPEED AC DRIVES FOR TURBOMACHINES

by

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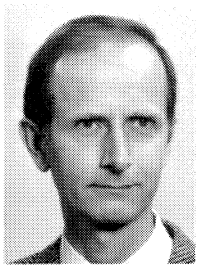
He has published a number of papers and is a member of various technical committees.

Werner Heil graduated (1962) from Darmstadt Institute of Technology, Federal Republic of Germany, with the Dipl.-Ing. degree in Electrical Engineering (power and apparatus).

He joined Asea Brown Boveri in Baden, Switzerland as a Design and Development Engineer in 1963. From 1970 to 1972, he taught design of electrical machines at the Brugg-Windisch Technical College, Switzerland.

Since 1990, Mr. Heil has been General Manager for Sales and Engineering AC machines.



Herbert Prenner graduated (1964) from Vienna University of Technology in Electrical Power Engineering. He joined Asea Brown Boveri, Switzerland, in 1964 and was engaged in development of DC and AC machines for industrial application. Since 1986, he has been Section Manager for Development of the AC-Machines Sales and Engineering Department of ABB Drives in Birr, Switzerland.

In March 1992, Dr. Prenner received the Dr. Ing. degree from Munich University of Technology for a contribution in the field of electromagnetic field calculation.

ABSTRACT

Converter-fed adjustable speed electric motors are now finding increasing use for driving turbomachinery in the power range from 1.0 to 20 (50) MW. The most important properties of these drives are explained, with the main emphasis on the frequency converter, the motor, and especially the shaft line.

INTRODUCTION

Because of ever increasing demands in request to overall process control, full capacity utilization, and high energy charges, the industry finds itself compelled to apply adjustable speed technology instead of fixed speed for machine drives, both for new installations and for retrofits. In particular, converter-fed adjustable speed electric motors are now finding increasing use in driving turbomachinery in the power range from 1.0 to 20 (50)MW. System upgrading can pay back in one to three years of operation.

An electrically adjustable speed drive usually contains a transformer, a static frequency converter, a motor, and the relevant control and monitoring equipment. This package is a high-technology product which must be designed by a team of specialists including experts in electronics, electric regulation, informatics, electrical machines, and applied mechanics.

The most important properties of these drives are explained, with the main emphasis on the frequency converter, the motor, and the shaft line.

STATIC FREQUENCY CONVERTERS

A static frequency converter is a high-efficiency device based on modern power electronics which change the line frequency without the use of moving (wearing) parts. Such a device for a capacity of 10 MW is shown in Figure 1. Static frequency converters are able to produce frequencies higher than that of the supply network (50/60 Hz), and consequently, the motor speed can be higher than the normal synchronous speed of 3000/3600 rpm, up to 7500, respectively, 9900 rpm depending on converter type.

A converter consists of a rectifier and an inverter. The various designs in current use for high speed turbomachinery drives with synchronous motors are:

- line-commutated rectifier with load commutated inverter.

with squirrel-cage induction motors:

- line-commutated uncontrolled rectifier with voltage-source inverter.

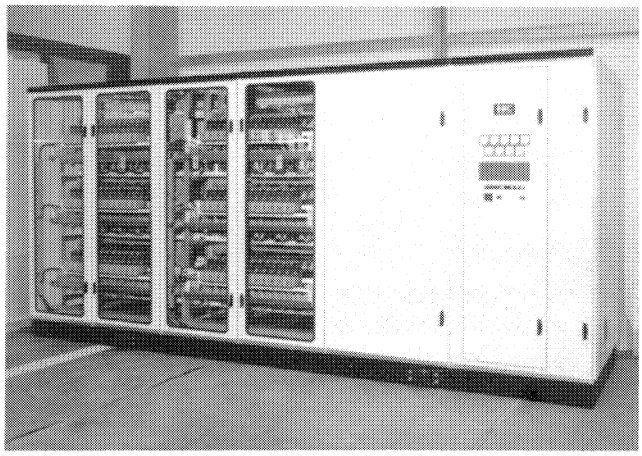


Figure 1. Water Cooled LCI-Type Converter, 10 MW.

· line-commutated controlled rectifier with load-commutated inverter.

· line-commutated controlled rectifier with current-source inverter.

An inverter phase module is shown in Figure 2.

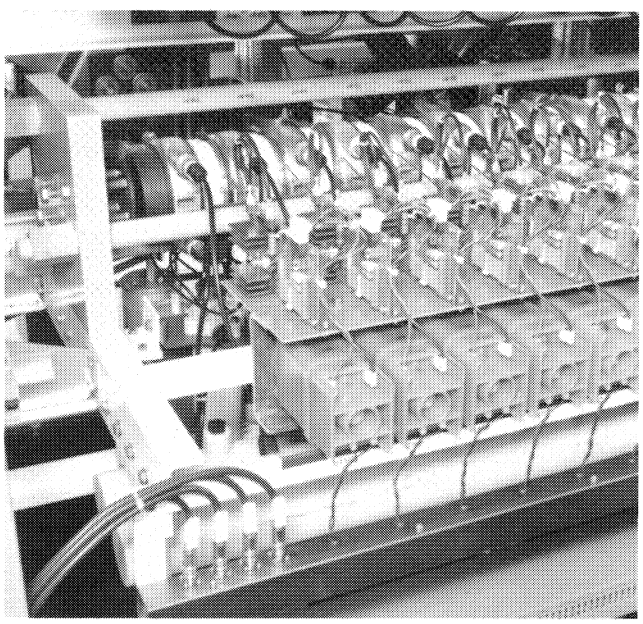
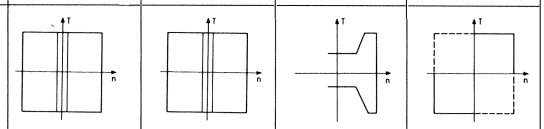
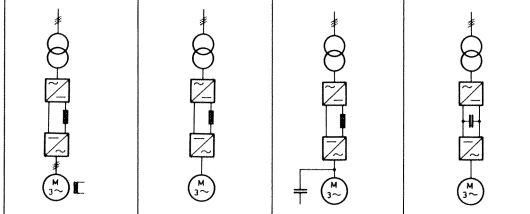
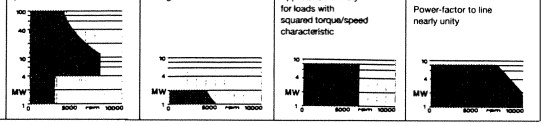


Figure 2. Water-Cooled Thyristor Column with Gte-Pulse Transformers.

