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Measuring Torque Correctly

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Contents

1 Introduction

Though torque is unquestionably an important mechanical quantity in the field of machine building, its significance is not confined to that area alone. The precise measurement of torque, particularly that which occurs in rotating components, places heavy demands on manufacturers and users of test benches. The situation is further complicated by the trend towards improving the mechanical performance of modern engines by increasing their speed of revolution, coupled with a desire for greater accuracy in such areas as the measurement of efficiency.

This challenge is met by continuous development taking into account the ongoing advances in the application fields. Whilst torque shafts according to the original design principle are still used for certain applications, the full range of transducers now includes torque measurement hubs and torque flanges. Innovations in contactless torque transducers concern the transfer of power from the stator to the rotor and the transmission of measurement signals.

But even the most advanced measurement technology can only show its strength when specific rules are followed. This book is a comprehensive revision of the 1989 HBM publication "The Proper Use of Torque Transducers". It gives an overview of important aspects concerning the use of torque transducers and provides a source of reference for resolving issues affecting applications.

The information given can be adapted and applied by everyone who uses torque measurement devices. On the other hand it is not possible to put forward suggested designs for highly specialized problems. There are many torque measurement tasks which can be solved only after the problem has been clearly defined and all parameters have been taken into account. However, this does not come within the duties of a component supplier. This book can therefore give no assurances about specific characteristics or fitness for purpose in the legal sense, and no responsibility can be accepted for the use to which products are put.

This book describes the principal methods of torque measurement with particular reference to the mechanical and electrical configuration of torque transducers based on the strain gage principle (also called the SG principle) which are those most commonly in use at the present day.

The main fields covered by this publication are selection criteria, the environment within which applications operate, installation, startup, vibration analysis, calibration, and the metrological principles applicable to measuring with torque transducers. It should be especially beneficial to those readers who do not have much practical experience in torque measurement.

Profound theoretical discussion has been avoided. However, for those approaching the subject afresh an appendix sets out the technical terms for the specification of torque transducers. There is also a brief outline of vibration engineering, together with the most important relationships in the form of tables and a collection of equations complete with short explanatory notes.

1.1 The significance of torque as a measured quantity

In a highly mechanized world, torque is among the most important of all the measured quantities. It plays a highly significant role not only in such products as gas turbines with 50 kN·m of nominal torque at 8000 min^{-1} and a mechanical output of over 40 MW, or Formula 1 test benches with nominal torque in the range 1 to 2 kN·m at 20,000 min^{-1}, but in fact in virtually everything including screw caps on medicine bottles. And for many products the permitted tolerances are mandatory.

There are countless applications for torque measurement in test bench engineering, process monitoring and control, drive and conveyor engineering, quality assurance and R&D.

Recent years have seen rapid market growth. Faced with consumer demand for vehicles which offer lower fuel consumption, higher levels of comfort, greater operating safety and longer-lasting reliability, the automobile industry is highly oriented toward innovation. The industry's requirement for metrological and test techniques to match this demand has therefore grown and is growing. This trend is being accelerated by ever stricter legal requirements for lower emissions.

Increasing importance is being attached to acquiring relevant data reliably and reproducibly. Torque is the key quantity in all investigations and refinement operations, particularly for the development of internal combustion engines and transmissions since, in combination with rotation speed, it provides the possibility to calculate mechanical power. Whereas at one time, particularly in the case of engine test benches, this measuring task was fulfilled by the use of braking devices with a measurement capability, nowadays the trend is toward in-line torque measurement with the aid of rotating torque transducers.

The main reasons for this are that the processes are always dynamic and the interplay between mechanisms such as the engine and the transmission is becoming an increasingly important consideration when it comes to optimization. And in the matter of the torque transducers used in power and functionality test benches, HBM is the worldwide market leader.

HBM has over fifty years of experience in the electrical measurement of mechanical quantities. Production of the first transducer for measuring the torque in a rotating shaft train began over forty years ago. Fig. 1.1 shows a first generation torque shaft in comparison with modern torque transducers. Even today first generation transducers are still being sent to HBM for testing, overhaul or calibration, having been faithfully carrying out their tasks for more than thirty years. This is testimony to the quality and durability of HBM products.

Fig. 1.1 Different generations of torque transducers

HBM was accredited as a DKD calibration laboratory for the measured quantity force as long ago as 1977. This made HBM the first calibration laboratory to be accepted into the DKD (German Calibration Service). Accreditation for the measured quantity torque followed on July 13th, 1990. For many years HBM was the only calibration authority for torque in Germany and practically set the national standard.

HBM now offers calibration steps from 2 N·m up to 20 kN·m which is the widest range available in the DKD. The equipment used possesses an extremely high level of accuracy thanks to mass-lever systems in which the force is directly generated by the action of a mass in the earth's gravitational field.

As a manufacturer of precision measuring instruments and also of sturdy industrial transducers, HBM takes its responsibilities for quality and reliability very seriously. Logically it was just a short step to a quality management system meeting the requirements of the relevant standards. In 1986 HBM was the first company in Germany to be accredited in accordance with ISO 9001. Then in 1996, in the context of a year-long active campaign for protection of the environment, HBM's environmental management system was accredited to ISO 14001.

2 Torque measurement methods

2.1 Calculation from electrical power

Torque can be calculated from the electrical power and speed of rotation. Modern measuring equipment makes it easy to determine the electrical power and rotation speed of electrical machinery. However, when calculating torque there can be relatively large errors since dissipated power and the operating status of the machinery have considerable influence.

Today's instruments with their advanced computerized features take an ever-increasing number of parameters into account in order to raise the level of accuracy and dynamic response. The key application areas, however, are more commonly to be found in process monitoring, such as mechanical agitators, rabblers and the like, since this is where it is important to monitor additional electrical parameters such as reactive power or efficiency. A significant advantage of determining torque by this method is that there is no need for any kind of mechanical intervention in the power train.

However, this method is suitable to only a limited extent for accurate, dynamic torque measurement. It cannot be used if the torque information is needed referring to another point on the train of mechanisms, for instance downstream of a transmission or some other power sink.

The uncertainty involved in measuring torque by purely computational means can be several factors worse than using torque transducers fitted with SG measuring systems. Due to the greater accuracy of SG transducers, they are also commonly used as transfer transducers when calibrating electrical machinery.

2.2 Measuring reaction torque

2.2.1 Measuring the reaction force on a lever arm

Measuring reaction force according to the principle that in-line torque equals reaction torque is a method very frequently used to determine power. Fig. 2.1 shows an industrial measurement configuration with a pendulum mounted braking device. The force acting on the end of the lever arm is measured using a force transducer. This solution calls for complex mechanical arrangements. To avoid measurement errors it is necessary to take due account of disturbing influences such as changes in the pendulum bearing over time, expansion of the lever arm due to temperature changes, and the different states of operation.

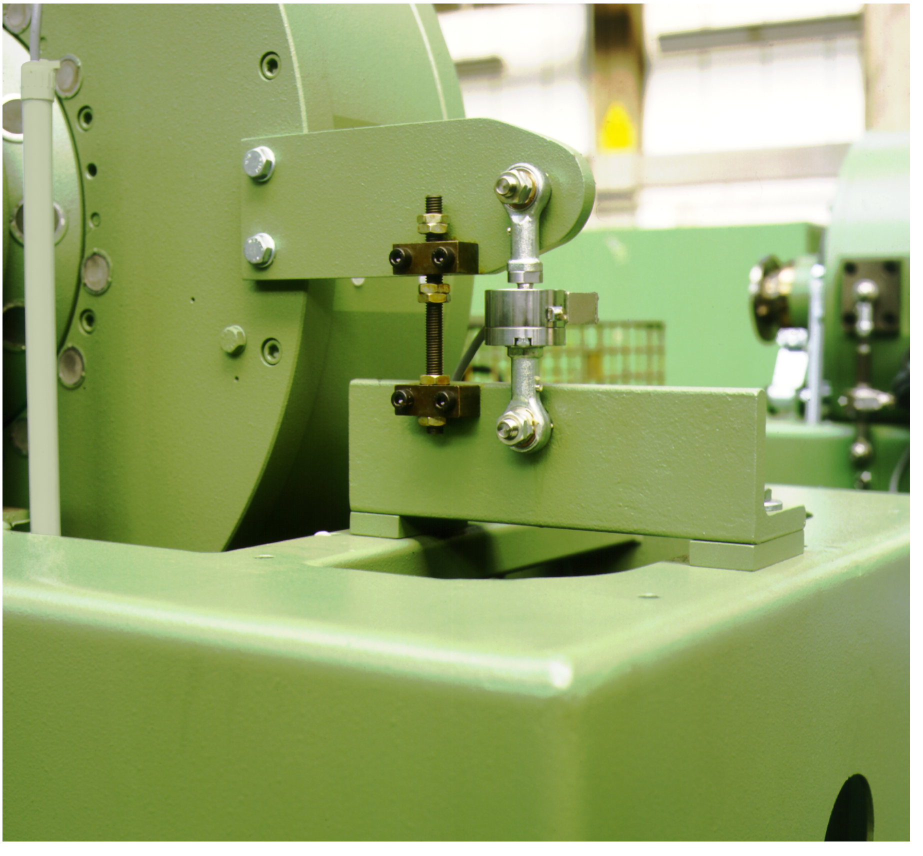


Fig. 2.1 Using a U2B force transducer to measure the reaction force acting on a lever arm

On the one hand, the inertia of the large masses involved make dynamic testing difficult. The mass moment of inertia acts as a mechanical low pass filter. On the other hand this characteristic can also be an advantage in cases where there is no necessity to measure dynamic moments. Dynamic torque components which are of no interest do not impinge on the force transducer. Another key application area for reaction force measurement is determining the viscosity of a medium for the purpose of process control via the supporting force of a motor in an agitator. Fig. 2.2 shows a simplified sketch of a suggested design.

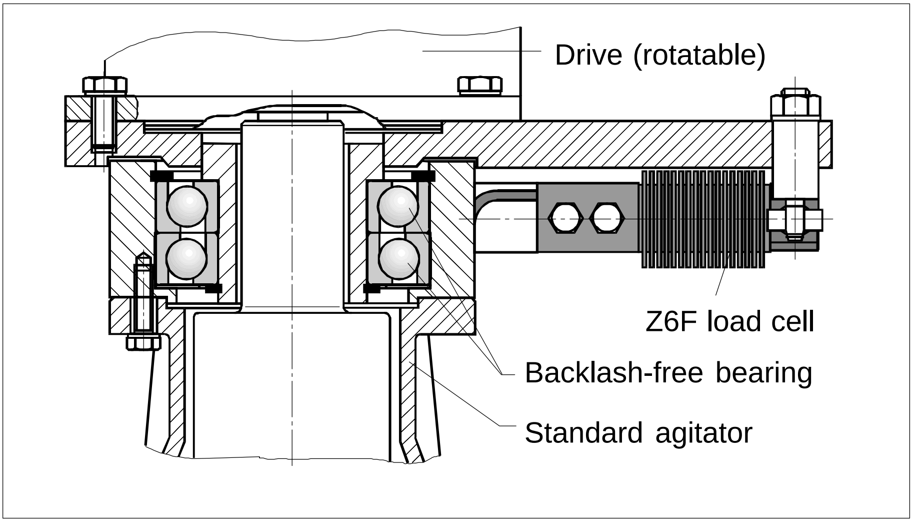


Fig. 2.2 Measuring viscosity with the aid of a Z6F load cell

The HBM range includes many forms of force transducers and load cells suitable for measuring reaction force by means of a lever arm. The main selection criteria are:

- Stiffness: a high degree of stiffness allows higher mechanical natural frequencies. Lower stiffness results in greater displacement during measurement, which can be helpful if overload stops or damping techniques become necessary.
- Design
- Direction of force: tensile and/or compressive force
- Required accuracy
- Cost

2.2.2 Reaction torque transducers

Reaction torque transducers combine into one device the functionalities which the bearing and the force transducer have in the case of the lever arm-based torque measurement described in the previous section. Their main application is non-rotating torque acquisition. Typical examples are process monitoring in agitators, rabblers and similar types of mixing equipment. In such applications the transducer is located directly between the container and the drive on the agitator. The drive shaft goes right through the transducer. Fig. 2.3 shows a suggested configuration for measuring viscosity on the basis of reaction torque measurement.

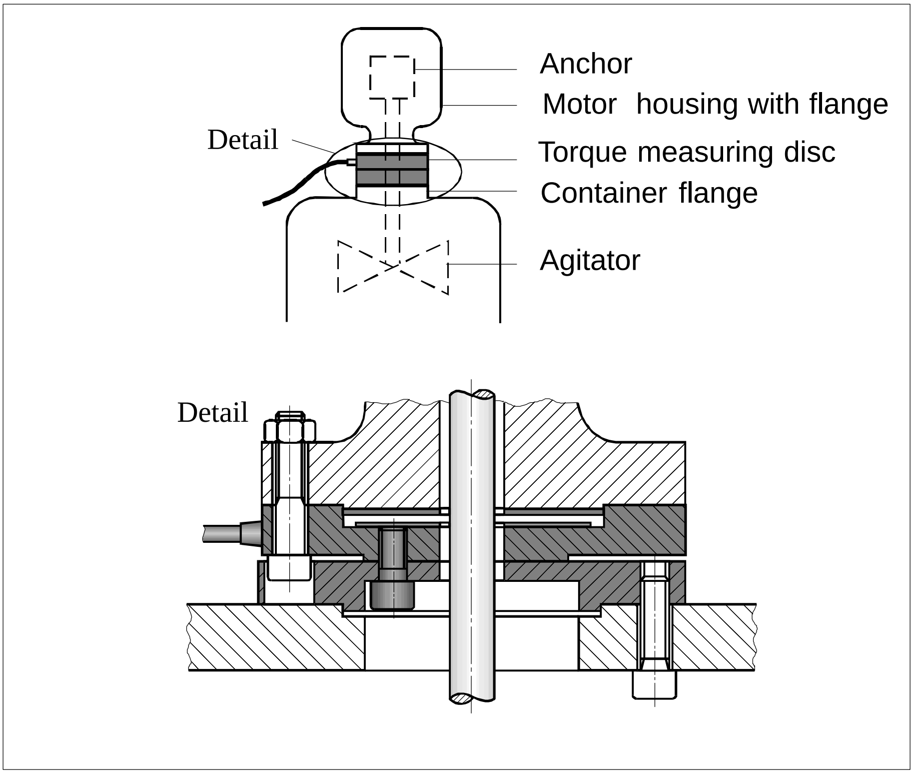


Fig. 2.3 Reaction torque measurement using a TB1A torque measuring disk between the motor housing and the container

The driving torque introduced into the agitator must be transmitted from the motor housing to the container flange in the form of a reaction torque. A TB1A torque measuring disk is fitted at precisely this point, between the motor housing and the container. The agitator shaft projects upward through the center hole and the motor is supported on the measuring disk. Interestingly enough the bearing friction in the motor, unlike the bearing friction on the bottom end of the agitator, does not give rise to measurement errors.

If a transmission is located between the transducer and the point on the drive train where the torque is actually intended to be acquired, the transmission ratio must be taken into account without fail in the choice of measuring range and in the scaling of the measuring amplifier. The torque that is actually to be measured will then be displayed with figures in the appropriate range. Torsion fatigue tests on components are yet another field of application. Fig. 2.4 shows a typical application of this kind.

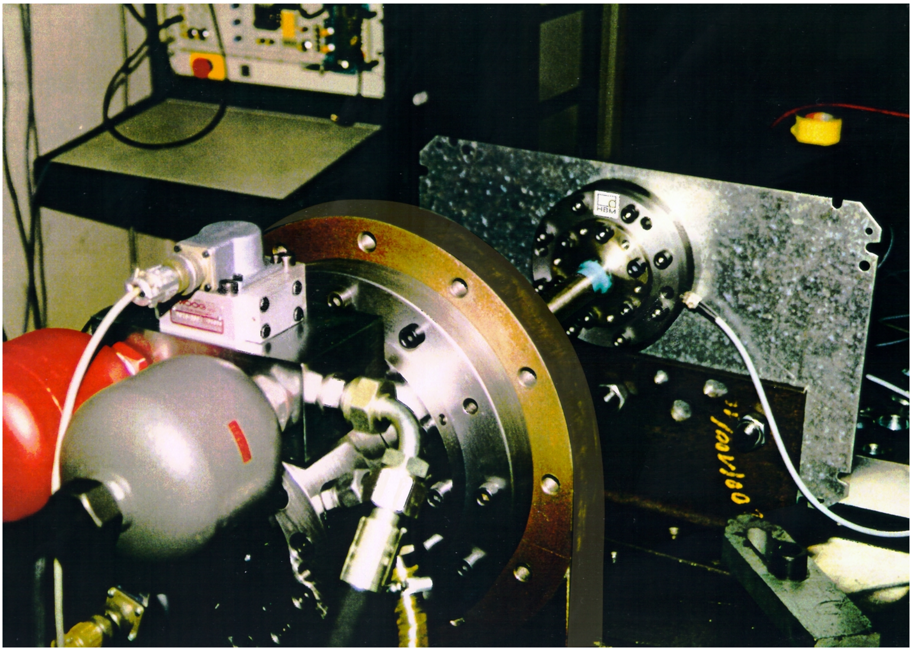


Fig. 2.4 Using a TB1A to conduct torsion fatigue tests on a rod

2.3 Measuring in-line torque

This method acquires the torque in a rotating train of shafts and is commonly known as in-line torque measurement. Fig. 2.5 shows the principle by which in-line torque measurement works. Torque transducers are conventionally divided into three product groups: torque shafts, torque hubs and torque flanges.

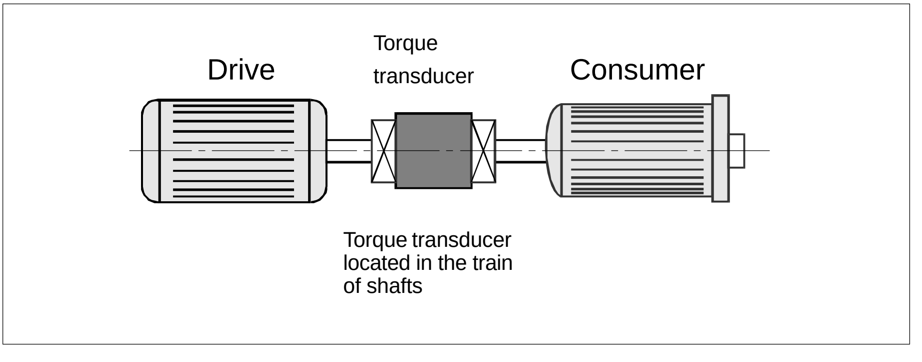


Fig. 2.5 The in-line torque measurement principle

Signals can be obtained by using a number of different physical principles:

- Hydraulic, pneumatic
- Translating an elastic deformation into a change in capacitance, inductance, resistance, permeability or phase.

Nowadays the most commonly used approach is to measure deformation with the aid of strain gages (SGs), which change their resistance in proportion to the strain involved. All torque transducers in the HBM range are constructed on this principle. This technology, together with the core skills of HBM in the field of the development and manufacture of measurement bodies (spring elements), has fully proven its advantages by producing the highest levels of accuracy and dynamic response.

2.4 SGs in torque measurement

SG torque transducers consist mainly of spring elements combined with strain gages (SGs) and compensation elements as well as adaptation accessories for the torque connections to input and output sides. The main features of the strain gage principle mentioned in [1] as being of importance to torque measurement are set out in concrete terms below:

- Strain gages used in the SG measuring bridge (or Wheatstone bridge circuit, named after the English scientist Sir Charles Wheatstone) together with their means of compensating for the effects of interference variables, have excellent characteristics with regard to linearity, hysteresis and reproducibility.
- Because SGs have negligible mass, the frequencies involved in processes under investigation can be very high (> 50 kHz). Centrifugal acceleration in excess of 10,000 m/s^2 is not critical.
- Static and dynamic moments can be acquired.
- SGs exhibit excellent strength in the presence of vibration, making them highly stable under alternating loads.
- Torque transducers with SGs exhibit excellent long-term stability when suitably configured for the application concerned.
- Because of the way they are manufactured and the fact that they are produced by the same company, SGs and measuring bodies (spring elements) can be individually adapted to work with one another to optimum effect.

- Due to the use of SGs specially adapted to show only minimal effects of temperature variation on the output signal, combined with the properties of the measuring bridge and the use of additional compensating elements, temperature has minimal effect on such devices. They can therefore be used in a wide range of temperatures.
- Torque can be measured in positive and negative directions regardless of whether the shaft train is rotating.
- The SG measuring bridge can compensate for the highly critical mechanical variables which can cause interference during torque measurement, namely bending moments, axial forces, lateral forces and rotational effects.

3 The structure of torque transducers

The following section describes the structure of torque transducers in which the measuring body forms part of the rotating transmission train and is elastically deformed by the torque being measured. The strain that occurs in these circumstances is acquired with the aid of strain gages.

A torque transducer of this kind principally consists of a rotating measuring body, known as the rotor, and a housing known as the stator. Torque transducers can differ structurally not only in the form of their signal transmission but also in their mechanical design.

Slip rings or contactless systems can be used to transmit the supply voltage and measurement signal. In addition some types of torque shafts have built-in bearings and others are without bearings. The design of the measuring body is important. A distinction is made between three types of torque transducers: torque shafts, torque hubs and torque flanges.

3.1 Mechanical structure

3.1.1 Measuring body designs

As a basic principle measuring bodies can take any shape. On the other hand they must have smooth surfaces on which the strain created by torque can be measured using strain gages.

Frequently used measuring bodies include versions with solid, hollow or square-section shafts. When these designs are used, torque produces torsional stress only.

In contrast, tubular measuring bodies with the same load-bearing cross-sectional area provide higher bending stiffness. A solid square-section shaft is often used for especially high torque measurement ranges. It is very simple to manufacture and bonding of strain gages is easy.

In the case of other measuring body shapes, such as spokes or cages, the applied torque generates a local flexural stress in sub-elements of the measuring body. Particularly for low torque values, cross-shaped measuring bodies offer the advantage of higher strain values combined with greater flexural stiffness.

Such are the present-day demands on torque transducers for power test benches that they cannot be optimally met by the measuring body versions mentioned above. That is why in the mid nineteen-nineties HBM was the first manufacturer of torque transducers to introduce the shear principle for torque measurement [2]. Characteristic of this principle are features known as half beams, which can be used as shear elements for metrological purposes [3]. Four radial I-profile beams (shear spokes) are built into the T10F torque flange. Not only is this advantageous in terms of measurement technology, but it also gives excellent ratios for the lateral stiffness in directions perpendicular to the direction of measurement. Fig. 3.1 shows an overview of the measuring body shapes most commonly in use at the present day.

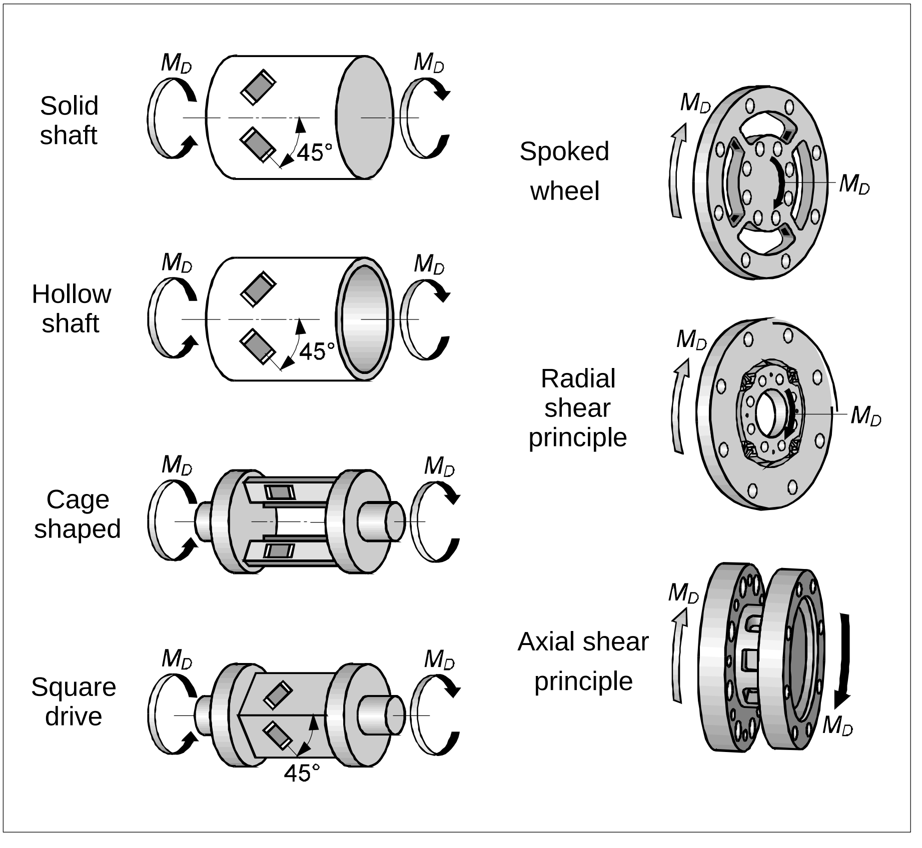


Fig. 3.1 Commonly used measuring body shapes

A careful choice of geometry for the strain gage application site and shear elements will make it possible to adjust the required properties of the torque flange within wide margins. Fig. 3.2 shows a series-manufactured T10F measuring body.

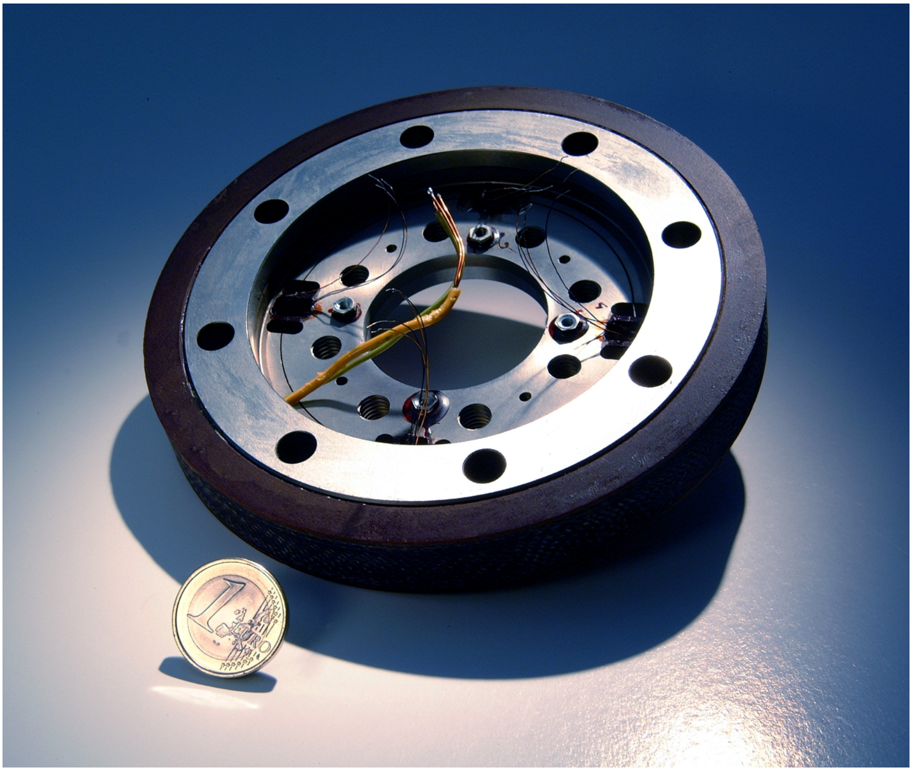


Fig. 3.2 T10F measuring body, 200 N·m

The measuring body of the T10FS torque flange represents yet a further development and modification of the above. In this case the U-profile shear elements are arranged axially rather than radially. The strain gages are fitted on the inside. This form of measuring body offers the possibility of a hermetically sealed version. HBM has applied for patents on the new radial shear principle and axial shear principle measuring body configurations. A number of patents have already been granted. Fig. 3.3 shows a series-manufactured T10FS measuring body.

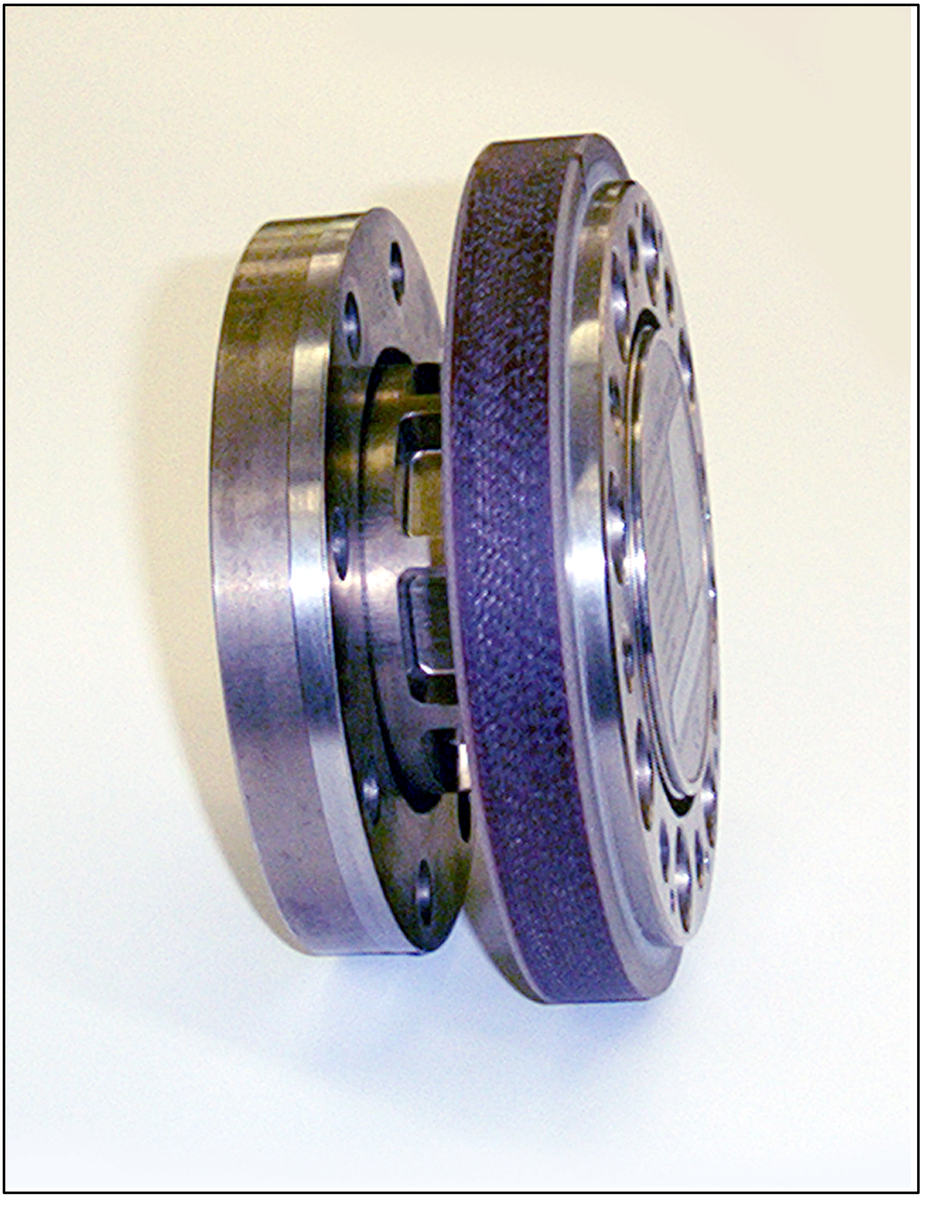


Fig. 3.3 T10FS measuring body, 500 N·m

3.1.2 Torque connections

Depending on the torque transducer concerned, HBM uses one of three possible methods of connecting the torque into the transducer:

- Square-drive plug-in connection to DIN 3121
- Clamping connection
- Bolted flange

HBM offers couplings for various torque transducers, tested in conjunction with the above and even ready mounted on the transducers if required. To facilitate torque connection by the customer, these are designed for different forms of friction locking or form locking connections and serve to minimize lateral and longitudinal forces as well as bending moments acting on the torque transducer.

As a result of mounting torque transducers with couplings, the quality of balance may change. At high operating speeds this may cause difficulties. Whenever possible the system should be balanced in its ready-to-operate state. Further information can be found in chapter 5.

In the case of the Renk curved tooth couplings® obtainable from HBM, the keyway system to DIN 6885 is used. This is a form locking connection. The customer has to specify the nature of the connection and the tolerance field for the fit. For preference, HBM supplies cylindrical drill-holes to tolerance H7 in accordance with DIN 7154, Sheet 1.

The necessary keyway grooves are preferably manufactured to DIN 6885, Sheet 1, P9 with back play. Typical connections for the different types of fit are illustrated in [4]. Since keyway DIN torque connections are usually not only subject to play but also have to be clamped in the longitudinal direction, they are unsuitable for use in the measurement of torque in which the direction of torque and rotation is alternating (in what is known as four-quadrant mode).

Square-drive plug-in connection

HBM torque shafts/screw torque transducers have a square section in accordance with DIN 3121. This standard contains specifications relative to driving squares for power socket wrenches. Dimensions comply with ISO 1174-2. Available designs are the form F external driving square, which is completely pierced by a pin hole, and the form G and form H internal driving square, depending on the nominal torque concerned. These square-drive connections are form locking connections, and thus they are not suitable for torque measurement in cases where the direction of the torque is alternating.

Clamping connection

Clamping connections work on the friction locking principle. Depending on the design and requirement of the application concerned, HBM recommends radial tensioning for simple requirements and the cone principle in cases where the demands are severe. Further notes and instructions about configuration and installation can be found in chapter 5.

Bolted flange

The flanged connections used on HBM torque transducers transmit the applied torque by friction locking. This ensures not only that flanged connections transmit very high torque values without any problem, but also – which is particularly important – that dynamic torque is correctly introduced into the transducer. Fixing bolts arranged at regular intervals around the circumference ensure reliably even pressure and totally uniform torque transmission around the circumference as a whole.

The number of fixing bolts required depends on the nominal torque of the measuring shaft concerned. Because of the need for standardization, the flanges used with some torque shafts have more securing holes than are actually necessary. In these cases some holes remain free. However, the bolts must be inserted at equally spaced intervals around the circumference. Instructions on the subject can be found in the documentation accompanying the products concerned. Chapter 5 goes into flanged connections in detail.

3.1.3 Torque shafts with bearings

In the case of torque shafts with bearings the stator is radially and axially fixed by the bearings in relation to the rotating measuring body. This version is used when:

- the operating speed lies in ranges which can still be easily handled by roller bearings
- the design of the torque shafts makes bearings absolutely necessary.

The latter situation basically applies to torque shafts with slip rings. In the case of T20WN type torque shafts the cost and effort involved in contactless measurement signal transmission and power supply are significantly reduced, since very small air gaps for power transmission can be specified and these are constant.

When properly installed by applying the torque that is to be measured to the designated side (the measuring side) the effect of the bearing roller friction on the measurement result is very small. The friction moment of the bearing on the measurement side of the T20WN/2 N·m amounts to approx. 0.00005 N·m.

In torque shafts with bearings the stator need only be lightly secured against rotation. The built-in bearings normally have only the weight of the stator to support and are therefore not subject to any significant wear. Nevertheless, depending on the device type concerned, the load capacity of the bearings is dimensioned to allow them also to act as support bearings. The installation options are described in greater detail in chapter 5. The special low-friction bearings guarantee maintenance-free operation over a very long period of time (20,000 service hours in the case of the T20WN).

3.1.4 Contactless and bearing-free torque transducers

In torque transducers for high operating speeds the stator is not mechanically connected to the rotating measuring body. The measuring body is fitted into the shaft train and the stator is mounted on the machine bed. Since the configuration includes no wearing parts, these torque transducers are absolutely maintenance-free and wear-free except for the optical speed measurement system.

During shipment the fixing elements bolted around the circumference of the stator on the T32FN provide transportation safety and are an aid to assembly. In the case of the T34FN a centering element acts as an assembly aid. Similarly, assembly aids are supplied for the T10FS torque flange with rotation speed module. With the assembly aids fixed, the stator and the rotor have to be installed without deformation or tensioning. Further information can be found in chapter 5.

3.2 Measurement signal transmission

In torque transducers the measurement signal is primarily generated on the rotor. The SG circuits are passive measuring systems and must therefore be supplied with an excitation voltage of several volts. The output signal corresponding to the measured quantity is in the order of several millivolts. There are two possible ways of transmitting the measurement signals and supply voltage between the rotor and the stator. These are:

- Transmission via slip rings
- Contactless transmission

HBM torque transducers use both these variants. Torque shafts with slip rings are mainly suited to short-term measurement with rotary speeds of no more than 4000 min^{-1}. They offer the advantage that the subsequent signal processing electronics can consist of any measuring amplifier suitable for SG measuring systems. The disadvantage is the unavoidable abrasion occurring in the slip rings and the consequent need for maintenance.

Contactless transmission systems are incorporated into torque transducers designed for long-term operation and high continuous rotation speeds. These transducer types have special electronic transmission components for the supply voltage and measurement signal.

3.2.1 Measurement signal transmission via slip rings

The voltage supply and measurement signal in HBM torque measuring shafts are transmitted over hard silver slip rings and silver graphite carbon brushes. This combination offers optimum interference-free signal transmission (low noise, low thermally-induced voltages), long service life ($3 \times 10^8 ... 6 \times 10^8$ revs) and therefore minimal maintenance effort.

Two sets of brushes are used in order to guarantee reliable contact in all operating conditions. The two sets are arranged so that two brushes are held by spring pressure against each slip ring, but offset at a certain angle. Fig. 3.4 shows the configuration of a SG measuring bridge in principle, together with the compensating components and measurement signal transmission via slip rings.

Although four slip rings are actually enough for signal transmission, HBM torque shafts are fitted with a fifth slip ring to equalize the potential between the rotor and the stator. Perfect equalization is generally not guaranteed via the bearings. Without this potential equalization considerable signal interference can result from differences in potential.

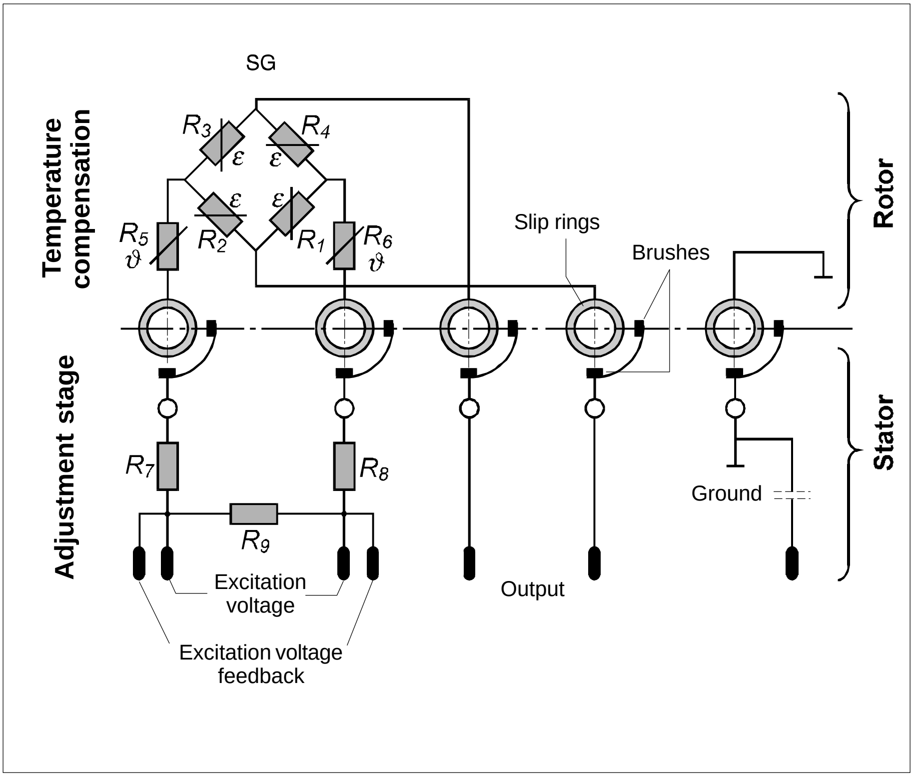


Fig. 3.4 Schematic diagram of slip ring transmission

3.2.2 Contactless energy and measurement signal transmission

HBM's decades of experience are also reflected by the contactless transmission of energy and measurement signals. The optimum solution has always been chosen from among the developments and technical requirements of the time. For this reason the technology used differs from one series type to another. Overall a distinction can be made between the following situations:

- Supplying torque transducers with AC voltage and using mechanically separated transmission paths for the transmission of energy to the rotor and the transmission of measurement signals
- Supplying torque transducers with DC voltage and using mechanically shared transmission paths

In order to ensure that the electronic connection conditions in torque flanges of the T10 family are compatible with earlier torque transducers, versions are available with AC and DC voltage supply in conjunction with measurement signal transmission over mechanically shared transmission paths.

When a torque transducer is being supplied with AC voltage and measurement signal transmission over mechanically separated transmission paths (see Fig. 3.5) an AC voltage with a frequency of 15 to 25 kHz, depending on the measurement amplifier system concerned, is inductively transmitted onto the rotor on the torque measurement shaft.

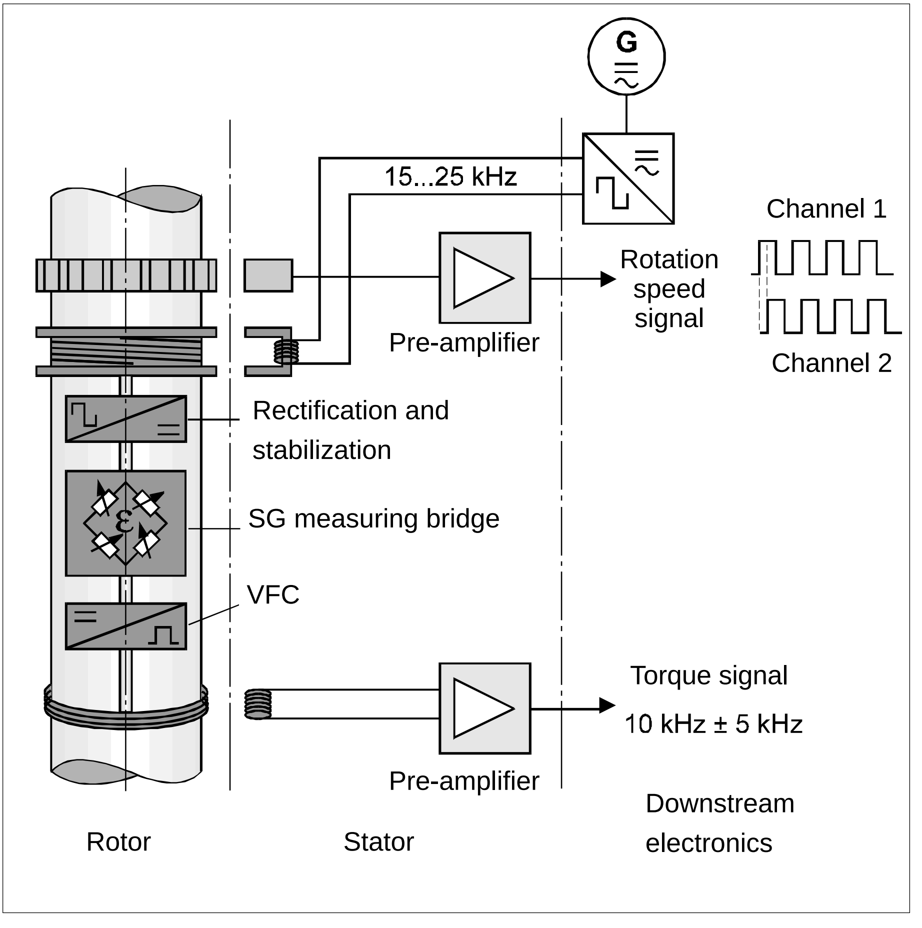
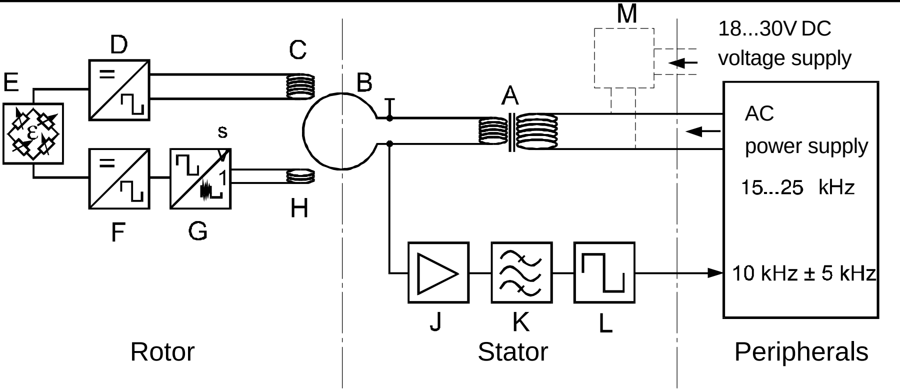


Fig. 3.5 The contactless transmission principle

The rectification and stabilization incorporated into the rotor transforms this into a stabilized DC voltage to feed the SG measuring bridge. The torque-proportional output voltage U_A from the measuring bridge goes to a voltage-frequency converter (VFC) in the rotor. This VFC generates a pulse frequency which is proportional to the output voltage of the bridge, and is thus also proportional to the torque. This pulse frequency is transmitted inductively to the stator, where it is boosted in the pre-amplifier. Amplitude losses are unimportant because the information regarding the torque is contained in the frequency.