The main properties of these drive systems are shown in Table 1. The application limit in power is determined mainly by the converter type, and the limit in speed mainly by the motor design. System diagrams in Table 1 show the general operating principles. In most cases, a transformer matches the line-side supply voltage into a voltage suitable for feeding the rectifier. Thus, the first part in the converter is the rectifier, where the alternating current (AC) power of constant frequency is transformed into direct current (DC) power. In the inverter, the DC power is transformed into AC power with variable voltage and frequency. Depending on the type of converter, the inverter is equipped with phase control thyristors or GTO-thyristors. It must be noted that the alternating current and voltage at the output of the inverter will have only an approximate sinusoidal form, i.e., the so called typical harmonics in current and voltage have to be considered for the performance of the system. A typical output for the load-commutated inverter is shown in Figure 3, and the same in case of a voltage-source inverter is given in Figure 4. A minor amount of harmonics in the output of the inverter is due to the influence of the imperfect rectifier causing the so called atypical harmonics. A reduction of harmonics can be achieved by applying twelve-pulse converter connections, instead of six-pulse connections and a proper design of the intermediate circuit between rectifier and inverter.

Table 1. Block Diagrams for Large Adjustable High Speed Drives with Synchronous and Squirrel-Cage Induction Motors.

Drive system	Converter-fed synchronous motor	Current-source inverter-fed induction motor	Load-commutated inverter-fed induction motor	Voltage-source inverter-fed induction motor (PWM)
Operating range torque/speed				
System diagram Transformer Rectifier D.C.-Link Inverter Motor				
Typical power range	1 000 ... 80 000 kW	up to 2 000 kW	500 ... 8000 kW	800 ... 8 000 kW
Maximum speed	7 500 rpm	5400 rpm	6000 rpm	7 500 ... (10 000) rpm
Typical speed range	(0) ... 10 ... 100%	2 ... 100%	65 ... 100%	0 ... 100%
Significant properties	Single-motor drive of high power High speed possible	Single-motor drive of medium power Maximum motor voltage: 660 V	Single-motor drive with nearly sinusoidal motor current and voltage Applicable normally for loads with squared torque/speed characteristic	Single- or multi-motor drive, Good performance also at low speeds, Power-factor to line nearly unity
1) economic limit				

The fundamental components of voltage and current produce the average torque in the air gap of the motor. The harmonics which unavoidably occur always act in the detrimental fashion, i.e., they raise the electrical losses, they produce pulsating torques in the air gap and thus induce additional stresses in the shaft line and foundation, they generate harmonics in the power system, they produce additional magnetic noise, and they create higher insulation stresses. During the design process for the whole drive, it is possible with particular measures to minimize these disadvantages coming from the converter operational principles. In this way, the user fully benefits from the many advantages that adjustable speed drives offer.

Torque Harmonics and Fault Torques

In order to make a torsional analysis, the torque harmonics in the air-gap of the motor must be known. When the power is supplied through a converter, calculation of the torque in the air-gap of the motor becomes more complicated than when it is supplied directly

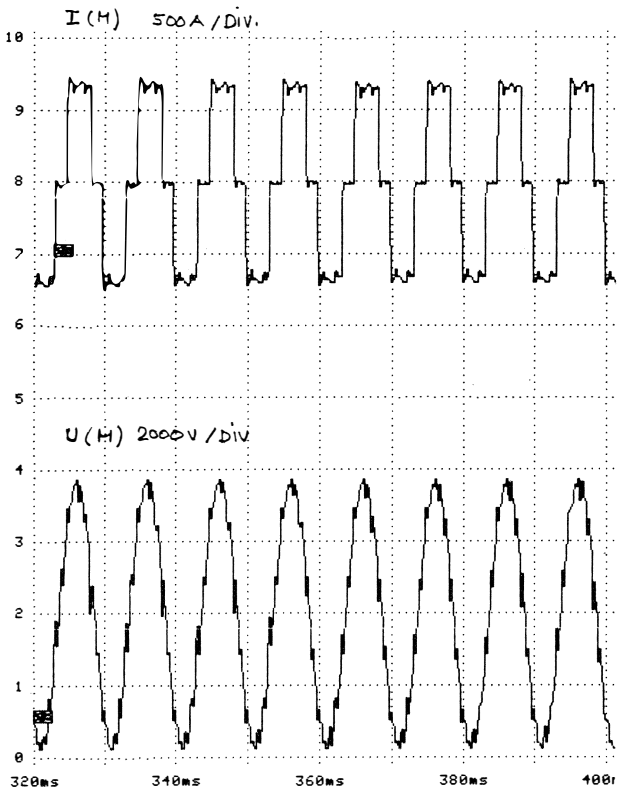


Figure 3. Current (above) and Voltage (below) of a Synchronous Motor Fed by a Load Commutated Inverter.

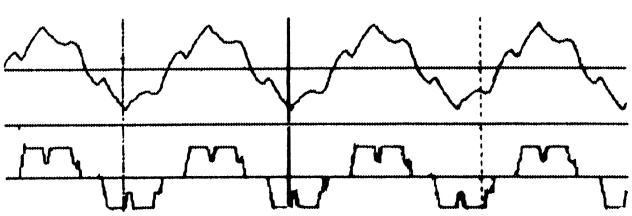


Figure 4. Voltage (above) and Current (below) of an Induction Motor Fed by a Voltage Source Inverter.

from the network. Modern methods for motor torque calculation, which also yield more reliable results, are based on computer programs which can include either all or only one part of the drive, beginning with the supply network and ending with the air gap or with the shaft line torque. Because of the great number of variations in construction details of these electrical power devices and the very fast development which is taking place, there are no "universal" programs. In the programs available, the models of the electrical part of the system are limited to certain types of equipment from a given supplier. For shaft line models, these programs require the use of lumped-mass systems, because they must be consistent with the electrical models applied. From the standpoint of the mechanical engineer, all of the programs worked with apply shaft line mechanical models, which are in some respect insufficient (degrees of freedom, nonlinearity).

In order to better illustrate the effect of these torques on the shaft line vibrations, they will be given here in simple, approximate analytical form. On the other hand, this simplified form of data will be accepted by most of the programs used merely for torsional analysis of the shaft line.

Because of the way the converter operates, the most important part of the disturbing excitation torque in the air gap is produced by the nonsinusoidal components of the motor supply current or motor voltage, respectively. Calculation of these torques is made under the assumption that ideal conditions exist in the DC link, i.e., that the current or voltage, respectively, remains constant.

$$M(n)_{gap} = M(n)_o + \Sigma M(n)_{hk} \sin\ [mk2\pi f_n t + \varphi_k] \quad (1)$$

where

$M(n)_o$ = average torque at speed n
$M(n)_{hk}$ = amplitude of the kth harmonic at speed n
m = 6 or 12 (pulse number of converter)
k = 1, 2, 3 . . . (integrals)
f_n = stator supply frequency at speed n
φ_k = phase angle
t = time

To give an idea of these torque curves, they are shown over the whole operating range in Figure 5 for runup of a synchronous motor having a load-commutated inverter. During steady-state operation in the load-commutated range, the torque harmonics are considerably smaller. With squirrel-cage induction motors having voltage-source inverters, the situation is more complicated, be-

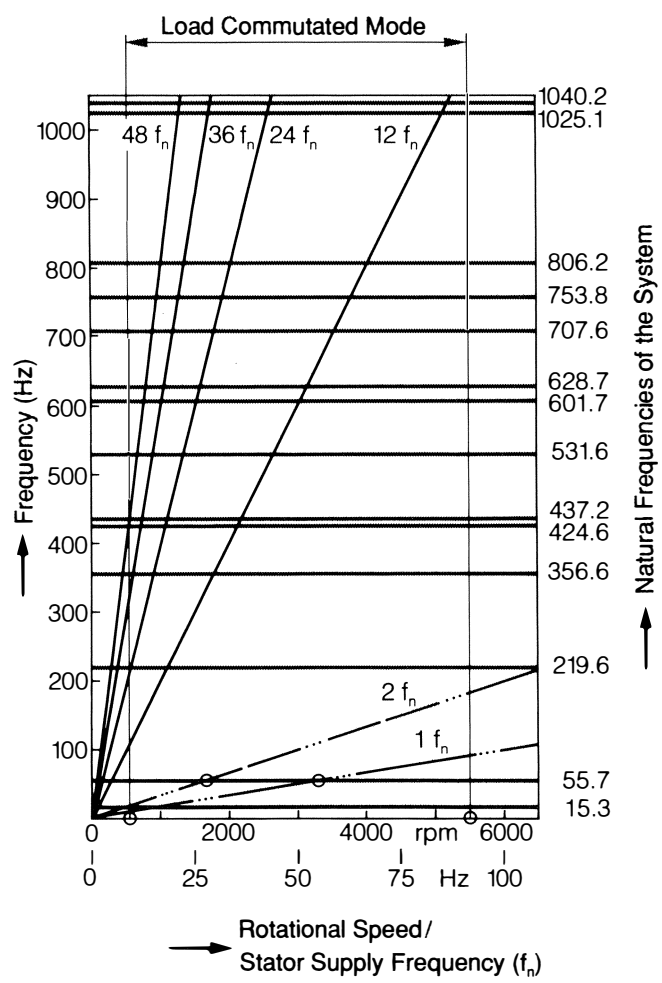


Figure 5. Torque Harmonics during Runup of a Synchronous Motor Fed by a Load Commutated Inverted.

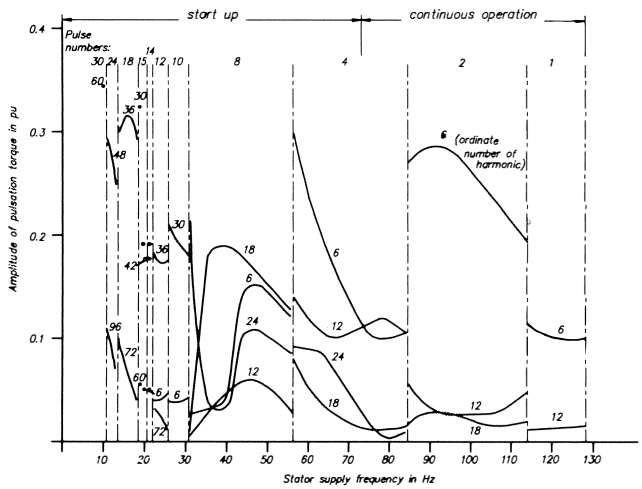


Figure 6. Torque Harmonics in the Whole Operating Range of a Squirrel-Cage Induction Motor Fed by a Voltage Source Inverter.

cause the operating speed range must be divided into several ranges with different control modes, as in Figure 6.

Another source of excitation torques, although not so important, is the pulsation of the direct current or direct voltage in the converter link. These additional torque components are defined in the following equation.

$$M(n)_{add\ gap} = \Sigma M_{ai} \sin\ [m2\pi(if_l \pm kf_n)t + \varphi_i] \qquad (2)$$

where the additional symbols are:

M_{ai} = amplitude of the i th harmonic
f_l = network frequency
i = $\pm 1, \pm 2, \pm 3 \ldots$
k = $0, 1, 2, 3 \ldots\ldots$

The most important component is the torque of 0-order (k=0, i=1).

In the case of voltage-source inverters, switching of the pulse width modulation modes can produce an instantaneous change in the magnitude of the particular harmonics, as can be seen in Figure 6. This shock-like process can be analytically described by

$$M(t)_{pij} = \Sigma M(n)_{ik} - M(n)_{jk}] \exp(-t/T_k) \sin\ [mk2\pi f_n t + \varphi_k] \qquad (3)$$

where the additional symbols are:

$M(n)_{ik}$; $M(n)_{jk}$ = amplitude course on either side of the point of change of voltage pulse numbers
i, j = pulse numbers of two adjacent ranges
T_k = time constant

The converter-fed drive offers some advantages in regard to protection of the shaft line against possible electrical faults in the line-side supply system. The converter causes a remarkable reduction in the effect of these disturbances. To fulfill the security requirements regarding all possible electrical disturbances, the highly improbable line-to-line terminal short circuit will be used as the dimensioning basis. The torque is a multiple of all torques that could arise from the disturbances on the line-side supply system, attaining over the converter the air gap of the motor. The short circuit equation is frequently given by:

$$M(t)_{sc} = M(n)_{sc1} \exp(-t/T_1) \sin\ (2\pi f_n t) - M(n)_{sc2} \exp(-t/T_2) \sin 2\ (2\pi f_n t) \qquad (4)$$

where

M_{sc2} = 0.5 M_{sc1}, the torque amplitude, which is practically independent of the stator supply frequency f_n

T_1; T_2: time constants

ELECTRIC MOTORS

For high-speed drives, the following types of electric machine are generally applied:

- squirrel-cage induction motors with laminated rotors (Figure 7)
- synchronous motors with laminated rotors (Figure 8)
- synchronous motors with solid rotors, as used in turbogenerator design (Figure 9)
- synchronous motors with salient poles (for special purposes).

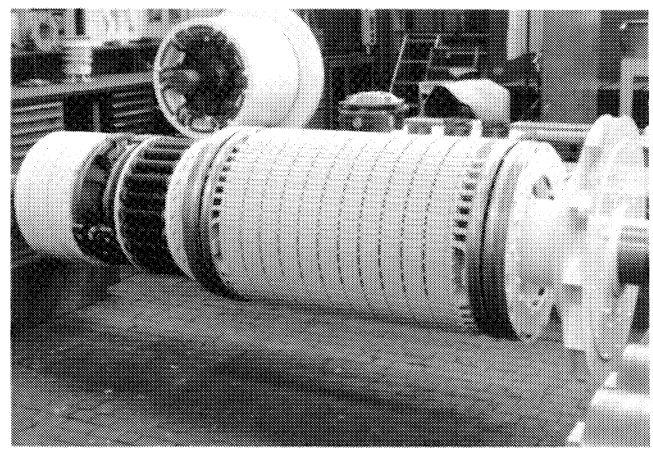


Figure 7. Laminated Rotor Squirrel-Cage Induction Motor.