All HBM torque transducers with contactless transmission of energy and measurement signals have a shunt calibration resistor on the rotor which can be activated by the subsequent electronics. This is wired in parallel to one branch of the SG measuring bridge. Activation is carried out by increasing the AC supply voltage increases from 54 V_{SS} in normal operation to 75...81 V_{SS} or in the case of a DC voltage supply, when an asymmetrical DC voltage of 5 V is applied to an extra connector pin dedicated to shunt activation. Further information on using the shunt calibration resistor can be found in chapter 5 and chapter 7.

Due to the extremely compact design of the T10 family and additional requirements for simpler assembly, new demands were made on contactless energy and measurement signal transmission. Fig. 3.6 shows the principle of contactless transmission in the T10 family.



A Transformer
B Antenna segments
C Rotor winding
D Rectification and stabilization
E SG measuring bridge
F Voltage-frequency converter
G Carrier signal
H Rotor winding
J Selective pre-amplifier
K Narrow band-pass filter
L Demodulator
M Switch-mode power supply

Fig. 3.6 The contactless transmission principle

In electrical configuration KF1, the generator for the amplifier supplies the antenna segments (B) of the stator with an AC voltage at a frequency of around 15 to 25 kHz. At the same time the transformer (A) provide adaptation of the voltage. The energy is transmitted inductively onto the rotor. The associated rotor winding (C) delivers the supply voltage, after rectification and stabilization in (D), for the rotor electronics and feeds the SG measuring bridge (E).

The output voltage from the bridge controls the frequency of the square-wave voltage (F). This square-wave voltage switches on and off the MHz-range carrier signal (G) that feeds the second rotor winding (H). The antenna segments (B) receive the carrier signal from which the frequency-modulated measurement signal is retrieved with the aid of a selective pre-amplifier (J), an ultra-narrow band-pass filter (K) and a demodulator (L). The measurement signal is then available at the connector in the form of a square-wave voltage of 10 kHz ± 5 kHz.

This new transmission method has the following features:

- A 12 V excitation to the SG measuring bridge results in a higher measurement signal to noise voltage separation.
- Frequency modulated measurement signal transmission provides the best transmission quality, regardless of spacing and displacement.
- The wide measurement frequency range of 0 to 1200 Hz enables measurements using higher dynamic band widths.
- Switchable shunt calibration reduces the effort involved in carrying out mechanical checking.
- Measurements can also be carried out in standstill conditions (nil rotation speed). The speed has no effect on transmission.
- The stator ring can be divided, which means it can be retrofitted without having to dismantle the mechanical shaft train.

In addition to an AC voltage supply, the T10 family also offers the option of an asymmetrical 18...30 V DC voltage supply. Conversion of the DC voltage into the AC voltage required for feeding the antenna segments is carried out by a switch-mode power supply in the stator of the torque flange. The measurement signal is transmitted as described above.

3.3 Measurement systems for speed and angle of rotation

Depending on the torque transducer concerned, the speed and angle of rotation are measured by either a magnetic or an optical system. When the rotor turns, voltage pulses are generated according to the number of teeth or slots per revolution, and the frequency of these pulses is proportional to the rotation speed.

The pickup heads or detector diodes are mutually offset so as to generate two pulse trains with a phase shift of $\pi/2$ (corresponding electrically to 90°).

This phase shift provides information on the direction in which the transducer is rotating. A pre-amplifier converts the pulse trains into pure square-wave voltages. It is particularly important to note whether systems can acquire the speed and angle of rotation without restriction. The important criterion in this situation is the minimum rotation speed for achieving adequate pulse stability. If the minimum rotation speed is nil, the system can be used without restriction for measuring the angle and speed of rotation. If the minimum rotation speed is greater than nil the system is suitable for measuring speed of rotation, but its suitability for measuring the angle of rotation must be checked from case to case.

3.3.1 Magnetic rotation speed acquisition

Magnetic rotation speed acquisition relates to systems with Hall sensors and to inductive systems. In inductive systems the rotor has a toothed ring, opposite which there are inductive pickup heads in the stator (see Fig. 3.7).

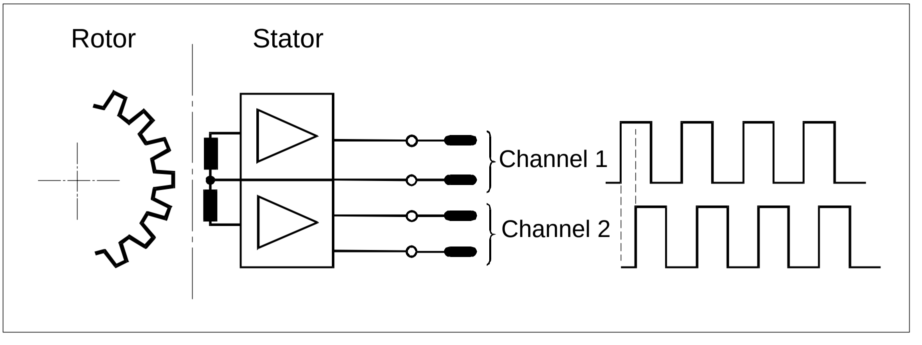


Fig. 3.7 Schematic diagram of contactless magnetic rotation speed measurement

Inductive pickup heads react to changes in metallic influences. Systems with Hall sensors react to changes in magnetic fields. The rotating part therefore carries one or more magnets, depending on the resolution required. Since both systems need a minimum rotation speed in order to achieve adequate pulse stability, their suitability for measuring angle of rotation is limited.

3.3.2 Optical rotation speed and rotation angle acquisition

In optical systems a distinction is made between transmitted light methods and incident light methods (see Fig. 3.8).

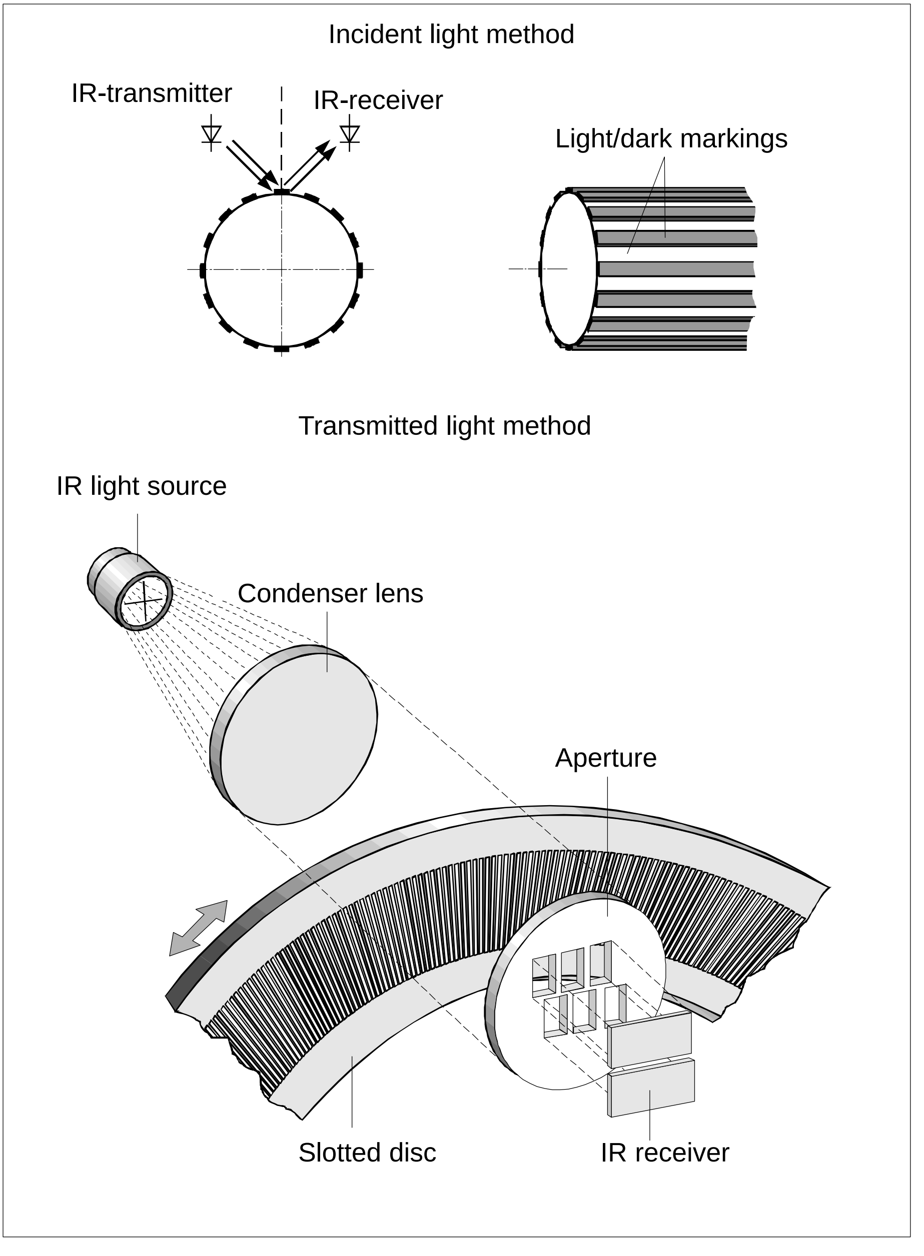


Fig. 3.8 Transmitted and incident light methods in optical measurement systems for speed and angle of rotation

The transmitted light method typically exhibits a high degree of reliability and therefore HBM uses only this type for torque transducers. Due to the required boundary conditions, only slotted metal disks are used. The number of mechanical increments per revolution is in the range 90 to 720, depending on the size and type of installation.

In certain circumstances, torque flanges that are constructed without bearings can appear to register periodic rotation speed signals even when they are at a standstill. This effect is due to radial vibration of the rotor, which causes the position of the slotted disk to change in relation to the optical sensor.

In the case of torque shafts with bearings, the minimum rotation speed for adequate pulse stability is nil, which means that they can be used without restriction for measuring the angle and speed of rotation.

3.3.3 Reference pulse

New-generation torque flanges possess not only a rotation speed measuring system but also a reference pulse. This is generated by a magnet in the slotted disk and a magnetoresistive sensor, creating one pulse per complete revolution of the rotor. This reference pulse is synchronized with the rotary speed output signal on channel 1. Fig. 3.9 shows the electrical position of the reference pulse. The pulse lasts as long as a rotary speed pulse.

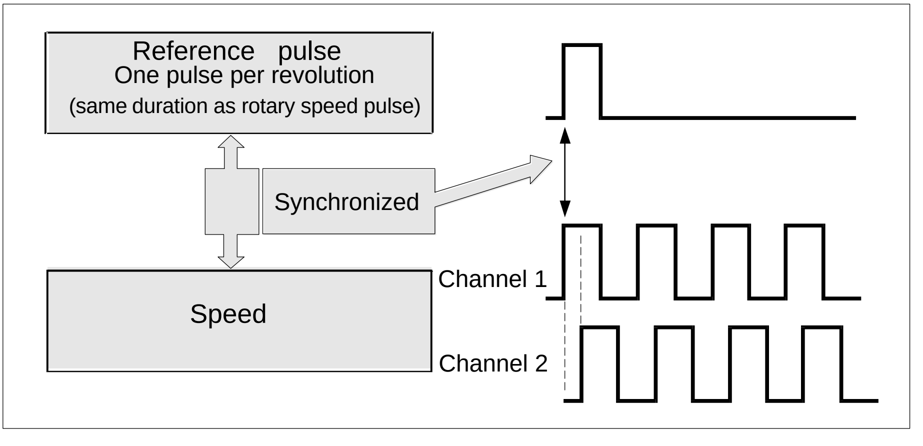


Fig. 3.9 Electrical position of the reference pulse

3.4 Electrical output signal types

3.4.1 mV/V torque signal

Torque shafts with measurement signal transmission using slip rings are purely passive transducers. Their torque signal interface is defined in mV/V. In order to generate a measurement signal, these transducers require an auxiliary voltage known as an excitation voltage. The output signal is very small - just 10 mV at 2 mV/V with an excitation voltage of 5 V.

This voltage is boosted in a downstream amplifier, then evaluated and converted into a suitable output signal. DC and carrier-frequency amplifiers are used for this purpose. It is impossible to give a clear-cut statement regarding which type of amplifier should be used, since both methods have system-related advantages and disadvantages [5].

Carrier-frequency measuring amplifiers are used for preference. With this type, any interference from brush arcing on the slip rings has no effect since it is outside the carrier-frequency range, and thermally-induced voltages have no effect on the measurement signal. Further information on measuring amplifiers and wiring can be found in chapter 5.

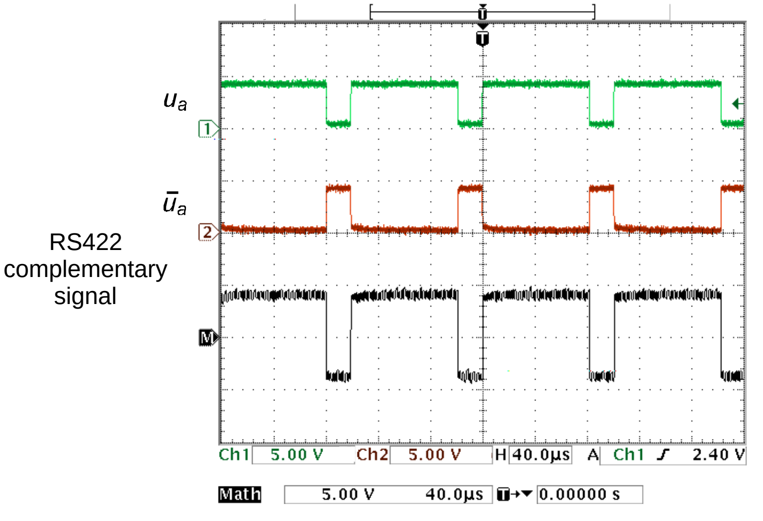
3.4.2 Torque frequency signal

In torque transducers with contactless energy and measurement signal transmission the standard interface for the torque signal is a frequency interface at 10 kHz ± 5 kHz. T20WN type torque shafts are exceptions to this (see section 3.4.3).

Frequency output signals are either symmetrical (RS422 complementary signals) or asymmetrical (12 V), depending on the version of stator electronics concerned. RS422 complementary signals need a differential input on the downstream processing side. Fig. 3.10 shows the symmetrical output signal from a T10F and Fig. 3.11 shows a suggested input wiring solution for the downstream electronics.

The recommended differentiating circuit receivers are LTC485, MAX485, MAX487 or IL485. While the terminating resistance Z_0 is not absolutely essential, it is recommended when cable lengths are significant in order to prevent end of line reflections due to mismatching.

Cable length 6m



Cable length 56 m

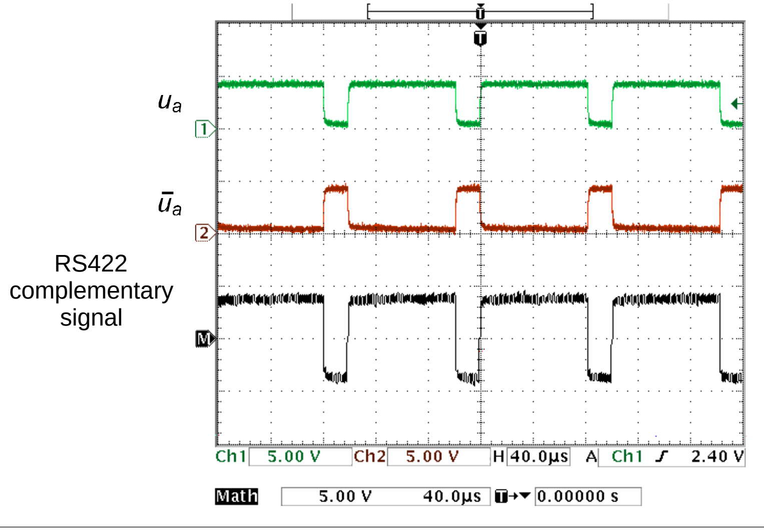


Fig. 3.10 Oscillogram of the symmetrical output voltage from the T10 family

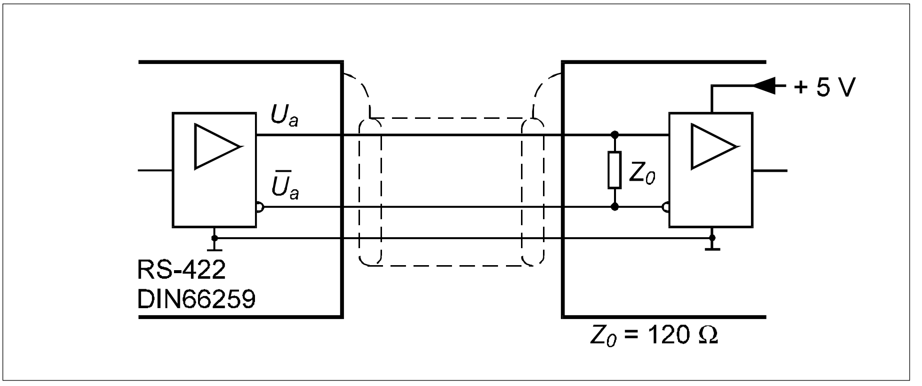


Fig. 3.11 Suggested input wiring solution for the downstream electronics

The asymmetric signal is referenced to earth. A corresponding oscillogram is shown in Fig. 3.12. The effect of cable length on signal shape can be clearly seen.

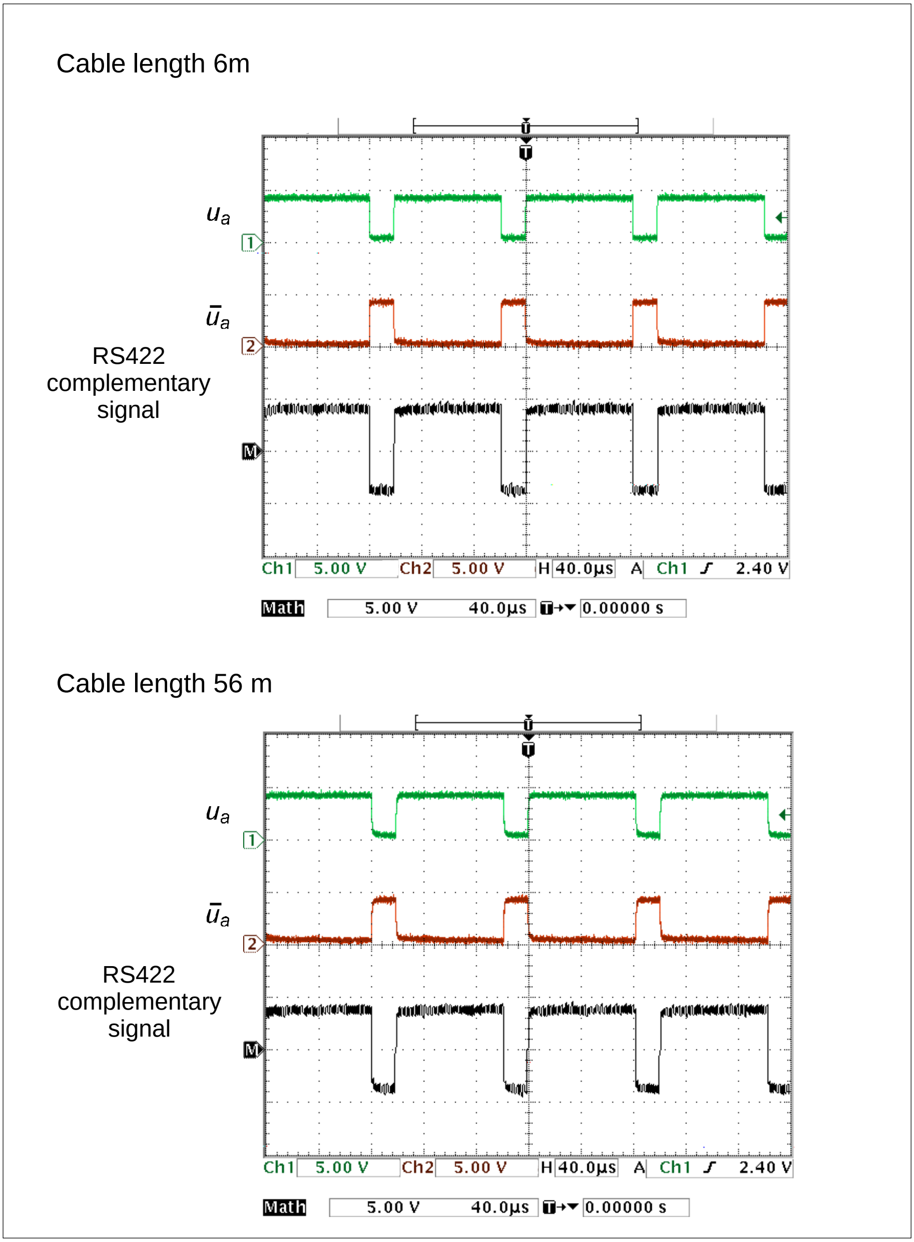


Fig. 3.12 Oscillogram of an asymmetric signal

3.4.3 Analog torque output ± 10 V

Members of the T10 family with DC voltage supply not only have a frequency output but optionally their stators can also have a ±10 V analog output for torque. T20WN type torque shafts have only a ±10 V analog output for the torque signal. The ±10 V analog output makes it possible to produce very cost-effective standalone applications, because no additional measuring amplifier is needed. On the other hand, variable signal evaluation (filter setting, scaling) is not possible.

3.4.4 Frequency output signal for speed and angle of rotation

Frequency output signals are either RS422 complementary signals, TTL level, or other asymmetrical 5 to 25 V signals. Usually there are two channels available, offset by $\pi/2$ (corresponding to a phase shift of 90°). These are used by the subsequent electronics during rotation speed measurement for the purpose of detecting the direction of rotation. Depending which channel takes precedence over the other, the downstream electronics will detect a positive or negative rotational direction.

When the angle of rotation is being measured, the pulses are added or subtracted. Members of the T10 family with the speed measurement option also have the ability on channel 1 to double the number of pulses in the stator electronically. Channel 2 in this case provides a static signal which shows the direction of rotation (see Fig. 3.13).

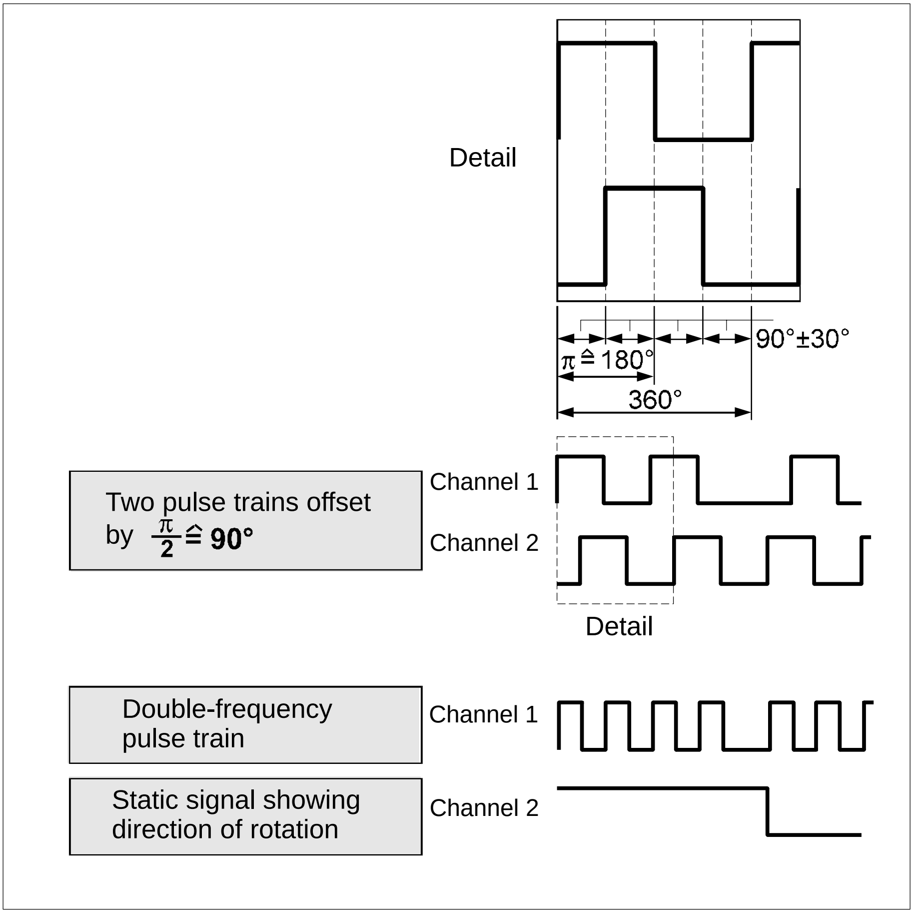


Fig. 3.13 Forms of the rotation speed output signal from the T10 family

4 Selection criteria and application environment for torque transducers

This chapter summarizes the main criteria which must be kept in mind when selecting a torque transducer and configuring an application (such as a power test bench). It describes the properties of the overall system and of torque transducers, and how these interact.

Various test questions can be asked in order to decide whether to define the desired properties of the torque transducer according to the given environment so that the overall system fits together, or whether to take the opposite approach, which is to design the environment after specifying the properties of the torque transducer so that the properties of the overall system fit together.

This discussion is founded on the basic properties of torque transducers explained in chapter 3 and is continued in the concrete design variants described in chapter 5.

The chapter is divided up into sections to match each of the basic properties of a torque transducer, that is to say its dimensions, nominal torque etc. Each section describes the concepts which can be used to define these properties for a planned measurement task. The concluding section is concerned with the topic of vibrations, since it has a bearing on many of the criteria discussed in the sections which precede it.

The chapter is structured so that each of the individual criteria that need to be kept in mind has a separate section of its own giving sufficient background to enable a practitioner to make well founded decisions with regard to selection. To provide an overview, these criteria are listed below. The list not only corresponds to the various sections in the chapter, but can also be used as a checklist.

- Mass and mass moment of inertia of a transducer
- Torsional stiffness, and stiffness with respect to other (parasitic) loads
- Maximum speed of rotation
- Measuring speed of rotation
- Maintenance requirements
- The size of the torque to be measured, from the point of view of quasi-static processes

- Dynamic torque from rotary acceleration and retardation
- Oscillating torque
- Dynamic torque peaks from electrical machinery
- Parasitic loads
- Required accuracy in the light of various aspects
- Environmental effects: dust, foreign bodies, fluids, chemicals, temperature, atmospheric humidity, EMC conditions
- Dynamic torque from torsional vibration
- Parasitic loads from bending and axial vibrations

4.1 Dimensions and basic mechanical properties

4.1.1 Mechanical installation

When a suitable torque transducer is being selected for a given or planned application, geometric criteria can often be quite significant. On the one hand these criteria refer to the external dimensions, which among other things can influence the choice of design. On the other hand the mechanical connection method is also significant, for instance in deciding between a flanged connection or a shaft stub connection.

The concepts for selection and configuration which can be deduced from these differences stem directly from the special features brought about by the fact that various designs can be used for installing torque transducers. The reader should therefore pay particular attention to chapter 5, in which the installation of torque transducers is a decidedly major topic. This discussion is based on the explanations given in chapter 3 on the subject of the various designs and connection methods used.

4.1.2 Mass, mass moments of inertia

Mass

The mass of a torque transducer has an effect on the amount of possible sag and, by the same token, the possible bending vibration. The greater the mass the greater the bending and consequently the lower the natural bending frequencies.

However, this effect cannot be discussed in isolation from other components. It is strongly dependent on the bending stiffness of the couplings and shaft sections supporting the rotor on the torque transducer. The number and arrangement of bearings supporting the shaft train required for the application has a determining influence on the effective stiffness of shaft sections. A stiff bearing arrangement can support a heavier mass and yet still fulfill the same specifications with regard to sag and mechanical natural frequencies. A stiff bearing arrangement comes about from using several closely spaced support points, or double-row bearings in a version which can also absorb bending moments. A detailed account of the contexts in which bending vibrations occur can be found in section 4.6.

Mass also influences the bearing forces occurring. This applies not only to the static component, which is influenced by the weight that has to be supported, but also to the dynamic bearing forces, which are crucially determined by the vibrational influences described above.

Mass moment of inertia

The mass moment of inertia is a measure of the resistance that a body presents to a rotary acceleration. In the case of a torque transducer, since the axis of the rotary motion in question is already specified it follows that the only information required is a unique mass moment of inertia. The mass moment of inertia is determined on the one hand by the mass of the body and on the other by the distribution of that mass with regard to its distance from the axis of rotation.

The greater the average distance of the mass from the axis of rotation, the greater the mass moment of inertia. The mathematical definition is given in Appendix C together with a compilation of the mass moments of inertia for typical bodies (such as cylinders with uniform mass distribution).

In rotating machinery the mass moment of inertia generally has an influence on the rotary acceleration which can be achieved at a given torque, or the amount of torque required to reach a desired rotary acceleration. See also section 4.3.2.

However, it should be noted that the mass moment of inertia that relates to this concept is the sum of the mass moments of inertia of a large number of components in the shaft train. In many applications, therefore, the influence of the torque transducer’s mass moment of inertia is not the deciding factor.

The mass moment of inertia of the torque transducer can have a profound effect in applications with high-performance speed control. The same applies to many items of production equipment in which there is a continuous succession of rapid acceleration followed by retardation and a change of direction, as in automated thread tapping.

Another significant matter is the effect which the mass moments of inertia of individual components have on torsional vibration properties. The mass moment of inertia plays the same role in torsional vibration as mass plays in bending vibration. Thus the greater the mass moment of inertia, the lower the natural frequencies for the same degree of stiffness. In many cases the mass moment of inertia of the torque transducer is negligible when compared with the moments of inertia of other components in the shaft train, for example the mass moments of inertia of internal combustion engines, electrical machinery, transmissions and flywheels.

4.1.3 Stiffness

Generally speaking, a torque transducer that operates on the strain gage principle must have a certain degree of elasticity, since torque measurement using SGs acquires the torque indirectly via the strain and a completely rigid measuring body would allow no strain. The elasticity is quantitatively described by information on the torsional, bending, radial and axial stiffness. It can be said in general that the compactly constructed torque flanges are stiffer in every respect than conventional torque shafts.

The significance of stiffness in terms of layout can be seen partly in the fact that some of the design concepts for installing a torque transducer in the shaft train are largely determined by the stiffness. For example on the one hand the parasitic loading that arises in a highly elastic transducer due to alignment errors when couplings or joint shafts are dispensed with is less than in the case of very stiff transducers.

On the other hand a very stiff torque transducer can more easily bear the intrinsic weight of certain components in the shaft section. A second characteristic quantity associated with each type of stiffness coefficient is its maximum loading capacity with respect to the corresponding force or moment. For more information the reader is referred to section 4.3, which discusses the subject of maximum torque and maximum parasitic loads.

Directional stiffness in a torque transducer also has a strong influence on the vibrational properties of the shaft train. Regardless of whether torsional, bending or axial vibrations are considered, it is always the case that higher stiffness leads to higher natural frequencies in the corresponding vibrations whereas less stiffness leads to lower natural frequencies. Another important influence quantity in addition to the various types of stiffness is the mass or mass moment of inertia. A mathematical description of these relationships including a discussion on which frequencies are favorable or unfavorable as natural frequencies can be found in section 4.6.

Torsional stiffness

Torsional stiffness describes the relation between torque and elastic torsion about the axis of rotation (for the mathematical definition see Appendix C). In the interplay between stiffness and masses/mass moments mentioned above, it has an effect on the susceptibility of the configuration to torsional vibration.

Compactly constructed torque flanges exhibit torsional stiffness which is very high in relation to the nominal torque.

Bending stiffness

Bending stiffness describes the relation between a bending moment and the angle of elastic bending. In the interplay between stiffness and masses/mass moments mentioned above, it has an effect on the susceptibility of the configuration to bending vibration. In contrast, in the context of flexible shafts the ratio between bending and radial force is also known as bending stiffness.

Radial stiffness

Radial stiffness describes the relation between a force in the radial direction and the radial, parallel movement in opposite directions which it causes in both front faces of the torque transducer. As in the case of bending deformation, this radial deformation causes eccentricity relative to the axis of rotation. In the interplay between stiffness and masses/mass moments mentioned above, it too has an effect on the susceptibility of the configuration to bending vibration. The effect on the natural bending frequencies is qualitatively the same as in the case of bending stiffness.

High stiffness in both the bending and radial direction often makes it possible to design elegant and space-saving variants in which the weight of the individual components in the shaft train is supported by the torque transducer, thus making the use of additional supporting bearings unnecessary. The details are explained in chapter 5.

Bending and radial stiffness are at their highest in compactly constructed torque flanges.

Axial stiffness

Axial stiffness describes the relation between a longitudinal force and the change in length which it causes in the torque transducer. In the interplay between stiffness and masses explained above, it has an effect on the susceptibility of the configuration to axial vibration.

4.2 Operating conditions and equipment features

4.2.1 Maximum operating speed

The torque transducer and coupling must survive undamaged at all operating speeds. The nominal speeds shown in the specifications must therefore correspond to at least the maximum operating speed that is likely to occur. The direction of rotation is not relevant.

4.2.2 Measuring the speed and angle of rotation

Many HBM torque transducers are fitted with a speed measuring system either as standard or as an option, and some are also fitted with a measuring system for angle of rotation (see chapter 3). In order to judge whether such a system is suitable for the application concerned, it is first necessary to consider whether the direction and possibly the angle of rotation need to be acquired in addition to the speed, and whether the system under consideration requires a minimum speed for a stable speed measurement signal. The resolution must also be taken into account. On the other hand the number of system-generated pulses per revolution must also be considered with regard to the speed, since at very high speeds combined with a high number of pulses the output signal is often at such a high frequency that it cannot be analyzed without a great deal of effort.

4.2.3 Maintenance requirements

Maintenance requirements play an increasingly important role. Torque transducers with slip rings require regular maintenance, especially since there is a limited service life for slip ring brushes (see chapter 3). The grease used as a lubricant in tooth couplings has to be changed periodically. Torque transducers with contactless measurement signal transmission and bearings have a much lower maintenance requirement thanks to the very high service life of their bearings, but cannot be said to be maintenance-free. Torque transducers that are constructed without bearings and have contactless measurement signal transmission are maintenance-free and wear-free, as are couplings such as the bellows or multi-disk type.

Open optical systems for speed measurement run the risk of contamination if used in unfavorable conditions. Cleaning instructions are shown in the technical documentation.

4.3 Measuring range and maximum torque

4.3.1 First rough estimate of the torque in an application

In almost all cases a rough idea of the torque expected to be present in an application will already be available at the design stage. This will be based on such things as the nominal torque of the machines that drive the configuration (internal combustion engines, electric motors) or the nominal torque of the driven application (pumps, compressors, stirrers, and commonly in the case of test benches, absorption dynamometers).

However, it should be noted that in most sectors of machine building, the expression nominal torque will be taken to mean an average torque that can be maintained over a fairly long term. Peak torque is often considerably higher. In the case of a torque transducer, on the other hand, nominal torque refers to the upper scale limit of its measuring range, in other words this is a limit that should not be exceeded in normal operation. A torque transducer with the same nominal torque as the machine to which it relates is under-dimensioned in most cases.

It is often possible to use the relation between torque, speed and mechanical power

$$P = M_D \, \Omega$$

to compute a nominal torque that has not been explicitly specified. In this case P refers to the power (W) and Ω to the angular velocity (s^{-1}, or - to be more descriptive - rad/s, see Appendix C). If the above formula is adapted for torque and the angular velocity is replaced by the speed n in $\min^{-1}$, we obtain

$$M_D = \frac{60\,P}{2\pi n} \quad \text{(numerical equation, } M_D \text{ in N·m, } P \text{ in W, } n \text{ in min}^{-1}\text{)}$$

A rough estimate based on nominal torque alone requires a very generous safety factor when configuring the measuring range for torque transducers, as will become clear below.

4.3.2 Dynamic torque

The anticipated dynamic torque must be known as precisely as possible when selecting a torque transducer, since very often the actual maximum torque is only known from the dynamic torque peaks. Nominal and maximum torque can now be used in combination with the appropriate safety factors to select a particular torque transducer or a particular nominal torque. If a large number of vibration cycles can be expected, the transducer must be selected for its fatigue strength. The specifications of the vibration bandwidth must then be compared with the dynamic loading that can be anticipated.

The next sections will discuss the actual mechanisms that can give rise to dynamic torque. Torsional vibration has been deliberately omitted, since vibration in its own right does not usually cause dynamic components in torque, but rather acts as an amplifying mechanism. In order to do justice to the all-embracing significance of vibration in the field of dynamic loading, this topic has been singled out for separate discussion in section 4.6.

Torque resulting from acceleration and retardation

To start a body rotating or change its speed of rotation, the theorem of moments from engineering mechanics dictates that a moment must be applied. In the case under discussion the moment concerned is always torque and the axis of rotation is always the rotation axis of the machine. The relation can therefore be considered as one-dimensional. Total active torque can be meaningfully represented as the difference between the driving end torque M_{Din} and the driven end torque M_{Dout}, as shown diagrammatically in Fig. 4.1.

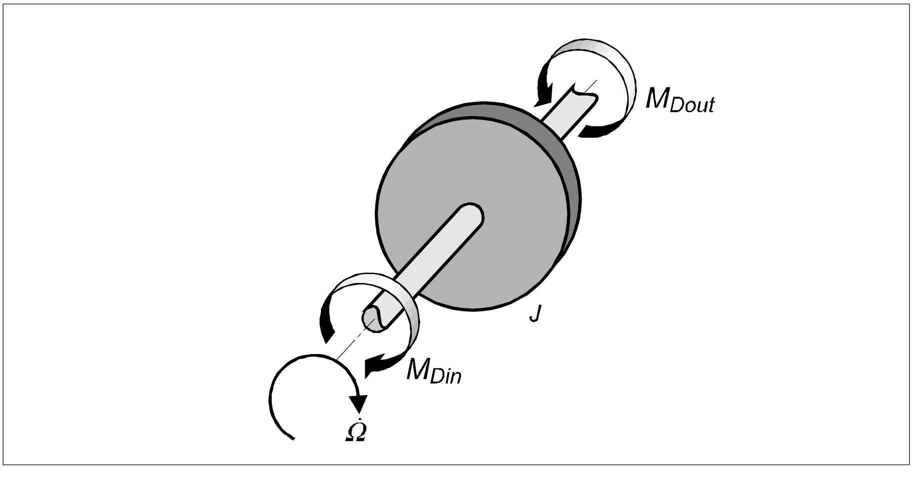


Fig. 4.1 Torque resulting from acceleration and retardation

$$M_{Din} - M_{Dout} = J\,\dot{\Omega}$$

In this case the dot on top of the variable designates the derivative with respect to time, whereby $\dot{\Omega}$ is the rotary acceleration. The mass moment of inertia J which should be used here depends on the section of shaft train for which the theorem of moments is evaluated. In that case the torque values to be substituted for M_{Din} and M_{Dout} are the intersecting moments acting at the respective end of the section considered as a unit. This means the internal torque acting at these points in the shaft train.

If for instance when starting up the machine the driven machine (brake) is idle, whereas the drive machinery is applying a given torque to the drive side, the mass moment of inertia of the rotating parts of the drive machinery should be included. On the other hand the drive machinery acts as the source of a torque. The section is therefore outside the machine and its mass moment of inertia does not enter into consideration. The reverse applies when the drive machinery is idle during retardation. However, if both machines are working against each other they are both acting as sources of torque, and only the section of shaft train that lies between them is considered, on the basis of the mass moments of inertia that are present in that area.

This line of reasoning deliberately ignores the fact that individual shaft sections can rotate at different speeds from one another due to torsional deformation. This aspect will be fully discussed in the section on torsional vibration below.

The described application of the theorem of moments can be generalized without any problem even for the case of transmissions in the shaft train. If we designate the various angular velocities of individual shaft sections as $\Omega_1, \Omega_{2,} \ldots$ and the mass moments of inertia of these shaft sections as $J_1, J_{2,} \ldots, \ldots$ the theorem of moments yields an expression for the torque in the following form:

$$M_{Din} - M_{Dout} = J_1 \; \dot{\Omega}_1 + J_2 \; \Omega_2 + \ldots$$

At the configuration stage in particular, the rotary accelerations operating in an application are often unknown. But if the change of speed and the time span within which this change takes place are known, a simple estimate can be made. Rotary acceleration is assumed to be constant:

$$\dot{\Omega} \approx \frac{\Delta\Omega}{\Delta t} = 2\pi \frac{1\,\mathrm{min}}{60\,\mathrm{s}} \frac{\Delta n}{\Delta t}$$

By way of example, torque is measured with the aid of a type T34FN torque transducer in an automated thread tapping machine. In this application the speed accelerates from zero to 4000 min^{-1} within just 14 ms.

Then the speed change Δn and the time span Δt required for this to occur are given as

$$\Delta n = 4000 \ \mathrm{min}^{-1} \qquad\qquad \Delta t = 14 \ \mathrm{ms}$$

The resulting expression for the rotary acceleration is then

$$\dot{\Omega} \approx 2\pi \frac{1\,\mathrm{min}}{60\,\mathrm{s}} \frac{\Delta n}{\Delta t} = 29.920\ \mathrm{s}^{-2}$$

Starting with the least critical case, the mass moment of inertia that has to be overcome during rotary acceleration is simply that of the torque transducer itself. The torque acting within the torque transducer during the given rotary acceleration at the measuring point then simply depends on the proportional mass moment of inertia exhibited by the portion of the torque transducer that faces away from the drive. For the HBM T34 FN torque transducer this is

$$J_1 = 3.1\ \mathrm{kg}\cdot\mathrm{mm}^2 \qquad \text{or} \qquad J_2 = 74.9\ \mathrm{kg}\cdot\mathrm{mm}^2$$

Using the rotary acceleration $\dot{\Omega}$ defined above the dynamic torque is derived as

$$M_{Ddyn\,1} = 0.09\ \mathrm{N}\cdot\mathrm{m} \qquad \text{or} \qquad M_{Ddyn\,2} = 2.24\ \mathrm{N}\cdot\mathrm{m}$$

In this case, therefore, the torque induced by the acceleration of the torque transducer exceeds the nominal torque of 2 N·m if an unfavorable situation is selected for the driving end. The effect is usually even more marked, since the mass moment of inertia of the transducer must be added to the mass moments of inertia of the other components in the shaft train.

When making an estimate of this kind, however, it must be kept in mind that constant rotary acceleration represents the most favorable case possible. If rotary acceleration is not constant but the same overall change of speed is reached within the same time span, an even higher rotary acceleration must occur temporarily. This is normally the case in practice. In the worst case there may even be impact-like torque peaks.

Oscillating torque

In this section, oscillating torque is understood to mean that an alternating torque component (dynamic component) is superimposed on a torque component which is constant or only slowly changing over time (average torque). The oscillating component may or may not be periodic. This section pays the closest attention to the sources which give rise to this kind of time-related behavior in torque. This illustrates that the subject is different from a consideration of torsional vibration, which is concerned with the reaction of the entire mechanical system when torque changes with time in the ways described. For a discussion of vibration phenomena and effects the reader is referred to section 4.6.

The principle of oscillating torque is shown in Fig. 4.2. In this example the average torque is constant and the dynamic component is periodic. A periodic or quasi-periodic dynamic component of this sort is typical of many effects which occur in practice:

- Forces of gas pressure in reciprocating engines
- Forces of mass in crankshaft drives and connecting rods
- Forces acting on teeth in transmissions
- Periodic aerodynamic forces that are prone to occur in the interplay between the rotors and stationary guide blades on ventilation fans or turbines

The first reason for the great significance of oscillating torque is that because it is superimposed on the average torque, it causes peak torque to be significantly higher than average torque. As a rule this means that when configuring a shaft train, and not least when selecting a torque transducer, if the dynamic torque components are not known they must be estimated.

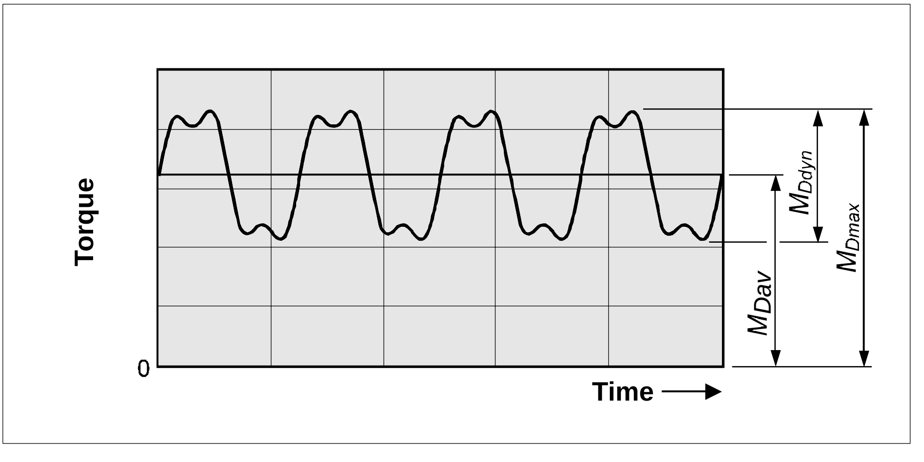


Fig.4.2 Superposition of the average torque and the dynamic torque component

Two practical examples are shown in Fig. 4.3. A low-pass filter was used to make the average torque visible. At this point it should be noted that it is not uncommon for such a filter to be used without the practitioner realizing that crucial dynamic components remain hidden. The effect of a mechanical low-pass filter is much the same, though in this case not all of the mechanical components in the test bench are actually subjected to the highly dynamic torque components.

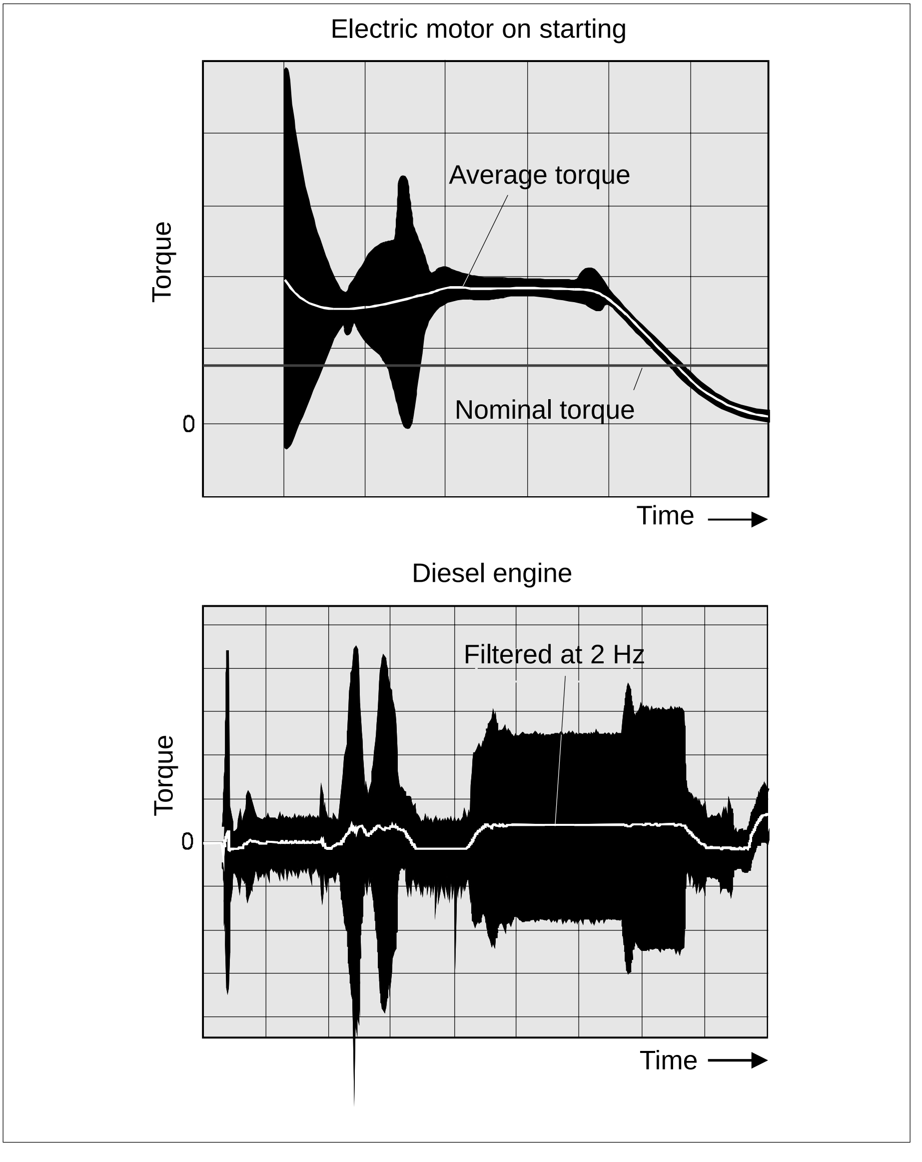


Fig. 4.3 Measured dynamic torque curve

A quantitative measure of the amount of torque oscillation relative to the effective average torque is the shock factor, defined as the ratio of maximum torque to average torque:

$$k = \frac{M_{D\max}}{M_{D\mathrm{av}}}$$

It is possible to make a generalized statement that shock factors in internal combustion engines are generally between 2 and 10. The background to this and the relationship between the number of cylinders and the principle (Otto or Diesel) will be explained below.

The second reason for the great significance of oscillating torque is its role as a possible excitation for torsional vibration. To consider this aspect it is necessary to examine both the amplitude and frequencies of the dynamic component. It should be noted that a periodically oscillating torque that is not in the form of a purely harmonic sinusoidal motion contains not only the fundamental frequency but also components with frequencies that are all multiples of the fundamental frequency.

A common application in which oscillating torque occurs to a particularly strong degree is that of the internal combustion engine. In connection with torque measurement, such applications are typically found in the various test benches within the automobile industry. The cause of torque oscillations is the conversion of the oscillating motion of pistons and connecting rods into rotary motion. The forces which occur in this process, and which are converted into torque by means of the connecting rods, are on the one hand the gas pressure forces that drive the pistons and on the other the mass forces required to accelerate the oscillating masses. Fig. 4.4 shows the torque curve for a four-cylinder diesel engine over two revolutions of the crankshaft (one complete operating cycle).

Torque can occasionally operate completely counter to the direction of the average torque. The effect of oscillations during use is generally greatly reduced with the aid of flywheels. Regarding their effect on torque measurement, it is therefore crucially important whether the torque transducer is located in the section of shafts between the engine and the flywheel or beyond the flywheel. The strongest torque oscillations operate between the engine and the flywheel.

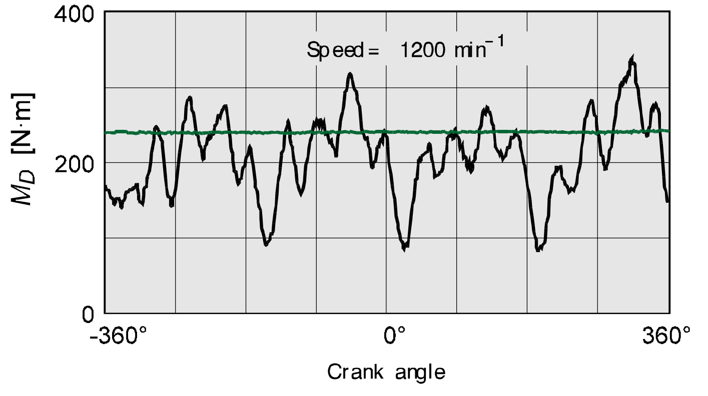


Fig.4.4 Torque in a four-cylinder diesel engine

The following points can often be helpful for the purpose of judging how different types of engines behave with regard to an oscillating torque:

- The ratio of the amplitude of the oscillating torque component to average torque is higher in engines with a lower number of cylinders than in engines with a greater number of cylinders.
- The oscillating torque component is greater in diesel engines than in comparable Otto engines.
- At low engine speeds the predominating influence comes from the gas pressure forces, whereas at higher speeds the dominant influence is from the acceleration of the oscillating masses. In a typical four-cylinder Otto engine the transition between the two regimes occurs at around 4000 min^{-1}.
- The fundamental frequency of the oscillation in a *k*-cylinder four-stroke engine is

$$f_{eng} = 60\,\frac{s}{min}\,\frac{k}{2}\;n$$

- The fundamental frequency of the oscillation in a *k*-cylinder two-stroke engine is

$$f_{eng} = 60\,\frac{s}{min}\;k\;n$$

- Components with considerable amplitudes are also usually present as higher harmonics of the fundamental frequency (see Fourier series in chapter 6).

Starting torque and switching operations in electrical machinery

The characteristic curve of an electrical machine, regardless of whether it is a motor or a retarder, shows that the active torque is dependent on the rotation speed. The torque designated as the nominal torque in such machines is the torque at nominal speed. Peak torque is significantly higher, for instance when starting up.

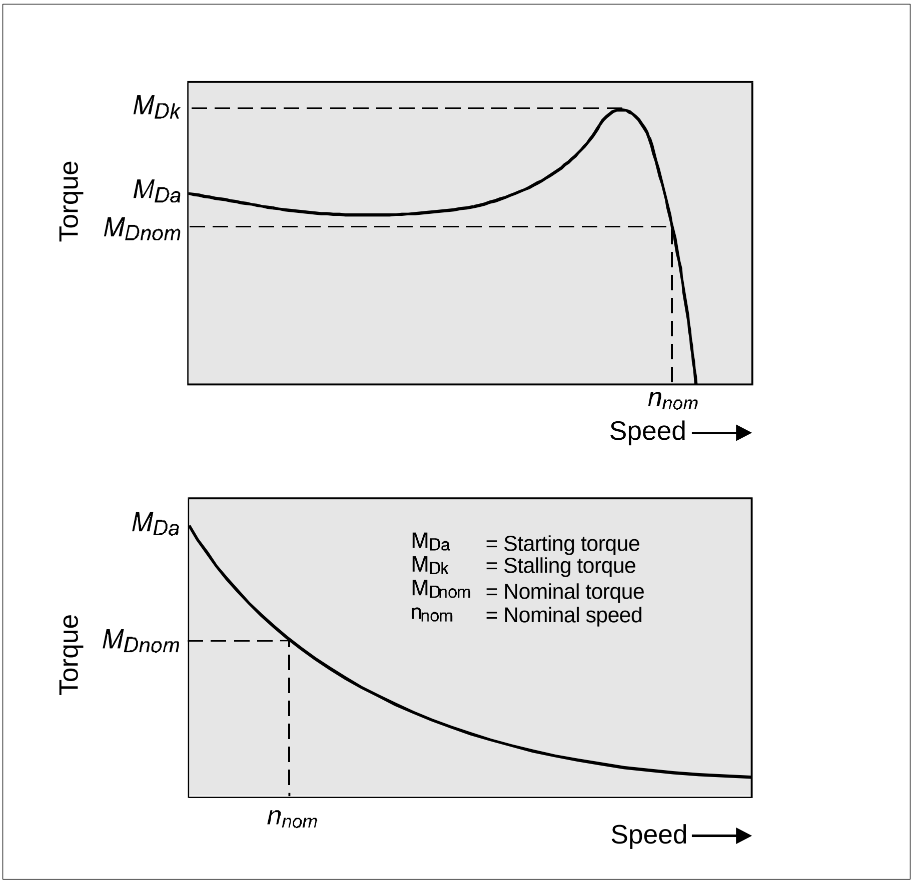


Fig. 4.5 Typical characteristic curves of an electric motor

As the characteristic curves in Fig. 4.5 show, the starting torque that a machine can apply at zero speed is often greater than the nominal torque. In the upper illustration showing the curve of a machine with shunt characteristics, another torque peak known as the stalling torque occurs just below nominal speed. Typical examples are DC shunt-wound motors, three-phase shunt-wound motors, all squirrel-cage motors and slip ring motors. The lower illustration shows an example of a machine with series characteristics in which the torque falls away continuously with increasing speed. Typical examples are DC series-wound motors, AC series-wound motors, three-phase series-wound motors and repulsion motors. In both cases it must also be reckoned that when starting up and retarding the machine, the effective torque exceeds the machine's nominal torque.

It should be noted that the characteristic curves discussed above all show the average torque for the speed concerned. Other effects that occur in electrical machines cause additional dynamic torque components. In all mains-fed three-phase machines, torque oscillations occur as a result of transient electromagnetic interactions between the machine and the mains. The starting torque in all synchronous motors oscillates at a rate of double the prevailing slip frequency. Thus when 50 Hz machines are starting up, they run through frequencies of 100 Hz to 0 Hz.

Another source of dynamic torque increase in electrical machines is represented by switching operations when an electrical machine is switched to another operating mode during the transition between starting mode and operating speed. A typical example is switching an electric motor from star connection to delta connection. Very irregular torque can occur in these circumstances.

4.3.3 Parasitic loads

Definition and significance

The term parasitic loads refers to all moments and forces which can act on a transducer in addition to the intended measured quantity. In the case of a torque transducer these are chiefly lateral and longitudinal forces, and bending moments. Definitions for these are given in Appendix A.

Torque transducers are designed so that ideally the deformation caused by parasitic loads has no effect on the measurement signal. This is achieved by configuring and wiring up the SGs in a special way which ensures that strain signals from the individual SGs are added together provided the strains concerned are caused by a torque, but cancel one another out if the strains are caused by parasitic loads.

Nonetheless, cross-talk onto the torque signal is a possibility if the parasitic load is very large, since there is always a certain amount of manufacturing variance.

But since such errors are statistically scattered, the influence a given parasitic load has on the torque signal is not predictable. If cross-talk onto the torque signal occurs, its magnitude and sign usually depend on the direction of the axis of the bending moment or the direction of the lateral force respectively.

Parasitic loads can often be large enough to cause the destruction of the torque transducer. This risk is increased by the fact that despite a certain cross-talk onto the torque signal, parasitic loads are frequently overlooked or their true order of magnitude cannot be estimated when only the torque signal is being analyzed.

Origin

On the one hand parasitic loads can be directly caused by the external forces and moments acting on the torque transducer. Typical examples are the weight of a joint shaft or the tensile forces on a belt pulley. On the other hand all the parasitic loads mentioned can also be caused by the mounting conditions, due to the distortion which occurs when compensating elements are missing (couplings, joint shafts) in conjunction with inadequate alignment of the components in the shaft train.

The effects of different mounting conditions on torque transducers and their consequences for parasitic loads are discussed in greater detail in chapter 5.

Permissible parasitic loads

The ability of a torque transducer to measure can consequently only be guaranteed if parasitic loads are kept in check to some extent.

It should be noted that even at the point where the upper limit is reached, some effect on the measurement signal may be perceptible. It must also be borne in mind that the effects of different kinds of parasitic loads are superimposed on one another and therefore the magnitude of the acceptable parasitic loads must be reduced for each individual load type (bending moment, lateral force, axial force) if more than one such parasitic load is active at the same time.

The permissible upper limits (parasitic load limits) for HBM torque transducers are stated in the relevant specifications. Detailed information on their definition and interpretation are given in Appendix A.

Spatially fixed and rotating parasitic loads

Like torque, parasitic loads are also dynamic as a rule. However, it is useful to single out two special idealized cases which in many cases represent the most important properties of the time-related curve.

On the one hand there are types of parasitic loads which rotate with the torque transducer. At constant speed such parasitic loads act statically upon the torque transducer, and quasi-statically when operating at gradually changing speed. If cross-talk onto the torque signal occurs in such a case, a constant component appears to be added to the torque.

Other parasitic loads act in a constant direction regardless of the angle of rotation, and are therefore spatially fixed. At constant speed, parasitic loads of this type act on the torque transducer as rotating loads. If cross-talk onto the torque signal occurs, the above-mentioned direction dependence will cause the parasitic load to act as a signal component that oscillates as a function of the angle.

This classification can be used for both radial forces and bending moments. On the other hand it is meaningless in the case of longitudinal forces.

In the case of direct external forces and moments it is usually easy to see whether a loading rotates with the transducer or is spatially fixed. A classic example of a jointly rotating force is an unbalance force, and typical spatially fixed forces are the weight forces or tensile forces in belt drives.

Even in the case of parasitic loads that result from distortion due to misalignment, both cases occur. If two shaft sections need to be linked and their axes are aligned but the connecting elements are not centered or correctly angled on the axis, the assembly process gives rise to a jointly rotating deformation that also causes jointly rotating forces or moments. The effect on other jointly rotating parts such as a torque transducer is therefore that of a static load. This situation will be referred to as a flange error or centering error (see Fig. 4.6).

By contrast if the axes of the two shaft sections that have to be linked are not in alignment, this misalignment gives rise to a spatially fixed distortion. This subjects all rotating parts to forces and/or bending moments which are rotating from the perspective of an observer rotating along with them. Such a situation arises for instance due to a static misalignment which was not fully eliminated at the assembly stage (see Fig. 4.7).

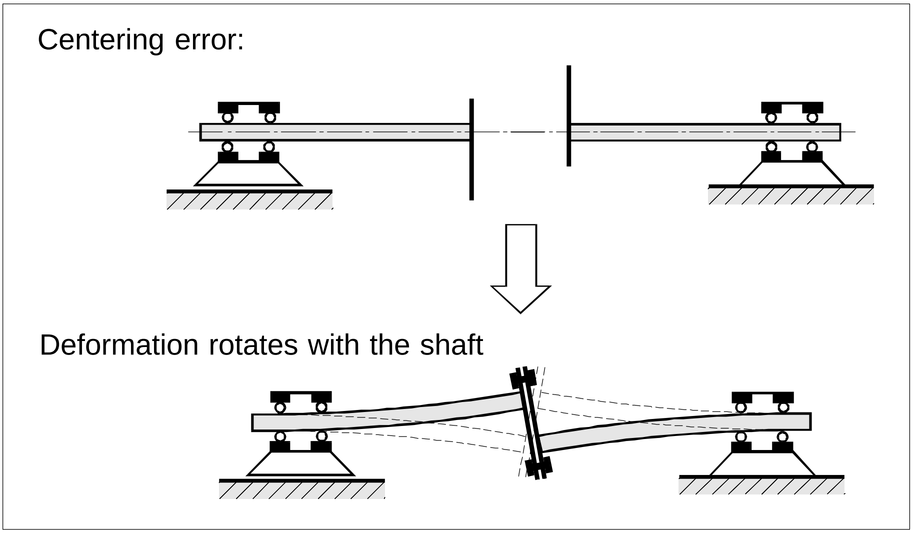


Fig. 4.6 Bending deformation due to a radial centering error

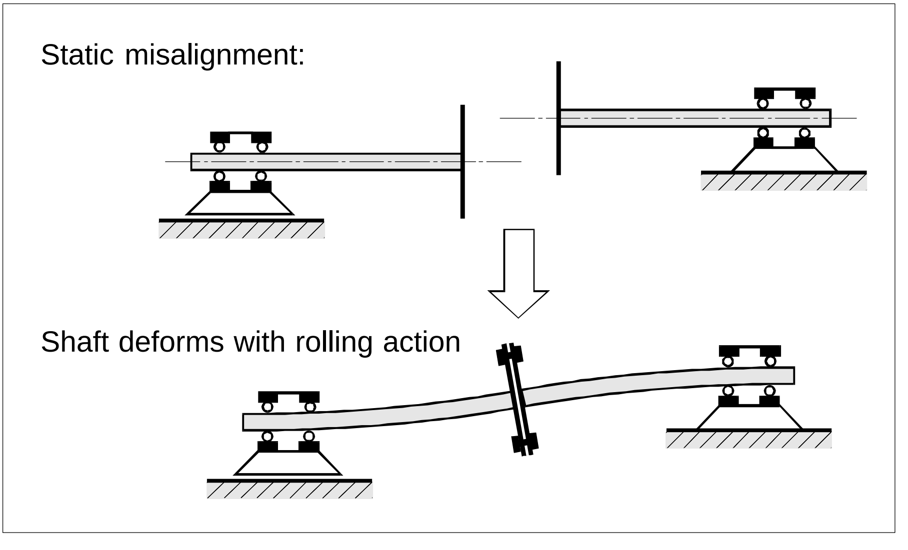


Fig. 4.7 Bending deformation due to static parallel offset

4.4 Accuracy

A discussion on the subject of accuracy divides up on closer consideration into a number of individual aspects. These are explained in some detail in chapter 7 and Appendix A. Quantitative determination of these aspects uses a large number of characteristic quantities that are to be found in a torque transducer's specifications.

- Accuracy class
- Classification under the calibration regulations
- Measurement uncertainty
- Repeatability and reproducibility
- Linearity
- Hysteresis
- Environmental influences (explained below)

When interpreting numerical information about the various properties listed, it must be borne in mind that these are generally given as a percentage in each case, but the reference quantity for this percentage is not standard. Two approaches are common: first, to refer to the nominal torque of the transducer concerned (also known as the full scale value, or full-scale related specifications), and second to the prevailing torque (also known as the actual value, or actual-value related specifications).

In the latter case it must be remembered that the respective definitions, guidelines or regulations contain the qualification that actual-value related specifications relating to accuracy need only be met with effect from a defined minimum value. This benchmark is at least 10 or 20 % of the full scale value.

Translation of these specifications into concrete computational expressions requires precise knowledge of the definitions on which they are based. For this reason the reader wishing to find out more about the quantitative evaluation of the various aspects of accuracy is referred to chapter 7 and Appendix A. The definitions and practical significance of the individual characteristic quantities are explained in those sections.

4.5 Environmental influences

Whilst torque transducers are sturdy, they are also sensitive electronic instruments. Certain environmental conditions must be fulfilled in order to ensure that they work perfectly. Harsh environmental conditions can reduce the accuracy of such a device or even impair its ability to function. In extreme cases, environmental conditions such as the harmful effects of chemicals can destroy a transducer.

Manufacturers generally define the environmental conditions for using their torque transducers, and these must be taken into account in the choice of transducer. These conditions can often be fulfilled by suitably designing the site where the transducer will be installed.

4.5.1 Dust and foreign bodies

The extent to which a device is protected against penetration by foreign bodies and dust is indicated in the first digit of the two-digit IP protection class according to DIN EN 60529. Dust and other foreign bodies small enough to find their way into a torque transducer can affect the proper operation of a torque transducer. It is not uncommon to find dust produced by mechanical braking systems, or even soot, in the vicinity of torque transducers. As well as the question of a transducer's general ability to operate, another matter which is significant for assessing the sensitiveness of a torque transducer to dust is the possibility of a mechanical or electrical shunt.

A mechanical shunt (torque shunt) causes a false reading in the measurement result due to the fact that not all of the torque is conducted into the transducer via the intended measuring point. An electrical shunt is brought about by electrically conducting particles such as metallic dust, soot or graphite causing a false reading when the electrical signals follow undefined pathways.

Assignment to a particular protection class comes about as a result of standardized test conditions.

4.5.2 Liquids

The extent to which a torque transducer can be exposed to liquids is indicated by the second digit of the IP protection class.

There is not only a question of how waterproof a transducer is, but also whether chemical interactions are likely, especially if liquids other than water are involved.

4.5.3 Chemicals

Torque transducers may come into contact with various chemicals during use. These can include cleaning agents, lubricants, fuels and hydraulic fluids.

Interaction with the materials in a torque transducer may occur in the SG (backing foil, covering), in the plastics materials on the rotor, in the electronic PCBs and even in the metals from which the rotor and stator housing are manufactured.

There are so many different chemicals that torque transducer manufacturers are not in a position to predict the effects of every possible chemical on every possible component or material in a torque transducer. This matter should be discussed and clarified between the user and the manufacturer in each individual case. It is a manufacturer's duty to disclose the materials used. The user can then make use of this information to check whether any incompatibility exists with the chemicals to be expected in the operating environment.

4.5.4 Thermal conditions

When estimating the thermal conditions encountered while using a torque transducer, part of the process is to fully consider the conditions in the external environment (outside temperature or room temperature). The other aspect is to take account of the temperature changes which arise when the application is running, including not only heat sources but also any cooling mechanisms which may be present.

Since the usual measurement principle based on SGs determines torque indirectly from a measurement of strain, it is important to take account of thermal strain interactions. These can only be compensated for to a certain extent. Electronic components are only usable and their properties are only thermally stable within certain temperature limits. The permissible temperature range for a torque transducer is therefore limited and is usually divided into different sections.

When thermal conditions in the torque transducer's operating environment are known, it is then necessary to deduce, depending on the metrological requirements, how much emphasis to place on the nominal temperature range (specifications apply), the operating temperature range (still possible to measure) and the storage temperature range. These ranges can be defined by comparing the specifications shown in the appropriate data sheets with the information given in Appendix A.

For the nominal temperature range, the upper limits for the influence of temperature on the output signal are stated in the specifications. A distinction is made between the influence of temperature on the zero point, which causes a parallel shift in the characteristic curve of the transducer, and the influence of temperature on the sensitivity, which causes a change in the slope of the characteristic curve. For a quantitative description of these effects see Appendix A.

A status with a changing or non-uniform temperature distribution exhibits a stronger temperature influence than one having a static, uniform temperature distribution when each differs from the reference temperature by the same amount. In addition to temperature fields that change over time, the type of status that does not change over time also falls into this group when, for instance, heat arising from bearing friction or from internal combustion engines is conducted via the shaft train to the torque transducer, whilst at the same time its external circumference is being cooled by the air sweeping over it as a result of its rotation.

The influence of changing or non-uniform temperature distribution can be minimized. One approach is to directly cool the components in which heat is being generated so that the heat flux to the torque transducer is minimized. When configuring the cooling system it is important to keep an eye on the symmetry in order to avoid non-uniform distribution. The effect on the torque transducer can also be minimized by thermal shielding, for instance by using suitable couplings such as the multi-disk type.

4.5.5 Humidity

Humidity does not refer to water in the form of droplets, but rather to the amount of water vapor dissolved in the surrounding air. As a rule this is expressed as a percentage of relative humidity, where 100 % relative humidity refers to saturation. When saturation is reached, no further water vapor can be dissolved in the air and it begins to condense. Particularly critical are conditions of condensing humidity in which electronic components can become covered with dew, since this can cause an electrical shunt. Humidity can also have an effect on electronic components and plastics. In temperatures below freezing point condensing humidity can even freeze, and this can lead to a mechanical force shunt or a torque shunt. Both effects can cause a false value reading.

4.5.6 Electromagnetic compatibility (EMC)

The electromagnetic conditions within the environment in which a torque transducer operates have an important influence on the way the transducer functions, due mainly to the electronics incorporated in the device (conversion of the rotor signals, transmission of signals from the rotor to the stator). Moreover, as with other transducers, there is the usual set of problems arising from interactions with cables.

Typical sources of electromagnetic interference in power test benches for motor vehicle engines or transmissions are the high ignition voltages in internal combustion engines, as well as the effects caused by large electrical machines (for more details on electromagnetic interference, see chapter 5). If interference effects exceed the permissible limit for the torque transducer, measures must be taken to reduce them at the source of the emissions. Otherwise some form of shielding or screening must be used.

4.6 Vibration

The following section summarizes the vibration engineering concepts that are relevant from the design and layout point of view. The general basic concepts of vibration engineering can be found in Appendix B, and methods of vibration analysis are discussed in chapter 6.

Vibration plays an all-embracing role in the overall field of dynamic mechanical loading on torque transducers, since even though it does not as a rule cause dynamic load components in its own right, it does act as an amplifying mechanism. Certainly it is often the deciding factor when trying to answer the question about whether a certain dynamic load will actually lead to a critical operating status, and if so in what frequency band. This applies both to a loading due to a dynamic torque, the possible sources of which are given in section 4.3.2. and to dynamically operating parasitic loads such as rotating bending moments, as discussed in section 4.3.3.

Vibrational properties are always system properties, for which reason the frequently asked question about the natural mechanical frequencies of torque transducers does not go far enough. On the contrary, a configuration such as a test bench must be investigated as a whole. This chapter therefore concentrates chiefly on the available options for computing and estimating natural frequencies.

A critical operating status is always to be expected when a natural frequency coincides with the frequency of an excitation, in other words when resonance is present. It should be noted in such a case that many periodic excitations are not precisely in the form of a sinusoidal motion. As can be verified with the aid of the Fourier series expansion, the fundamental frequency of the excitation can be accompanied by multiples of this frequency in the excitation spectrum. This will be discussed further in the context of order analysis (see chapter 6).

Whether a resonance actually leads to a critical operating status depends on the amplitude with which the frequency concerned occurs in the excitation spectrum. The second deciding factor is the damping of this natural frequency, which is a system property. Damping is in any case very difficult to take into consideration in advance at the layout and design stage, since a decisive component often stems from complex effects such as in the fitting points. This section will therefore not dwell further on damping, but notes on design measures that can be taken to increase damping are included in chapter 5.

4.6.1 Torsional vibration

Simple mathematical mechanical models

In the case of torsional vibration, the vibrational motion is a torsion about the axis of rotation. A particularly simple system that can exhibit torsional vibration is shown in Fig. 4.8, where a massless torsion rod with torsional stiffness c_T is firmly fixed at one end, and supports a disk with mass moment of inertia J at the other end.

As mentioned above, damping is initially disregarded during modeling, since quantitative determination on the basis of physical effects such as bearing friction, air friction and processes within joints and couplings is very difficult.

The vibrational motion of the system, expressed by the torsion angle φ, is described by the following differential equation:

$$J\ddot{\varphi} + c_T\varphi = M_{Din}$$

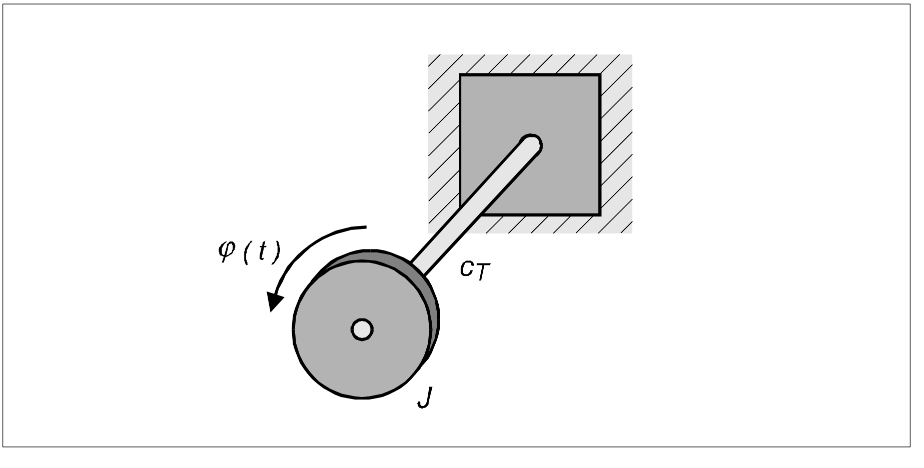


Fig. 4.8 Simple torsional oscillator: massless torsion rod with disk

In this equation $\ddot{\varphi}$ represents the rotary acceleration, c_T the torsional stiffness of the shaft, J the mass moment of inertia of the disk and M_{Din} a possible external torque that acts upon the disk and represents a vibration excitation.

At this point it is not the intention to go into detail about the general solution of the differential equation and discuss it further. The solution method is exactly the same as for the standard system of a single mass oscillator, which is itself fully discussed by way of an example in Appendix B.

As in the case of a single mass oscillator, the natural frequency is derived as

$$f_{T0} = \frac{1}{2\pi}\sqrt{\frac{c_T}{J}}$$

Yet another model for torsional oscillators, which is still fairly simple and to which it can be shown that many practical systems approximate closely, is the massless torsion shaft with two disks, shown in the drawing in Fig. 4.9.

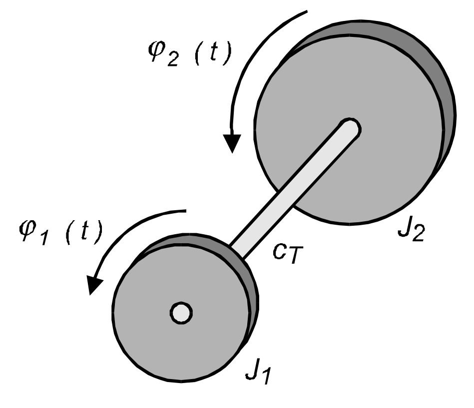


Fig. 4.9 Simple model of a practical torsional oscillator: massless torsion rod with two disks

This model also shows that when there is torsional vibration in rotating machines, generally the vibrational motion is superimposed by a rotary motion of the system as a whole. In the equation describing the motion of the two-disk model this is reflected by the fact that it applies to the difference $\Delta\varphi$ between the two torsion angles φ_1 and φ_2:

$$J_\Delta \, \Delta\ddot{\varphi} + c_T \, \Delta\varphi = \frac{J_\Delta}{J_1} M_{D1} - \frac{J_\Delta}{J_2} M_{D2}$$

with the short forms

$$J_\Delta = \frac{J_1 \ J_2}{J_1 + J_2}$$

and

$$\Delta\varphi = \varphi_1 - \varphi_2$$

The derivation of this equation will not be discussed further here. For the associated torsional natural frequency we obtain

$$f_{T0} = \frac{1}{2\pi}\sqrt{c_T\left(\frac{1}{J_1} + \frac{1}{J_2}\right)}$$

Even shaft trains that include drive transmissions can in principle be reduced to a mathematical mechanical substitution model based on disks and massless shaft sections. For this purpose the shaft train is thought of as being divided into different sections each operating at a different speed. One of these sections is defined as the reference section and its angle of rotation is the reference angle of rotation, that is to say its speed n_{ref} is chosen as the reference speed of the substitute system. The various occurrences of stiffness and inertia in this section are taken into account for the calculation with a correction factor of one. For a section with deviating speed n_{act} with the transmission ratio

$$\mu = \frac{n_{act}}{n_{ref}}$$

the substitute mass moment of inertia and substitute torsional stiffness

$$J^* = J\mu^2 \qquad \text{und} \qquad c_T^* = c_T\mu^2$$

must be used. The derivation of this method can be found in [6].

Estimating natural frequencies for real technical systems

In order to be able to estimate whether vibration problems are likely to be encountered in an application it is necessary to take the approach that the whole shaft train must be understood as a complete system and suitably represented as a mathematical mechanical model. Thus the frequently asked question about the natural frequencies of the actual torque transducer is not particularly meaningful, which is why such natural frequencies are not given in HBM torque transducer specifications.

When representing real systems in such simplified models it must first be noted that in reality each component of a rotating machine has inertia and finite stiffness, whereas in the model, stiffness and mass moments of inertia are each concentrated on separate components. The disks are idealized as rigid, and the shafts as massless. Simple substitute models can be applied on the one hand by neglecting stiffness or inertia of components if this can be justified, and on the other hand by combining the mass moments of inertia of adjacent components into one mass moment of inertia (or doing the same for stiffness coefficients).

Summarizing the mass moments of inertia of adjacent components is always meaningful if very little torsion is possible between them, in other words if there is a high degree of stiffness. Mass moments of inertia are then simply added algebraically. If a whole series of components with strongly differing mass moments of inertia are strung together, those with the relatively largest mass moments of inertia are decisive for the mass moment of inertia overall. Rarely is this the mass moment of inertia of the torque transducer.

When considering occurrences of torsional stiffness, the information of significance includes not only the stiffness of torque transducers, couplings and other components, which must be found out from manufacturers' specifications, but also the torsional stiffness of simple shaft sections. The latter can easily be computed according to the rules of engineering mechanics on the basis of the material constants, cross-section and length of the shaft section concerned. In practice all applications of any relevance to this case are shaft sections with a circular or ring-shaped cross-section. The computation for this special case can be found in Appendix C.

Summarizing the torsional stiffness of adjacent components is always meaningful if there are no components with relevant mass moment of inertia between them. In the case of torsional stiffness in components connected axially to a shaft train, this amounts to a series connection of springs, and it is then possible to use

$$\frac{1}{c_{Ttot}} = \frac{1}{c_{T1}} + \frac{1}{c_{T2}} + \dots$$

to compute the total torsional stiffness. If a whole series of shaft sections with strongly differing torsional stiffness are strung together, those with the relatively lowest stiffness are decisive for the total stiffness. If a shaft section has a segment with torsional stiffness 1 kN·m/rad and another segment with torsional stiffness 100 kN·m/rad, the total torsional stiffness is 0.99 kN·m/rad, which is almost exactly equal to the smaller of the two stiffness values. Depending on the design of the torque transducer and the other components, the stiffness of the torque transducer can be crucially significant.

However, if mass moments of inertia that are not negligibly small are so arranged that torsional stiffness occurs between them which is not negligibly large, it cannot be meaningful to combine them. For the purpose of estimating in this case, a more complex model with more than two disks must be invoked. The reader should refer to the literature for details of the computations involved [6], [7].

In certain special cases, individual sections can be considered separately. This situation is known as decoupled vibrations. Such a case exists where there is a break in the transmission of torque. Even though a complete break defeats the purpose of a shaft train, there are circumstances in which an extensive interruption can exist. This is the case in gear transmissions with play, or in components which are torsionally very elastic and have torsional stiffness which is lower than the rest of the shaft train by several orders of magnitude. If the link does in fact exist but is highly elastic, it is possible to obtain a close approximation for high torsional natural frequencies by considering the two halves of the shaft train separately. Where the natural frequencies under consideration are low, however, a close approximation comes from considering the system in total but simplifying it by assuming that all components are completely rigid except those with particularly high torsional elasticity.

Practical example of estimating a torsional natural frequency

The following section will use the above computation methods to estimate the lowest natural frequency of the typical engine test bench illustrated in Fig. 4.10. As already mentioned, there is no need to take damping into account for this particular task.

The test bench consists of an internal combustion engine, a Kuesel element (proprietary name referring to a torsionally elastic coupling for vibration damping, see chapter 5), a joint shaft, the T10FS torque flange from HBM (nominal torque 1 kN·m) and an asynchronous electric generator acting as an absorption dynamometer. For the sake of simplicity the intermediate flanges required in each case are regarded as parts of the engine or motor. The Kuesel element and the measurement flange are each thought of as being divided into two parts in such a way that the component that undergoes the main distortion is located between them.

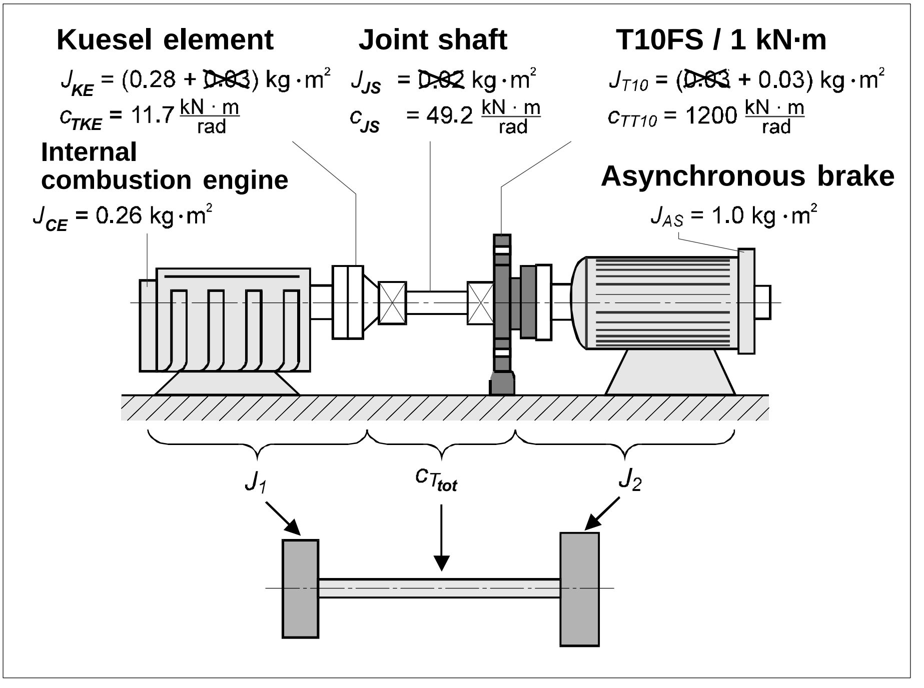


Fig. 4.10 Engine test bench and mathematical mechanical substitution model

The lower part of the figure shows a reduction to the simple mathematical mechanical model of a shaft and two disks.

The left-hand disk in the substitution model combines the mass moment of inertia of the internal combustion engine with that of the left hand part of the Kuesel element:

$$J_1 = (0.26+0.28)\ \mathrm{kg}\cdot\mathrm{m}^2 = 0.54\ \mathrm{kg}\cdot\mathrm{m}^2$$

In the substitution model, the torsional stiffness of the shaft is derived as follows from series connection of the individual stiffness values in the Kuesel element, joint shaft and torque flange:

$$\frac{1}{c_{Ttot}} = \left(\frac{1}{11.7} + \frac{1}{49.2} + \frac{1}{1200}\right)\frac{\mathrm{kN}\cdot\mathrm{m}}{\mathrm{rad}} \quad \Rightarrow \quad c_{Ttot} = 9.38\ \frac{\mathrm{kN}\cdot\mathrm{m}}{\mathrm{rad}}$$

The right-hand disk in the substitution model combines the mass moment of inertia of the right-hand part of the torque flange with that of the asynchronous machine:

$$J_2 = (0.03 + 1.0)\ \mathrm{kg \cdot m^2} = 1.03\ \mathrm{kg \cdot m^2}$$

In this simple substitution model, the mass moments of inertia of the components classed as belonging to the elastic shaft must be disregarded. These are struck out in Fig. 4.10. A comparison by order of magnitude shows that it has very little effect in this case. In cases where this is not so unambiguous, it is possible to manage by adding the moment of inertia proportionally to the right-hand and left-hand disk in the substitution model. Conversely but by the same token, the elasticity of components assigned to both disks is disregarded. These components are approximately rigid.

The natural frequency for the example system is then given approximately from the formula specified above for the natural frequency of the model of a shaft and two disks:

$$f_{T0} = 26.3\ \mathrm{Hz}$$

If the excitation frequencies that occur in relation to speed are also known, this information can be used to deduce which operating speeds may be critical. If the rotation frequency itself is present as the excitation frequency (once-per-revolution excitation), a simple conversion from Hz to min^{-1} gives the result that resonance is likely to occur in the region of 1580 min^{-1}.

As well as the rotation frequency itself the excitation spectrum often also contains higher harmonics, of which the triple is quite common. This leads to the appearance of resonance if, for instance, the triple harmonic of the rotation frequency corresponds to the natural frequency. To put it another way, if the speed corresponds to 1/3 the natural frequency, then in this example resonance could be expected in the region of 527 min^{-1}.

This concept delivers a usable approximation of the lowest natural frequency. The next higher natural frequency will indeed be significantly higher in this case, since very few disregards were needed to represent it in the model that has only one natural frequency.

However, if a significantly higher frequency range is of interest, then what was said above regarding an approximate decoupling takes effect: the Kueselelement has the effect of a decoupling point and the torsional vibration processes within the internal combustion engine which occur at significantly higher frequencies remain largely uninfluenced by the other test bench.

Further possible computations

In practice the computation methods mentioned here can be used for rough approximation only. This means that rather more profound computation is required when it comes to design and layout with regard to the problem of vibration.

In many cases it is still sufficient to take the modeling approach with ideal, massless shaft sections on the one hand and ideal, rigid disks with mass moments of inertia on the other. However, due to increasingly complex geometry, or a striving for greater accuracy, the point is reached where chains composed of such shaft sections and disks can be any length at all. Depending on the number of disks, such systems have more degrees of freedom and accordingly many torsional natural frequencies and associated mode shapes. The greater the effort expended on modeling, the greater the number of higher natural frequencies and the greater the increase in computational accuracy at low frequencies.