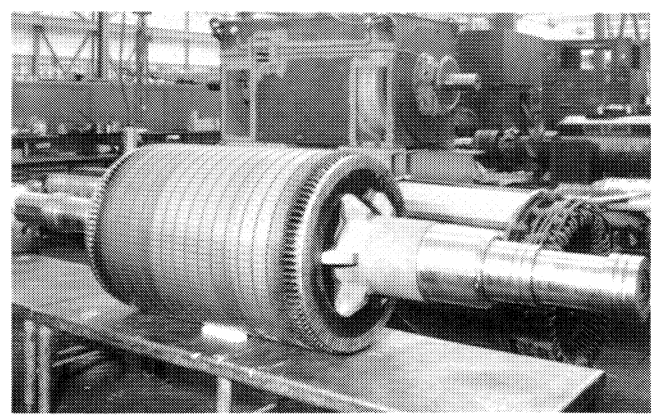


Figure 8. Laminated-Rotor Synchronous Motor, with Exciter.

The synchronous motors generally have a brushless exciter.

In design, these motors do not differ essentially from the machines for fixed-speed operation. For the electrical design of adjustable-speed motors, the following additional items should be considered:

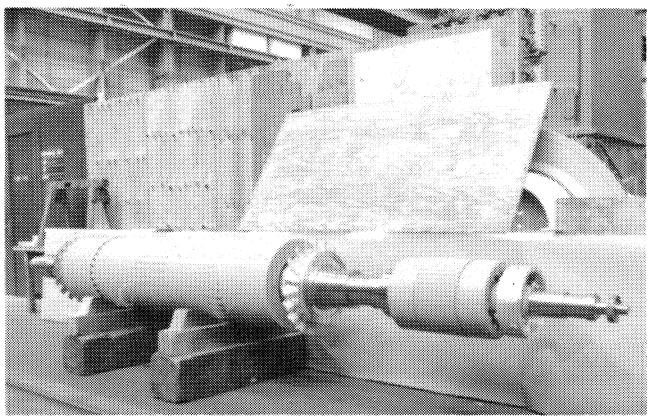


Figure 9. Solid-Rotor Synchronous Motor, with Exciter.

· the motor cooling system, which must be effective over the whole of the speed control range, at the specified loads;

· the harmonics in voltage and current, which lead to higher losses;

· the motor insulation system, which must be selected to withstand the special voltage waveforms, including surges and du/dt effects;

· the motor and converter voltages, which have to be selected so that the entire drive system can be designed to its economic optimum, taking into account all factors such as cable costs and total efficiency;

· the motor reactances, which must be matched to converter operation;

· for synchronous machines, for which the excitation system must be designed to supply a field current at any speed, including standstill;

· the short-circuit conditions, which differ from those for fixed-speed machines;

· the noise level, which also differs from that of conventional AC machines;

· the motor protection and supervision system, which must be engineered specifically for converter operation;

· adjustable-speed AC motors can be designed and certified for operation in hazardous areas.

In regard to the mechanical design, the following problems will be discussed here:

- Critical speed
- Allowable vibration
- Rotor strength

Critical Speed of the Motor Alone

In avoiding steady-state operation at a critical speed, one of the following three situations can exist, depending on the motor size:

- All critical speeds lie above the maximum operating speed.
- The whole operating speed range lies between two critical speeds (e.g., 1st and 2nd).
- One critical speed lies within the operating range; in this case, it must be blocked out (e.g., by not operating in the range of n critical ± 60 rpm).

The problem in trying to shift the value of a critical speed is that the design normally allows only very small changes in dimensions and mass distribution, with the result that the possible shift in the relevant critical speed is rather small (values under 100 rpm). In some cases, a critical speed can be shifted by producing an unsymmetry between the driving and nondriving shaft ends, sometimes also including the bearings. Solutions of this kind have been applied in the authors' turbogenerator at times during the past 10 years.

Allowable Vibration

The vibration level of these machines is evaluated in exactly the same way as with fixed-speed motors. For this purpose, there is a large number of relevant standards and specifications. The documents that the authors are aware of are listed in Table 2. There are, of course, some differences between them, but international efforts towards unification are under way.

Table 2. Standards and Specifications for Allowable Vibration Level of High-Speed Drives $0.3 \leq P \leq 50$ MW.

STATOR SUPPLY FREQUENCY	Hz	15	30	60	120
Harmonic of 6th order	Hz	60	180	360	720
12th order (18th)	Hz	180	360	720	1440
24th order (30th, 36th, 42nd)	Hz	360	720	1440	2880
48th order (54, 60, 66, 72, 78, 86, 90th)	Hz	720	1440	2880	5760
96th order	Hz	1440	2880	5760	11520

Frequency values beyond the normal confidence range of calculations

In regard to the measurement of vibration, the largest differences in practice are found between the motor manufacturers and the compressor manufacturers. The vibration level of a motor is normally judged by bearing housing measurements. For compressors, it is mainly shaft measurements that are used. The difficulty is that there is no simple correspondence between the absolute bearing housing vibration and the relative shaft vibration. On the other hand, the acceptability of a vibration level is mainly a matter of experience. In many cases, a Salomon's solution is accepted and both types of vibration probe are installed on the motor.

The vibration behavior of motors is generally good, as the balancing procedures for "elastic" shafts are well developed. Sometimes, difficulties can arise if the base frame is not stiff enough, in which case a "horizontal critical" may be found in the operating speed range, detectable as an increase in vibration amplitude. This "horizontal critical" is not what the authors call a shaft line critical speed, but is rather a natural frequency of the structure composed of the motor housing and rotor, resting on the flexible spring defined by the base frame design. The solution is to apply appropriate stiffening of the base frame.

Rotor Strength

Because of the increase in speed above the usual "synchronous" level of 3000/3600 rpm, the stresses in the rotor and its components due to the centrifugal forces will also increase. This means that the maximum allowable speed for an adjustable-speed motor is determined by the material and the design of the rotor. For solid rotors, material can be obtained with a maximum yield point of 800 MPa. For the pertinent rotor end-winding retaining rings, steel manufac-

turers offer material having a yield point of 1000 MPa. The situation for electric steel sheet is not so good; the guaranteed values lie between 300 to 400 MPa. Checks on material as received show that this value is exceeded very often. If one could be selective, it would be possible to build rotors using sheet having higher allowable stress levels. Because of the small tonnages involved, however, steel suppliers will not accept orders for such better material.

The strength of the copper alloys for windings and for rings for cages can also dictate the speed limit.

Because of the strength and design considerations relating to rotors for supersynchronous machines, the resulting speed-power limit curves are not smooth, and consequently, the purchaser's project engineer would be well advised to consult the motor supplier before fixing the maximum speed. It is sometimes possible to deliver more than what is stated in the standard brochures.

On the other hand, if a drive with a gear unit is unavoidable, then it is better to choose a supersynchronous motor with a speed that lies below the highest possible speed. This does not present any difficulty in gear unit design, but has the great advantage that the motor can be of standard material and standard design.

SHAFT LINE

Turbomachinery drives are usually planned by a project team or consultant, but the components are supplied by different manufacturers. The essential technical relationships can be seen in the diagram (Figure 10), which shows such a drive with the relevant interconnections. Examine such a situation, taking the example of a boiler feed pump drive as shown in Figure 11. This drive consists of a high-pressure boiler feed pump, a gear coupling with a floating spacer, a synchronous motor with a solid rotor, a second gear coupling, a reduction gear, a gear coupling with a floating spacer, and a double-suction booster pump. In this case, the shaft line has been built by three different suppliers. To make the necessary rotordynamic calculations, a large amount of machine data must be exchanged. The time required for this exchange can take at least one month, or up to about six months when the preparations by the project team are insufficient. The authors have found it best to fix the information exchange channels before the work begins, in a

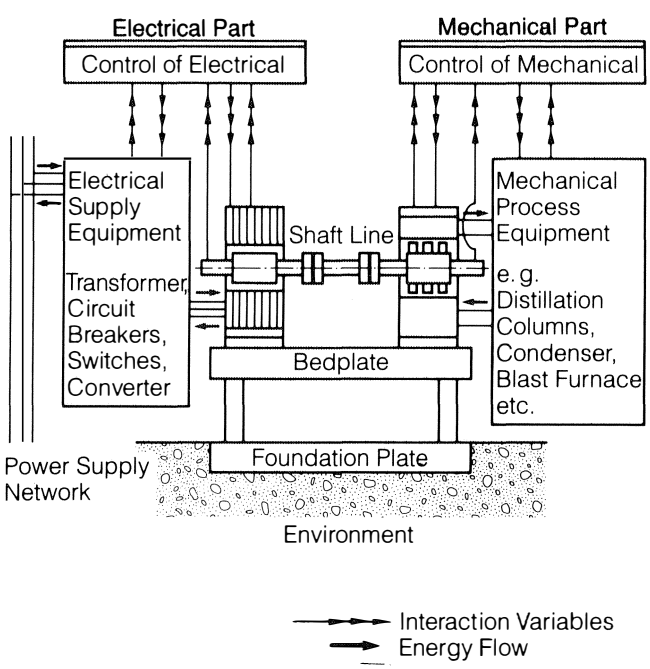


Figure 10. Arrangement of a Drive with Relevant Interconnections.

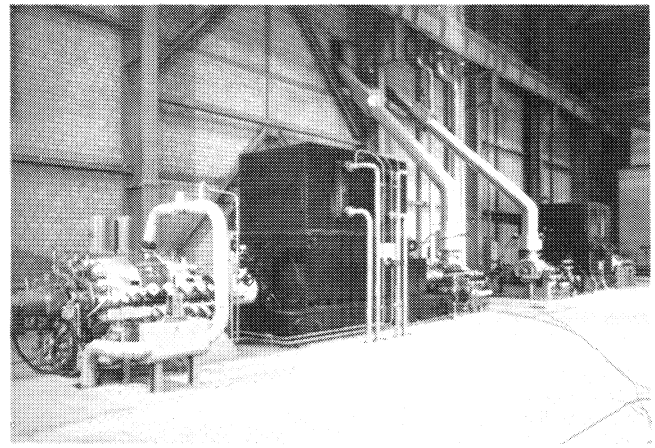


Figure 11. Boiler Feed Pump Drive, 10 MW/6000 RPM.

personal meeting of all those who are directly involved, i.e., the design and project engineers. It is essential in all cases that the technical representatives of all subcontractors supplying the couplings, gear units, torsion measurement systems, etc., are present from the beginning.

Rotordynamics calculations can be made only for a mathematical model of the shaft line. This means that the model must include all of the important vibration properties of the actual machine. The relevant physical properties for the shaft line model are mass, mass moment of inertia, stiffness, and damping distribution. Various rotordynamics programs can determine these principal properties from different sets of input data; e.g., the element stiffness input can be given either as a diameter or as a moment of inertia of the section. The most common input data set includes the following: the geometry (dimensions) of the shaft element, the material properties (including density, Young's modulus, Poisson's ratio or shear modulus, and damping coefficient), and the additional masses that are mounted on the element (mass, inertia diameter). For the kinds of drives discussed here, the model is constructed with from 50 elements, for very small drives, to about 250 elements for large ones. If the input does not differentiate between the element dimensions for mass and stiffness, and the sections being dealt with are not exactly round, faults will occur in the model description.

In constructing such a shaft line model, a number of uncertainties in regard to the mass and stiffness distribution will have to be resolved (e.g., force flow in shrink fit couplings). In extensive experience, it has been found that this problem can best be solved using the Method of Two Limit Cases (TLC-Method), see APPENDIX. When dealing with torsional vibration, it is much easier and the deviations between the results of independent sets of calculations will be reduced, when one defines the possible limit values for stiffness and mass instead of trying to guess their actual values. In this way, the investigator can straddle the frequencies and torques that arise in the system examined. The confidence value of such calculations is very high.

Flexural Vibration

The model for flexural vibration of the shaft is composed of two main parts: the shaft line and the bearings. The model for the bearings usually has three parts: the running surfaces and the "representative" so called modal mass and modal stiffness for the bearing support structure. The running surface for journal bearings is formed by the oil film. Nowadays, its model is given by a (2×2) stiffness and a (2×2) damping matrix. In the case of antifriction bearings, there is just an isotropic stiffness and a (small) damping coefficient.

The normal scope of flexural vibration calculations covers

- critical speeds (real eigenvalues),
- unbalance response, and
- complex eigenvalues.

The most important features concerning critical speed were reported under *Critical Speed of the Motor Alone.*

The unbalance response calculation must show the sensitivity of the shaft to unbalance over the whole operating range. For test unbalances, Grade G 6,3 is normally applied, as per the standard ISO1940. This is about two grades worse than what is actually reached in our rotors with the usual balancing procedure. The balancing grades for 6000 rpm are shown in Table 3.

Table 3. Balancing Grades for 6000 RPM, Per ISO Standard 1940.

ISO1940 BALANCING GRADE	6000rpm UNBALANCE gmm/kg
1.0	1.6
2.5	4.0
6.3	10.0

The complex eigenvalues are required in order to determine the running stability limit. This is a well-known effect with compressor rotors, but has the same significance for motors running at supersynchronous speeds.

Experience shows that these calculations must be made with a model for the whole shaft line. If the shaft line includes a gear unit, it is permissible to divide it into two parts: the model for the slow-running shaft and the model for the fast shaft. It is not permissible to make a division at any point modelled as a joint (e.g., a gear coupling); the reason for this will be given in an example at the end of this section.

The results of complex eigenvalue calculations depend largely on the oil film coefficients and to an equal extent on the often neglected seal coefficients of turbomachines. The latter coefficients are only starting to become introduced, but for high-pressure machines they are indispensable. Oil-film coefficients either can be calculated with the Reynolds equation, applying different assumptions, and different boundary conditions, or can be measured on a test rig. Both methods have certain limitations. To be on the safe side, in this case, faithful application of the TLC-Method is recommended once again. In the most usual present day approach for calculating oil film coefficients, they are given only as functions of the loading number for the bearing (in Europe, its reciprocal value is called the Sommerfeld Number). The calculations are made with two limit values for the loading numbers. The loading number, Sommerfeld Number

$$So = f\ (F, \delta, D, B, \tau, \omega) \qquad (5)$$

is a function of static load F, clearance δ, bearing diameter D, bearing width B, oil viscosity η and angular velocity ω. In order to more easily interpret the physical sense of the differing values of the Loading Number, only these factors are varied:

- static bearing load (max 1.6 F to min 0.4 F)
- clearance (max-min, in accordance with the tolerance range),
- oil viscosity (min-max).