From this it can be deduced that a much greater effort is required if several natural frequencies (critical speeds) are to be expected in the excitation frequency ranges. For a general explanation of the behavior of systems exhibiting multiple degrees of freedom the reader is referred to Appendix B. Natural frequencies and their associated mode shapes are calculated with the aid of a computer in such cases.

Where detailed computations are required it can be meaningful to use the finite element method and software for the digital simulation of drive train dynamics.

Subcritical and supercritical operation

In the case of forced vibration, the behavior when the excitation frequencies are smaller than the natural frequency is in principle different than when they are larger. This behavior, which is explained in Appendix B by the example of a single mass oscillator excited by a force, also applies to torsional vibration. In torque measurement this has special significance for a number of contexts and relationships. In general the excitation frequencies that generate torsional vibration are almost entirely proportional to speed, as was shown above in the example of the oscillating torque in internal combustion engines.

Thus if the occurrence of resonance can be attributed to particular speeds, we refer to these as critical speeds, and here especially we say that such speeds are torsionally critical.

The effect that torsional vibration has on the torque actually operating in the shaft train, and what effect if any it has on the measured torque, depends on various factors. These factors will be discussed in the light of the example shown in Fig. 4.11.

The source of the torque is the internal combustion engine on the left of the diagram. It loads the shaft train with torque M_{Din}, which is composed of the average torque and the dynamic component. For the purpose of idealization let us assume that the average torque is constant and the dynamic component is purely harmonic.

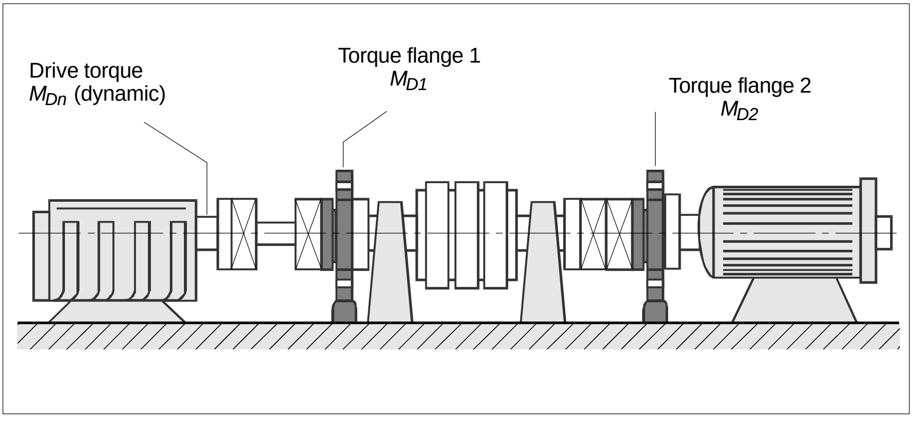


Fig. 4.11 Test bench with low natural frequency

The component identified in the sketch as torque flange 1 is separated from the torque connection to input side by just a short joint shaft that chiefly causes elasticity but exhibits a very low mass moment of inertia. The response characteristic therefore corresponds largely to that of a simple elastic spring, and any torque that is introduced has the same effect on it at every point:

$$M_{D1} \approx M_{Din}$$

By contrast, torque flange 2 in the sketch is separated from the torque connection to input side by a complete vibrating system that includes both rotational elasticity and a mechanical component with considerable mass moment of inertia, namely a flywheel.

The transmission response is therefore that of a vibrating system, as shown in Fig. 4.12. Here the input quantity is the torque from the internal combustion engine, and the output quantity is the torque at the position of torque flange 2, that is, the measured torque M_{D2}.

A general discussion of the topic can be found in Appendix B. The dependency of the measured torque on the excitation frequency or speed is clear to see. In the subcritical range, the measured torque still closely matches the engine torque. In the resonance zone there is a sharp increase. And lastly in the supercritical range, measured torque is lower than the torque produced by the engine. The mechanical low pass filter effect has appeared.

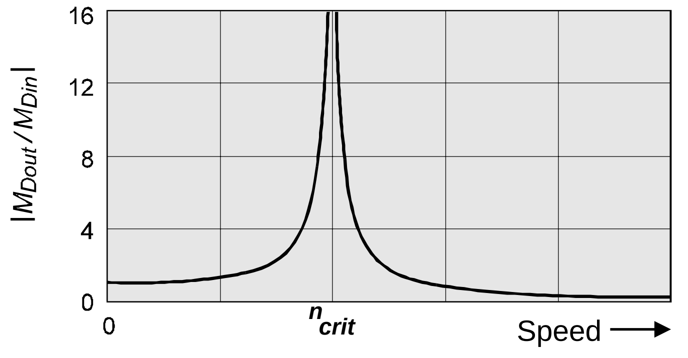


Fig.4.12 Dynamic torque for measurement flange 2 dependent on speed

Depending on the objectives of the torque measurement concerned, the decision to be taken is whether to opt for the position identified as torque flange 1 or torque flange 2 in the sketch.

Quite apart from the question of which torque should be measured, such an arrangement with a very low natural frequency can often have advantages with regard to the mechanical loading on the shaft train. Since in fact at low speeds the dynamic components of the excitation torque and unbalance forces are generally very slight, the vibration amplitudes are only relatively small when passing through the zone of resonance.

The second field in which it is possible to reach a conclusion by considering the subcritical or supercritical question is that of reaction torque measurement with the aid of pendulum mounted machines. A fundamental problem in this case is the mechanical low pass filter effect already discussed in chapter 2.

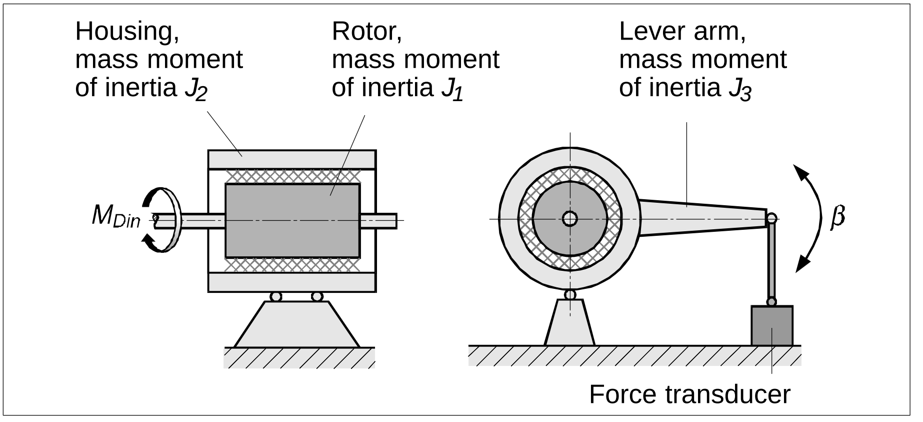


Fig. 4.13 The reaction torque measurement principle and pendulum mounted machine

This becomes clear on comparing the equations which quantitatively describe the mechanical effect of such a configuration. Using the symbols introduced in Fig. 4.13, we obtain the following equation for the dependency of the measured force F on the introduced in-line torque M_{DA} in the static case:

$$M_{Din} = Fl$$

In the dynamic case, on the other hand, the speed variation $\dot{\Omega}$ and an angular acceleration $\ddot{\beta}$ occur due to travel in the force transducer, which means that the mass moments of inertia of the machine and lever arm have an effect:

$$M_{Din} = Fl - \left[J_1 \dot{\Omega} + \left(J_2 + J_3\right)\ddot{\beta}\right]$$

In practice, however, when determining torque from the force measurement, only the equation for the static case is evaluated and the influence of the rotary acceleration acts as a measurement error. Due to the generally very high mass moments of inertia in the machines, which move on the pendulum mounted bearing, the natural frequencies found in this movement are normally low even in the case of extremely high stiffness allowing only minimal movement. This means, however, that dynamic torque at a frequency higher than the torsional natural frequency of the pendulum mounted machine does not get through to the force transducer. The average torque on the other hand, apart from the bearing friction from the pendulum mounted linkage, is correctly indicated by measuring with the aid of the pendulum mounted machine.

When planning to operate beyond the first critical speed, it must be generally borne in mind that passing through the zone of resonance places additional requirements on the performance of the drive, since part of the drive energy drains away in the vibrational motion. In order to prevent unacceptably large vibration amplitudes it is often necessary to pass through the resonance zones as quickly as possible, increasing the performance requirement even further.

Effect of damping on amplitude magnification during resonance

The damping which is actually at work in a shaft train is often not quantified, because as a rule damping acts on different components in the run and by different effects. These effects include bearing friction, shaft linkage friction, material damping and air friction.

On the other hand, damping which is indeed quantified is that from special components which are intended to introduce planned damping, such as torsionally elastic couplings (Kuesel elements). The latter are described in detail in chapter 5. As a rule if such elements are present in a shaft train, all the other damping effects are negligible in comparison. Nevertheless there are problems in using the concept in the simple mechanical model described here. The first reason for this is that, as explained above, the elasticity of the shaft train is a combination of the elasticity of the individual components acting in the manner of torsion springs connected in series. The stiffness and the damping of the torsionally elastic coupling are connected in parallel to each other, but in a series connection to the other torsionally elastic components in the shaft train. To compute the effective damping it would then be necessary to know the torsion of both sides of the torsionally elastic coupling relative to one another.

These facts are shown in the form of a spring-mass vibrating system in Fig. 4.14. It can be seen that in order for the mechanical modeling to return the damping correctly, a further degree of freedom is required. In the diagram, this degree of freedom is the displacement of point P_1. The literature speaks about half a degree of freedom in this connection, since no mass or mass moment of inertia is assigned to the point and its motion is described by a differential equation of the first order. To a close approximation the problem can be ignored if the stiffness of the torsionally elastic coupling is very low compared to the other stiffness values occurring in the shaft train, as is very commonly the case.

The second reason why it can be difficult to use damping quantitatively in the above model is that occasionally the damping in such mechanical components is friction damping. Compared to viscous damping proportional to speed, this has different properties which make mathematical modeling an extremely tedious matter. However, the manufacturers of torsionally elastic couplings give appropriate configuration advice in their documentation.

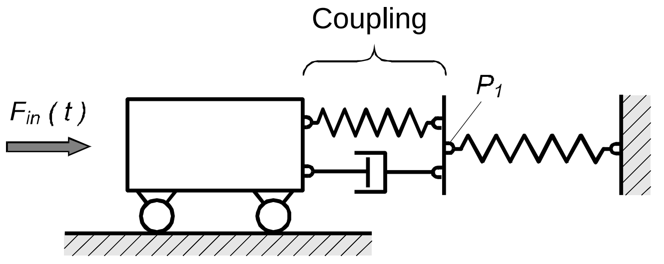


Fig. 4.14 Series connection/parallel connection of damping element

4.6.2 Bending vibration

Bending vibration in rotating machines frequently leads to considerable difficulties. However, it has secondary importance when torque is being measured, because its effect on the measurement signal is small. Although this is still commonly believed, there is practically no relevant link between bending and torsional vibration (except in special cases such as crankshaft drives, where the link is obvious), as can be seen from the article [8]. However, bending vibration has a strong influence on parasitic loads, especially bending moments and radial forces. Bending vibration is thus frequently the cause of initially inexplicable components in the torque measurement signal which can be traced back to the cross-talk explained above, and is often also the cause of overloading, and occasionally even the mechanical destruction of the transducer.

Some of the possible excitation mechanisms for bending vibration are:

- Unbalance
- Shaking motions
- Acceleration of the oscillating masses in crankshaft drives and connecting rods
- Periodic aerodynamic forces
- Dynamic instability (self-excited vibration)
- Periodic sag as a consequence of rotationally asymmetrical bending stiffness

As with oscillating torque, excitation frequencies are often proportional to the speed. Not uncommonly a particular or frequently occurring ratio of excitation frequency to speed can be the result of a special excitation mechanism. Further information can be found in chapter 6, where this topic is discussed in detail as an aspect of order analysis.

Regarding resonance frequencies and subcritical or supercritical operation, the same largely applies to bending vibration as to torsional vibration. The deliberate introduction of damping for bending vibration presents particular difficulties. The most practicable solution is damping at the bearing points (though this requires a relatively elastic bearing in order to bring about the necessary vibration amplitudes). Even in the case of bending vibration the energy of vibrational motion must be applied by the drive so that a certain minimum drive power is needed in order to pass through the resonance zones. Here again it is best to pass through the resonance zone as quickly as possible, which increases the requirement for drive power still further. Valuable pointers on this subject can be found in the literature on rotor dynamics such as [9],[10] .

Simple mathematical mechanical models

The simplest example of a mathematical mechanical model for computing bending vibration is the Jeffcott rotor shown as a diagram in Fig. 4.15, consisting of a flexurally elastic shaft assumed to be massless and a disk that has mass. This model is also known as a Laval rotor.

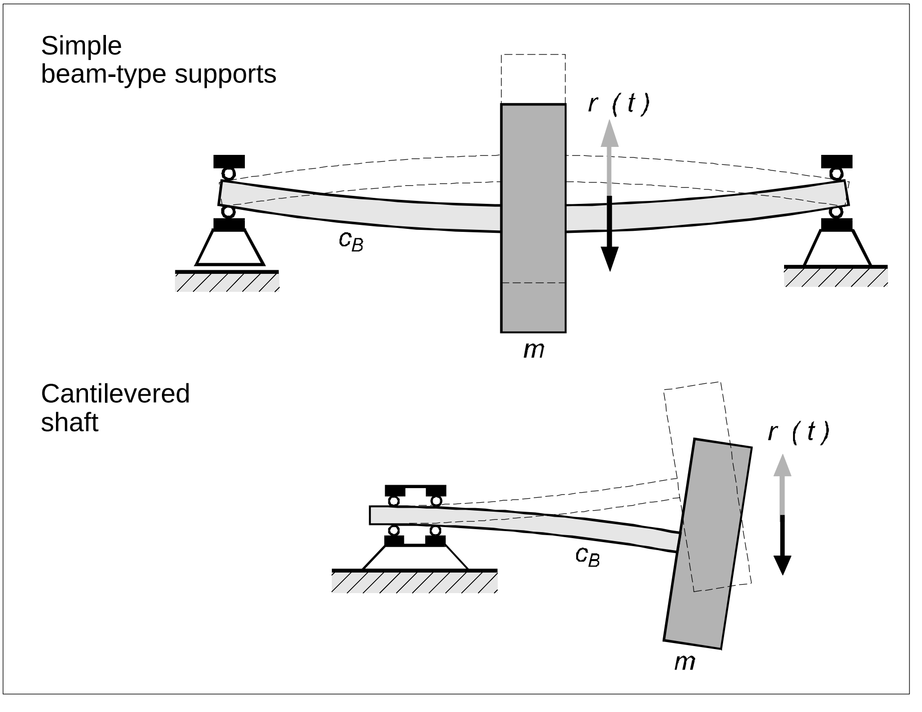


Fig. 4.15 Jeffcott rotor, different types of bearing configuration

The system is mathematically described by the bending stiffness c_B and the mass m of the disk. The bending stiffness is dependent on a material constant, the shaft length and cross-section, the shaft bearing configuration and the position of the disk (see Appendix C).

The bearing configuration is therefore important for two reasons. In the first place it determines how much shaft deformation is required for a given displacement of the disk. For example the same shaft is much stiffer if it has three bearing points rather than two. But secondly, bearings have a certain elasticity of their own, and this contributes to total elasticity according to the principle of the series connection of springs. Even so, bearing elasticity is secondary in many applications, especially when roller bearings are used. Yet the bearing blocks can also have an effect if they are designed to be highly elastic. In most cases the stiffness will be less in the horizontal direction than in the vertical direction, a condition known as anisotropic bearing stiffness. The Jeffcott or Laval rotor model ignores the effects of mass moment of inertia arising from disk tilting, which can be seen occurring in the lower example in Fig. 4.15.

The Jeffcott rotor is (considering the mechanical mathematical model) a single mass oscillator. Only the shape of the spring is different than in the standard model, and it has two degrees of freedom since it can bend horizontally as well as vertically. As is the case with a single mass oscillator (see also Appendix B), the natural frequency is derived from

$$f_{B0} = \frac{1}{2\pi}\sqrt{\frac{c_B}{m}}$$

In the case of the anisotropically elastic bearing configuration, the stiffness differs in the horizontal and vertical directions. Consequently there are also two different natural frequencies, but these are normally very closely adjacent. In practice this reveals itself as a wider resonance zone, since the ranges in which both natural frequencies resonate are overlapping.

As in the case of the models for torsional vibration, the mathematical mechanical model illustrated in the present case is characterized by an abstraction of a concentrated mass and stiffness. But in real systems, mass and stiffness are always continuously distributed over all the components.

A complete contrast is the model of the continuously uniform rotor, in which it is assumed that mass and stiffness are distributed uniformly over the entire length. This idealization, shown in Fig. 4.16, often delivers a usable model when a model with individual concentrated masses is not practicable even for the purpose of an approximation.

The natural frequencies obtained for this example are:

$$f_{Bi} = \frac{1}{2\pi}\frac{i^2\pi^2}{l^2}\sqrt{\frac{EI}{\rho A}} \qquad \text{where } i = 1, 2, ..., \infty$$

For each natural frequency there is a mode shape. The figure shows the two lowest mode shapes.

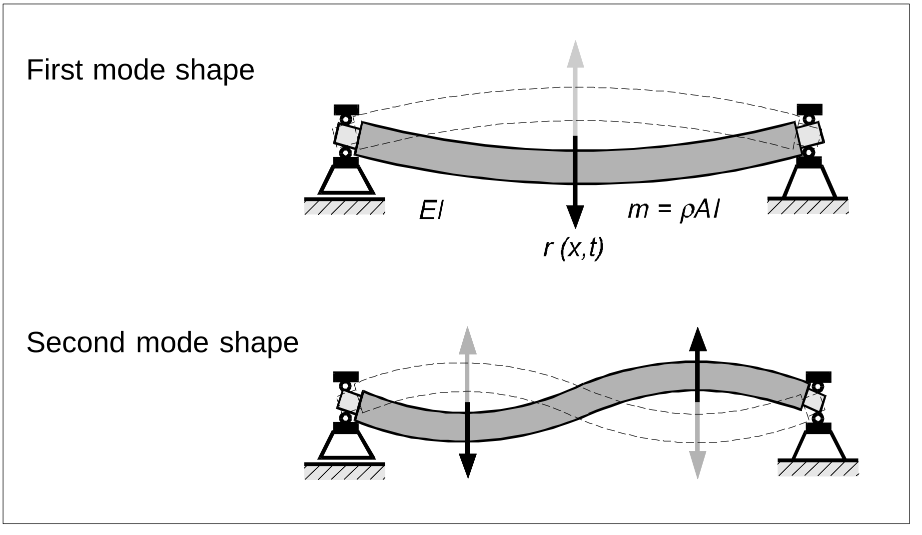


Fig. 4.16 Shaft with uniformly distributed mass and stiffness

A characteristic of all continuous vibrating systems is that in theory there is an infinite number of natural frequencies. Often only the lowest of these are relevant. If the first natural frequency is the only one of interest, the Jeffcott rotor is often a feasible approximation. On comparing the shaft in Fig. 4.16 with the Jeffcott rotor shown in Fig. 4.15 above, it can be seen that for the same mass and cross-section there is a difference of 30 % between the natural frequencies of the two rotary systems.

Applying mathematical mechanical models to real technical systems for the purpose of approximation

Representing real technical systems in the most simple mathematical mechanical models is generally more complex in the case of bending vibration than in the case of torsional vibration. This is because the bearing configuration has a crucial influence on the stiffness properties. It is almost always more complex than in the simple models.

It is more often possible to decouple individual sections of the shaft train than is the case with torsional vibration. This can always be done if two conditions are met. The first is that in the separation point between sections it must not be possible for bending moments to be transmitted, perhaps by interposing a joint or a coupling with high bending elasticity. Secondly the shaft must also be secured against radial displacement by having bearings at the point of separation. Moreover, as with decoupling in the case of torsional vibration, it is also possible to achieve an approximately valid decoupling for high frequencies if the couplings used have very high bending and radial elasticity. By this token, many joint shafts can be treated as decoupled. The shaft with uniformly distributed stiffness and mass which was introduced above then becomes a suitable computation model.

Systems with a belt pulley, where the torque transducer is protruding as shown in Fig. 4.17, are very good to model. They can be modeled as a cantilevered Jeffcott rotor in accordance with the lower figure in Fig. 4.15. The bending stiffness is then a combination of radial stiffness and bending stiffness of the torque transducer.

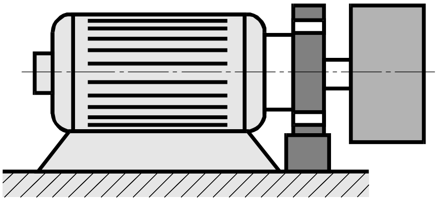


Fig. 4.17 Torque flange with cantilevered pulley wheel

Unbalance

When discussing unbalance the first intention is to give consideration to rigid rotating bodies (from here on referred to as rotors). Unbalance can be divided into two basic types which always occur more or less superimposed in real technical systems. They are known as static unbalance and dynamic unbalance.

In the case of bodies with static unbalance, as shown in Fig. 4.18, the center of gravity of the rotor is not on the axis of rotation. A radial force is necessary to keep the rotor in rotation about its axis. In the formal terms of mechanics: the principal axis of inertia of the rotor is shifted in parallel relative to the axis of rotation.

A quantitative measure that can be used is the distance of the center of gravity from the axis of rotation, known as the eccentricity ε. If we think of the unbalance shown in the sketch as an individual unbalance mass m_U located eccentrically on the otherwise perfectly balanced body with mass m at radius r_U, the eccentricity is derived as

$$\varepsilon = \frac{r_U\, m_U}{m + m_U}$$

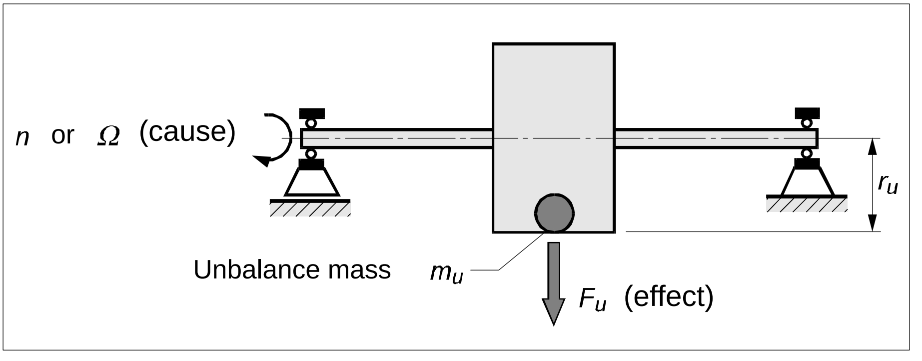


Fig. 4.18 Static unbalance with symmetrical body with an unbalance mass

A further quantitative measure of unbalance is directly based on the pattern of the unbalance mass model. The unbalance (here understood as a quantitative measure) is defined as the product of unbalance mass and the radius, giving the following:

$$U = m_U\, r_U$$

From this it is an easy matter to compute the unbalance force depending on the angular velocity of rotation:

$$F_U = m_U\, r_U\, \Omega^2 = U\, \Omega^2$$

This is the force with which the unbalance mass pulls outward on the body. Since the unbalance force is the quantity that can be measured for the purpose of balancing, and the unbalance (quantitatively understood) can be directly and unambiguously computed from the unbalance force in connection with the angular velocity or speed when known, this is the interpretation most commonly used in balancing.

As well as static unbalance, there is also dynamic unbalance (or more correctly: kinetic unbalance). This time the center of gravity of the body is in fact located on the axis of rotation but the symmetry is disturbed in such a way that the body cannot rotate about the axis of rotation unless an external moment is applied. In the formal terms of mechanics: the principal axis of inertia of the body is tilted relative to the axis of rotation. Clearly this can be represented by the conceptual combination of an ideally balanced rotor with two unbalance masses, as shown in Fig. 4.19.

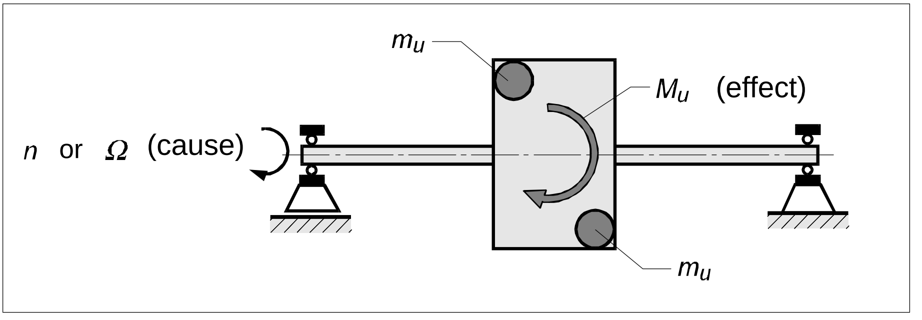


Fig.4.19 Dynamic unbalance with body with two equal unbalance masses

This case involves two equal unbalance masses which cancel out each other's effect on the center of gravity. However, they differ with respect to their axial positions. In the parlance of balancing technology there are said to be two balance planes, thought of as planes orthogonally intersecting the axis of rotation. On noticing that during rotation each of the two unbalance masses pulls away from the rotor with the same force but from different planes, it becomes clear that a tilting moment or bending moment remains.

As with static unbalance, a quantity known as dynamic unbalance can be quantitatively defined and can be used to directly compute the unbalance moment if the angular velocity is known. Another quite common representation is in the form of individual unbalances (unbalance understood as a quantitative measure) in two different planes, the positions of which are also specified.

It can be formally proved (see [6]) that each instance of unbalance in a rigid rotor is completely described by two pieces of information:

a) Superposition of one static and one dynamic unbalance each or

b) Information on two unbalance masses complete with their angular position for any two non-identical planes (omitting special restrictions concerning the angular position and quantity ratio of each of the two unbalance masses, as is the case in Fig. 4.19)

In practical balancing formulation (b) is used. Balance planes are chosen which are accessible to subsequent reworking and can be used to compensate the unbalances that are identified. This can be done by removing material (drilling) or by adding material (balancing weights, bolts, mastic). Typically the unbalances are identified by measuring the bearing forces, the phase relationship to the angular rotation of the rotor being captured at the same time.

With very short, disk-shaped rotors it may be enough to balance in one plane only. Balancing is carried out on machines such as the universal balancing machine from Schenck ROTEC GmbH shown in Fig. 4.20. As can be seen the rotor drive is a belt drive. The bearing blocks are kept narrow to allow accurate axial tracking of the measured bearing forces or displacements. The data processing and analysis unit informs the balancing technician directly as to the position at which material must be removed or added and the amount of material involved.

A residual unbalance may remain, depending on the effort expended. On its own, however, quantitative information about a residual unbalance is not suitable for describing the quality of a balancing process. A better basis is the division into quality classes in accordance with ISO 1940. The measure according to which the division is carried out is the product of the permissible residual eccentricity and the angular velocity Ω.

Fig. 4.20 with horizontal axis
Illustration by kind permission of Schenck ROTEC GmbH

It therefore shows the peripheral speed of the center of gravity. The numerical reference of the quality class corresponds to this peripheral speed, expressed in the unit mm/s. Choosing this quantity as the criterion ensures that roughly the same balance quality can be claimed for rotors on which comparable demands are made by the environment and by the application, even if the rotors concerned exhibit different masses and speeds.

The effect of this is that when considering whether a given rotor such as a torque transducer has achieved a desired quality class, different residual unbalances are permissible depending on the planned speed. In the case of a rotor that has to be balanced in two planes, the standard likewise defines how the permissible residual unbalance shall be shared between those planes.

The standard lists quality classes from G4000 to G0.4 and assigns typical applications in each case. The most relevant quality classes for torque transducers are G6.3 and G2.5.

As already mentioned at the start of this section, in the first instance these considerations apply to rigid rotors. As understood within balancing technology, rotors can be regarded as rigid if their first natural frequency occurs at more than double the highest occurring speed. The practical implication of this for shaft trains normally used for carrying out torque measurement is that individual components such as torque transducers and couplings can be balanced extremely well in this way. However, the shaft train as a whole will not automatically have satisfactory unbalance properties, because further geometric inaccuracies occur when the individual components are being put together, for example due to fitting tolerances.

In order to fulfill the most demanding requirements or allow the highest speeds, it may be necessary not only to balance individual components before they are assembled but also to balance the operational shaft train in its entirety. Since the shaft train as a whole is usually an elastic rotor, the restriction in principle to two balance planes only cannot be applied in this case. On the contrary, the balance planes must be defined having regard to the mode shapes that can be excited into producing unbalance vibrations at operating speeds.

The significance of balancing which is the method for reducing unbalance lies on the one hand in the fact that unbalance is a direct cause of forces that act upon the rotating components and the bearing. Then again it is also a fact that unbalance in combination with flexurally elastic shaft trains is the most important source of bending vibration.

Since the unbalance forces and moments rotate in common with the shaft train, the deformation relative to the shaft train is quasi-static. Nevertheless it gives rise to the phenomenon of resonance and is therefore rightly classified as a vibrational process. Resonance frequencies are in the main natural bending frequencies. From the type of excitation it follows that the frequency of vibration excitations due to unbalance is always the rotation frequency of the shaft train. It therefore follows that the critical speeds associated with bending are roughly equal to the natural bending frequencies.

4.6.3 Axial vibration

Axial vibration in rotating machinery is rather rare because the chief mechanisms in such machines have few links to axial forces and motion. Thus there are few excitation mechanisms. A possible excitation, however, can stem from lever mechanisms that convert an axial force component into a torque.

There have also been reports of vibrations arising from parameter excitation. This is due to the fact that the axial stiffness of elastic couplings is dependent on bending pre-tensioning. If this bending pre-tensioning is due to an alignment error, its spatial orientation relative to the rotating system is changing over time. This oscillating, time-related stiffness parameter can give rise to vibrations, as discussed in [11].

If axial vibrations occur, they can very easily lead to problems because on the one hand they are often not acquired by the measurement technology and remain unnoticed until damage occurs, and on the other because many bearing designs can accept only a low axial end thrust. Also many couplings, such as concentric ring couplings, are critical relative to axial vibration since they exhibit low axial stiffness. High-amplitude axial vibrations can therefore lead to jolting and may result in irregular loading.

5 Using and installing torque transducers

5.1 Mechanical prerequisites

5.1.1 Principles of installation

Torque transducers can be installed in any suitable orientation with respect to the direction of gravity. For several series types HBM can provide tried and tested couplings that are matched to the torque transducers concerned. When used with couplings in an oblique or vertical configuration, restrictions may apply with regard to the orientation with respect to gravity direction in which the couplings are used.

Moreover when torque transducers are installed obliquely and vertically, any masses belonging to the test bench must be supported by design so as not to exceed the longitudinal forces said to be permissible in the documentation of the respective torque transducers.

In the case of torque transducers with built-in bearings, as well as torque transfer transducers and reference transducers, the torque being measured should preferably be introduced on the measuring side. In this way the measured quantity is hardly affected by effects such as bearing friction. Further information can be found in chapter 3. Installation is described in detail in section 5.2.

When operating with torque transducers that are constructed without bearings it is essential to ensure that the rotor motion relative to the stator remains within permissible limits at every possible operating status. The permissible radial and axial static alignment of the rotor and stator is between ±1 mm and ±2.5 mm in the radial direction, and between ±2 mm and ±3 mm in the axial direction, depending on type and whether or not fitted with a speed measuring system. In accordance with DIN 45670 /VDI 2059, the relation

$s_{max} = \frac{4500}{\sqrt{n}}$ (numerical equation with s_{max} in µm and n in min $^{-1}$)

defines the permissible relative vibrations of the rotor. See also section 5.1.5. If the limits are exceeded it could even lead to the destruction of the transducer.

When using the types with assembly aids that were mentioned in chapter 3, the rotor needs to be mounted first. The stator must then be adjusted and aligned (for example by using spacers for the height and sliding the unit in the transverse direction) and then secured without deformation or tensioning relative to the rotor. The mounting aids must then be removed without fail. They should be stored safely until needed later for dismounting, repositioning or transporting the unit.

To ensure trouble-free operation of the transmission of electrical power, the T10 family of torque flanges needs the area free of metal parts specified in the documentation. If the power transmission is disrupted, the output signal is 0 Hz and the connected HBM measuring amplifiers go into negative overload. Most critical are the limiting ranges where the transmission is only sporadically interrupted and the fault is not noticed immediately.

In types T32FN and T34FN there is an arrow on the stator (Fig. 5.1) which makes it possible to determine the direction in which the torque is acting as well as the direction of rotation.

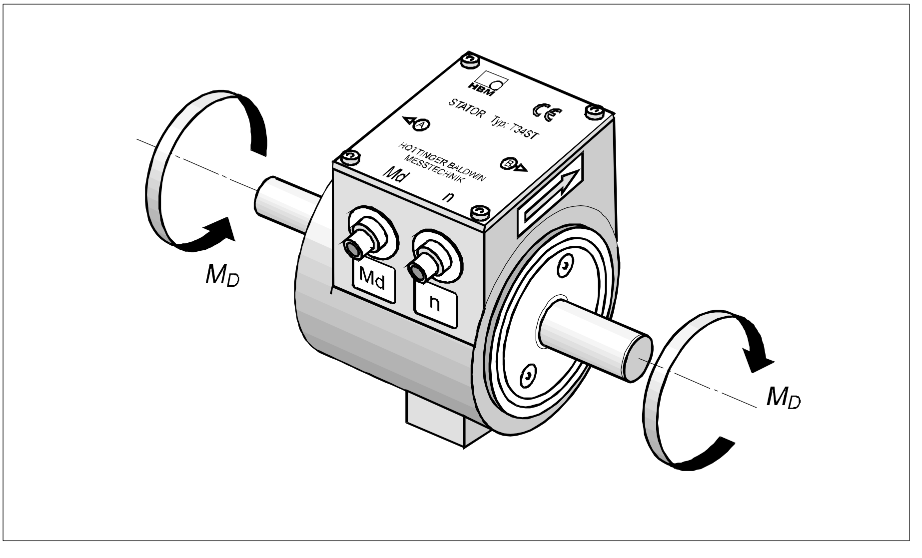


Fig. 5.1 Arrow on T34FN torque transducer

If the torque is operating in the direction of the arrow, a clockwise torque is introduced and the output frequency is 10 to 15 kHz (Fig. 5.2). Connected HBM measuring amplifiers deliver a positive torque signal unless the amplifier has been adjusted to do a sign inversion.

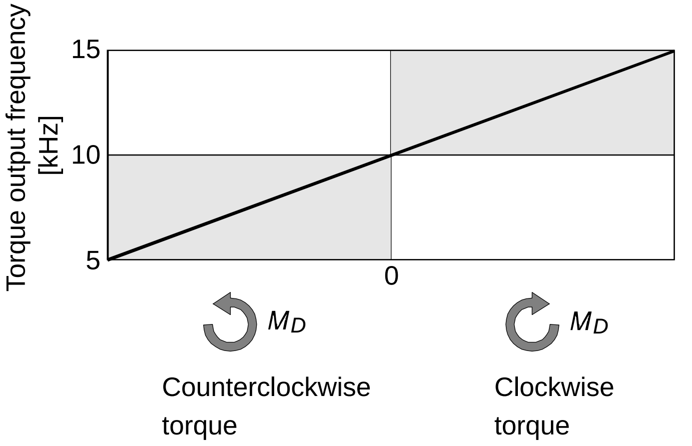


Fig. 5.2 Output frequency over direction in which torque is acting

If the torque transducer is rotating in the direction of the arrow, HBM measuring amplifiers determine a positive direction of rotation on the basis of the phase relation as described in chapter 3.

Contrary to widely held opinion, the direction in which the torque acts does not change if the transducer is rotated by 180° about the *z* axis when it is installed, as shown in Fig. 5.3. This follows from Newton's law of action and reaction. This can be demonstrated if two people manually apply a clockwise torque to a rod, for example. If they then change places they will quickly see that the direction in which the torque acts does not change.

In the T10 family the arrow on the speed measuring system serves chiefly to determine the direction of rotation, since the effective direction of the torque is independent of the installation position. Unlike the torque, the speed does change its sign when the transducer is rotated 180° about the *z* axis.

It is important to adhere to the general conditions on site regarding protection against dirt particles, dust, oil, solvents and moisture. See also chapter 4. Transducers should be mounted at a point where axial and radial motion is slight, especially as they are vibration resistant to only a limited extent. For a correct installation it is essential to take the load limits of torque transducers into account. Likewise all relevant safety rules and regulations concerning the protection of personnel must be complied with when the system is operating. This also contributes to the reduction or prevention of damage to property.

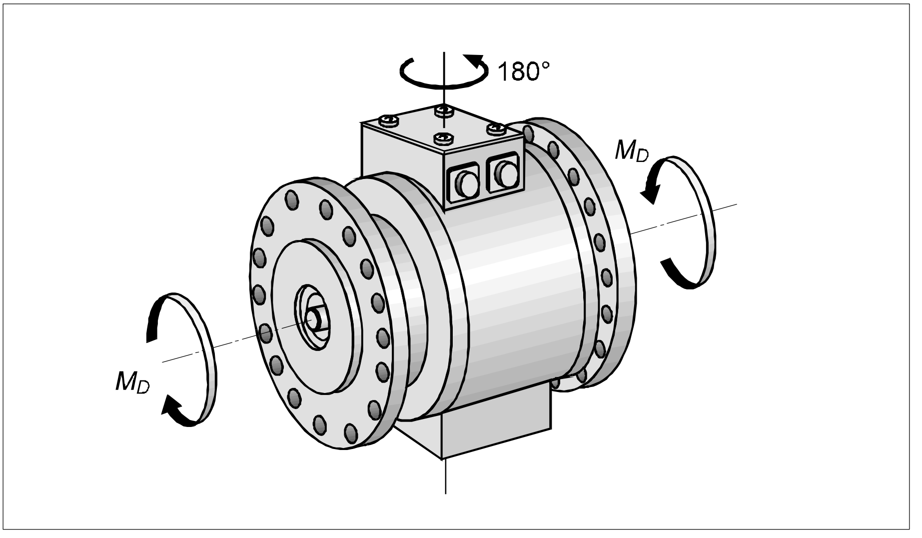


Fig. 5.3 Direction in which torque acts on reversing the installation position

Ever increasing requirements are being expected of torque measurement accuracy. Thus the methods which customers use in the layout of the mechanical torque connection also play an increasingly important role. The following are examples of things which can affect the correctness of a torque connection: concentricity and run-out tolerances, flatness of torque connection surfaces, surface quality, fitting tolerances, bolt tightening torque, materials hardness or strength and the geometry of the adapter flanges used in the application. Information on these matters can be found in the transducer documentation.

5.1.2 Checking torque transducers

Often torque transducers have to be checked or monitored after they have been built in over fairly lengthy periods. The calibration signal delivered by torque transducers can be used for this purpose, as can observation of the zero point. If there is reason to assume that the transducer has been overloaded, a sure indication of this is the presence of a shift in the zero point relative to its status at the time of supply.

Since the calibration signal is generated on the rotor, strictly speaking it applies only to the frequency output. When considering the ±10V voltage output it is important to take account of any settings that may be present in the amplifier on the stator. The calibration signal is additive. No load should therefore be applied to the torque flange when measuring the calibration signal.

The identification plate is often no longer accessible or legible after installation. HBM therefore supplies adhesive labels showing the rotor's identification plate details. These labels should be applied to the stator or other suitable place (e.g. control cabinet, measuring amplifier).

If the calibration and zero signal for checking and monitoring T10F torque flanges is used, the individual zero point must be determined and logged first. The zero point is balanced to between 9,960 Hz and 10,040 Hz as standard.

It is recommended to have an inspection carried out by HBM after changes of the following types:

- Zero signal error > ±1 % (> ±50 Hz) relative to the individual zero point in the unfitted state.
- Zero signal alteration due to installation > ±3 % (> ±150 Hz) in the absence of unacceptable parasitic loads.
- A further change in the zero signal (after adjusting the zero signal alteration due to installation) when operating > ±1 % (> ±50 Hz).
- Calibration signal error > ±0.1 % compared with the information on the identification plate or test report.

These specifications are valid for stable reference temperature conditions and a 15-minute transducer warm-up phase. When testing a device that is already installed it is important to ensure that no additional torque is introduced into the transducer, for instance due to distortion in the shaft train. Since the calibration signal is additive, the possibility of a zero shift needs to be taken into account.

Depending on the application or quality requirements, checking with the aid of the calibration signal and observing the zero point is no substitute for mechanical calibration (see chapter 6). Shunt calibration is a purely electrical process. It is theoretically possible for a strain gage (SG) or a compensation element in the bridge circuit to become detached without any change in its resistance. This failure would not be noticed by checking the calibration signal. Practical experience shows, however, that this is not the case for an SG applied in the correct manner and that as a rule an SG only comes loose due to an irregular operating state such as a breakage at a measuring point.

Every transducer is measured at each stage of production to ensure that all SGs are correctly applied. This is documented in the test report among other places. As a result, with the aid of the calibration signal and the zero point it is very easy to assess whether torque flanges are operating within the specified data.

Since plastic deformation of the rotor due to overloading by bending moments and/or lateral forces does not necessarily show up in the zero signal or calibration signal, it is important to ensure correct installation to prevent overloading of this type. If any suspicion of an overload exists, however, it is often easy to come to a conclusion by checking the concentricity and parallelism of the torque connections to input side and output side.

5.1.3 Dimensions

When designing a torque measurement task, the external dimensions of the torque transducer are a major consideration. The dimensions of the torque transducer are determined by the desired nominal torque and other strength requirements, but also crucially depend on the design of the measuring body. Further information on the various measuring body designs can be found in chapter 3.

Where space for the torque transducer is limited, dimensions can be the deciding factor on whether to use a conventional, long-format torque shaft or a torque flange. For instance if it is planned to fit a rotating torque transducer to an existing power test bench that has previously been operated without in-line torque measurement, there is often little room available particularly in the axial direction. Often in such cases the only possible solution is to use torque flanges with their specially short design.

If the width or height of the available space is so restricted that it crucially limits the possible diameter of the torque transducer, an appropriate answer is not only the conventional torque shafts but also the T10FS torque flange with a reduced external diameter. The T10FS torque flange is only slightly longer than the T10F.

If the available height is so restricted that the stator or cable junctions cause a space problem, this can often be resolved by turning the stator round before installing it.

5.1.4 Types of mechanical connection

On the basis of the mechanical torque connections outlined in chapter 3, the connecting elements which are particularly important for test bench applications, namely

- Clamping connection
- Bolted flange

are discussed in this chapter.

The main methods used in clamping connections are uni-axial radial tensioning or the cone principle. Fig. 5.4 shows the principles. Uni-axial tensioning is used for simple requirements for maximum speed and when the torque that has to be transmitted is relatively small.

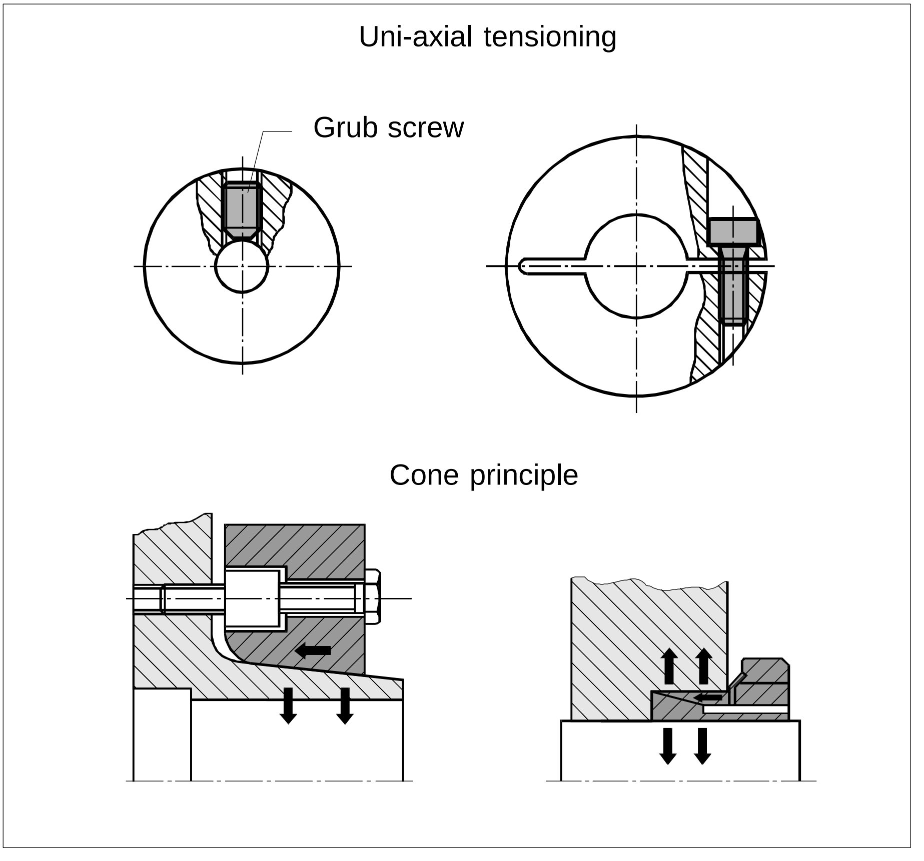


Fig. 5.4 Clamping connection for shaft stub

With this method, clamping is not as a rule symmetrical but tends to be more or less pointlike. Where there are pressing requirements for speed and torque transmission the cone principle is utilized. According to this principle, as Fig. 5.5 shows, oblique surfaces and axial tensioning are used to create a clamping force which ensures the optimum torque connection if handled correctly. In the event of very high torque values it is even possible to use multiple cone tensioning units in series.