The speed must always be taken at its actual value, because this is what defines the application point. For the viscosity variation, consider the influence of the possible spread of the viscosity index as per standard ISO 3448, and the influence of the extreme values of the oil inlet temperature. The static bearing load variation should take into account the load shifts due to changes in running conditions, especially in the case of shafts with statically indeterminate conditions (more than two bearings). The load variation cannot be defined quite as precisely as some other variables, therefore a safety margin can and should be included. This involves no difficulty, and the implications of any force variation can be more realistically evaluated than if the allowance were added to any other variable.

A very impressive example is provided by an incident that happened in May 1987. The subject was an adjustable speed drive of 13.4 MW, n max = 6400 rpm, for an ethylene compressor being commissioned in a petrochemical plant. The block diagram of the machine set is given in Figure 12. The motor was connected to the low-pressure compressor through a membrane twin coupling with a long, slender, floating spacer-shaft. This torsionally soft shaft was required in order to reduce the stresses in the shaft line during runup. The calculations of critical speeds, unbalance response and complex eigenvalues were acceptable, and indicated no cause for concern. The oil-film coefficients for the motor bearings were determined, as usual at the time, for mean values of bearing load, clearance and viscosity. After about two weeks of test runs, the shaft failed abruptly, as shown in Figure 13. After a thorough investigation, accompanied by extensive vibration measurements, the final solution was found: an auxiliary bearing was added to the floating shaft and the membrane coupling was replaced by a stiff flange coupling. Since then, the machine has been running satisfactorily, more or less continuously.

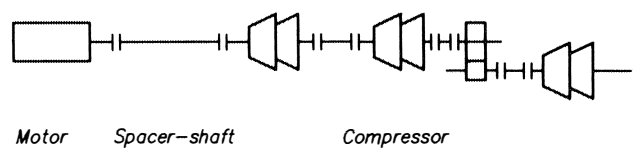


Figure 12. Block Diagram of a Compressor Drive, 13.4 MW/6400 RPM.

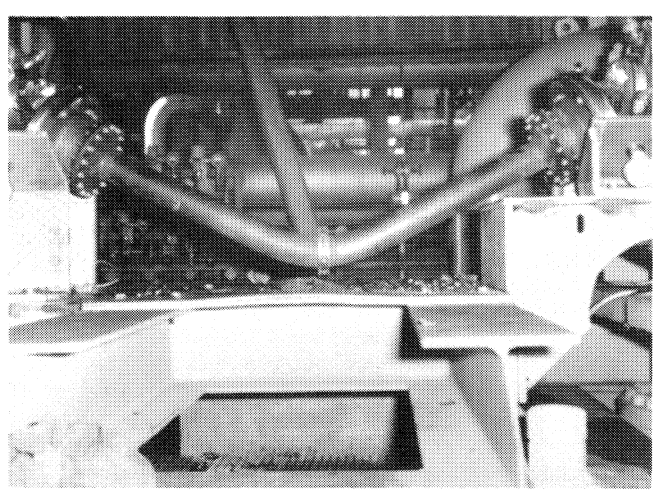


Figure 13. Damaged Floating Shaft of a Compressor Drive, 13.4 MW/6400 RPM.

The theoretical research into this incident has shown that in such calculations, not only the complete shaft line, but also the possible scatter of the values for the oil-film coefficients must be taken into consideration. The TLC-Method was applied, as explained earlier, and the conclusion was that, in this case, the nonconservative oil-film forces had given rise to a latent instability that manifested itself through the kinematically weakest link: the floating shaft. The complex eigenvalues calculated for the two limit values of oil-film coefficients for a speed of 5400rpm (maximum operating speed 6400 rpm) given in Table 4 show the begin of the possible instability rise due to (positive) negative damping coefficients. In this case, it was not necessary to replace the existing four-lobe bearings by tilting-pad bearings.

Table 4. Complex Eigenvalues for a Compressor Drive, 13.4 MW/6400 RPM. Eigenvalue calculations with Hessenberg-Method. Gyroscopic-effect: Rel. 5400 rpm; maximum number of eigenvectors: 30; right eigenvectors are calculated frequency for calculation: 300 Hz; equilibrium control for 30 eigenvectors, list of all 248 eigenvectors, list of purely real eigenvectors (imaginary part = 0), list of complex conjugate eigenvalues.

```
EIGENWERT-BERECHNUNG MIT DEM HESSENBERG-VERFAHREN

KREISELWIRKUNG:   REL. DREHZAHL = 5.4000E+03

MAXIMALE ANZAHL DER EIGENVEKTOREN  :  30
ES WERDEN DIE RECHTS-EIGENVEKTOREN BERECHNET
FREQUENZ-GRENZE FUER DIE AUSWERTUNG:   300.000 HZ
GLEICHGEWICHTS-KONTROLLE FUER   30 EIGENVEKTOREN

LISTE ALLER 248 EIGENWERTE

LISTE DER REIN REELLEN EIGENWERTE
      NR       REALTEIL

       1     -28976.347
       2     -24499.504
       3     -23810.279
       4     -20932.460
       5      -3798.798
       6      -2494.565
                 α          ω
LISTE DER KONJUGIERT KOMPLEXEN EIGENWERTPAARE
      NR       REALTEIL     IMAG.TEIL

       7     5.4203E-01   3.2114E+01
       9    -1.3125E+01   9.1923E+01
      11     8.5254E+00   1.2110E+02
      13    -4.1776E+00   1.2564E+02
      15    -3.4356E+01   1.4304E+02
      17     1.7114E+00   1.6994E+02
      19    -4.3336E+01   2.5397E+02
      21    -1.1507E+01   3.1528E+02
      23    -1.2940E+01   3.2940E+02
      25    -9.7595E+00   3.9884E+02
      27    -3.3943E+01   4.7184E+02
      29    -9.6008E+00   4.8755E+02
      31    -4.2478E+01   5.9716E+02
      33    -9.6237E+02   5.9947E+02
```

EIGENVALUE

$\lambda = \alpha \pm i\,\omega$

CALCULATION FOR FLEXIBLE OIL FILM

```
LISTE ALLER 248 EIGENWERTE

LISTE DER REIN REELLEN EIGENWERTE
      NR       REALTEIL

       1    -105331.823
       2     -88835.242
       3     -79336.756
       4     -45003.147
       5     -42399.098
       6     -39591.327
       7     -37789.209
       8     -37188.461
       9     -29962.824
      10      -8849.825
      11      -2582.858
      12      -2094.032

LISTE DER KONJUGIERT KOMPLEXEN EIGENWERTPAARE
      NR       REALTEIL     IMAG.TEIL

      13    -1.8771E+00   3.9690E+01
      15    -1.6256E+01   9.9541E+01
      17    -4.4237E+00   1.2671E+02
      19    -3.3851E+01   1.4864E+02
      21    -4.4093E-01   1.8841E+02
      23    -2.6857E-01   1.9928E+02
      25    -8.6816E+00   3.4161E+02
      27    -5.3571E+00   3.6827E+02
      29    -1.4601E+03   3.7896E+02
      31    -7.3505E+00   4.0018E+02
      33    -6.3966E+00   4.4243E+02
      35    -2.4440E+03   4.7647E+02
      37    -2.5264E+03   4.8163E+02
      39    -2.8299E+00   5.2106E+02
      41    -1.0340E+01   6.1075E+02
```