On the subject of clamping connections, customers often ask HBM what fit tolerance they should choose. It is impossible to give a general answer to this question, since in such cases the demands of every application need to be tested very precisely, and the assembly conditions and capabilities available to customers must also be taken into consideration. Opinions can be given based on HBM components. For instance in the case of the bellows coupling for the T20WN torque transducer, the bore hole diameter is produced with tolerance H7.

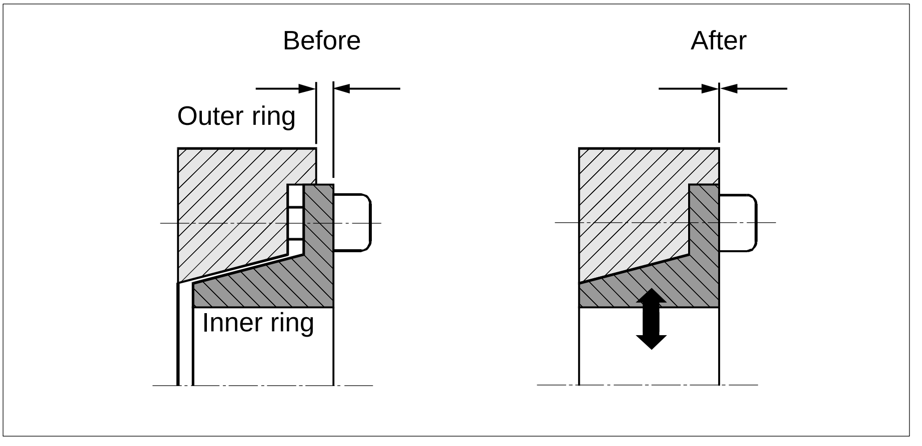


Fig. 5.5 Aligning the faces

For a correct torque connection, shaft diameters should be produced in tolerance j6. This gives the preferred fit H7/j6, which is defined as a sliding fit according to [4]. As a rule a sliding fit has to be additionally secured against rotation. This is unnecessary in the case of a clamping connection, since clamping with the prescribed bolt tightening torque gives a correct torque connection that not only has frictional resistance but is also free from play.

In the case of bolted flanges and conventional torque transducers, the bolts can be inserted in any orientation. Fig. 5.6 makes this clear using the T32FN torque transducer as an example. When bolts are inserted from the inside facing outward, threaded bore holes for the bolts can be provided on the application-side flange.

Another possibility is the through bore hole. This method uses a combination of a bolt and a lock nut. Such a combination is also needed when the bolts are introduced from the outside. In this event the customer's flange must have a through bore hole in all cases. Since torque is transmitted entirely by friction locking in the case of HBM torque transducers, adapting bore holes are not necessary and in fact are not permitted due to the prevailing tolerances.

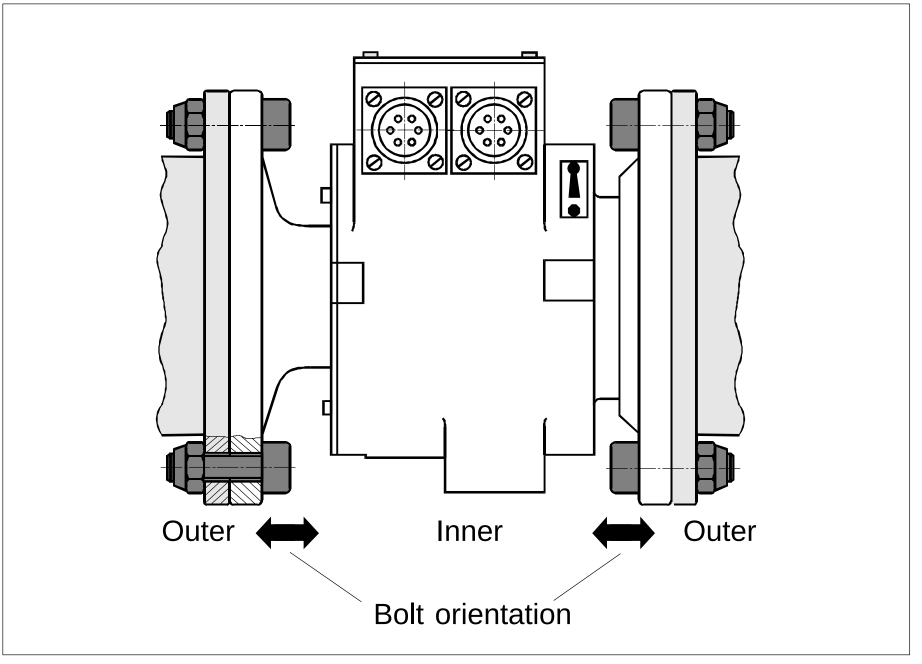


Fig. 5.6 T32FN torque shaft with flange connections

Torque flanges are bolted from one side. One side of the flange has threaded bore holes while the other side has threaded bore holes and through holes. Fig. 5.7 shows this using the T10F torque flange as an example. On the left-hand side is the connection flange for the application. The measuring body of the T10F torque flange is bolted to it. If an intermediate flange is needed because of differing flange designs, it is bolted to the adapter flange. In this example, all bolts are connected into the screwed joints from the right-hand side.

This type of screwed joint calls for the assembly to be mounted with great care. Even though the assembly recommendations contain clear instructions, a number of typical mounting errors are made time and again. Fig. 5.8 shows the critical areas.

Over long bolts on the adapter flange side

This causes axial prestressing in the torque flange which can be severe enough to overload the measuring body. Visible effects range from a strong offset in the zero signal through to mechanical failure in severe cases.

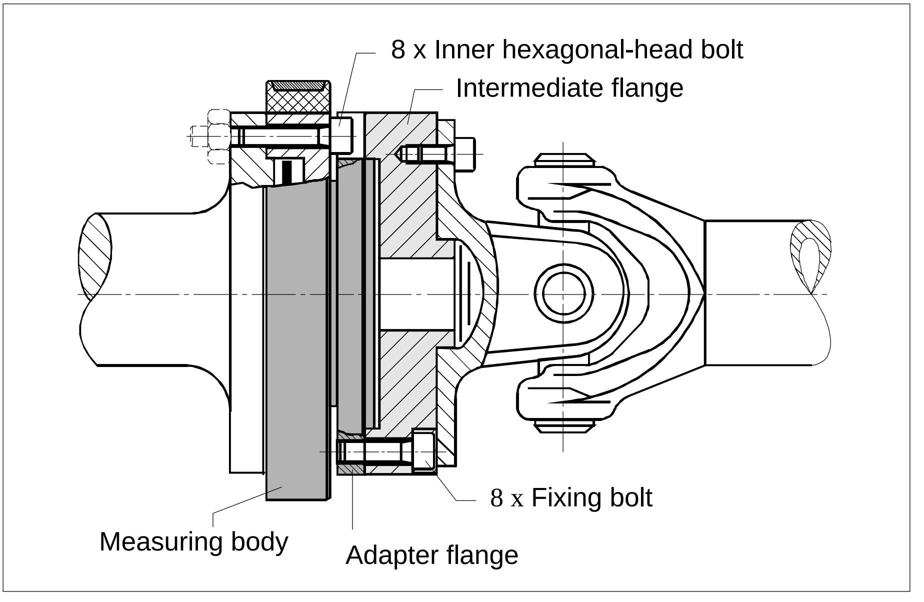


Fig. 5.7 Screwed rotor joint for a T10F torque flange

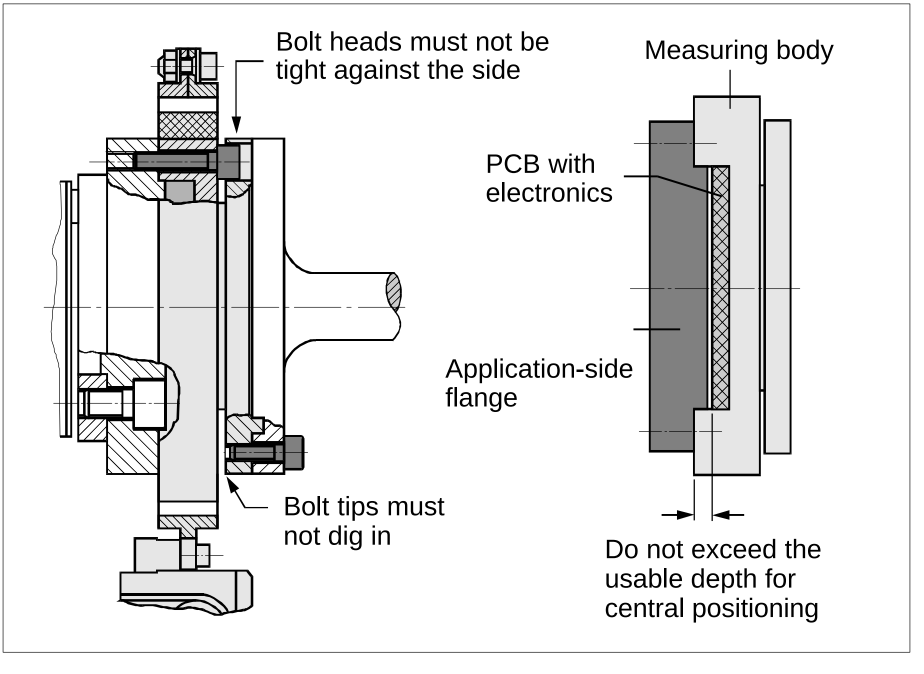


Fig. 5.8 Critical areas when mounting the T10F torque flange

Exceeding the depth that can be used for central positioning

Axial distortion can be severe enough to lead to plastic deformation of the circuit board holders and may result in damage to the rotor circuit board. In addition the measuring body may become prestressed. This leads to zero point shift and possibly also to sensitivity deviation compared to the original calibration.

Tightly fitting bolt heads in the adapter flange

This can give rise to undefined torque shunts and incorrect measurement.

In the case of the T10FS torque flange the above mentioned points are not critical since, as Fig. 5.9 shows, bolt heads cannot be a tight fit and over long bolts can easily be seen.

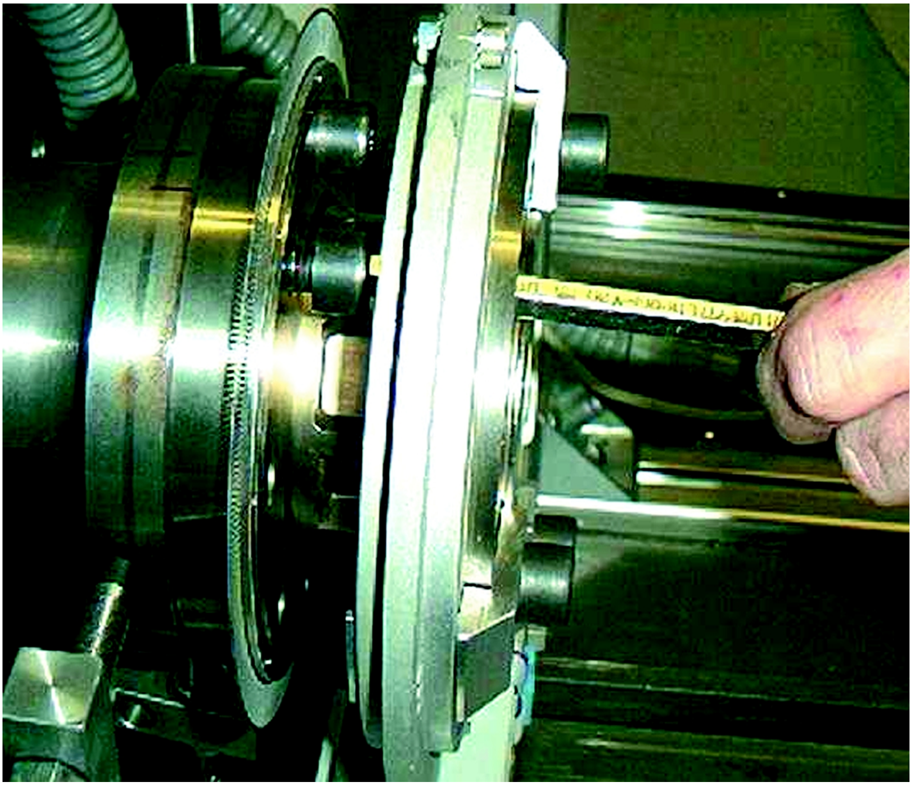


Fig.5.9 Mounting the T10FS torque flange

5.1.5 Balancing

Balancing improves the mass distribution of a rotating body in such a way that it rotates in its bearing free from the effects of centrifugal forces. Chapter 4 discusses the rotor dynamic effects of unbalance. Unbalance reveals itself as bending moments in the shaft train, as forces in the bearings and foundation, and as vibrations in the machine with the rotational frequency of the unbalance. Information about the required balance quality levels and the associated permissible levels of unbalance is detailed in Guideline DIN ISO 1940. Further notes can be found in chapter 4.

Balancing is carried out according to different principles, depending on the combination of torque transducer and coupling delivered by HBM.

In the case of the combination of a torque shaft and Renk curved tooth couplings® the coupling housings are mounted and then the compound is dynamically balanced. Due to the coupling structure, only the coupling housings can be balanced together with the torque shaft. The coupling hubs would become misaligned, so they are balanced separately. Users must therefore balance their equipment with the coupling hubs attached, but without the torque transducer and coupling housings. In principle the coupling housings and torque transducer should not be dismantled again. If this is unavoidable, however, the following points must be observed: all parts (flanges, bolts, etc.) must be labeled so that they can be reassembled in precisely their delivered configuration. The prescribed tightening torque for the bolts must be taken from the dimensional drawings of the coupling.

A shaft and coupling combination which has been taken apart and reassembled cannot be operated at the maximum speed specified by HBM until it has been balanced again. However, practical experience shows that with careful assembly there is no need to rebalance for speeds of up to 3000 min^{-1}. It may be necessary to determine the actual permissible speed by measuring the amplitude and speed of the vibrations. The amplitude of any vibration that occurs is influenced by the mass of the sympathetically vibrating housing and foundation, the stiffness of the bearing or foundation, how close the operating speeds are to resonance and in short the dynamic behavior of the entire equipment configuration.

By way of example, Fig. 5.10 shows the dynamic application limits that apply to the T32FN torque transducer and within which the mechanical loading capacity is guaranteed. See also section 5.1.1. The relative shaft vibration in the area of the connection flange s_{max} is in accordance with DIN 45670/VDI 2059. The threshold values relate only to the T32FN and do not apply to connected machinery and equipment. If these limits are permanently exceeded in the course of operation, mechanical damage to the measurement shaft may result.

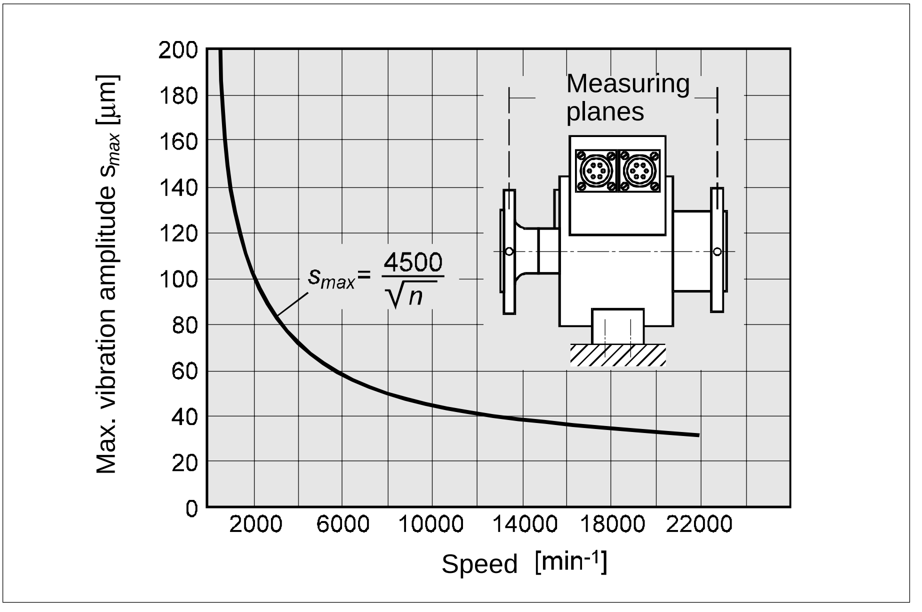


Fig. 5.10 Vibration amplitude over speed for the T32FN torque transducer

In the case of multi-disk couplings assembled and delivered by HBM, no further balancing with the overall system is needed after fitting. They are designed to make this unnecessary. Torque transducers and couplings are individually balanced in accordance with their specifications during production.

The clamping hubs used in BSD-Thomas couplings for T34FN torque shafts have exact centering as well as mechanical connections which are absolutely free from play, ensuring negligible influence on the balance quality.

BSD MODULFLEX® torsionally stiff couplings for the T10F torque flange provide a bolted-flange connection. These couplings are balanced complete with their bolts. They are designed to ensure that misalignment of the bolts during assembly is minimal and therefore has no significant influence on balance quality.

Although each torque transducer and coupling combination supplied is balanced for the specified maximum speed, in some cases the balance of the whole shaft train must be checked after installation. Balancing the operational shaft train ensures optimum smooth running.

5.2 Installation in the shaft train

This section deals with designing the shaft train in respect of the correct mechanical transmission of torque and all the forces and moments which act upon the bearings. A correct mechanical transmission must firstly satisfy the fundamental machine building requirement for a statically determinate bearing configuration. Furthermore the loads imposed on the torque transducer should be minimized. The design of the shaft train can influence not only parasitic loads but also, depending on the environment and objectives of the measurement, the possible torsional vibrations.

The couplings and other compensating elements play a major role in the fulfillment of these objectives. This is because when combined with the appropriate bearing configuration they help to prevent forced deformation.

5.2.1 Compensating elements in the shaft train

Flexurally elastic single joint couplings

Flexurally elastic couplings are intended to compensate for geometric errors. There may be static offset resulting from alignment errors, or an error jointly rotating with the shaft train resulting from flange errors or unsatisfactory centering. Dynamic deformations are also a possibility. The restoring forces tend to be weak overall. In the case of flexurally elastic couplings it should be borne in mind that whilst they are elastic relative to bending deformation, as a rule they are deliberately designed to be stiff relative to torsional deformation. They are sometimes known as torsionally stiff couplings.

A single joint coupling, or half coupling, can compensate for radial angular offset. Most designs of single joint coupling also allow slight axial offset. As the name single joint coupling suggests, it kinematically fulfills the function of a joint (such as a universal joint), but unlike the ideal joint it often causes restoring forces or moments. The permissible limits for angular and axial offset can be obtained from the specifications. These limits are quite small as a rule, for instance the maximum permissible angular offset is typically much less than one angular degree.

Two single joint couplings can also be used to compensate for parallel offset, and thus perform the functionality of a double joint coupling. In conjunction with rather long shaft sections that are joined into the shaft train by a single joint coupling at each end, the functionality of a joint shaft can also be emulated.

Typical examples of actual versions are the multi-disk coupling or the curved tooth coupling, as shown in Fig. 5.11 and Fig. 5.12. In the case of the curved tooth coupling, the torque is transmitted by locking of the indentations between the outer component and the inner component. Because the external indentations on the inner component are rounded in the longitudinal direction, angular motion is possible. In multi-disk couplings, angular flexibility is provided by thin, ring-shaped steel plates which are set perpendicular to the axis of rotation so that they are very stiff relative to torsional deformation.

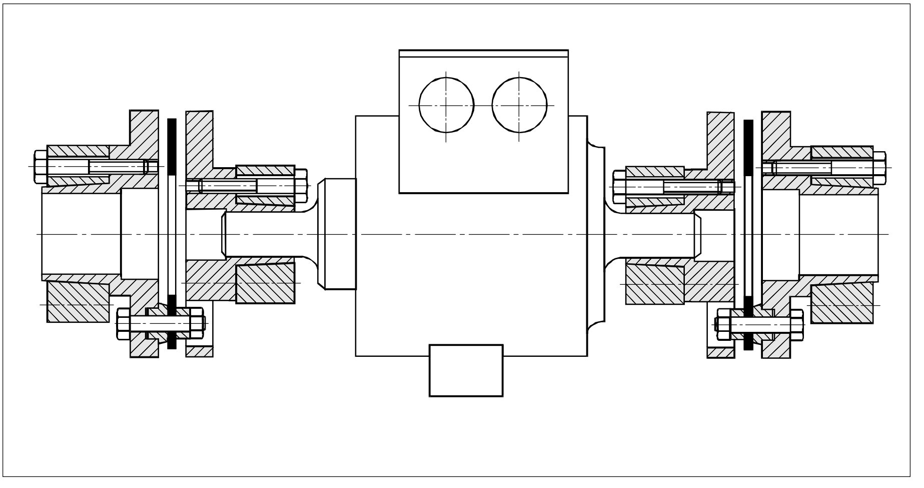


Fig. 5.11 T34FN torque transducer with steel multi-disk couplings

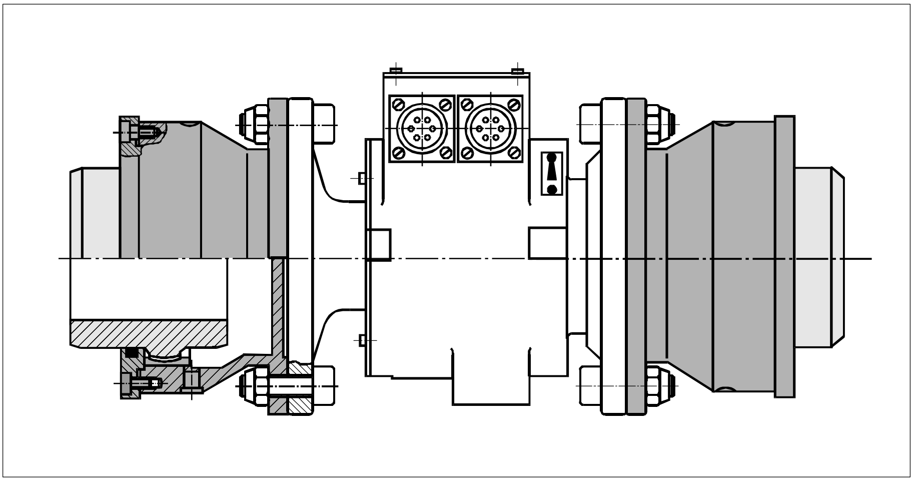


Fig. 5.12 T32FNA torque transducer with Renk curved-tooth couplings®

Flexurally elastic double joint coupling

A double joint coupling or full coupling can compensate for radial angular offset and radial parallel offset. Most designs also allow slight axial offset. A double joint coupling kinematically fulfills the function of two joints (such as universal joints) that are interconnected by a short shaft section. Their degrees of freedom are therefore the same in number and type as with a joint shaft.

Unlike the ideal joint shaft, a double joint coupling as a rule generates restoring forces or moments. As in the case of single joint couplings, the permissible limits for angular, parallel and axial offset are small. Such couplings are therefore used to best advantage where compensation is confined to the inaccuracies that can never be fully avoided during the mounting process, in order to make sure that forced deformation and the associated internal distortions can be prevented.

As is the case with single joint couplings, these couplings also exhibit extremely high stiffness against torsion, especially in conjunction with torque transducers.

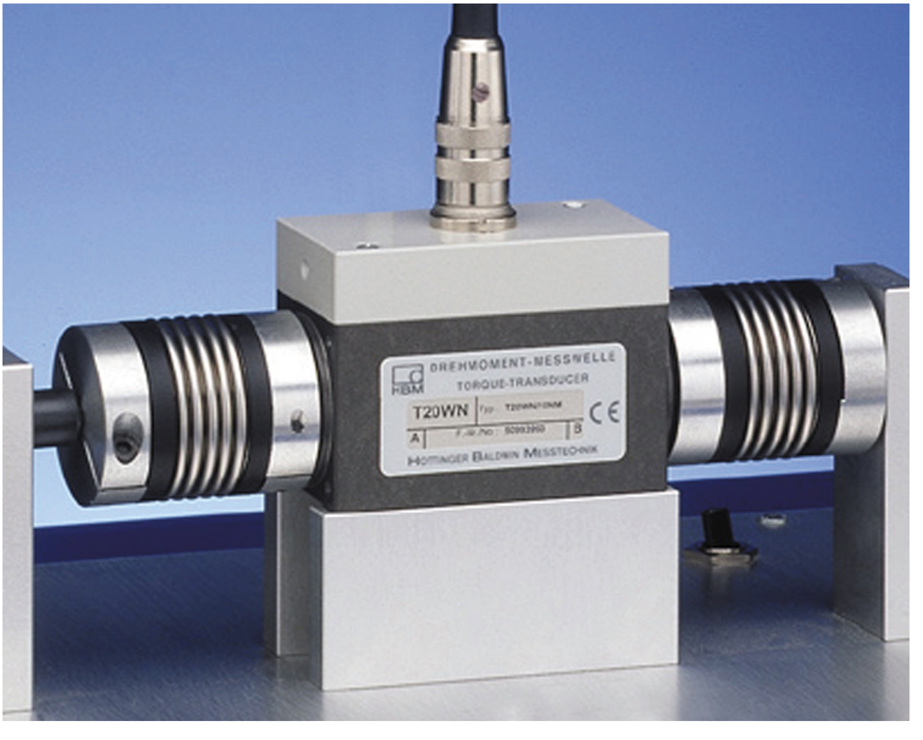


Fig. 5.13 T20WN torque transducer with bellows couplings

The principles of construction used for full couplings employed in conjunction with torque transducers are steel multi-disk couplings and couplings that can be deformed in more than one direction. Steel multi-disk couplings need two sets of plates or plate packages in order to emulate the functionality of two joints. Examples are shown in Fig. 5.13 and Fig. 5.14.

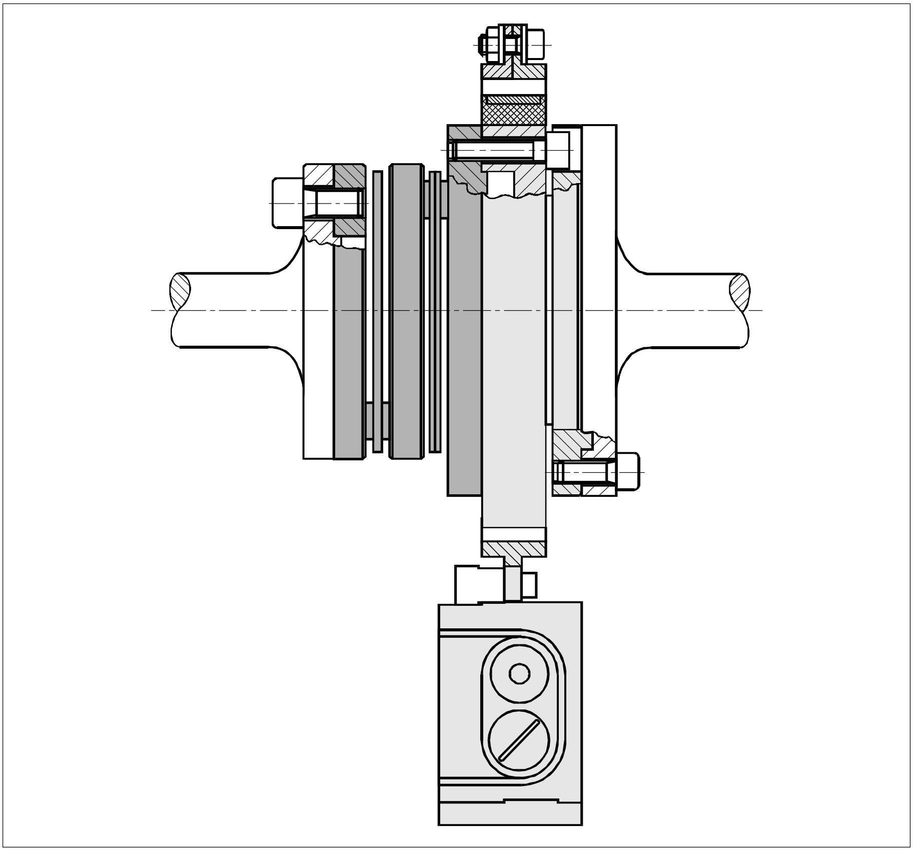


Fig. 5.14 T10F torque flange with steel multi-disk coupling

Joint shafts

The two joints and the straight shaft section between them can compensate for both radial angular offset and radial parallel offset. The working principle is therefore the same as for a double joint coupling. Apart from friction effects, however, joints cause no restoring forces or moments.

The joints in joint shafts permit greater displacement than the elastic elements in couplings, and joint shafts are therefore able to compensate for greater radial displacement. This is the main difference in the fields of application of double joint couplings and joint shafts.

One should plan to use joint shafts where individual sections of the shaft train are deliberately designed to be offset or mobile relative to one another. They are also used when elastic couplings are not suitable because the design of the application or machine does not allow them to be assembled with sufficient accuracy to ensure that the deformation limits are complied with. Even if vibrational motion is to be expected from individual mechanisms in the shaft train, the use of joint shafts is usually advisable.

The universal joints used in the conventional type of joint shaft cause non-uniform rotary motion when the deflection angle is large. The angle of rotation φ_2 on the driven side of the universal joint does not correspond exactly to the angle of rotation φ_1 on the driving side, but instead leads during one half of each revolution and lags during the other half. In the case of an individual universal joint as illustrated in Fig. 5.15, the formula

$$\tan\varphi_2 = \frac{\tan\varphi_1}{\cos\beta}$$

expresses this fact quantitatively as a function of the deflection angle β (which for the sake of simplicity is constant). For torques M_{D2} and M_{D1} and speeds n_2 and n_1 on the driven and driving sides respectively this gives

$$\frac{M_{D2}}{M_{D1}} = \frac{n_1}{n_2} = \frac{1-\cos^2\varphi_1\ \sin^2\beta}{\cos\beta}$$

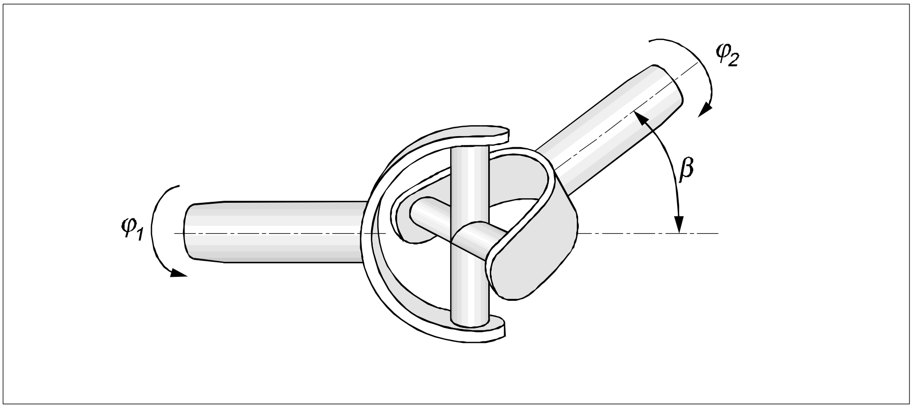


Fig. 5.15 Kinematics of a deflected universal joint

This means that the torque on each side always passes through the relative maximum when the speed of the side concerned passes through its minimum. If the deflection angles of both joints in a universal joint shaft are the same, the non-uniformity arising from each of the two joints is canceled out, and the same angular velocity and torque exist on the driven side of the joint shaft as on the driving side. Constant velocity joint shafts, or homokinetic joint shafts, avoid any non-uniformity.

Different manufacturers define various styles of joint shafts depending on the intended application. There is often a standard version and a heavy duty version. Some versions of joint shafts meet the need to compensate for an axial offset. The conventional method is the sliding collar. With some types of constant velocity joint shafts, however, a certain axial displacement is possible in the joints.

Another way of classifying joint shafts can be according to the design of the connection flanges. Flange versions from Germany and the rest of Europe are usually to DIN standard, whilst those from the USA comply with SAE standards. Even within a given group there are many variants.

Flexurally elastic torsion rod

Flexurally elastic torsion rods offer another possible method of compensating for slight angular offset and radial displacement. These rods are very slender shaft sections which exhibit very low bending stiffness because they are relatively long. They are connected to the shaft train either by flanges or by splines or gear-tooth profiles. An advantage of flexurally elastic torsion rods which is frequently found to be crucial is that they are virtually free from play with respect to torsional motion.

Torsionally elastic couplings

Torsionally elastic couplings help to reduce sharp peaks in dynamic torque. The first part of the task is to prevent severe dynamic torque peaks from reaching the torque transducer. The causes and general properties of torque peaks are described in a discussion about dynamic torque in chapter 4. The second part of the task is to prevent torsional vibration resonance. Both objectives can be achieved through a combination of torsional damping and torsional elasticity.

Torsional damping can be achieved by material damping of elastic polymers and by friction elements inside the torsionally elastic coupling. Often both methods are used together. On the one hand torsional elasticity is a precondition for the required torsion to take place in the torsionally elastic coupling so that its torsional damping can have any effect at all. Yet on the other hand this torsional elasticity causes the torsionally elastic coupling to act as a mechanical low pass filter. As can be inferred from the explanations on torsional vibration in chapter 4, an estimation of the natural frequency of the whole shaft train section must be taken into account in order to come to a quantitative estimate of the damping effect that a torsionally elastic coupling will provide.

However, it may be that an accurate observation of the torque generated by a test object complete with all its dynamics is required. In that case, and in the light of the problem to hand, a decision needs to be taken whether the information loss due to the damping and the mechanical low pass effect exhibited by a torsionally elastic coupling is acceptable. Otherwise it might be better to select a torque transducer with a higher measuring range that is capable of tolerating and measuring the dynamic torque peaks.

Overload protection

When considering overloads and the appropriate protection measures to be taken in a shaft train, a distinction has to be made between two different cases. In one case the overload is part and parcel of normal operation from the torque measurement point of view, and is therefore quite often the quantity that is actually of interest. For example abrasion processes can be tracked by measuring torque and when critical values are reached either the drive can be shut down by limit value switches or an alarm can be sounded. As a rule the overload builds up slowly in such cases and switching off the drive is fast enough.

In the second case, however, the overload results from a malfunction such as seizure of the brake or transmission, or a short circuit in the excitation winding of a DC motor. In a malfunction, shutting off a drive on the basis of the measurement information would usually take too long. Protecting the torque transducer and the rest of the shaft train against this kind of overloading requires the shaft train to be mechanically disconnected. Depending on the circumstances of the application, this may be carried out by mechanical overload protection devices known as safety couplings.

Commonly used versions include:

- Friction couplings
- Couplings in which the torque is transmitted by the engagement of form-locking driving elements held in place by a flexible spring which ejects them if the torque becomes too severe
- Pneumatic or hydraulic couplings

- Magnetic couplings
- Protection devices with shear bolts

Whether a mechanical device of the type listed above will provide the desired protection depends on the configuration of the shaft train and the type of malfunction likely to be encountered. The question of which type to select is also heavily dependent on the application.

One particular difficulty is that the load peaks which occur in the types of malfunction described are highly dynamic, but the mechanical overload protection devices respond with a certain delay. In particular, such load peaks run through the shaft train in the form of a shock wave of torsional deformation, so that the timely triggering of mechanical overload protection depends on whether this shock wave reaches the protection device first, or the torque transducer, or some other component in the shaft train which is then damaged by the overload.

An estimation of whether it makes sense to use a mechanical overload protection device must be made by the person responsible for the design and layout of the application as a whole.

5.2.2 Effects of geometric errors in the shaft train

Due to the manufacturing and assembly tolerances that are always present, the various components in a shaft train are never totally aligned with one another, and never totally centered with one another. The first-mentioned shortcoming is known as an alignment error or static offset and is considered to be an error fixed in the non-rotating reference system. The second-mentioned shortcoming is known as a flange error or centering error and is considered to be an error which is fixed within the rotating reference system. Offset plays a significant role because it leads to forced deformation if the components are connected to a shaft train. Occurrences of forced deformation have the effect of reaction forces and moments, as discussed in the section on parasitic loads in chapter 4. In order to prevent these reaction forces and moments, or at least to minimize their effect, the compensating elements introduced earlier must be used and must be used in the proper way.

Radial angular offset

An example of the principle of radial angular offset is shown in Fig. 5.16, where two simple, straight shaft sections are to be joined together by a flange.

As the illustration of the unjoined shaft train shows, the center lines of the two shaft sections intersect but are not parallel. The illustration of the shaft train joined without any compensating element shows the forced deformation, which causes a bending moment. If a torque transducer is included in either of the shaft sections, a bending moment has the effect of a parasitic load on it. The magnitude of the bending moment depends on the bending stiffness of the transducer and shaft. Bearings are also loaded by bending moments if they offer no tilting degree of freedom.

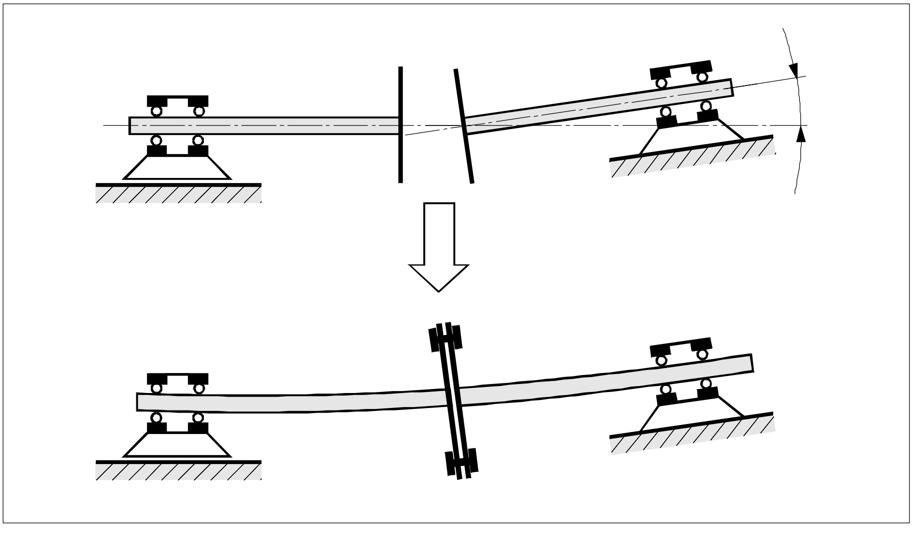


Fig. 5.16 Radial angular offset without compensating element

A single joint coupling is adequate to compensate for radial angular offset, as Fig. 5.17 shows. It goes without saying that double joint couplings and joint shafts can also be used and in practice are even the normal choice, because after all several types of offset can occur at the same time. A joint shaft can compensate for the largest offsets.

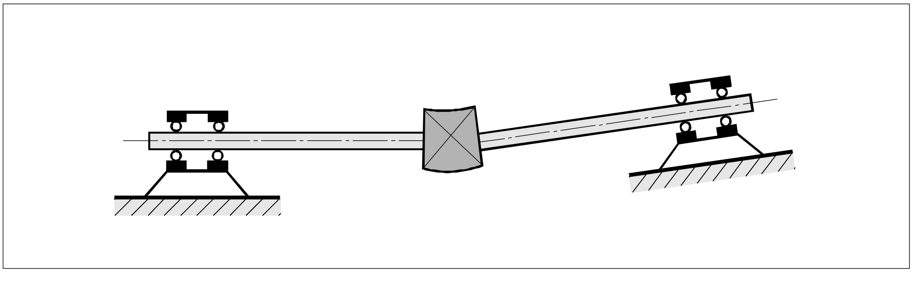


Fig. 5.17 Radial angular offset with single joint coupling

Radial parallel offset

An example of the principle of radial parallel offset is shown in Fig. 5.18, where two simple, straight shaft sections are to be joined together by a flange.

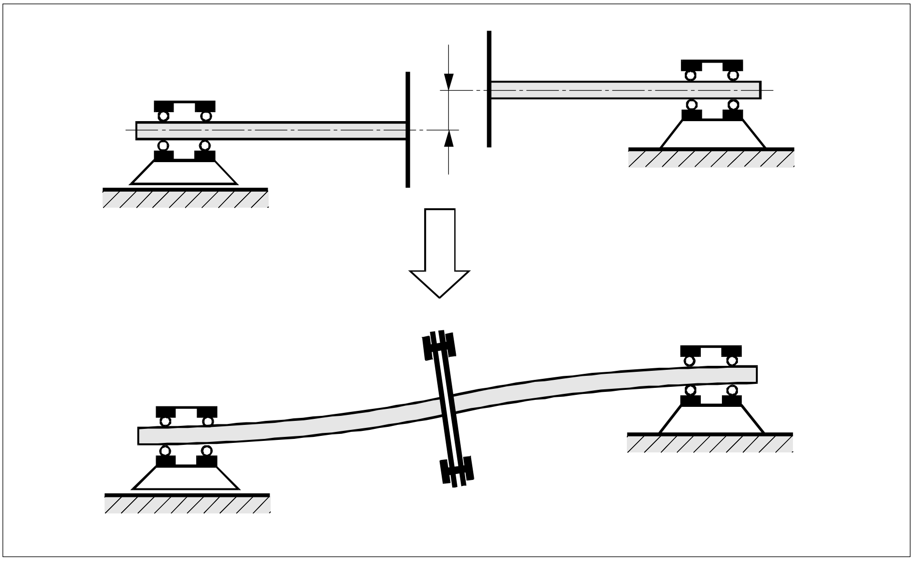


Fig. 5.18 Radial parallel offset without compensating element

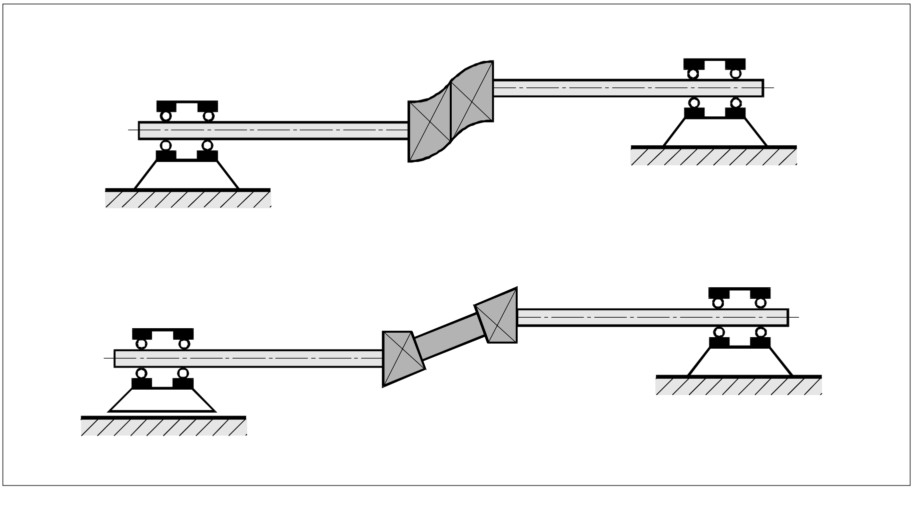


Fig. 5.19 Radial parallel offset with compensating element, double joint coupling above, joint shaft below

It can be seen that before the shaft train is joined, the center lines of the two shaft sections are parallel but do not intersect. Joining the shaft train leads to forced deformation, in this case in the form of bending and radial force.

Compensation for radial parallel offset requires a double joint coupling, a combination of two single joint couplings or a joint shaft, as Fig. 5.19 shows. As in the case of angular offset, a joint shaft can compensate for the largest offsets.

Axial offset

An example of the principle of axial offset is shown in Fig. 5.20, where again two simple, straight shaft sections are to be joined together by a flange. The center lines of both shaft sections are identical, but the two flanging surfaces that have to be joined are not in the same axial position. The shafts are either too short or too long.

If a shaft train of this type is fitted together by force, the components are subjected to axial strain in the form of elongation or compression. If no balancing elements or longitudinal sliding bearings are installed to absorb this elongation or compression, other components will become deformed. This is demonstrated in Fig. 5.20 by the deformation of the bearings. Since the axial stiffness is generally very high, even a small deformation requires strong internal forces. Strong axial forces can cause slender shafts to buckle, reducing the length of the shaft by bending deformation.

As already mentioned, most designs of flexurally elastic couplings allow a certain amount of compensation for axial offset. In the case of joint shafts, special types are available allowing compensation even for quite significant levels of offset.

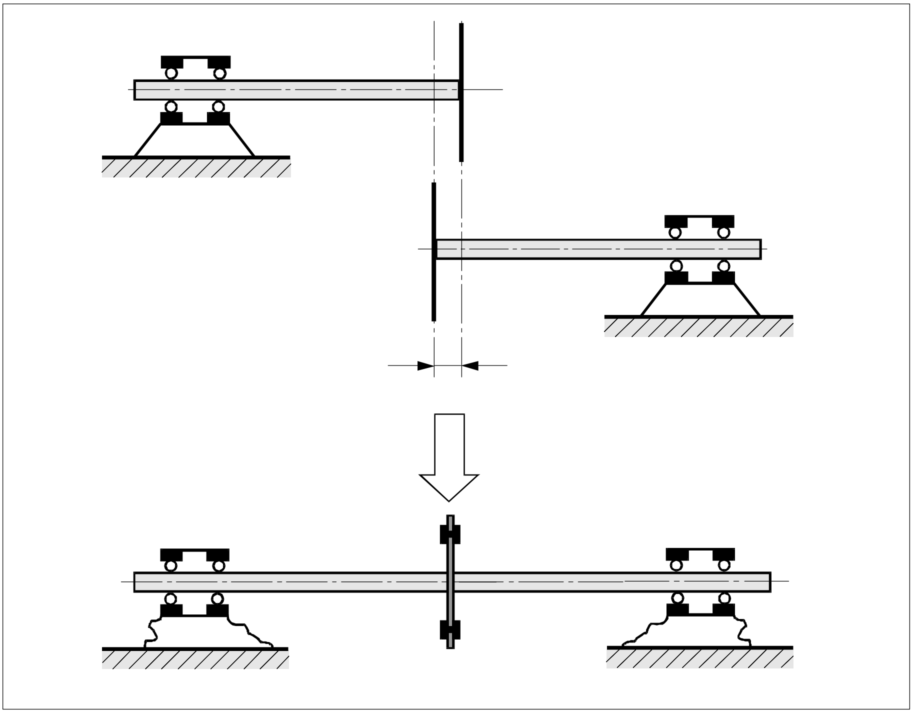


Fig. 5.20 Axial offset without compensating element

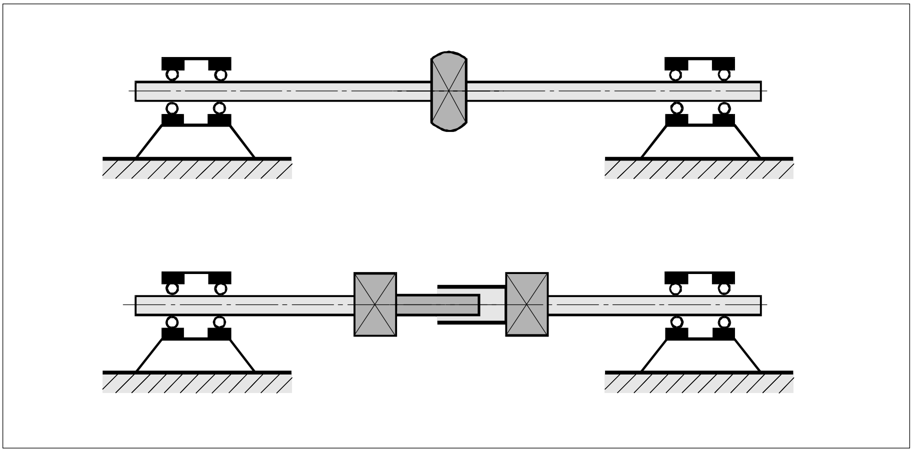


Fig. 5.21 Axial offset with compensating element, double joint coupling above, joint shaft below

Superposition of different forms of offset

In practice there is almost never just one kind of offset. Fig. 5.22 shows the superposition of radial angular and parallel offset with axial offset.

Generally a compensating element that can compensate for all the indivdual forms of offset can also compensate for superposition. Double joint couplings and joint shafts are suitable elements for this purpose, however in both cases it has to be a type which allows axial compensation (see Fig. 5.23).

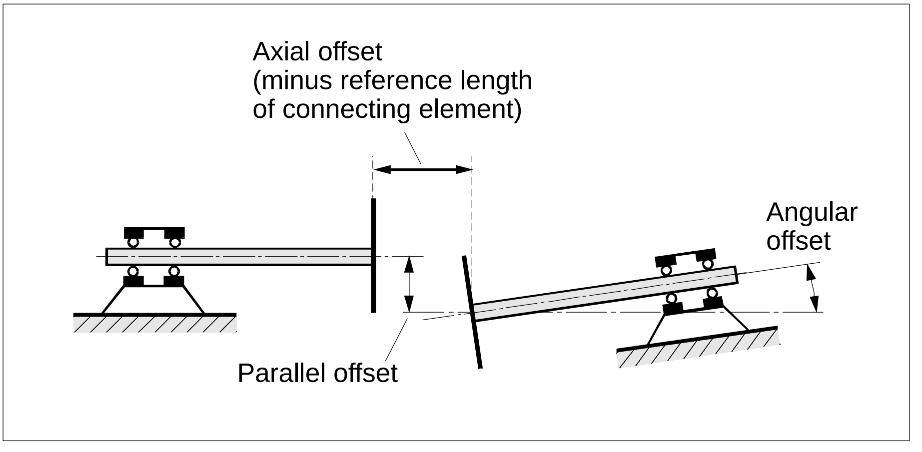


Fig. 5.22 Superposition of different forms of offset

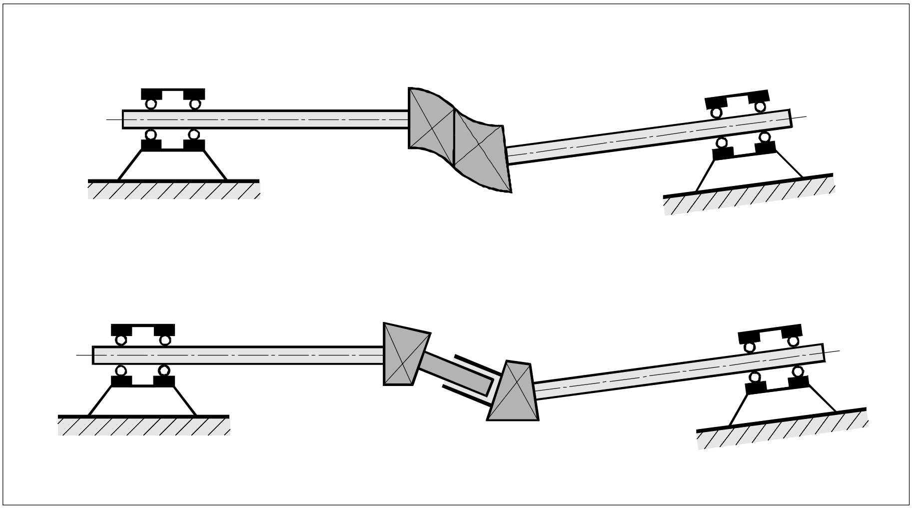


Fig. 5.23 Superimposed offset with compensating element, double joint coupling above, joint shaft below

5.2.3 Designing the shaft train to include torque transducers

Statically determinate bearing configuration

A statically determinate bearing configuration should be a basic requirement in the design of a shaft train. This is a requirement of proper mechanical design, not of torque measurement. According to the definitions of engineering mechanics, a bearing configuration is statically determinate when all bearing forces and moments can be determined solely from knowledge of the external forces and moments by applying the laws governing statics, without at the same time needing to determine the deformation status. Thus no kind of forced deformation must be applied for the purpose of assembly, even if individual components are not ideally aligned and centered with respect to one another.

There is also a requirement for kinematic determination, which is that no shaft section or other component can move or tilt without deformation being required in order for this to happen. To put it plainly, every component must be properly supported.

A bearing configuration that is statically determinate with respect to axial forces requires and permits in an axially rigid section of the shaft train precisely one bearing that can give support against axial forces. This is known as a fixed bearing. In the case of all other bearings, axial loading must be prevented by means of a slight degree of mobility. These bearings play the role of what is known as a loose bearing. If several sections of the shaft train are joined together by couplings which permit axial motion, each section must have one fixed bearing of its own.

In order for a shaft train to be supported in a manner which is also statically determinate relative to transverse forces and bending moments, it is important not to include too many bearings. Fig. 5.24 shows two classic examples in which a statically determinate bearing configuration is achieved without the use of compensating elements: (a) a shaft supported like a simple beam, in which both bearings offer a degree of freedom relative to tilting angles, (b) a cantilevered shaft with a bearing that has no degree of freedom relative to tilting angles.

However, in torque measurement applications (test benches) the driving and driven machines each represent a cantilever-type support (a bearing configuration with no degree of freedom relative to tilting angles). An arrangement combining these two machines into one shaft train can therefore only be made statically determinate by inserting compensating elements with the functionality of joints that enable a tilting motion.

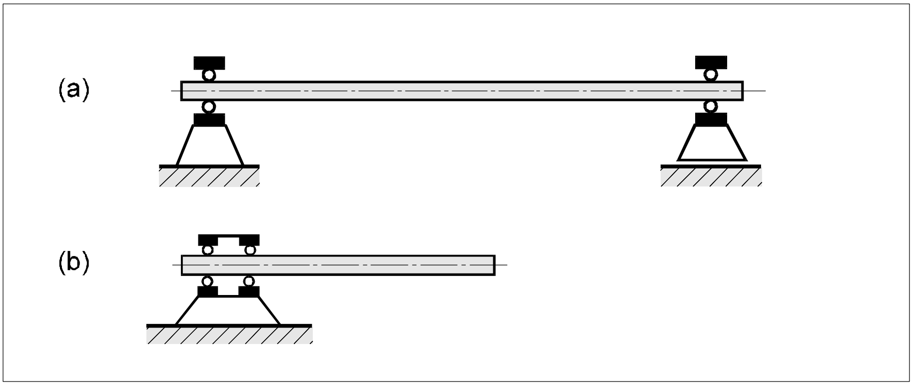


Fig. 5.24 Statically determinate bearing configurations without compensating elements

From the standpoint of engineering mechanics this equates to dividing the shaft train into several bodies. Some examples are shown in Fig. 5.25: (a) and (b) for shaft trains without support bearings, (c) for shaft trains with support bearings with a tilting degree of freedom, (d) for shaft trains with support bearings without a tilting degree of freedom. As discussed in the section on the various compensating elements, the functionality of an individual joint is fulfilled by a flexurally elastic single joint coupling, whereas double joint couplings and joint shafts always have the functionality of two joints that are mounted more or less closely adjacent to one another. In the diagram, each box with a cross symbolizes a joint with a tilting degree of freedom.

It must be emphasized that inserting a support bearing always requires additional couplings or joint shafts. In practice support bearings themselves have no tilting degree of freedom in most cases, and therefore a full coupling or joint shaft must be provided on both sides of the support bearing.

There is also a risk in using too great a number of couplings or couplings with too many degrees of freedom. This is because the requirement for kinematic determination is violated. In practice it is easy to overlook a violation of kinematic determination, especially since real couplings in contrast to ideal joints usually apply restoring forces to some extent, which means that all components are properly supported in standstill conditions. But since these restoring forces are very slight compared to the strength of the shaft train, such an arrangement leads almost inevitably to problems such as extremely large displacements in rotating conditions, even when there is only the slightest amount of residual unbalance.

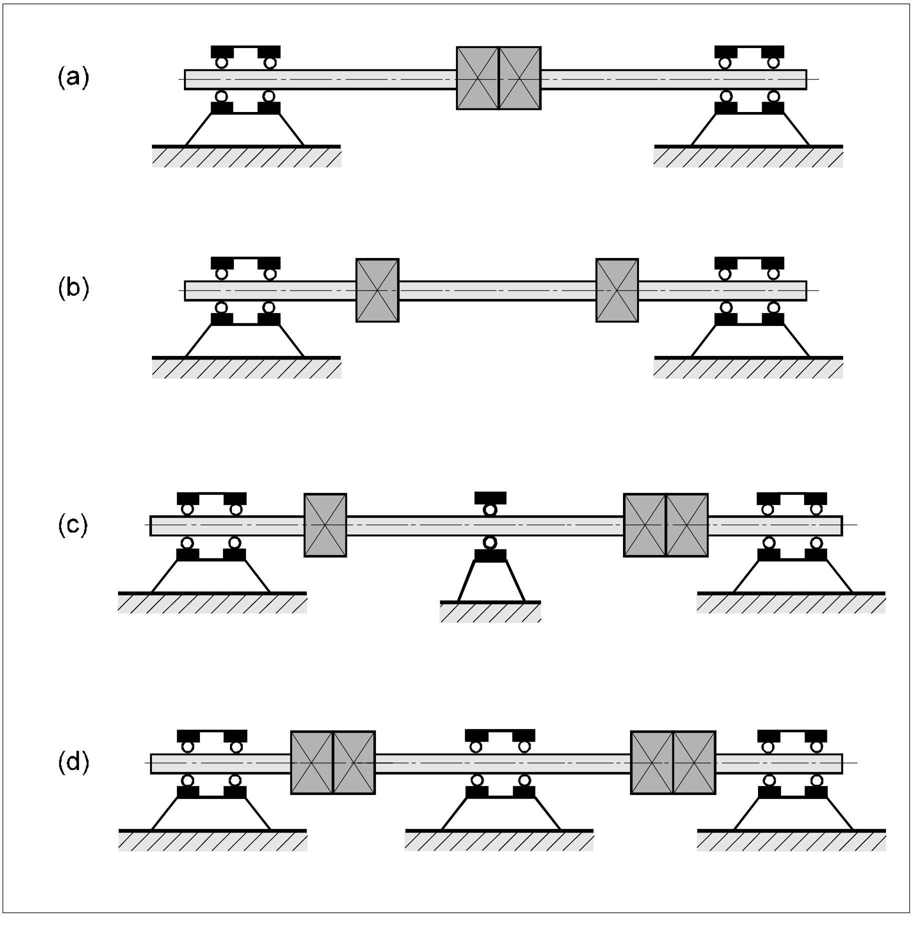


Fig. 5.25 Shaft trains supported on statically determinate bearing configurations with compensating elements

Installing torque transducers with integrated bearings

For torque shafts with integrated bearings three different ways of installing them into the shaft train are possible. They are illustrated in Fig. 5.26.

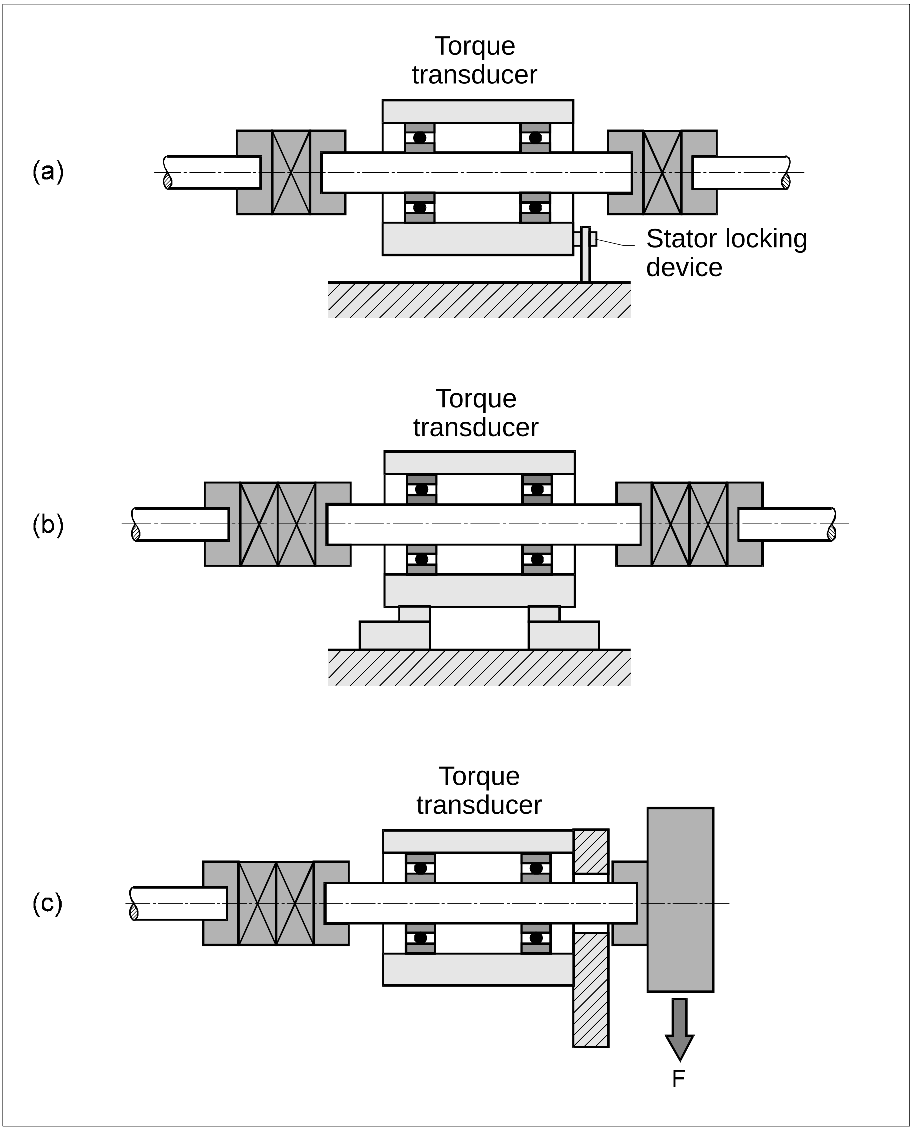


Fig. 5.26 Installing torque transducers with integrated bearings

(a) Freely suspended torque transducer installation. In this case the shaft train supports the housing via the bearings. A double joint coupling or single joint couplings at each end must be provided in order to ensure a statically determinate bearing configuration for the shaft train. Figuratively speaking a torque transducer with two couplings has the function of a joint shaft. In any event a device to prevent the stator from rotating together with the rotor must be provided.

(b) Torque transducer installed as a support bearing between two sections of the shaft train. In this variant the housing is bolted to the floor. Couplings or joint shafts are provided in order to ensure a statically determinate bearing configuration. Generally it is important to bear in mind that this type of application places an additional loading on the torque transducer bearings and is therefore accompanied by a heating effect.

(c) Torque transducer installed as a bearing block for a toothed gear or pulley wheel. In this instance a compensating element has to be provided on the side where the torque transducer is connected to the shaft train. In most cases this is a double joint coupling or a joint shaft, but in a few cases it may even be a single joint coupling. The choice depends on the requirements for a statically determinate bearing configuration. In this installation variant it is necessary to keep in mind not to exceed the permissible parasitic loads that act on the torque transducer due to weight forces, gear tooth forces or belt forces.

In all three variants it is important to remember that long structural components between the torque transducer and the coupling or joint shaft can lead to additional loading of the torque transducer due to transverse forces and bending moments. These need to be smaller than the respective parasitic limit loads. When quantifying these parasitic loads it is important to be aware that additionally dynamic components may occur due to effects such as unbalance.

Installing torque transducers without bearings

Installing torque transducers which are of the conventional, long design but not equipped with internal bearings equates to the freely suspended installation of torque transducers with built-in bearings. This means that the torque transducer is supported at each end by half couplings and therefore acts as a joint shaft (see Fig. 5.27).

However, unlike the case of torque transducers with integrated bearings, the stator must be fixed separately to the machine bed or other non-rotating system component. Accurate alignment of the stator and rotor is required in both the radial and axial directions. Tolerances are type-dependent and can be found in the respective technical documentation.

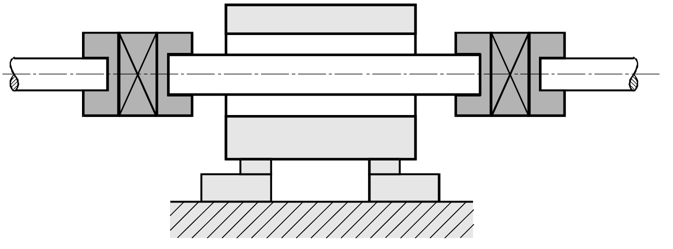


Fig.5.27 Installing torque transducers without integrated bearings

Installing torque flanges

The variant with coupling shown in Fig. 5.28 (a) corresponds to the situation in Fig. 5.27 for long-form torque transducers without bearings. Instead of a half coupling at each end a full coupling is used at one side. This presents no problem for a torque flange because the radial force loading and bending moment loading due to weight or vibrations are of less consequence.

The high tolerance shown by torque flanges relative to parasitic loads means that in many applications it is possible to use a variant that is entirely without support bearings, as shown in Fig. 5.28 (b). In this instance the measurement flange is supporting the joint shaft. However, keeping to the load limits with respect to parasitic loads must always be kept in mind (see Appendix A).

In certain applications, again due to the high tolerance with respect to bending moments and transverse forces, a cantilevered configuration can also be chosen. In such applications the measurement flange typically supports a pulley wheel or toothed gear, as shown in Fig. 5.28 (c). A very successful application of this type is presented in [12] for the field of frictional power measurement in cylinder heads.

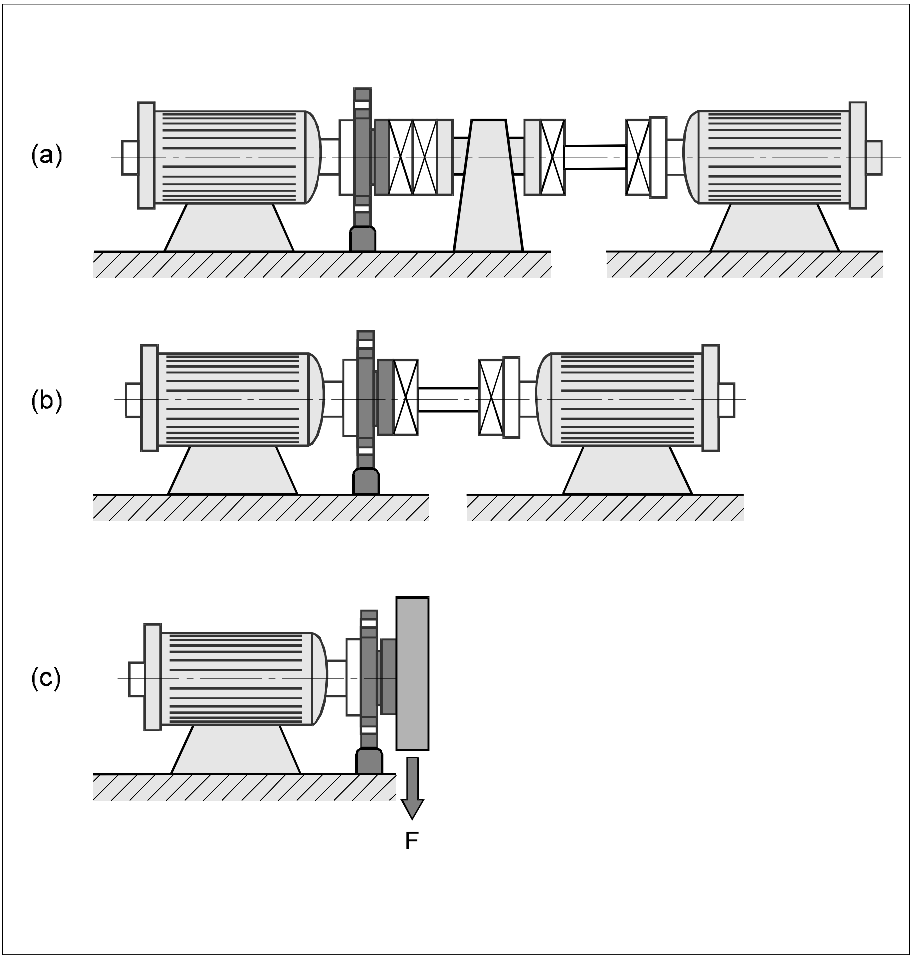


Fig. 5.28 Variants for installing torque flanges

5.2.4 Aligning the shaft train

Good alignment can minimize static offset from the outset. This is true of angular offset, parallel offset and axial offset. Advanced measuring instruments for checking alignment operate on the laser triangulation principle. If an instrument of this kind is not available, alignment can be carried out less accurately with the aid of dial gages. Fig. 5.29 shows the alignment principle.

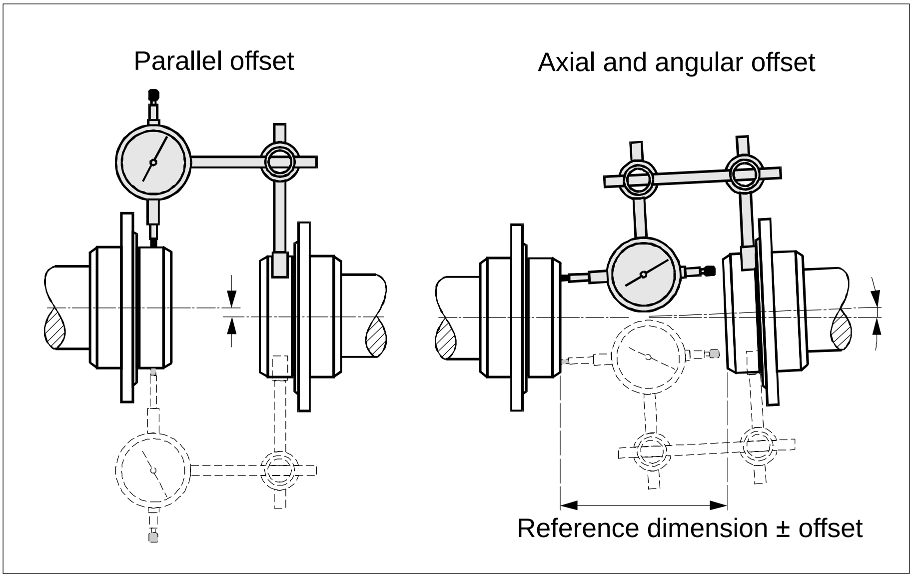


Fig. 5.29 Alignment principle using dial gages

The drawings of the dial gages make it clear which measurement points need to be measured for comparison. There is no need to follow the same measuring setup to the letter. If for example it is not possible to set the foot of the dial gage on the flange part, separate measurements can be taken for the two opposing flanges relative to a common reference surface and a calculation can then be made by subtracting. Alignment by laser triangulation is also based on measuring the position of the same points as those shown in Fig. 5.29.

Radial parallel offset is identified by measuring the circumferential surfaces on both flange parts for comparison. It must always be identified in two axes, for example horizontal and vertical. An additional measurement on the opposite side (the dial gage position shown by dotted lines in Fig. 5.29) makes it possible to distinguish between static offset and run-out/eccentricity. This distinction can also be made by rotating one of the shaft train sections and repeating the measurement at the same point, or by slowly rotating a shaft train section while measuring.

Radial angular offset is identified by measuring the faces on both flange parts for comparison. Angular offset must also be identified in two axes. An additional measurement is needed on the opposite side (the dial gage position shown by dotted lines in Fig. 5.29) to distinguish between radial angular offset and axial offset. To distinguish angular offset from flanges with an angular error (rotating geometric error) the measurement must be repeated at the same point after rotating one of the shaft train sections, or a shaft train section must be slowly rotated while measuring.

Axial offset is identified by measuring the faces on both flange parts for comparison. If the measurement is not taken in the middle of the flange which means directly on the geometric axis of rotation, an additional measurement is needed on the opposite side in order to distinguish it from radial angular offset. Here again it is necessary to rotate the opposing flanges to make sure that the effect of a skewed flange (rotating geometric error) has not been incorrectly identified as axial offset.

Since in practice not only the three forms of static offset but also rotating geometric errors (eccentricity) are mutually superimposed, a very large number of measurements is usually needed for precise alignment. Because the effect of corrective measures can seldom be accurately calculated in advance, several cycles of measuring, correcting and checking are usually required.

If elastic couplings are provided in the shaft train, the alignment can often be finely adjusted when the shaft train is fully installed. This is done by measuring the deformation imposed on the coupling as a result of installing the shaft train.

5.3 Electrical connection

Shielded, low-capacitance measuring cables from HBM are recommended for making the electrical connection between a torque transducer and a measuring amplifier. In transducers with mV/V output these cables make it possible to use the six wire circuit described below, and in torque transducers with integrated electronics they increase operating safety and provide protection against electromagnetic interference. Cable lengths depend on the transmission system employed for the torque transducer and the measuring amplifier being used.

Torque transducers with measurement signal transmission via slip rings or other transducers with mV/V output use a six wire circuit. In this the cable contains sensor circuits which sense the excitation voltage in the transducer and feed it back to the measuring amplifier. In the amplifier, comparison with the preset voltage causes the generator voltage to be re-adjusted. This method corrects all resistances and the changes that occur in them [13].

HBM delivers these cables both ready-fitted with suitable connector plugs, and by the meter for individual cabling. In view of the special technical requirements of the different signals, other forms of cable are not recommended.

If using cable extensions it is essential to ensure a perfect connection with the lowest possible contact resistance and good insulation. The connectors which

HBM supplies fulfill these requirements. All connectors (or sleeve nuts) must be firmly tightened.

Measuring cables should not be laid parallel to power lines and control circuits. If this is unavoidable (for instance in cable ducts), a minimum separation of 50 cm must be maintained and the measuring cable must additionally be fed through a steel conduit. Stray fields from transformers, motors, circuit breakers, thyristor controls and so forth must be avoided.

5.3.1 Protection against electromagnetic interference

The often considerable electromagnetic environmental load which has been continually increasing in recent years, particularly in the industrial setting, creates ever greater demands for immunity from interference in torque transducers, measuring amplifiers and entire measurement chains.

Spurious radiation is generated by:

- Telephone equipment, radio sets and mobile telephones
- Solenoid valves in hydraulic equipment
- AC converters
- Welding equipment
- Power lines and control circuits
- Thunderstorms
- Measuring instruments themselves

These sources of electromagnetic fields can couple interference voltages by inductive or capacitive interaction via connection cables or housing enclosures into unprotected measuring circuits and impair function.

This becomes particularly critical due to continually increasing measurement signal resolution and demands for higher measurement accuracy. EMC problems can be detected as a rule by the existence of non-reproducible errors such as measured values that fluctuate with no recognizable cause.

As an approximation the following order of rank can be given for the immunity from interference of analog signal forms: 10 kHz ± 5 kHz frequency signal (symmetrical better than asymmetrical), 20 mA current outputs, 10 V voltage signals, mV/V (carrier frequency), mV/V (DC voltage). Digital bus systems offer the advantage that the measured value is either absolutely correct or is recognized as being in error.

However, measuring instruments themselves also emit interference voltages and fields. In so doing they can adversely affect the function of other instruments in their vicinity.

To give the practitioner a sense of security in both directions (protection against interference – no unacceptable emissions), all HBM products conform to the EMC guidelines published by the Commission of the European Union or to the EMC regulations of the Federal Republic of Germany and carry the CE mark.

Compliance with the guidelines is thus documented on the product itself. Proof of compliance with the guidelines is provided by testing to relevant EMC standards. HBM measuring instruments and transducers comply with EMC standard EN61326 for measuring instruments and accessories, which lays down both the minimum immunity from interference and the permissible level of emissions.

5.3.2 Shielding

The cable shield is connected so that it extends over the surface of both the torque transducer and the measuring amplifier in accordance with the Greenline shielding design. By this means the measurement system is enclosed in a Faraday cage. When doing this it is important to make sure that the shield at both cable ends is connected so that it extends over the surface of the housing ground. Fig.5.30 shows a view of the Greenline screwed cable gland as an example.

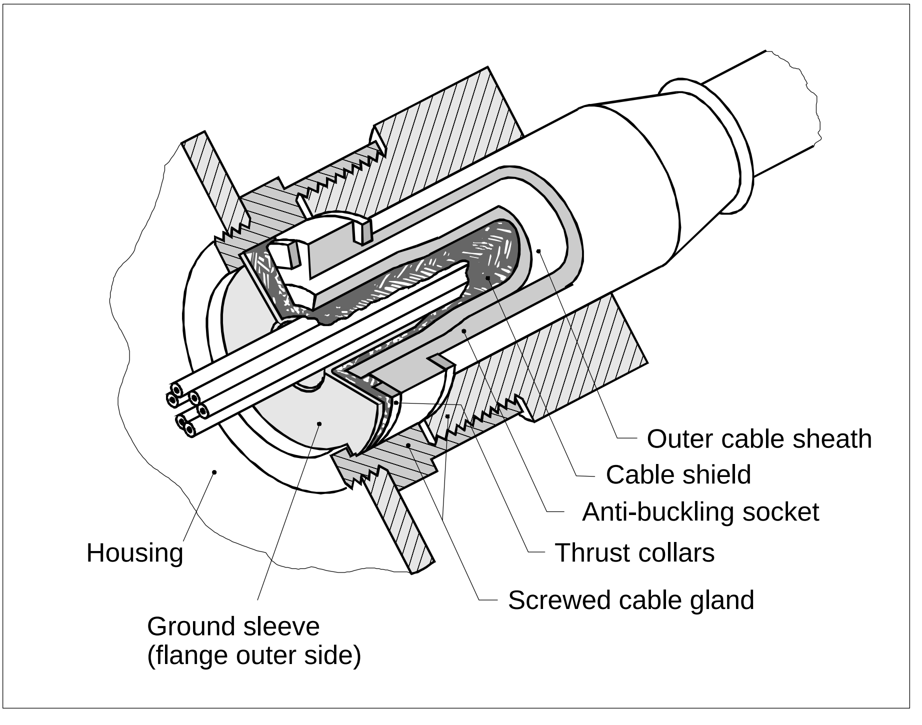


Fig. 5.30 Simplified diagram of a Greenline screwed cable gland

Also with control cabinets a grounding terminal is not enough for the shield connection. The connection must extend over the surface. The measurement signal will then be shielded against electromagnetic interference. In torque flanges, the transmission path and rotor are extensively protected by special electronic encoding methods against electromagnetic influences.

Unfortunately even a shield that is connected so that it extends over the surface at both ends (Faraday cage) complete with potential equalization is only effective against capacitive, high-frequency magnetic and electromagnetic fields. Low-frequency magnetic fields caused by alternating currents are hardly impeded at all. Additional protection can be gained by laying the measuring cable and if necessary also the power cable in steel conduits. The best protection against magnetic fields is provided by conduits made of soft iron or mu-metal.

If there is interference due to potential differences (compensating currents) the connections between operating voltage zero and the housing ground on the measuring amplifier must be separated and a potential equalization line must be laid between the stator housing and the measuring amplifier housing. This should be made of copper and have a minimum line cross-section of 10 mm^2.

A further recommended measure is the installation of a grounding strip between the stator and the machine ground, since a perfect ground connection is not always guaranteed. In a torque transducer with slip ring transmission, this additional conductor can be clamped to the stator to prevent it from rotating with the rotor.

In torque transducers with contactless measurement signal transmission, interference may arise from potential differences between the rotor and stator in the machine, possibly caused by unchecked leakage. In this event, grounding the rotor fully by means of brushes usually helps. The stator must likewise be fully grounded.

5.4 Measuring amplifiers

The measuring instruments that can be used for processing the output signals from torque transducers which are proportional to torque, speed and angle of rotation are dictated by the nature of the transmission technique used in the torque transducers concerned. Above all the measuring amplifier type must be suited to the transducer output signal.

Torque transducers fitted with slip rings can be connected to any measuring amplifier that is suitable for SG transducers. As already described in chapter 3, carrier-frequency measuring amplifiers should be used for preference, since they are designed to be very much less sensitive to any interference that is outside the carrier frequency of the amplifier.

Also thermally induced voltages which occur for example in the slip ring and brush combination are eliminated by carrier-frequency measuring amplifiers. Due to the band-pass behavior of a carrier-frequency measuring amplifier, interference signals with frequencies that differ from the carrier frequency have no effect on the measurement signals. In Fig. 5.31 it can be seen that neither DC voltages (thermocouples, galvanic voltages) nor mains interference are transmitted. However, it should be noted that the frequencies used in present-day frequency converters must not be identical to the carrier frequencies. Otherwise susceptibility to interference could actually be increased. In contrast, a DC amplifier would amplify all interference within its bandwidth to the same extent as the useful signal.

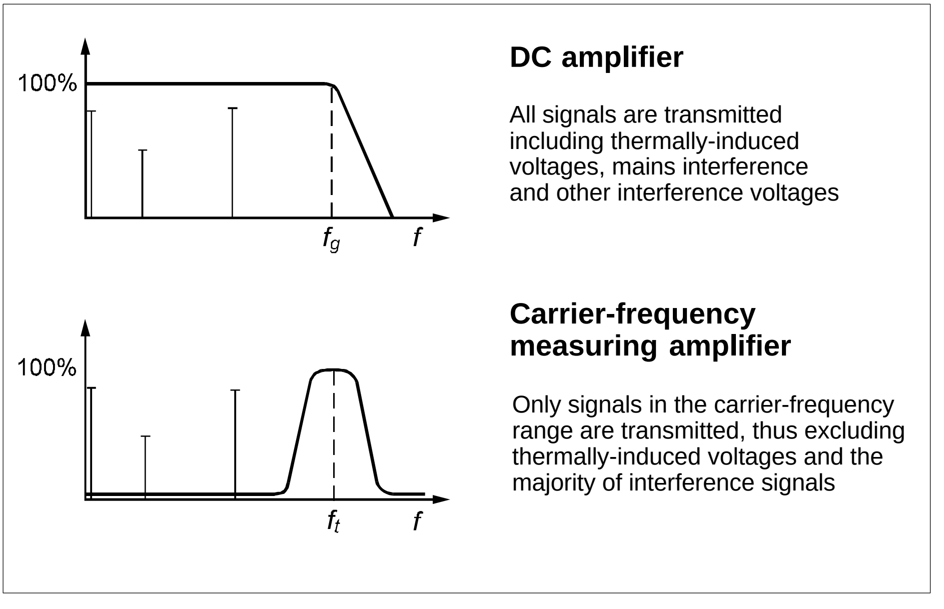


Fig. 5.31 Differences between DC and carrier-frequency amplifiers

On the other hand, torque transducers that use contactless methods to transmit measurement signals or have measuring systems for speed/and or angle of rotation require special electronics. The HBM system solutions available for this purpose are the MGCplus, the Spider 8 and the PME family. Follower electronics units that are put to use in control circuits include application interfaces such as the frequency input of a PLC.

The conversion method used to transform a frequency into an analog voltage or into a digital signal is particularly significant. Here HBM uses the combined frequency count/cycle duration method described in section 5.4.1, since simple count methods or frequency voltage converters are no longer sufficient to meet the demands for accuracy and resolution of present-day measurement technology.

For transducers with a ±10 V torque output signal, an additional amplifier is needed if the signal is to undergo further conditioning. This is the weakness of integrated amplifiers that are set up for one fixed measuring range and one low-pass signal bandwidth.

5.4.1 Features of measuring amplifiers

HBM measuring amplifiers achieve very high accuracy classes (for example the ML60B frequency measurement module at 0,01) and possess the following functions depending on the version concerned:

- Automatic zero adjustment and tare functions, calibration signal activation
- Analog output scalable over wide ranges, making it possible to switch over the measurement chain to sub-ranges
- Selectable low-pass filter with typical cut-off frequencies of 0.05 to 2000 Hz and Butterworth or Bessel characteristic
- Multiple limit value switches (can be used for overload protection)
- Peak-value memory or current value memory

These features can relieve the PLC of simple signal conditioning functions such as limit-value building and peak-value storage.

HBM measuring amplifiers provide an almost unlimited choice of output signals and interfaces:

- Analog voltage output (±10 V)
- Analog current output (0 to 20 mA, 4 to 20 mA or ±20 mA)
- Profibus DP
- CANopen
- Interbus-S
- RS232C
- RS485
- IEEE488

- Centronics
- Ethernet TCP/IP

Which interface to use depends on the follower system and the operating environment. Where possible, digital outputs should be used in preference to analog. It is important to bear in mind not only the sensitivity to noise which analog signals possess, but also the cumulative error due to subsequent conversion procedures. With HBM the amplifier signal already has 24-bit resolution. Converting this to an analog signal only to convert it back to digital in the process control system makes no sense. An A/D converter in a PLC typically has an accuracy of only 0.5 to 1 % at 12-bit resolution.

Since present-day measuring amplifiers operate internally on a digital basis and are digitally adjustable, options exist for adapting to another measuring range, or altering limit values, or varying the filter, and so forth. Complete settings can be stored as a parameter set in the measuring amplifier or PC.

Measuring amplifiers are available in the widest variety of housing versions for laboratories, test shops and industry:

- Desktop housing with or without display
- 19-inch rack frame with or without display
- Front panel housing with digital display
- DIN rail housing
- Die-cast housing with IP65 protection class

The choice depends on the operating circumstances and needs no further comment.

Other tips, for example concerning the choice of filter, can be found in section 5.5.

5.4.2 Combined frequency count/cycle duration method from HBM

The combined frequency count/cycle duration method always works on the same principle, regardless of whether the measured quantity to be evaluated is torque or speed. The amplifiers measure frequencies by means of a purely digital method [14]. This method not only achieves excellent temperature stability as well as long term stability, but is also highly suitable for measuring low frequencies. These measuring amplifiers can also be used as counters for incremental angle of rotation transducers. Fig. 5.32 shows the signal conditioning structure.

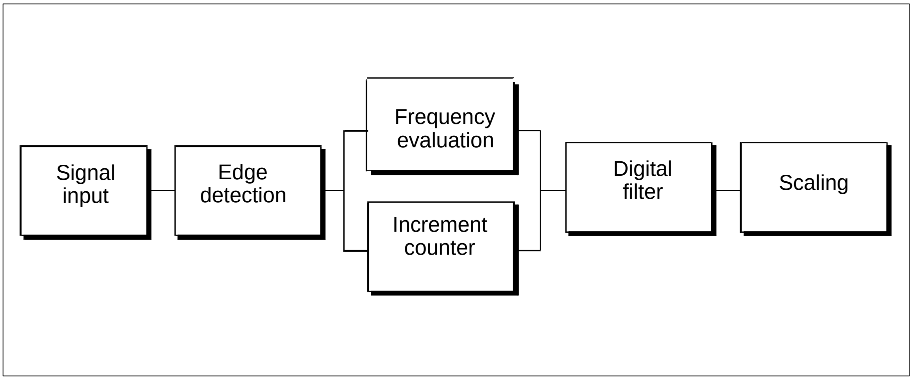


Fig. 5.32 Structure of signal conditioning

Input signals

The amplifier operates with 1 or 2 frequency input signals (F1, F2). Only the positive edges of frequency input signal F1 are analyzed for the purpose of evaluating torque. Both frequency input signals are used for evaluating speed and/or angle of rotation, and in appropriate cases the pulse is quadrupled. The second signal is used for determining the direction of rotation by analyzing the phase position of the two signals relative to each other. In addition a zero-index signal input and a transducer error signal input are available.

The zero index is used as a reset signal in incremental counter mode. The counter is reset to zero when this signal occurs. The transducer error signal is used by incremental transducers, where available, to indicate an internal error (such as failure of the light source in an optical system). If this signal presents with a level of 0 V, the amplifier module identifies the measured value as incorrect.

Signal analysis

In principle the amplifiers analyze only the edges of the frequency input signals. There is a choice between a fixed amplitude of typically 5 V, maximum 30 V, and automatic amplification which can process signals from 100 mV to 30 V.

If both frequency input signals are used, for incremental counting or frequency measurement there is a choice between analyzing only the positive edges of input F1 or evaluating all positive and negative edges of inputs F1 and F2.

If clean signal edges cannot be relied on, the remedy is a switchable digital glitch filter. This ensures that only signal states which stay at a constant level for at least 1.6 µs are taken into consideration. Fig. 5.33 shows how the function works.