CALCULATION FOR STIFF OIL FILM

Torsional Analysis

The motor/converter supplier should advise the torsional analyst of all excitation torques having amplitudes above three percent of the rated torque. According to present knowledge, this could also include harmonics of the 120th order. Numerical values of frequencies for some selected orders of harmonics arepresented in Table 5, and the magnitude is shown of the frequencies which could arise and which should be considered in the calculations. The consideration of such high frequencies also requires the use of much larger models than what until now has been considered usual and necessary. The first question which arises is: how accurate is the torsional theory of deSaint-Vénant as presently applied in this frequency range above some hundreds of Hertz. Another question concerns the size of lumped-mass models. In torsional analysis,

Table 5. Frequency Values for Selected Harmonics.

BEARING HOUSING VIBRATION		SHAFT VIBRATION	
ELECTRIC MACHINES	TURBO-MACHINES	ELECTRIC MACHINES	TURBO-MACHINES
VDI 2056 Gruppen G/T ISO 2372/1974 Class III/IV ISO 3945 DIN/ISO 3945 IEC 34-14	VDI 2056 Gruppen G/T ISO 2372/1974 Class III/IV ISO 3945 DIN/ISO 3945	VDI 2059-3 ISO/DIS 7919-3	VDI 2059-3 ISO/DIS 7919-3
API 541 API 546		API 541 API 546	API 611 API 612 API 617
NEMA MG1-20.52			
under way ISO 2372-3	under way ISO 2372-3		

lumped-mass models permit the solution of the equations of motion by numerical integration. Numerical integration permits the consideration of variable mass, variable stiffness, variable damping, and variable frequency and amplitude of excitation. The problem in numerical integration, however, is the selection of step size and the number of these integration steps to be performed. The step defines the amount of calculation required and the storage capacity of the computer, but the crucial point is: how accurate are the results after accounting for the accumulation of minute errors in each step? These errors result from the absolute step value and the limitation in numerical capacity of a computer, also when double (fourfold) precision is applied. The error estimation is a matter which in practice can be evaluated only in relative terms. Finite-element programs which use modal analysis do not have this problem, but they are not yet capable of taking into account the nonlinearities that could occur in the systems being studied.

In view of what has been said, the authors consider that the limit for a system applied for analysis is not more than 20 masses, but with about 1300 Hz maximal natural torsional frequency.

The frequencies and the mode shapes have been calculated for a system of 20 masses, constructed for a shaft having continuous, equal mass and stiffness distributions over the whole length. The calculations were made in parallel, with both a finite-element program and a lumped-mass program. The comparison with known theoretical results shows clearly that after the first 10 eigenvalues, the accuracy of frequencies and mode shapes was diminishing. Theoretical proof of this result can be found in the literature.

Torsional analysis should be made whenever operating conditions are such that the torsional excitation is greater than three to five percent of the rated torque. Because of the operating principles of currently used converter types, an analysis must be made for all three important operation modes, namely

- runup,
- steady-state operation,
- electrical faults.

Mechanical disturbances such as those which occur in turbomachinery could also be added to the list.

During runup, the excitation frequency and amplitude change with time. Very few of the available torsional analysis programs have algorithms which are adequate to take this into account. Until now, the authors have almost always calculated that the most important stresses in the shaft line were induced while running through the first torsional natural frequency. Torque peaks of up to about 70 percent of rated torque were accepted. A runup procedure for the drive for a compressor test rig, 18 MW 5500/20132 rpm, is

shown in Figure 14. During runup, a certain amount of hammering of the teeth in the gear cannot be avoided.

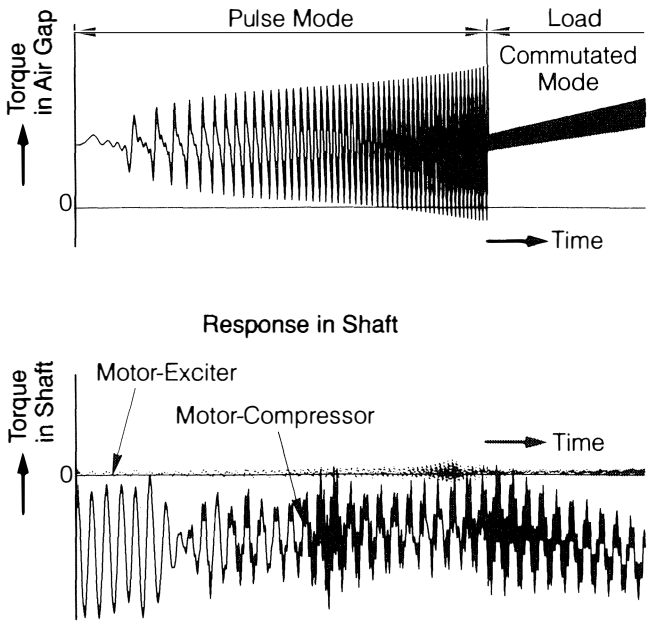


Figure 14. Runup of a Compressor Test-Rig Drive, 18 MW/5500 RPM.

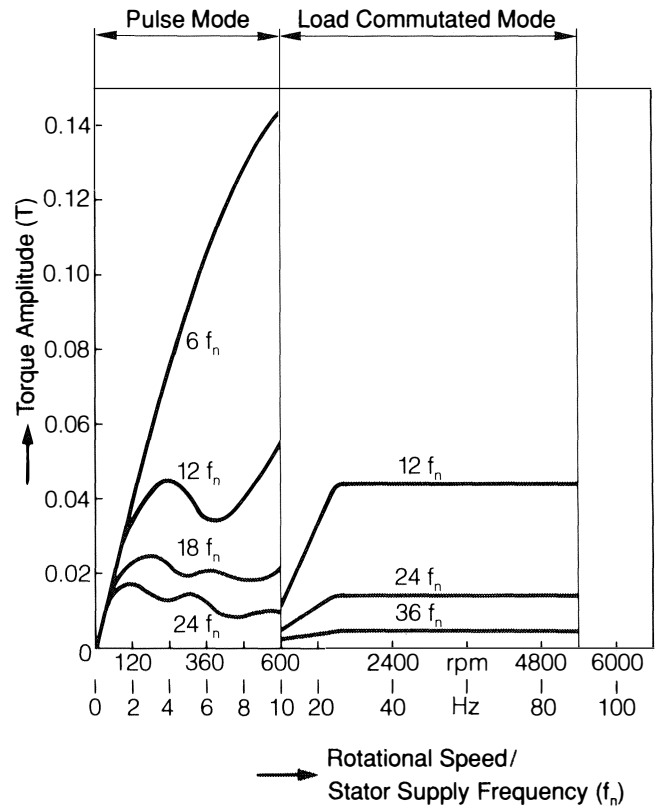


Figure 15. Campbell Diagram for the Whole Operating Range of a Compressor Test-Rig Drive, 18 MW/5500 RPM.