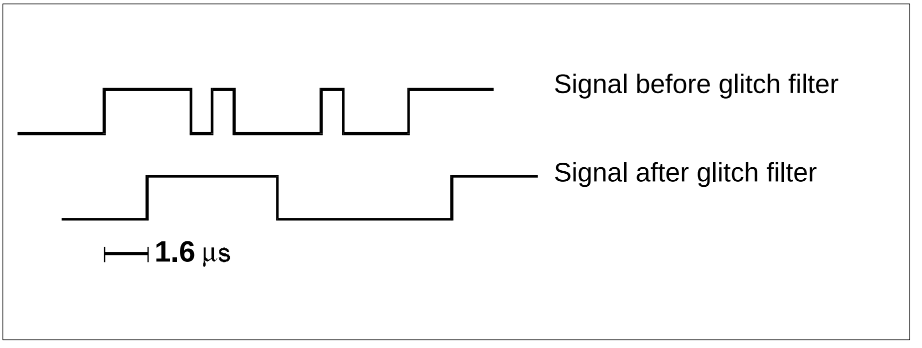


Fig. 5.33 Function of a glitch filter

Frequency evaluation

The method uses a combination of an event counter with a fixed gate time and a cycle duration counter. In principle the analysis operates at a gate time of 1/9600 seconds. Within this gate time, however, not only are the signal edges counted, but also the length of time between the last event within the previous gate time and the last event in the gate time just expiring (the cycle duration). This enables the frequency to be determined, as Fig. 5.34 shows.

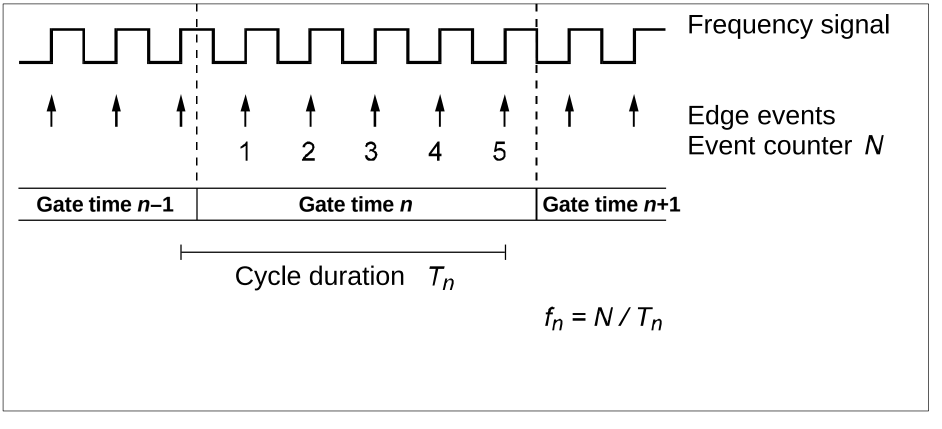


Fig. 5.34 Frequency analysis

The resolution of the method results from the gate time and the counting accuracy with which the cycle duration is determined. At a counting frequency of 39.3216 MHz, such as is used in the ML60B, the frequency resolution is then 39.3216E6 · 1/9600 = 4096 parts. This basic resolution can be increased, however, if the results are submitted to digital filtering. The prerequisite for this is satisfied since the method operates in such a way that every edge event and cycle duration is taken into consideration without interruption.

If the input frequency drops below 9600 Hz, it is possible for no edge events at all to be observed in certain gate time periods. Should this be the case the next gate time is simply started and analysis is only carried out, with a correspondingly longer cycle duration, when a gate time period within which an edge event is acquired occurs once more. Meanwhile the old frequency value is output as the measured value or fed to the subsequent digital filter.

But this can lead to the persistence of certain values if the frequency suddenly drops to zero, so in this case a new frequency value is determined even though there are no new signal edges. This is possible in the form of a maximum estimated value since if no edge events are detected in a particular time period, the maximum possible frequency value can be derived from this as the reciprocal of this time period. If the frequency suddenly drops to zero, this algorithm leads to a hyperbolic drop in measured values (see Fig. 5.35).

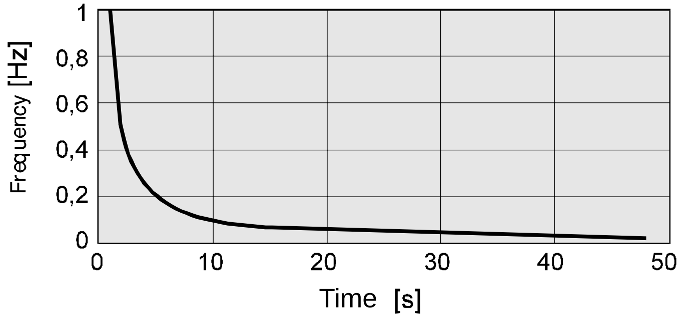


Fig. 5.35 Curve showing the measured values when a 1 Hz frequency signal suddenly drops to zero

In principle the low pass cut-off frequency can be selected irrespective of the anticipated input frequency. If the chosen low pass cut-off frequency is high, however, changes in the measured value and the event of the re-determination of the measured value after each edge event in the case of low measure frequencies will have the effect of a step in the analog output.

Increment counting

Increment counting is possible at input frequencies up to 2 MHz. This means that when the customary incremental transducers are in use there is no need for speed restrictions. The second frequency input is used here for detecting the count direction so that it is also possible to produce an incremental/decremental counter. The counter can be reset via the zero-index input, making it also possible to carry out absolute, incremental angular measurement.

For this type of function it can be a good idea to use a low-pass filter when intending for instance to acquire a characteristic curve for torque and angle of rotation in a multi-channel application. If so, the same low pass cut-off frequency should be chosen for both channels, so that the signal phase delay is the same in each.

5.5 Adjusting the measurement chain

5.5.1 The aim of adjusting the measurement chain

Adjusting the measurement chain means setting up the measuring amplifiers and other follower electronics. The first step is to make sure that all incoming signals can be acquired, and then that the display is correct in respect of the unit and numerical value. With present-day measuring amplifiers the task is mostly menu-driven. Entering the information over a computer interface is another particularly convenient method. When the adjustments have been made, amplifier systems such as MGCplus or the modules in the PME family have the capability to store the setup values permanently. Storing different parameter sets makes it possible to switch between a number of different setup variants without having to enter the settings or adjustments all over again.

Many torque transducers can be used to measure not only torque but also speed and angle of rotation. Each of the three measured quantities being acquired must have its own measurement channel.

5.5.2 Basic measuring amplifier settings

Without claiming to be exhaustive, this section will deal briefly with some important settings which have to be made before making the adjustments themselves. A detailed explanation of the individual steps can be obtained from the product documentation concerned.

Excitation voltage and supply voltage

For all transducers that will be directly operated as SG transducers, the first task is to define the excitation voltage. This applies to non-rotating reference torque transducers and torque transducers with slip rings, among others. Different excitation voltages are available depending on the measuring amplifier, but care should be taken to ensure that the excitation voltage used is suitable for the transducer. It would be preferable to choose the excitation voltage used for determining the sensitivity at the factory or for calibration. In this connection it is important to ensure that the transducer type set up in the amplifier is SG full bridge.

The voltage supply for contactless HBM torque transducers must be ensured by using the correct HBM amplifier (or amplifier module). As explained in chapter 3, the voltage supply for HBM torque transducers is either a square wave voltage supply or a DC voltage supply.

Defining the input quantity and measuring range of the amplifier

Input quantities are usually voltage ratio (mV/V), frequency (Hz) or voltage (V). The correct input quantity is not guaranteed in every case by choosing the right HBM amplifier (or amplifier module). For example the HBM Spider8 amplifier offers the option of using the same channels for measuring analog voltage signals (such as ±10 V) or voltage ratio for SG transducers and pulse counting.

ML60B and MP60 amplifier modules can be used for pulse counting or frequency measurement according to choice. Which of these input quantities is intended needs to be set up in the measuring amplifier before starting to make the actual adjustments. In the case of the frequency signal it is especially important to make sure that the amplifier expects symmetrical or asymmetrical frequency signals to suit the transducer. This may be already guaranteed by choosing the correct amplifier module or may actually have to be set up in the amplifier, depending on the design of the amplifier concerned. In certain types of measuring amplifier the switching threshold level must be set up to suit the amplitude of the input signal.

Frequently the user also has to define the measuring range for the input quantity concerned. Quite often this is not continuously adjustable. For optimum resolution it is always best to choose the smallest of the available steps that cover the full range of expected input signals.

Example 1:
HBM torque transducers using contactless signal transmission and frequency output have a zero signal of approx. 10 kHz and a nominal sensitivity of ±5 kHz, depending on the direction in which the torque acts. For use in the range from positive to negative nominal torque, input quantities of 5 kHz to

15 kHz can accordingly be expected. It follows that suitable ranges are those from 1 kHz to 20 kHz (in the case of the HBM amplifier ML60B to the modular system MGCplus) or from 10 Hz to 100 kHz (in the case of the HBM amplifier MP60 from the PME family).

Example 2:
HBM torque transducers that are non-rotating or fitted with slip rings have a nominal sensitivity ranging from ±1 mV/V to ±2 mV/V depending on the type. In many HBM measuring amplifiers the measuring range is derived automatically when the excitation voltage is chosen. In other types of amplifier it must be explicitly chosen, for example ±3 mV/V to cover a transducer sensitivity of 2 mV/V.

Example 3:
The speed signal is usually a pulse train with a frequency proportional to the speed. The input signal for the measuring amplifier is therefore a frequency. If the expected maximum speed is 1000 min^{-1} and the number of pulses per revolution delivered by the speed module is 360, the maximum number of pulses per minute is 360,000 or 6000 per second. Therefore the chosen measuring range must go at least as far as 6 kHz.

Filtering

Before measurement can begin it is necessary to set up a careful choice of the filters most commonly incorporated into amplifiers. The concepts for selecting the right settings are discussed in chapter 6 in connection with measurement data conditioning for vibration measurement.

Defining the output unit

Defining the physical output unit for torque, which is here usually N·m or kN·m, is important for showing the correct unit on the display or for storing the correct unit on data media. It provides the final piece of information that makes the actual adjustment of the measurement chain, that is, the entry for the characteristic curve, complete. The measuring amplifier imposes no restrictions and carries out no plausibility checks, which means for example that it will even accept mbar for a torque transducer.

5.5.3 Entering the characteristic curve for torque measurement

Defining the characteristic curve according to the measurement task

Entering the characteristic curve for torque measurement is the pivotal task in adjusting the measurement chain. Before beginning adjustment it is important to be clear about certain questions regarding the way in which it is planned to use the measurement chain.

The first thing to be clear about is whether the transducer is going to be operated in its nominal measuring range or only in a sub-range. Use in a sub-range means when a torque transducer with, say, a nominal measuring range of ±1 kN·m is used in the range ±800 N·m. In order to optimize accuracy for sub-range applications, the sub-range can be taken into account when defining the characteristic curve on which the adjustment will be based. There are two possible ways of going about this.

In the first, which should be preferred in the case of strong accuracy requirements, a working standard calibration or DKD calibration is carried out with the measuring range restricted to the sub-range. Thus the information contained in the calibration certificate is specially suited to the sub-range application. The transducer can safely be used over its full nominal measuring range, it is merely that the calibration is valid for the sub-range only. The situation is different in exceptional cases where the sensitivity is readjusted so that the entire span of the transducer's nominal output signal is assigned to the torque span of the sub-range.

The second method is based on a test report or calibration certificate for the whole of the nominal measuring range. However, the calibration results for torque levels outside the desired sub-range are disregarded for the purpose of defining the characteristic curve. The principle is illustrated in Fig. 5.36 with strongly exaggerated linearity deviation. The figure shows the characteristic curve for the sub-range from zero to 40 % of nominal torque and the characteristic curve for the whole nominal measuring range using a setting based on zero point and end point.

In the example, straight lines through the zero point and end point are used as approximations for the characteristic curve for the entire range and for the sub-range respectively. However, in principle a straight line of best fit or a best fit polynomial can be generated specially for the lower sub-range. However, such lines or curves of best fit for sub-ranges are not made available on a calibration certificate. Instead the user must take responsibility for generating them from the calibration results for the individual torque levels.

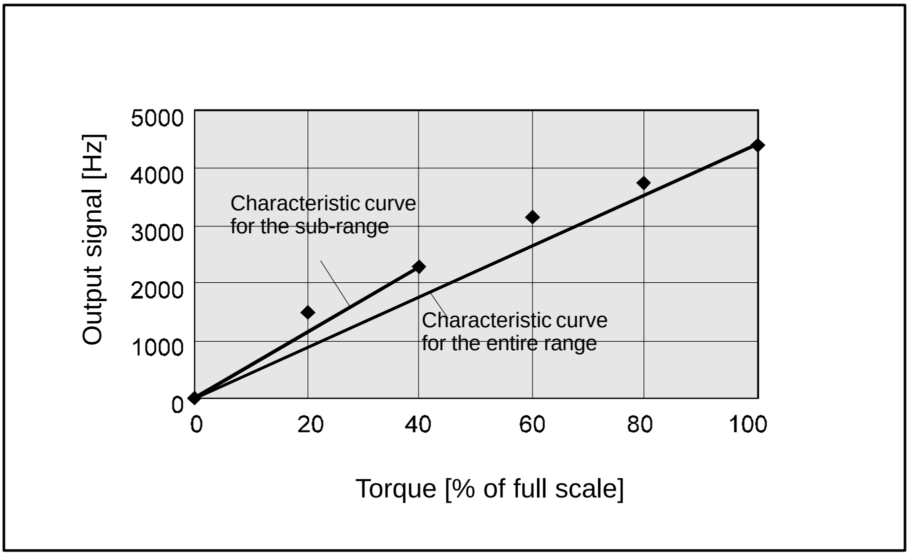


Fig. 5.36 Characteristic curve for a sub-rangefrom 0 to 40 %

If possible it should also be clear whether the application is intended to measure clockwise torque or counterclockwise torque, for in most cases a different sensitivity (or best fit polynomial) will apply to clockwise and counterclockwise torque for the same transducer. A single sensitivity value is defined only in the case of calibration for alternating torque (see chapter 7).

Zero balance, zero offset and tare

The first step in entering the characteristic curve is normally the zero balance. The zero signal varies from one individual to another in a single type of torque transducer and moreover the value of the zero signal is prone to change during mounting. In connection with monitoring torque transducers, section 5.1.2 mentions that even when the correct mounting conditions are observed, the zero signal may change by as much as 3 % of nominal sensitivity. Lastly, it is also possible that there could be continual loading by an initial torque which reproducibly is always superimposed on the torque actually being measured.

In order to avoid falsification of the measurement result due to these effects, a zero balance must be carried out before starting to measure. The transducer must be free of load (except for any initial torque which may be present). The current value of the input signal is then calibrated in the amplifier (using, that is, the output signal from the transducer in Hz, mV/V or V) and assigned to the output value zero in the physical unit (which is usually N·m or kN·m). For measurements of greater accuracy it is recommended that the zero balance should be carried out only after a first loading of the built-in torque transducer under operating conditions in order to anticipate the effects of mechanical settlement.

Complementing the zero balance, a zero offset can be set up and the tare function can be used, in order to adapt the zero point after adjusting the measurement chain. As opposed to defining the zero point during actual adjustment, in this case the zero offset or tare values are recorded so that the correction procedure can be reversed. The value concerned is recorded in output units (for example N·m). For torque measurement the procedure and concept of zero offset correspond to the physical background. On the other hand the concept of tare, which is a term originating from weighing technology, works in conjunction with the concepts of gross and net. A parallel in the meaning of the word with regard to torque measurement is seen only in the rare case of a constant initial torque signal.

Measuring amplifiers are capable of displaying the gross or net signal according to choice. Whereas the zero offset affects both, changing the tare value affects only the net signal. A suggested use of the zero offset and tare functions could be along the following lines. After mounting, the zero point is defined by applying the zero balance procedure. For everyday balancing the zero offset or tare functions would be used.

This type of short-term zero balance can be used to minimize the effects of temperature on the zero signal (see appendix A) by carrying it out at the actual operating temperature. Then by checking the value of the zero offset (or of the tare) the long-term zero point drift can be monitored, even though it is compensated for in measuring mode. Details of the criteria to be used in assessing variations in the zero point can be found in section 5.1.2.

Entering the characteristic curve in the form of a straight line (two-point adjustment)

In most cases the characteristic curve is entered in the measuring amplifier as a straight line defined by a two-point adjustment. The slope of the straight line follows from the sensitivity, which represents the transducer's output signal span between nominal torque and actual torque as zero. The calibration signal can also be used for entering the slope of the line, since the associated torque span corresponding to the value determined electrically by the calibration signal is separately specified for each transducer.

Even better than using the nominal sensitivity from the data sheet is the individual sensitivity of the transducer, which can be found from the test certificate or from a working standard or DKD calibration certificate. In this way it is also possible to take into account the deviations depending on the measuring range and on the direction in which the torque acts for which it is intended to define a straight line.

The values that should be used from the respective certificates are discussed in chapter 7 in connection with the detailed explanation of the test certificate and calibration certificate. Depending on the application it may be even better if the characteristic curve is defined by an on-site calibration of the torque transducer in the planned application (e.g. test bench). The reader is referred to the discussion on the subject in chapter 7.

HBM measuring amplifiers offer two practical options for inputting the characteristic curve. The first option is to enter the nominal value, as it is known, which involves entering first the physical value of the display or output (for example in N·m) for the full measuring range and then the associated input signal (corresponding to the output signal from the torque transducer typically in Hz). To do so, the input signal has to be entered as a span relative to the zero signal. This procedure is chosen when, for example, the sensitivity shown in the test certificate is used as the basis for the adjustment. The sensitivity is the input signal assigned to the full scale of the nominal measuring range (usually in N·m).

The second option consists of specifying any two points on the straight line. Here the physical output values (typically in N·m) must always be entered as numbers. The assigned output signals can be either entered or captured by measurement. It is not mandatory for one of the two points to represent the zero signal. The entered or captured input signals are understood by the measuring amplifier as absolute numeric values, not as a span relative to the zero signal. An example to illustrate this method is adjusting the torque transducer with the aid of the calibration signal. In this case the first point to be captured by measurement is the zero signal. In order to define the second point, the output value assigned to the calibration signal is entered (see test certificate or calibration certificate) and the assigned input signal is captured when the calibration signal has been activated.

Multiple-point adjustment and the use of best fit polynomials

High-precision measuring amplifiers offer not only two-point adjustment but also the option of multiple-point adjustment, in which the characteristic curve is represented in the form of a series of straight lines. In appropriate cases even greater precision is offered by using best fit polynomials, in which the results of a calibration over several torque steps and possibly multiple measurements are taken into account.

In both cases however it is important to consider the point that the highly accurate reproducing by a characteristic curve acquired by calibration only makes sense if random influences on this characteristic curve are slight, in other words if there is an average over an adequate number of measurements. This usually requires the existence of a DKD calibration certificate.

As a rule, implementing the characteristic curve in the form of a polynomial of the second order or higher is not possible in the measuring amplifier, but requires data processing in a computer. Suitable best fit polynomials for instance are part of a DKD calibration certificate (see chapter 7). A polynomial must be selected in which the torque is expressed as a function of the electrical output signal from the transducer.

5.5.4 Entering the characteristic curve for measuring the speed and angle of rotation

Setting up the measuring amplifier for speed measurement

The output signal for speed measurement in all torque transducers consists of two pulse trains at a frequency proportional to the speed. In order to adjust the measurement chain the zero signal and nominal value must be entered, as was seen in the case of the torque characteristic curve. The zero signal is 0 Hz in every case. The nominal value that has to be entered is the highest possible operating speed n_{max}. Since the pulse train is generated so that a defined number N_{imp} of pulses is delivered per revolution, this results in the assigned input frequency f_{in} for a given maximum speed n_{max} in accordance with the formula

$$f_{in} = \frac{n_{max} \; N_{imp}}{60} \quad \text{(numerical formula with } n_{max} \text{ in min}^{-1}\text{, } f_{in} \text{ in Hz)}$$

It should be noted that in this formula the number of pulses delivered electrically by the transducer must be used, and this is not identical to the number of mechanically generated increments in every case, for example the number of slots in the slotted disk in an optical speed measuring system. In some cases there are also setup options in the stator of the torque transducer for varying the number of electrical pulses.

In addition to the characteristic curve itself there are some important options for speed measurement which have to be set up in the measuring amplifier. These include the analysis of the second pulse train. This is used either for detecting the direction of rotation or for increasing the number of pulses per revolution (doubling or quadrupling the frequency).

Setting up the measuring amplifier for angle of rotation measurement

For measuring angle of rotation the amplifier must be set up on pulse counting. In order to adjust the measurement chain the zero signal and nominal value must be entered, as in the case of the torque characteristic curve. The nominal value that has to be entered is the maximum angle of rotation φ_{max} and the assigned input signal is the associated number of pulses $n_{\varphi max}$ received by the measuring amplifier. The pulse train is generated from a given number N_{imp} of pulses per revolution. the assignment is calculated in accordance with the relation

$$n_{\varphi max} = \frac{\varphi_{max} N_{imp}}{360°}$$

In order to exclude an angle of rotation display greater than 360° even though rotation in excess of a full revolution occurs in the application, the transducer must deliver a reference pulse. This pulse can then be analyzed as a zero index by measuring amplifiers with the appropriate function.

6 Analysis of vibrational processes

This chapter deals with recognizing, measuring and interpreting vibration as it typically occurs in rotating machinery in which torque measurement technology is applied.

Chapter 4 gives a basic description of the relevant instances of vibration, how they are computed and their significance for the choice and layout of equipment. A brief introduction to the most important general concepts of vibration engineering is given in Appendix B. There is a concise explanation of the basic concepts of signal analysis, particularly those on which the data analysis techniques described below are founded.

A comprehensive treatment including a full mathematical derivation can be found in the specialist literature on the theory of vibrations, for example [15]. On the subject of carrying out and evaluating vibration measurements on rotating machinery the reader is referred to VDI Guideline 3839.

6.1 The aim of vibration analysis

The first aim of vibration analysis against the background of torque measurement is to distinguish between genuine mechanical vibrations and apparent vibrations due to effects caused by inadequately adjusted measurement equipment or incorrect data analysis. Secondly, vibration analysis can be used to assign mechanical vibrations to causative mechanisms (such as unbalance). Assignment of this kind is particularly necessary if the vibrations constitute an excessive loading and effective steps must be taken to reduce them.

On the other hand vibrations can also be investigated as a method of diagnosing certain types of defects. The emphasis is then placed not so much on whether the vibrations constitute an unacceptable loading as such, but rather whether the underlying cause of the vibrations represents an unacceptable failing in the machinery itself. For example, continuous vibration monitoring can detect the early stages of failure in roller bearings or transmissions, giving the opportunity to stop the machine before it crashes.

6.2 Measuring vibrations in rotating machinery

6.2.1 Suitable transducer types and how to arrange them

As explained in chapter 4, vibration in the shaft train can take various forms such as torsional, bending or axial. To obtain a complete picture of the vibrational processes in a rotating machine, it is usually necessary to use several different types of transducer. This section will briefly outline which transducers are suitable for the various tasks involved in vibration measurement. It will not go into detail about how to handle and operate different types of transducer, nor will it discuss such matters as the amplifier electronics needed in each case. For information of this kind the reader is referred to the specialist literature and to the technical documentation on the products concerned.

For measuring and identifying torsional vibration torque transducers are very suitable. Other measurement principles are also used as alternative or supplementary methods. One such principle is the simultaneous acquisition of the angle of rotation at different points on the shaft train in order to determine the extent of its torsion. Another method consists of high-resolution measurement of the rotation speed using a device such as a slotted disk. In addition special instruments based on laser interferometers are available for measuring torsional vibration. Unlike the usual laser vibrometer, in which a single laser beam is directed onto the test object, in this case the laser light is divided into two parallel beams. Analysis of the interference between the reflected components of each beam gives the advantage that the effects of superimposed bending vibrations are canceled out.

Particularly suited to the measurement of bending vibration are contactless displacement sensors (such as eddy current sensors) arranged around the circumference of the shaft train. In the case of bending vibration of the shaft train, bending can occur in two spatial directions. In order to acquire a complete picture it is therefore necessary to use at least two transducers (see Fig. 6.1).

By measuring in this way, the runout – that is, the geometrical lack of roundness and unevenness of the surface as well as the inhomogeneity of the material – is measured as a component which is periodic with the rotation frequency. Modern systems offer various means of compensation. The axial position is highly significant for this type of vibration measurement, since bending vibration occurs in different mode shapes (see chapter 4).

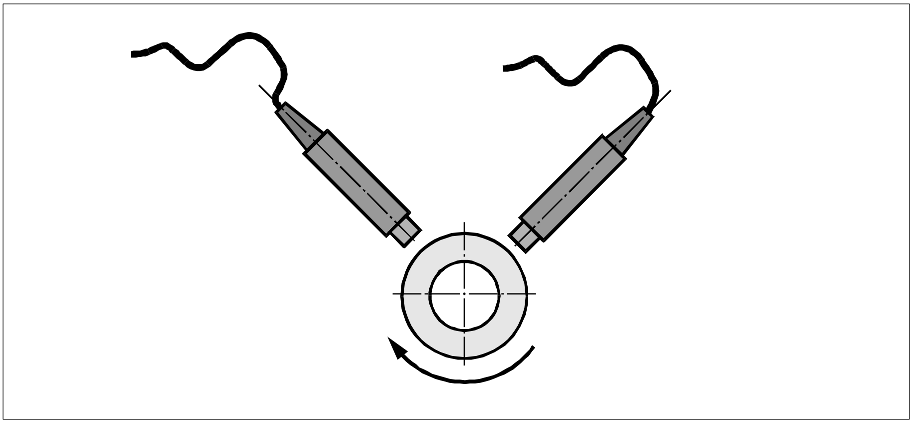


Fig. 6.1 Contactless eddy current transducers for measuring bending vibrationContactless eddy current transducers for measuring bending vibration

If the vibration mode shapes are known approximately in advance, it is only necessary to ensure that the point of measurement does not lie at a vibration node. However, if the intention of taking measurements is to determine the mode shapes, it may be necessary to measure bending deformation at several axial positions. Detailed information on how to configure a measurement application of this type is contained in standard DIN 45670.

Another possible method of identifying bending vibration is to measure the vibrations at the bearing blocks. However, this method is highly dependent on bearing block stiffness. If too stiff, the usually low-frequency vibration components will hardly be transmitted. The measurement signal is then often dominated by frequencies arising from movements of the rolling elements in the bearing.

Piezoelectric accelerometers are very widely used as transducers for this type of measurement because they are so simple to use. One great advantage, particularly during troubleshooting, is that it is often enough just to fasten the accelerometer to a smooth surface on the bearing block with wax. Accelerometers also exhibit a very wide band of frequencies. Signal frequencies can range from 5 Hz to several kHz. However, at low frequencies it is important to consider that very low accelerations are at work even when displacements are quite large, so that low-frequency vibration components are easily overlooked when acceleration is being measured.

Another type of transducer for taking measurements at the bearing block is the inductive vibration speed transducer. The frequency band for this transducer type ranges from 10 Hz to 1 kHz. Speed transducers were among the first vibration transducers produced. Many specifications concerning vibration limits are still based on the vibration speed. The great reliability of the principle is reflected among other things in the fact that it is used in many types of balancing machines.

Contactless displacement transducers or vibration transducers mounted to the bearing blocks are also the right choice for measuring axial vibration. However it is often difficult to find suitable, accessible measurement points.

6.2.2 Data conditioning and recording

Filters

Filters are an important data conditioning tool. They are used to filter signal components with certain frequencies out of a signal in order to isolate those components with the frequencies deemed to be physically relevant. The resolution of a signal into components of various frequencies is explained in section 6.3.2, which deals with analyzing data in the frequency domain. In particular the choice of filter frequencies is highly dependent on the expected vibration frequencies and on which of the possibly contained frequencies the measurement results are intended to return. These concepts are therefore considered in this section.

The task of a high-pass filter is to suppress frequencies below a specified value. Such a filter can be useful for filtering out a static offset of the measurement signal when it is intended to take only vibrations into consideration. If for example the intention is to measure bending vibration using contactless displacement transducers pointing toward the shaft, the constant component is determined only by the distance between the transducers and the shaft, and therefore has no information content concerning the processes it is intended to measure. The use of a high-pass filter with a very low frequency eliminates this effect from the measurement signal without the need for tedious zero balancing. This method is also known as measuring the AC signal. This kind of option is often made available by the measuring amplifier without being called a filter.

The task of a low-pass filter is to suppress frequencies above a specified value. This can be required in order to smooth the signal, for instance if the useful signal is submerged beneath high-frequency noise or similar interference. Low-pass filters are highly important, for if the signals are not known it is only with their help that defined upper limits for the frequencies contained in the signal can be obtained. This is absolutely necessary in order to avoid the alias effect.

This will be discussed in more detail in the next section, which deals with analyzing data in the frequency domain. That section also describes how the sampling rate and frequency for the low-pass filter are determined according to the signal frequencies that are important in the measurement concerned. The decision about which of the various frequencies contained to bring out also has an effect. The basic outlines relating to this decision are explained in chapter 4.

It needs to be mentioned that no real filter works ideally. The suppression of frequencies beyond the limiting frequency of the filter does not take place instantaneously, but increasingly strongly with increasing distance from the limiting frequency. The effect of the filter is often defined by using the 3 dB limit which specifies the frequency at which the amplitude of a single-frequency signal is reduced by 3 dB. In addition, all filters have certain signal delays. In the terminology of frequency domain methods it is therefore said that the phase position of the signal components is shifted.

Different priorities for minimizing as far as possible the unwanted secondary effects of filtering make it necessary to choose between filters with different characteristics. The commonest of these are the Butterworth characteristic and the Bessel characteristic, which are also incorporated in HBM measuring amplifiers. The Butterworth characteristic provides a very steep transition between the pass band and the stop band. However, such filters have the disadvantage of relatively long settling times. In this regard filters with the Bessel characteristic are more favorable. Bessel filters are therefore recommended in situations where the relevant frequencies can be kept at an adequate distance from the limiting frequency of the filter. By contrast, Butterworth filters are advantageous when a sharp separation between the pass band and the stop band is important. However, the factor that most decisively influences the settling time is the bandwidth of the filter, that is, the width of the frequency band that constitutes the pass band. The narrower the bandwidth, the longer the settling time.

When feeding the measurement signal into a control loop, it is important to be aware that the transfer properties of the filter can cause an instability in the control loop, depending on the response characteristic of the other components concerned. For instance under certain circumstances long settling times can destabilize a control loop.

Choosing the sampling rate and avoiding the alias effect

Recording data so that it can be analyzed later always involves dividing the time into discrete intervals. This digitizing process is known as sampling, and is inevitably accompanied by limitation of the frequency resolution. Clearly a digitized signal cannot return frequencies that are approximately the same as the sampling rate. It is easy to imagine how a 100 Hz sine-wave signal would appear if we looked at only 100 points per second, in other words if the sampling rate and the signal frequency were both 100 Hz. Since sampling is repeated after exactly one complete period, the same output value would appear each time and we would see a horizontal line instead of a sine curve.

Far worse, however, is when the high frequencies contained in the original signal (before sampling) show up in the digitized signal as low-frequency components (at alias frequencies), as illustrated in Fig. 6.2. This figure shows a 40-Hz sine-wave signal sampled at a rate of 50 Hz. The result appears to be a vibration signal with a frequency of 10 Hz.

This means in practice that if high frequencies are not acquired because they are not considered to be technically relevant, it is not simply a case of doing without information on the high-frequency signal components. On the contrary, these signal components generate misleading information which has an effect on the lower frequency band that is being studied. In order to avoid this alias effect we have to invoke the Nyqvist/Shannon sampling theorem. This prescribes that the sampling rate must be at least twice the highest frequency contained in the signal. For the purpose of reaching meaningful quantitative conclusions about the signal components in the highest frequencies present, however, practitioners recommend a factor of 5 rather than 2.

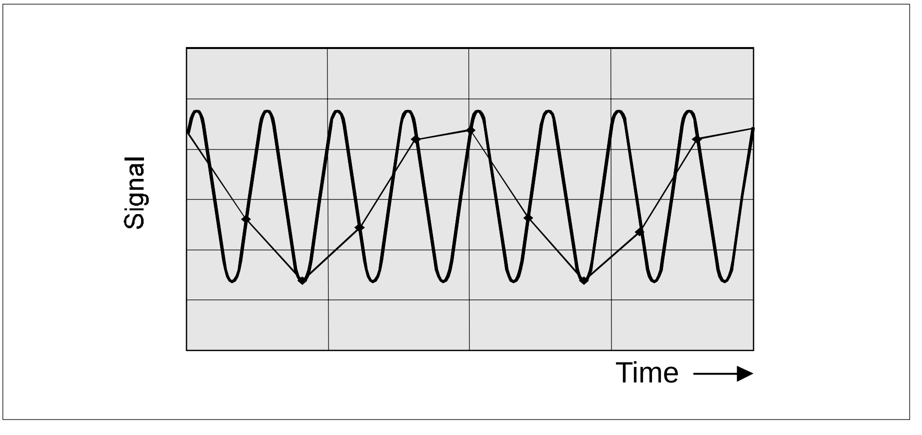


Fig. 6.2 Example of the alias effect

On the other hand, it is not uncommon to find that a signal contains very-high frequency components that are not technically relevant. Acquisition at the minimum sampling rate prescribed by the theorem would generate an unmanageable flood of data. Also generally it is quite impossible to reach a reliable conclusion about the highest frequencies present in a signal. In order to avoid alias effects despite these considerations, it is often necessary to limit the signal frequency in the upward direction by using a low-pass filter. Naturally this cannot be done by applying digital filtering to the signal once it has been sampled.

6.3 Data analysis

The aim of data analysis in this connection is to use data obtained from vibration measurement in order to derive information about the actual behavior and properties of the technical system under investigation. This includes interpreting the behavior disclosed in this way. At this point data analysis passes over to a consideration of the backgrounds and relationships of the vibrational processes shown in chapter 4, where vibrations are discussed with regard to design and layout.

6.3.1 Time domain

In signal analysis, procedures and methods are divided into two main categories: those that operate in the time domain and those that operate in the frequency domain. In methods that operate in the time domain, data stays in the sequence in which it was acquired.

Identifying extreme values

The first and always highly important consideration in the analysis of vibrational processes is to determine the absolute amplitudes. For this purpose only the extreme values need to be identified. Often all that is needed is to visually check a graphical display of the plotted measurements. If the data is available in digitized form, the task of finding the extremes can easily be given to a computer. Depending on the situation, it can be important to identify the absolute extreme values for a given series of measurements, or the local maxima and minima (peaks).

The amplitudes make it possible to draw conclusions about the loads to which individual components in the setup were exposed during operation. These conclusions are drawn more or less directly depending on the measured quantities acquired. It is obvious that torque measurement provides a very direct assessment of the torsional stresses acting on the torque transducer and other components. Measurement of the radial accelerations makes it possible to reach direct conclusions about a part of the accelerations acting upon the components in the shaft train, which can be important in cases where electronic components are rotating as well. However, it is also possible to use the same measurement data to obtain evidence about loads arising from bending deformations. This is done by using the acceleration data to gather evidence about displacements, and then considering this in conjunction with the bearing configuration and the possible deformation of adjacent components to determine the bending deformations.

Differentiation and integration

Using differentiation it is possible to calculate velocity from a measurement of the displacement or acceleration from the velocity. Using integration it is possible to determine velocity from the acceleration and displacement from the velocity. Both methods can be carried out using a data processing computer program (e. g. catman® from HBM).

Since differentiation describes the way an output signal changes over time, a data plot with an irregular trend line will always result in a data plot with an even more irregular trend line due to differentiation. This effect also amplifies noise components, which can never be entirely avoided. As a rule therefore, digital differentiation is more problematical than digital integration.

However, digital integration has the disadvantage that initial conditions are necessary in order to determine a state unambiguously. For instance, if it is intended to use the measured speed of a point to determine that point's absolute position, the position must be given for a reference point in time, that is, an initial condition is required for the position. Even if this information is available, the task of incorporating it into automated, computerized integration is extremely tedious and requires highly detailed intervention. For vibration measurement, however, this limitation has virtually no great practical significance because the oscillating component of the signal is almost always sufficient for the purpose.

Drawing conclusions from the shape of a curve

At times a great deal of important evidence about the type or cause of oscillating measurement signals can also be gathered from examining the shape of the curve.

An example of this is severely flattened, virtually horizontal trend lines at the positive or negative peaks, giving the impression that they have been truncated, as shown diagrammatically in Fig. 6.3. Such curve shapes suggest that a saturation effect or an obstruction may be present in the system.

There may be an overload or electrical saturation in the measuring system, or there could be a mechanical stop that occurs when there is play in components such as roller bearings. It would be possible to make the distinction in this case by investigating whether any other effects are present which are typical of impacts, since these most commonly occur in connection with play. Also the sources of oscillating torque components which are special to internal combustion engines, as already described in detail in chapter 4, can often be identified quite easily by comparing the data plots with the typical properties of the engine type concerned.

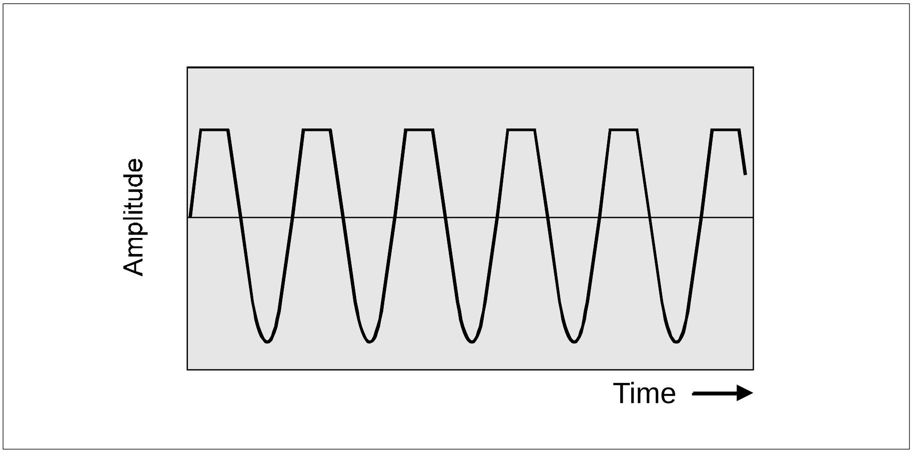


Fig. 6.3 Example of saturation effects or obstructions in the system

Particularly if frequency domain methods of analysis are planned for the next stage, it can be useful to carry out a plausibility check of the chosen sampling frequency by examining the curve in the time domain. Too low a sampling frequency shows up as an extremely jagged line such as that shown in Fig. 6.4. Often there is no third data point between two peaks or extreme values.

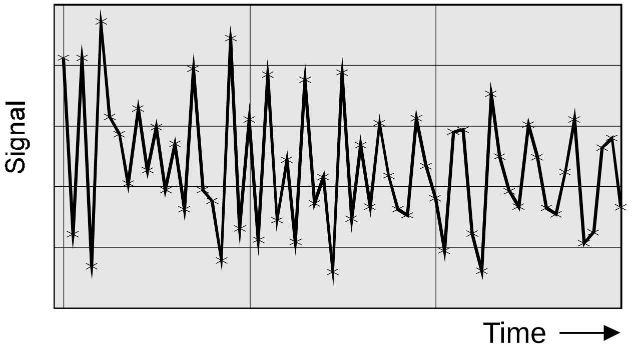


Fig. 6.4 Example of data recorded at too low a sampling rate

Investigating bending vibrations by considering orbit shapes

As already mentioned, to acquire a complete picture of the bending displacements of a point under investigation on a shaft train, its movements need to be measured in two directions. For example if a coordinate system is set up in such a way that the *x* axis coincides with the longitudinal axis of the shaft, bending must be measured in the *y* and *z* directions.

The display can be set up not only in the form of two separate time-dependent curves, but in the case of periodic or quasi-periodic movements, also in the form of orbital plots. In this type of plot the displacements in each of the mutually orthogonal axes are superimposed and traced as shown in Fig. 6.5.

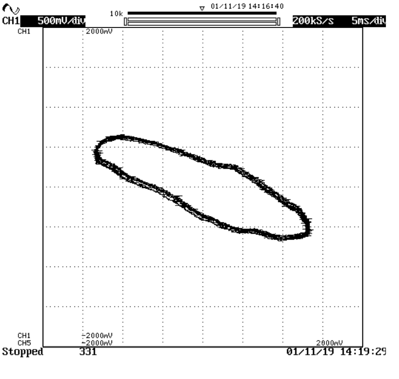


Fig. 6.5 Orbit of a rotating shaft

Orbit displays are easy to generate on an oscilloscope during online measurement. Since they also provide a great deal of information about bending vibration without any computer involvement, they are an extremely popular method of taking observations while measurements are in progress.

Whereas the orbit of a completely stationary test point is itself no more than a point, the real orbit of a test point on a rotating shaft exhibits a curve of more or less greater extent. Closed curves mean periodic motion. However, it should be noted that the image generated by an oscilloscope depends upon the duration of the afterglow from the point of light impinging on the focusing screen. Bearing in mind the short time frames that exist due to these practical concerns, most observed orbits are likely to be closed. Deviations from periodicity show up as gradual changes in the shape of the orbit over time.

The basic shape of an orbit is usually an ellipse. As a general rule the rotor center point travels on the orbit and is always synchronous with and in the same direction as the rotation of the rotor about its own axis. The elliptical shape results mathematically from the superposition of oscillating movements in both axial directions of the orbit. These oscillating movements are approximately harmonic, and thus take the form of a sine curve when plotted against the time axis. They come about as the result of an excitation at rotational frequency arising from unbalance. The elliptical shape is most clearly seen close to resonance, where vibrational displacements are at their greatest and therefore dominate other effects.

The fact that the ellipse does not as a rule take on the special shape of a circle can be explained because in reality the stiffness of a rotating machine is never the same in both the horizontal and vertical direction. In particular there are significant differences at the bearings, where vertical stiffness is usually significantly higher than horizontal stiffness. Thus not only are the vibrational displacements different in each of the axial directions, but so too are the natural frequencies.

This follows from the relationship shown in Appendix B between natural frequency and system stiffness. In subcritical operation the amplitudes are greater in the direction of least stiffness, whereas the reverse is true in supercritical operation. When passing bending resonance therefore there is not only an increase in size due to the high amplitudes close to resonance, but also a change in the shape of the ellipse because the directions of the major and minor axes are changing places.

Also the deviations of the orbit from the ideal elliptical shape can be traced back to technical properties of the system and technical processes. First there is the previously mentioned runout, that is the lack of roundness, or surface roughness, of the shafts or disks, which shows up in the measurement signal when displacement transducers are used. The electrical saturation effects or mechanical obstructions discussed earlier and illustrated in Fig. 6.3 also make themselves felt in the orbit. They then show up as flattened sides.

All these deviations from an elliptical shape for the orbit mean that the oscillating movements in each of the axial directions of the orbit are not purely harmonic and therefore higher harmonic components must be present, in the sense of the Fourier series expansion which is explained in more detail below. This relationship can be used to make predictions about the frequencies of the individual components in the measurement signal, even without explicitly deploying the frequency domain methods described in the following section.

If the orbit is in the form of a loop in which the orbit intersects itself, as shown in Fig. 6.6, this indicates subharmonic components in the shaft oscillations with respect to the rotation speed (see also Appendix B). In this case the shaft makes two or more complete revolutions about its own axis before returning to the same position in its orbit. The illustration shows a computer simulation of the motion of a rotating shaft that rubs asymmetrically against the housing.

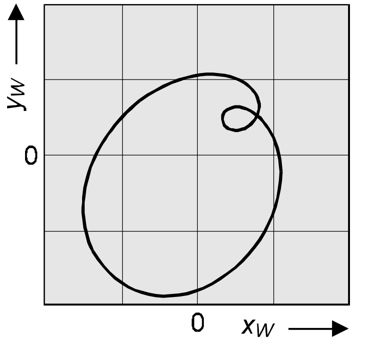


Fig. 6.6 Typical orbit for subharmonic shaft movements

6.3.2 Frequency domain

Basic ideas and concepts

The basic idea of frequency domain methods for signal analysis is to divide any time signal into a sum of harmonic signals, that is, signals that take the shape of a sine curve when plotted over time.

Naturally such an approach is excellent for the analysis of vibrations, for the response of a vibrating system can be most easily understood when the excitation forces or moments are represented as superimposed harmonic functions, as fully explained in Appendix B. The vibrational response of the system then automatically consists of harmonic components.