With adjustable-speed drives, steady-state operation brings an absolute novum in machine design, namely steady-state operation in torsional resonance. The Campbell diagram in Figure 15 shows that it is practically impossible to build a variable-speed machine that would not produce resonances. The lucky circumstance is that in the operating range, it is possible to have only those resonances having higher-order natural frequencies. Normally, these frequencies do not have very large amplification factors. The calculation can be made as for steady-state vibration, as has been done for years in internal-combustion engine calculations, or as for a transient vibration in resonance, until the steady state is reached. The largest resonance amplitude calculated up to now has not exceeded 30 percent of the rated torque. The damping considered was Q = 100 for shafts and, in the case of elastomeric couplings, the magnification factor of M=4 to 10 as given by the coupling supplier was used.

The next problem in steady-state operation concerns running at, or crossing over of, pulsing-number limits by drives with squirrel-cage induction motors fed by voltage-source inverters (Figure 6). Here the calculation could be done with a fixed frequency and only with a sudden amplitude change, as can be done with many torsional programs. When making calculations for the running-through, taking into account the change in frequency, the hysteresis in the process can also be included.

Finally comes the consideration of the phase-to-phase short circuit. This is a transient phenomenon that most torsional programs can deal with. The problem is only the variable stator feed frequency, which makes it necessary to calculate various operating points. These lie at the beginning and the end of the operating speed range, and at all possible resonances with the terms $1f_n$ and $2f_n$ (see also Figure 15 and Equation (4)). Sometimes, when the calculation points are concentrated at each end of the range, intermediate frequencies should also be examined. The torques in the shafts can be rather high, especially when the driven masses are higher than the motor mass. These high torques must be evaluated in the light of the probability of occurrence. This brings us to the last technical aspect to be considered here—the recommendations for sizing the shafts.

In sizing the shafts, three groups of events can be distinguished:

- Events that can occur continually, i.e., an "infinite" number of times. In this case, the shaft must be sized for the fatigue limit.
- Events that can occur only a certain predictable number of times. Sizing then depends on the fatigue strength for a finite number of load changes.
- Events that can occur only under extraordinary conditions, perhaps only once or a few times in the life of the machine. The shaft is then sized in relation to the yield point or material strength.

CONCLUSIONS

An outline has been presented of problems and solutions concerning the mechanical calculations in connection with electrically adjustable high-speed drives in the power range of 1.0 to 20 (50) MW. This work reflects the experience of the authors over a period of about 20 years, based on a total of more than 30 different vibration calculations made for motors which have actually been manufactured, representing a total capacity of over 800 MW.

APPENDIX

Method of Two Limit Cases (TLC-Method) As Applied in Torsional Analysis

Assumptions for Model Construction

- The shaft line model will be represented by a series of shaft elements.

· A shaft element is a shaft part having constant stiffness and mass distribution

· Element length for FE-program input: length ≤ equivalent stiffness diameter ≤ 250 mm

· Two inputs to describe mass quantity: largest and smallest possible values

· Two inputs to describe stiffness: one as flexible and one as stiff as possible

· Conservative values of damping

· Upper limit values of excitation, electrical and mechanical

Natural Frequency Calculation

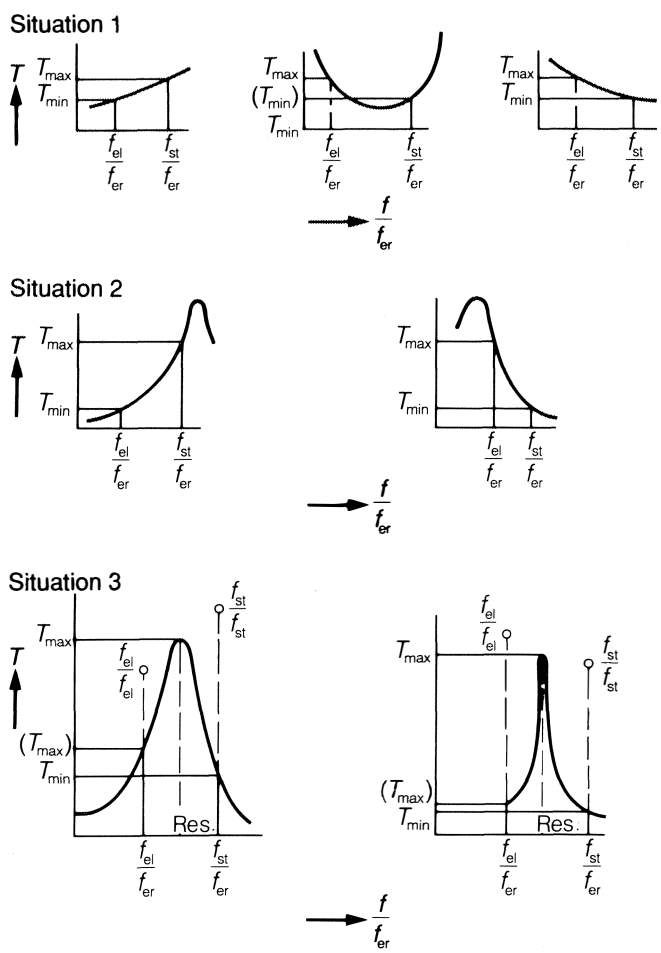
Natural frequencies are calculated for two models:
Lower limit: mass: largest possible value, stiffness: as flexible as possible
Upper limit: mass: smallest possible value, stiffness: as stiff as possible

Torsional Response Calculations

Response calculations are made for the two models used in the calculation of natural frequencies. Three situations can arise (Figure A-1):

Situation 1: Exciting frequency sufficiently remote from the natural frequency $\Delta T_1 = T_{max} - T_{min}$

Situation 2: Exciting frequency near the natural frequency $\Delta T_2 = T_{max} - T_{min} \approx$ (2 to 3) ΔT_1



Situation 3: Exciting frequency straddled by a natural-frequency scatter-band $\Delta T_3 = T_{max} - T_{min} \approx$ (4 to 7) ΔT_1
(it is easier to make resonance calculations by changing the excitation frequency instead of the natural frequency of the model, i.e., $f_{er} = f_{el}$; $f_{er} = f_{st}$), f_{er}: excitation frequency, f_{el} to f_{st}: natural frequency scatter-band, T: torque in the shaft

REVIEW OF THE TLC-METHOD

Advantages

· Assumptions for stiffness are simple and clearly interpretable.

· The available spread of the mass moment of inertia is covered.

· Actual natural frequencies will be straddled.

· Torques in the shaft parts are defined limit values.

· For cumulative-damage calculations, the higher of the calculated shaft torques has a very high confidence limit.

Disadvantages

· Two parallel calculations must be made for each load case. When a resonance point is straddled, two calculations are made, using the natural frequency as the exciting frequency.

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