Frequency range methods are suitable not only for identifying system properties, particularly natural frequencies, but also for describing the excitation mechanisms which act upon the system, such as oscillating torque components. This section is concerned solely with a concise presentation of the most important aspects in connection with vibration measurement on rotating machinery. For detailed information the reader is referred to the literature. A fundamental introduction to signal analysis can be found in [16] for example. A description giving special emphasis to the application of vibration measurement on rotating machinery can be found in Guideline VDI 3839.

Fourier series expansion

It can be proven mathematically that each periodic signal can be expanded into a Fourier series. A Fourier series is the sum of an infinite number of harmonic functions.

The lowest frequency in the series is known as the fundamental frequency. It is given as the reciprocal of the cycle duration of the signal. Other possible individual frequency components are a multiple of the fundamental frequency in each case. To specify the output signal component which has a given frequency, the component is characterized by an amplitude and a phase position. It can be represented either by explicitly stating the amplitude and phase angle, or by dividing it into a sine component and a cosine component. Finally in the general case a Fourier series also contains a constant component, technically designated a component of frequency zero. Thus for each periodic signal $s(t)$ there exists a notation in the form

$$s(t) = \frac{1}{2} c_0 + \sum_{n=1}^{\infty} c_n \cos\left(\frac{n\, f_0}{2\pi} t + \alpha_n \right)$$

where f_0 refers to the fundamental frequency and c_0 , c_1 , ... as well as α_0 , α_1, ... refer to the Fourier coefficients, which are different for each function. The Fourier coefficients of a periodic function can be graphically plotted by frequency, as shown for example in Fig. 6.7. Since the coefficients take non-zero values only at discrete frequencies, this gives what is known as a line spectrum. A complete graphical representation also requires, in addition to the amplitude spectrum shown here, a separate graph for the phase angle known as the phase spectrum. Because it has little practical significance for the application under discussion, this diagram is not shown here.

It is not intended to describe in detail how the coefficients are calculated, since the appropriate software will undertake this task for the practitioner in any case.

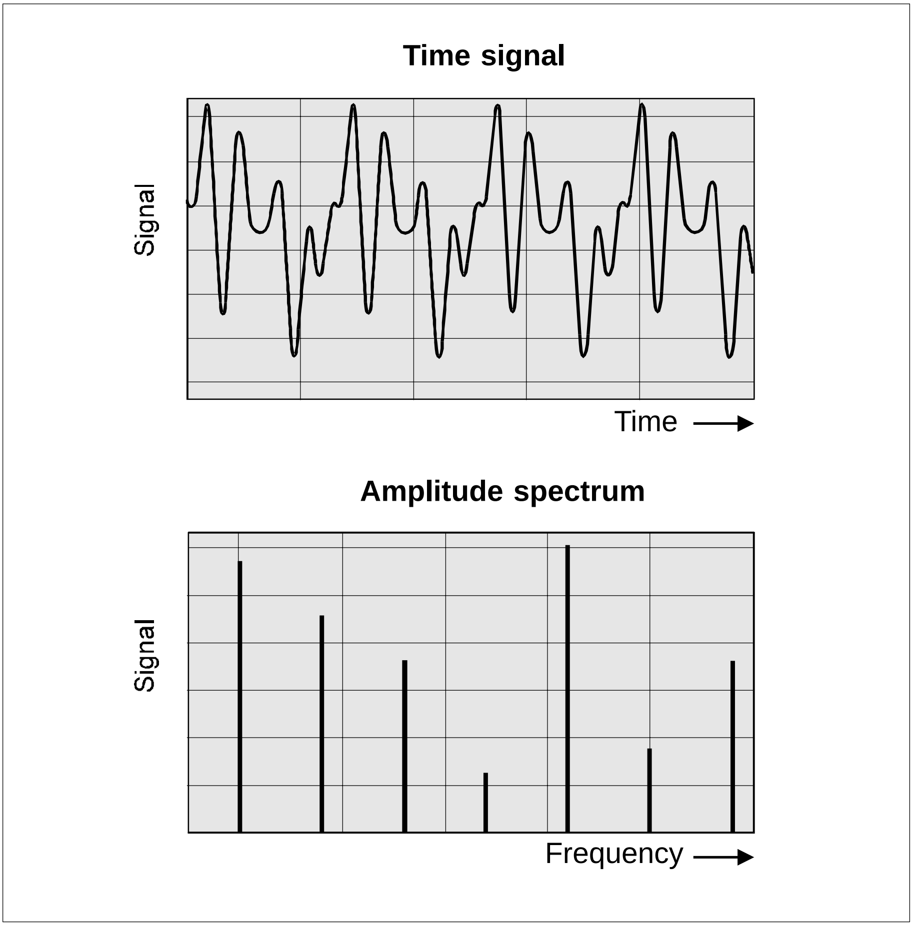


Fig. 6.7 Periodic time function and its associated amplitude spectrum (line spectrum)

Fourier transform

For non-periodic time signals a generalization of the Fourier series gives rise to the Fourier integral. For almost any time signal $s(t)$ a representation can be given in the form

$$s(t) = \int_{-\infty}^{+\infty} \underline{S}(f)\, e^{i2\pi f \cdot t}\, df$$

Here again, a time signal is represented as a superposition of an infinite number of harmonic functions. The frequencies concerned are infinitely close to one another. This gives a continuous frequency spectrum. An example is shown in Fig. 6.8. In order to assign to each frequency the associated amplitude and phase (that is, the counterpart of the Fourier coefficients in the case of periodic time signals), a frequency-dependent function is needed with the Fourier transform. The sum sign is replaced by the integral.

In order to conveniently summarize amplitude and phase information, it is customary to use complex numbers for notation. Since the Fourier transform assigns a frequency function $S(f)$ to a time function $s(t)$, it is also said to transform a time domain function into the frequency domain. It is not intended to describe in detail how the transform is mathematically calculated. The algorithm which is used on computers is known as the Fast Fourier Transform (FFT). Occasionally it can be helpful to use a logarithmic form of notation for the amplitude axis. This allows relative amplitude peaks, which are small in comparison to other peaks, to be clearly detected.

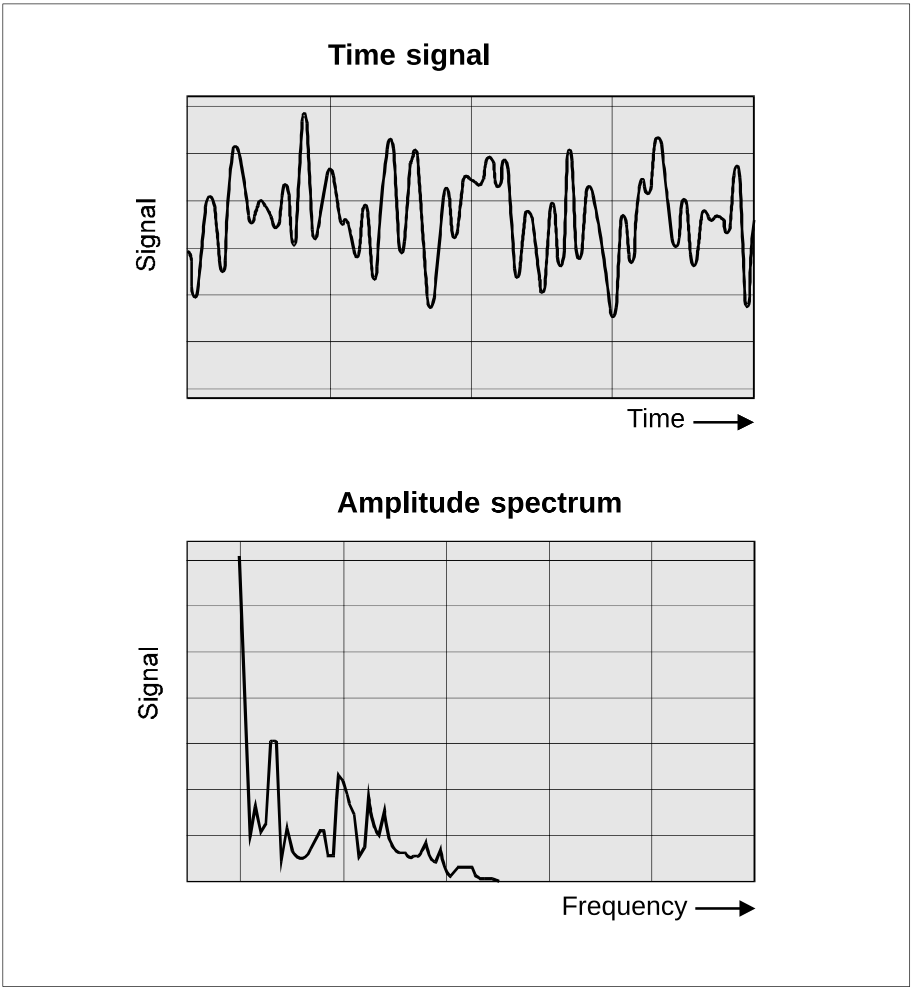


Fig. 6.8 Non-periodic time function and its associated amplitude spectrum (continuous spectrum)

Carrying out practical frequency analysis by computer

When performing practical frequency analysis on a computer, transition to the frequency range is always carried out as a Fourier transform. This course is chosen even when de facto periodic or near periodic time signals are concerned, because it is then possible to get by without tediously incorporating a priori information such as the fundamental frequency. A distinction is made between a real-time FFT (online) and an FFT based on recorded data (offline). The first case requires the use of special equipment (FFT analyzers) due to the considerable computing power involved. In the second case it is possible simply to use a suitable software package on a PC. This can be a mathematical program or a data acquisition and analysis program such as catman® from HBM.

Without going further into the mathematical background, here are some practical tips and hints:

- The transform can only be carried out by computer if the number of data points n_{sample} is a power of two. The usual values are 512, 1024 or 2048.

- The highest frequency f_{max} and the resolution Δf, obtained in the frequency range depend directly on the number of data points n_{sample} and the time span T of the signal excerpt being transformed.

$$\Delta f = \frac{1}{T}, \quad f_{\max} = \frac{1}{2} \Delta f \; n_{sample}$$

Since a computerized Fourier transform, unlike the mathematically exact form, cannot handle infinitely long signal excerpts, truncation errors occur. These can be avoided by using a window function. In this, the time signal is multiplied by a function before the actual transform. This ensures that the time signal at the beginning and end continually falls to the value of zero. Common window functions that every computer program offers in connection with the FFT are the Hanning window or the Hamming window. The term rectangular window is used as a synonym for no window function.

Order analysis

Investigation using frequency domain methods principally makes it possible to draw conclusions in two different areas. On the one hand the natural frequencies of the system can be determined. Ideally a known excitation that includes the whole of the frequency band being studied is specified for this, such as a sinusoidal force or a sinusoidal moment with a slowly increasing frequency or even a pulse (the Fourier transform of the ideal impact includes all frequencies with the same amplitude). The frequencies that then show up as maxima in the system response are the natural frequencies of the system. On the other hand frequency range methods can be used to analyze the frequencies contained in the excitation spectrum in a given application. Here the problem arises that in practice the forces and moments that excite vibrations often cannot be isolated from the forces and moments that are brought about in the vibrating system due to resonance amplification.

Appendix B explains the relationship between natural frequencies and resonances in a more general context. It comes to the conclusion that it is always necessary to reckon with vibration problems if a natural frequency of the system coincides with an excitation frequency. However, in order to be able to trace back to the causes, it is crucial to identify both in isolation from one another. In rotating machinery the technique of order analysis can be used for this purpose. This takes advantage of the fact that the majority of excitation mechanisms for vibration in rotating machinery stand in directly proportional relation to the speed of rotation. A wide range of rotation speeds is traversed and at each speed a frequency analysis of the measurement signal is carried out. These are then compared in a three-dimensional graphical display. Fig. 6.9 shows a representation where an amplitude spectrum for each speed has been plotted and in which, in the same way as in the usual representation in Fig. 6.7 and Fig. 6.8, the frequency is plotted on the *x* axis. This type of representation is known as a waterfall diagram or Campbell diagram.

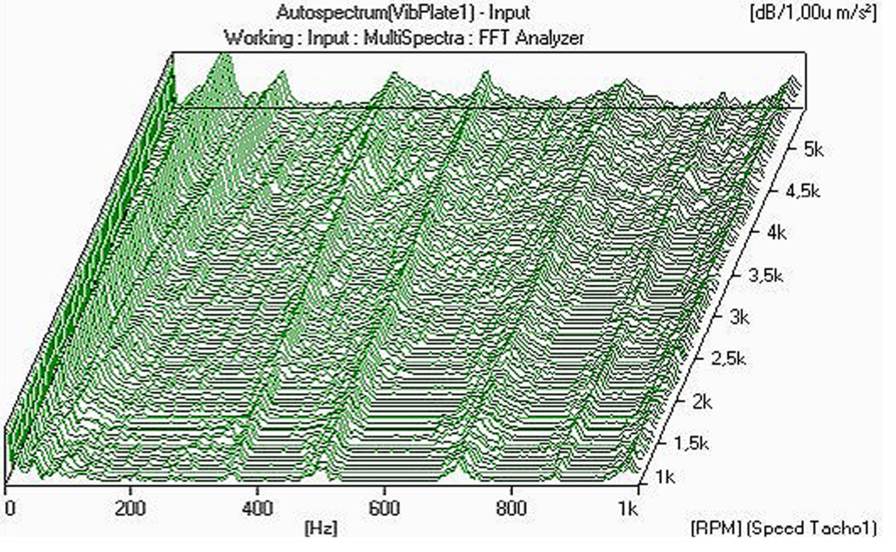


Fig. 6.9 Displaying the amplitude spectrum by frequency
Illustration by kind permission of Brüel & Kjær

In this diagram amplitude maxima which occur at a particular frequency independently of speed, and therefore point to natural frequencies, lie on a vertical line. Amplitude maxima that are proportional to the speed and therefore point to speed-induced excitation frequencies lie on diagonal lines. A series of natural frequencies can be clearly recognized in Fig. 6.9. Furthermore it seems likely that the particularly strong excitation at rotation frequency is caused by an unbalance. Also of note are higher harmonics at around twice the rotation frequency. These suggest excitation mechanisms giving rise to non-harmonic excitations, the Fourier series expansion of which accordingly contains components of a higher order.

An alternative diagrammatic form is shown in Fig. 6.10, which is based on the same data. Here too the frequency is marked off on the *x* axis, but as an order ratio related to the speed in each case. Another difference compared to the diagram in Fig. 6.9 is the fact that the third dimension is not shown in perspective as the *z* axis, but displayed in the form of a color scale.

This diagram shows a vertical line representing an amplitude maximum that occurs at a frequency proportional to the speed, in this example approximating to the maxima at ordinal numbers 1 and 2. An amplitude maximum occurring at a constant frequency can be seen as a curved hyperbolic plot.

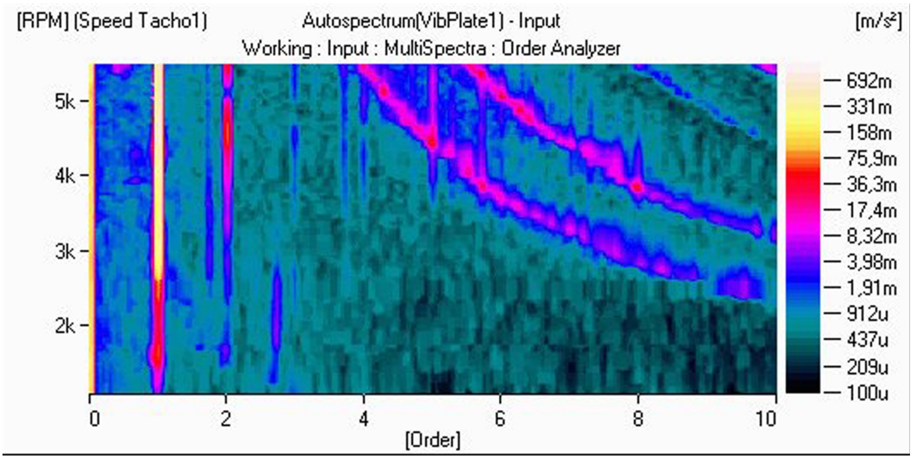


Fig. 6.10 Displaying the amplitude spectrum by order ratiollustration by kind permission of Brüel & Kjær

On the basis of the information obtained in this way concerning the order ratios of the dominant excitation frequencies, it is then possible to reach conclusions about the causative mechanisms that are at work. Typical instances of phenomena relating to torsional vibrations are the oscillating torque components associated with internal combustion engines that were discussed at length in chapter 4. In the case of bending vibrations the dominant factor is mainly unbalance, which always causes excitation that is exclusively synchronous with rotation, though here too, due to their kinematics, crankshaft drives can generate nonharmonic force trends for which the Fourier series expansion accordingly contains components of a higher order. If amplitude maxima occur at a constant frequency throughout a wide speed range, this points to mechanisms which generate dynamic instability or self-excited oscillations. Comparison should be made with the notes in Appendix B. The assignment of typical order ratios to relevant excitation types and vibration phenomena can be found in the diagnostic table in the following section.

6.4 Diagnostic table for vibration problems

Vibration type	Possible excitation causes	Characteristic properties	Transducers and methods for diagnosis	Tips for reducing or preventing vibration
Bending vibration	Static or dynamic unbalance	• Deformation is rotating with the shaft, sinusoidal measurement signal in each axis direction. Circular or elliptical orbit • The frequency equals the rotation frequency • In the case of anisotropic bearings there are two critical speeds • In the case of damping, these ranges coalesce at around the critical speeds into a broad, shared resonance range	• Accelerometers or speed transducers at the bearing housing in two axis directions perpendicular to the axis of rotation • Displacement transducers in two axis directions at various points on the shaft train between the bearings	• Balance the components individually • Balance the operational shaft train in its entirety • Shift the critical bending speeds, for instance by changing the bearing configuration, replacing flexurally elastic couplings, changing the mass distribution (see chapter 4) • Increase the flexural damping, for instance by fluid film bearings (caution: in the case of machines operated subcritically do not incorporate rotating damping, instability possible)

Table 6.1 Shaft train vibration problems and their analysis, part 1

Vibration type	Possible excitation causes	Characteristic properties	Transducers and methods for diagnosis	Tips for reducing or preventing vibration
Bending vibration	Geometry error in couplings due to plane parallelism error or centering error when connections too stiff	• As for unbalance vibrations, deformation is rotating with the shaft • If the measuring point is close to the geometry error, displacements are measurable even during very slow rotation (which is different from the case of unbalance)	• As for unbalance vibrations, but displacement measurement preferable in order to obtain measurement signals even at very low speeds if necessary • When there are strong suspicions, additional static test measurements, for instance using a dial gage	• Use elastic couplings or improve the plane parallelism and concentric properties of the shaft train • Shift the critical bending speeds (see bending vibration / unbalance) • Increase the flexural damping (see bending vibration / unbalance)
	Magnetic excitations in the case of asymmetrically or eccentrically structured rotors in electric motors or generators such as electric absorption dynamometers	• As for unbalance vibrations, deformation is rotating with the shaft, but cannot be reduced by balancing	• As for unbalance vibrations	• Replace the rotor of the electric motor or generator • Shift the critical bending speeds (see bending vibration / unbalance) • Increase the flexural damping (see bending vibration / unbalance)

Table 6.1 Shaft train vibration problems and their analysis, part 2

Vibration type	Possible excitation causes	Characteristic properties	Transducers and methods for diagnosis	Tips for reducing or preventing vibration
Bending vibration	Distorted shaft, e.g. due to thermal warpage during startup of power test benches for internal combustion engines	• As for unbalance vibrations, deformation is rotating with the shaft • Displacements measurable even during very slow rotation (unless distortion occurs only when operating)	• As for unbalance vibrations, but displacement measurement preferable in order to obtain measurement signals even at very low speeds if necessary • Check on thermal effect by checking the dependency of the measurement results on the length of operating time (cold vs. warm machine)	• Equalize radially asymmetric temperature loading of the shaft train • Replace permanently warped shaft section • Shift the critical bending speeds (see bending vibration / unbalance) • Increase the flexural damping (see bending vibration / unbalance)
	Dynamic instability, self-excited vibration, for instance stemming from internal damping or due to whirling of fluid film bearings (see Appendix A)	• Harmonic, resonant vibrations • Occur above a minimum speed which is usually in the vicinity of a critical speed of a natural bending frequency • No clearly defined phase position between the vibrations in the two axis directions • The frequency of the vibrations is always a natural frequency of the system and is therefore not proportional to the speed	• As for unbalance vibrations	• Minimize internal damping in the shaft train. Typical sources of internal damping are connecting elements with internal friction such as form locking couplings or inadequately degreased clamp fixtures • Increase the flexural damping (external damping), for instance by fluid film bearings

Table 6.1 Shaft train vibration problems and their analysis, part 3

Vibration type	Possible excitation causes	Characteristic properties	Transducers and methods for diagnosis	Tips for reducing or preventing vibration
Bending vibration	Anisotropic shaft-train	• Sinusoidal vibration at double the rotation frequency, that is resonance occurs when the speed is equal to half the natural bending frequency	• As for unbalance vibrations	• Correct the geometry with respect to the stiffness anisotropy (then rebalance!)
Torsional vibration	Oscillating torque due to kinematic and dynamic excitations, such as in the case of internal combustion engines, piston pumps, toothed gears	• Frequencies are proportional to speed. The order ratios of the fundamental frequencies relative to the speed can differ depending on the machine type (see chapter 4) • Often very many different higher harmonic components • Highest amplitudes at speeds where one of the excitation frequencies coincides with a torsional natural frequency (see chapter 4)	• Unfiltered output signal from the torque transducer • Analysis of angle of rotation signals or speed signals, when speed is not constant suitable methods could be differentiation or determining the phase differences of two measurement points at both ends of the shaft train • Comparison with bending or bearing vibrations of the same frequency	• Shift the torsionally critical speeds, for instance with the aid of torsionally elastic, damped couplings in the shaft train (see chapter 5) • Increase the torsional damping, for instance by installing torsionally elastic, damped couplings (see chapter 5), damped bearings for the driving and retarding machinery • If possible reduce excitation by modifying the machine from which it stems

Table 6.1 Shaft train vibration problems and their analysis, part 4

Vibration type	Possible excitation causes	Characteristic properties	Transducers and methods for diagnosis	Tips for reducing or preventing vibration
Torsional vibration	Accelerations due to startup and braking processes	• Leads to the appearance of vibration only when acceleration or retardation is extremely jerky • Individual torque peaks in which the point in time at which they occur is closely related with the command pulse from the drive or braking device • Depending on the damping in the shaft train, these peaks are followed by post-pulse oscillations	• Unfiltered output signal from the torque transducer • Analysis of angle of rotation signals or speed signals, when speed is not constant suitable methods could be differentiation or determining the phase differences of two measurement points at both ends of the shaft train	• Shift the torsionally critical speeds (see torsional vibration / oscillating torque) • Increase the torsional damping (see torsional vibration / oscillating torque) • Avoid jerky acceleration, for instance by changing the control concept or using an alternative electric drive or retarding unit
	Switching events	• Individual torque peaks in which the point in time at which they occur is closely related with the switching event • Depending on the damping in the shaft train, these peaks are followed by post-pulse oscillations	• As for torsional vibration due to startup and braking processes	• Shift the torsionally critical speeds (see torsional vibration / oscillating torque) • Increase the torsional damping (see torsional vibration / oscillating torque) • Avoid switching events or reduce their negative effects, for instance by changing the control concept or using an alternative electric drive or retarding unit

Table 6.1 Shaft train vibration problems and their analysis, part 5

Vibration type	Possible excitation causes	Characteristic properties	Transducers and methods for diagnosis	Tips for reducing or preventing vibration
Torsional vibration	Non-uniformity in universal joint shafts, (see chapter 5)	• Sinusoidal oscillating torque • The frequency corresponds to twice the rotation frequency	• Unfiltered output signal from the torque transducer	• Depending on whether the displacement of the joint shaft is static or dynamic • Improve the alignment of the shaft train • Always install universal joint shafts in a Z configuration so that the deflection angles at both ends compensate one another • Replace universal joint shafts with constant velocity joint shafts
	Apparent oscillating torque due to cross-talk of bending deformations from static misalignment, (see chapter 4)	• Since the deformation here is constant in the spatially fixed system the torque transducer follows a rolling action with the rotation frequency (see chapter 4) • The frequency equals the rotation frequency • The amplitude is practically independent of the speed	• Unfiltered output signal from the torque transducer • Static measurement of the misalignment, for instance with the aid of dial gages or laser alignment devices (see chapter 5)	• Improve the alignment of the shaft train • Provide compensating elements, equalize the alignment tolerances (flexurally elastic couplings, or joint shafts for greater alignment tolerances)

Table 6.1 Shaft train vibration problems and their analysis, part 6

Vibration type	Possible excitation causes	Characteristic properties	Transducers and methods for diagnosis	Tips for reducing or preventing vibration
Torsional vibration	Apparent oscillating torque due to cross-talk of oscillating bending deformation (see chapter 4)	• Bending vibration of the type where the distorted shaft rotates in a circular orbit generates no oscillating cross-talk component • Directional bending vibration or vibration rotating in strongly elliptical orbits generate an oscillating cross-talk component • The frequency equals the frequency of the bending vibration concerned	• Unfiltered output signal from the torque transducer • Simultaneous direct measurement of the bending vibration (see above)	• Take measures to correct the bending vibration concerned (see above)
Other forms of vibration	Mechanical loosening of items such as • Bearing covers • Flange connections in the shaft train • Form locking shaft connections	• Sinusoidal vibration with truncated peaks or very jagged curve shapes • The fundamental frequency equals the rotation frequency • Directional (non-rotating) motion • Large numbers of superimposed even-numbered and odd-numbered high-frequency harmonics • Superimposed harmonics also on the torque signal	• Accelerometers or speed transducers at the bearing housing in two axis directions perpendicular to the axis of rotation • Displacement transducers in two axis directions in the vicinity of the suspected cause • Monitor the dynamic torque signal, especially in the case of loose flange or shaft connections • Conduct acoustic test using measurement equipment or by listening	• Check the bolt connections of • Bearing covers • Housing fastenings for engines, motors and bearings • Shaft flanges • Check the form locking shaft-to-hub connections (for instance broken or missing involute gear teeth)

Table 6.1 Shaft train vibration problems and their analysis, part 7

Vibration type	Possible excitation causes	Characteristic properties	Transducers and methods for diagnosis	Tips for reducing or preventing vibration
Other forms of vibration	Defective roller bearings • Damage to the ball race on the inner and outer rings • Damage to the bearing balls or rolling elements • Damage to the cage • Bearing play	• Normally low-amplitude sinusoidal vibration • Frequencies in the event of damage to inner or outer bearing shells approx. 40 % to 60 % of the product Number of Bearing Balls * Rotation Frequency • Fundamental vibration frequency in the event of massive damage to the cage often less than the rotation frequency • Partially stochastic vibrations	Speed transducers and accelerometers at the bearing housings	Continuous monitoring or regular investigation of the bearings by means of frequency analysis of the vibrations concerned

Table 6.1 Shaft train vibration problems and their analysis, part 8

6.5 Assessing vibration

Considering what really counts at the end, it is always the responsibility of the manufacturer or user of the complete system to judge what level of vibration can be tolerated for the operation of machinery such as power test benches. The specifications for HBM torque transducers contain several items of information relating to robustness against vibration.

The specified vibration resistance documents the criteria used for testing the rotors and stators of torque transducers. Testing is carried out in accordance with DIN IEC 68, Part 2-6, which refers to the resistance of electronic devices relative to such environmental conditions.

In the case of HBM torque transducers, permissible vibration with regard to the smooth running of rotating machinery is specified by maximum vibration amplitude s_{max} (see chapter 5 and the specifications of the transducer concerned). The dependency of the maximum vibration amplitude on speed corresponds to DIN 45670/VDI 2059, in which the maximum vibration amplitudes for turbine engines are defined.

A number of factors can make it necessary to deviate from the specifications set out in standards. Particularly in connection with the metrological task to hand, other requirements can arise which may necessitate setting lower or higher limits for tolerable vibration.

Finally, the specified vibration bandwidth in accordance with DIN 50 100 refers to the continuous vibrating mechanical load of the measuring body in the event of oscillating torque (for further explanations on the subject see Appendix A).

7 Calibrating torque transducers

7.1 Defining the concept

7.1.1 Calibration

The purpose of calibration

Calibration determines under predefined conditions the relationship between the measured value of the output quantity and the input quantity which is the measured quantity, in this case torque. This means that a comparison is made with a reference in the same unit of measurement. It also means that torque calibration must only be carried out using a traceable reference torque. Proof of traceability for the measured quantity of force alone is not enough, since the method by which the force is converted into torque by means of a lever arm would then be undocumented.

The predefined conditions include not only those of the environmental type, such as temperature and relative humidity, but also the mounting conditions and loading sequence within which the transducer is put to use. The torque needed for the input quantity must be mechanically generated and also quantified. The required level of accuracy is always higher than that needed for the object of the calibration. Thus calibration establishes a relationship between the input quantity (i.e. the actual torque in the context of the accuracy of the calibration procedure) and the output quantity. The object of the calibration may be a torque transducer or even a measurement chain that includes not only a transducer but also amplifier electronics and a display unit. Different output quantities are involved, depending on the type of transducer and the settings in the measuring amplifier:

- Voltage ratio (mV/V) in passive SG transducers (torque transducers which transmit their signals by means of slip rings)
- Frequency signal (Hz) in many torque transducers using contactless signal transmission
- Voltage (V) in torque transducers with integrated measuring amplifiers
- Digital output signals
- In adjusted measurement chains: display in the physical torque unit (N·m); to distinguish the displayed values from the input quantity the calibration certificate refers to them as displayed units

Validity of the calibration

Strictly speaking a calibration is valid only for applications in which the torque transducer is used in accordance with the calibration conditions. As far as official DKD calibration is concerned, this point is clearly established in the appropriate standards DIN 51309 and EA-10/14 [17].

This also means that the same type of measuring amplifier and cable must be used for the calibration and for running the application. The ideal situation for a satisfactory transfer to the application is the calibration of a measurement chain, when the transducer is calibrated with the amplifier with which it will be used in the application. If this is not possible the instruments should as far as possible be the same kind, and preferably the same type with the same technical properties. For torque transducers outputting, say, voltage ratio as the output quantity therefore, the upshot is that the calibration is not fully valid if for example a transducer that was calibrated using a DC supply voltage is then operated on a carrier-frequency supply. The same applies to changing between different carrier frequencies.

Limiting the validity of the calibration to the same on-site conditions also has far-reaching consequences regarding the mechanical mounting conditions. There are different ways of making the transfer to the conditions on site in a subsequent application.

On the one hand it is possible to bring the conditions on site into line with the calibration conditions. This can be done by emulating the environmental conditions and adaptation accessories of the calibration laboratory as accurately as possible and minimizing the parasitic loads. On the other hand the on-site influences to which the transducer is exposed can be transferred into the calibration situation. This is done by including the adaptation accessories directly adjacent to the torque transducer in the calibration. A third possibility is to use the on-site calibration method. That way, the transducer is in exactly the same mounting conditions during calibration as it is in the course of normal daily operation. This last-named method is discussed in section 7.5. There is also some merit in combining all three methods. In any case, whenever the measurement uncertainty in a given application is to be determined these influences should be taken into consideration.

Regarding how much time to allow before recalibration, it must be kept in mind that the purpose of the calibration is limited to documenting the properties of the calibrated object at the actual time of calibration. Accordingly it can be no part of the calibration to make pronouncements about properties the object may possess in the future. No period of validity can therefore be inferred. It is always the responsibility of the user to decide on the next calibration date, since it depends on the amount of use, the importance attached to the measurement medium, the effects of errors and other similar factors.

The interval between calibrations should be part of the quality manual. A useful hint is the fact that the national standard on which DKD calibration is based, namely DIN 51309, prescribes a maximum recalibration interval of 24 months (plus two months for carrying out the calibration). As a manufacturer, HBM recommends that electronic instruments should be recalibrated about once a year and transducers every two years at least. In any case recalibration is necessary whenever a torque transducer has been subjected to a loading beyond permitted values, or after maintenance or after inappropriate handling that can have an effect on the measurement properties.

7.1.2 Terms that are often confused with calibration

Testing or inspection

Testing means establishing to what extent a requirement is fulfilled. Usually this involves comparing whether the limit values are exceeded and an assessment is then made on the basis of this comparison.

Adjustment

Adjustment means setting up or balancing a measuring instrument in order to remedy known systematic errors for the planned application. In contrast to calibration, this always involves making an intervention that brings about a lasting change. It should be stressed that adjusting a measuring instrument destroys its calibration history. In order to obtain reliable documentation in terms of the calibration of newly set up measurement properties, recalibration is necessary when an adjustment has been completed.

Examples of adjustments to torque transducers include setting the sensitivity to a value close to the nominal sensitivity and minimizing the effect of temperature on the zero signal. Both are part of the normal process of manufacturing torque transducers at HBM. For this purpose balancing resistors are soldered in position, or compensating strips are modified in or separated from the bridge circuit.

Testing, adjustment and calibration are often closely interconnected, for instance if an unacceptable sensitivity error is detected in a transducer during testing, this is immediately remedied on site by means of compensating measures (that is, an adjustment is made) and the measurement properties of the transducer that have been set up in this way are then measured and recorded (that is, calibration is carried out).

Legal verification

Legal verification covers quality testing and marking in accordance with the verification regulations and is the reserved domain of legal metrology (verification office, verification official). It is prescribed for such applications concerned with establishing a price (letter scales at the post office, gasoline pumps, etc.).

Adjusting the measurement chain

Adjusting the measurement chain means using the known measurement properties of the transducer (typically ascertained from calibration) to establish correct settings in the measuring amplifier, enabling it to produce a display from which the input quantity (in this case torque) can be easily inferred. This procedure is described in detail in chapter 5.

Shunt calibration

In shunt calibration the calibration signal is activated in the transducer and then measured and stored in the measuring amplifier. A known input quantity value (typically ascertained from calibration) is then assigned to this signal. This is therefore a special method of adjusting the measurement chain (see also chapter 5). The particular significance of the calibration signal lies in the fact that it can also be used to continuously inspect the measurement chain.

7.2 Calibration machine designs for torque transducers

By and large, equipment and machines for calibrating torque transducers can be subdivided according to the reference standard used.

In lever-arm-mass systems a precisely defined torque is generated when the weight force of calibrated or legally verified masses acts on the test specimen by means of a lever arm of known length. This type of equipment is known as a dead weight calibration machine. A characteristic in this case is that the reference property comes about because the torque is generated at a precisely quantified value. Calibration equipment that works according to the principle of the lever-arm-mass system is the technology currently used in all calibration laboratories for the high and highest grades of accuracy. Fig. 7.1 shows a calibration machine for torque up to 20 kN·m which is one of the machines that HBM uses for DKD calibration.

Fig. 7.1 Calibration machine based on the lever-arm-mass principle

The second principle is to use a torque reference transducer to provide the reference torque. In principle, systems with torque reference transducers can use any mechanism to generate the torque. The generated torque is then measured with the aid of a torque transfer transducer or reference transducer. Features of both principles can be combined in a force reference transducer with a lever arm.

The high degree of precision nowadays demanded of torque transducers, combined with the necessity for the reference to be always more accurate than the desired smallest measurement uncertainty that can be determined for the objects being calibrated, makes great demands on the calibration equipment. A few selected aspects will therefore be briefly mentioned at this point.

With any design of calibration equipment it is particularly important to ensure that the load acting on the torque transducer being calibrated is actually pure torque. But since torque generation very often involves significant lateral forces, it is often necessary to use appropriate bearings to ensure that lateral forces, axial forces and bending moments are supported without acting on the transducer.

A bearing between the reference torque and the torque transducer can falsify the measurement results due to the friction it causes. Every effort is therefore made to ensure that the bearings in calibration equipment are as free of friction as possible. This even goes as far as using air bearings despite their extremely high capital and running costs.

In lever-arm-mass systems there are further difficulties specifically connected with the design of the machine itself. Firstly, weight loading depends on the earth's gravitational field. But since this exhibits regional differences, the local acceleration due to gravity must be measured in calibration laboratories. Secondly the length of the lever arm must also be accurately quantified, which means that highly specialized designs are required for the application of force to the lever arm. But the effective length of the lever arm also depends on whether it is aligned precisely horizontally. However, since the deformation of a torque transducer under load allows a certain amount of torsion to occur, it is impossible to ensure that the lever arm is horizontal by hardware design of the calibration machine. An electronic closed loop control is therefore needed.

7.3 Orders of rank in calibration – the calibration pyramid

Every company that is or wishes to be certified to ISO 9000 must prove that its measurements can be traced back to national standards. This requirement extends to all measurement and test equipment used for quality assurance of the company's own products. The aim of the traceability requirement is to ensure "correct" measurement.

The background to this requirement is that even when the accuracy of the test equipment is satisfactory, initially it is unknown whether a torque display of, say, 128.6 N·m is actually correct or whether in fact it should be 133.1 N·m. The only way to be clear about this is to make a comparison with a reference – in other words, calibration.

Proof of traceability is provided when the test equipment or measuring device is calibrated against a reference that itself has been calibrated in unbroken succession back to the national standard. Depending on the level in this traceability chain, the reference is called the national standard (reserved for national metrological institutes, which in Germany is the PTB), a reference standard (reserved for accredited calibration laboratories, in Germany those in the DKD) or a working standard. The orders of rank are shown in the form of a diagram in Fig. 7.2. Depending on the order of rank of each standard, different demands are imposed on calibration for the purpose of passing calibration on to the next standard in the chain (measured quantity transfer). This is why there is an order of rank in calibration.

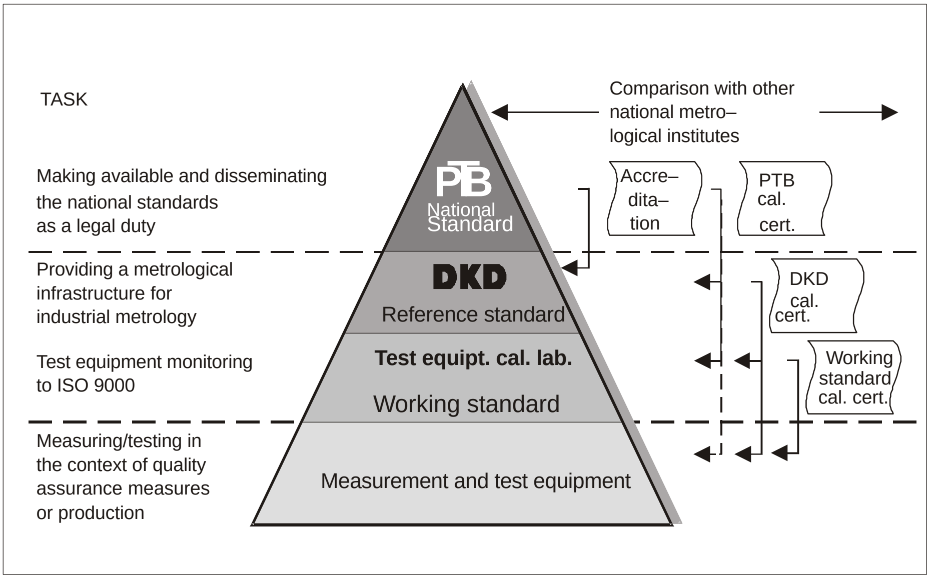


Fig. 7.2 Orders of rank in measurement media and calibration

In addition to the formal order of rank it should be noted that there is no point in disseminating the calibration unless the accuracy of the measuring equipment acting as a reference is higher than the accuracy that needs to be demonstrated in the equipment being calibrated. In practice this has resulted in a factor of 2 to 5 as the minimum difference between each level of accuracy.

Calibration by the PTB

The highest rank of calibration is occupied by the national metrological institute, which has the legal duty to hold the national standard in a state of readiness and to disseminate it in the context of calibration. In Germany this body is known as the PTB (German Metrology Institute).

Calibration within the DKD

The next rank of calibration is that provided within the DKD (German Calibration Service). The DKD is an association of calibration laboratories that are accredited by the PTB and therefore entitled to carry out DKD calibrations. This means in particular that such high ranking calibration demands more than merely that the reference standard has been calibrated by the PTB. On the contrary, all of the calibration procedures and evaluation algorithms are bound by mandatory national and international standards. At the present time the standards relating to the measured quantity torque are European Guideline EA-10/14 [17], German standard DIN 51309 and the amending DKD recommendation DKD-R 3-5 [18]. Correct implementation of the above regulations is overseen by the PTB in the context of accreditation and continuous monitoring. Accreditation covers not only qualifying the reference standard and other equipment but also the qualification of personnel responsible for calibration. From accreditation it directly follows that DKD calibration certificates are unquestionable proof of traceability.

Working standard calibration

Compared with DKD calibrations, working standard calibrations are usually carried out more simply and are therefore less costly to perform. They can be carried out not only by DKD accredited laboratories, but also by suppliers without accreditation or even by the user concerned. The responsibility for determining the procedures and evaluation algorithms of the process rests with the laboratory performing the calibration.

The traceability of a calibrated object can easily be proved if the working standard calibration took place in a calibration laboratory with DKD accreditation. Otherwise proof of traceability requires additional information about the calibration of the reference standard used. HBM refers to standard DIN/ISO 10012 when documenting the traceability of its working standard calibrations.

Factory testing in the case of a manufacturer of measuring equipment

In a general sense factory testing simply involves the use of measuring equipment. However, if the product being tested is itself an item of measuring equipment, its metrological properties will naturally be tested. If the test is documented in a manufacturer's test certificate, this often includes conclusions which amount to a calibration, provided the test result is not confined merely to go/no-go opinions but also includes quantified documentation of the measurement results.

The important difference with regard to calibration in the true sense is that factory testing is part of the production process, whereas calibration presupposes a completely finished product. Also in a more general sense, the information given in a test certificate is placed at the end of the order of rank for calibration.

7.4 Test certificates and calibration certificates from the various orders of rank

7.4.1 Testing in production and the manufacturer's test certificate

At HBM the contents of a test certificate are equivalent to an acceptance test certificate in accordance with EN 10204. The certificate provides documentation of a test for fulfillment of specific technical specifications. The data to be tested and the sequence of the test are laid down and it is the manufacturer who determines these matters for each product. The certificate goes beyond the task of purely testing in accordance with the above definition, to the extent that it documents not only the fact that the limit values for certain technical properties are respected, but also the actual values ascertained for the characteristic quantities concerned. Provided these characteristic quantities describe metrological properties, the test certificate can be regarded as a calibration certificate. The information on the test certificate for a torque transducer from HBM is enough to form the basis for measurement chain adjustment, but does not include any conclusions about the measurement uncertainty.

Test sequence

The test sequence for drawing up a test certificate on HBM torque transducers includes the following steps:

- Preloading to nominal torque three times in order to deal with settlement effects
- Measuring the output signal at the load steps specified for the respective transducer with increasing torque (ascending series of tests)
- Measuring the output signal at the load steps specified for the respective transducer with decreasing torque (descending series of tests)

This sequence is carried out separately for clockwise and counterclockwise torque. It is illustrated in Fig. 7.3 for a torque transducer for which the number of specified load steps is two. In certain types with especially high levels of accuracy a larger number of load steps will be tested.

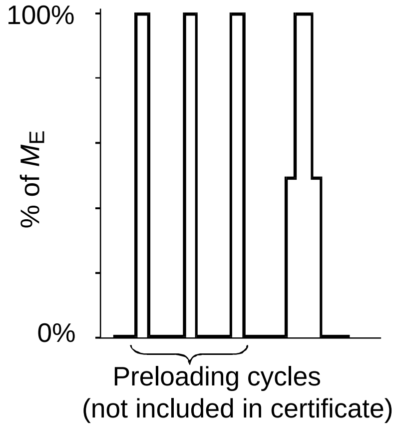


Fig. 7.3 Test sequence for drawing up a test certificate

Contents of the certificate

To go into detail, the test certificate for HBM torque transducers shows the following characteristic quantities separately for clockwise and counterclockwise torque:

- The measured output quantity delivered by the torque transducer at the load steps laid down in the test sequence
- Sensitivity
- Linearity deviation
- Linearity deviation including hysteresis
- Relative reversibility error

Precise definitions of these quantities can be found in Appendix A. The three preloading cycles are ignored in making the evaluation.

The measured values are quoted in zero-compensated form. The signal span is therefore specified relative to the initial torque signal, that is, the output signal after the last preloading and before the start of the actual measurement cycle (see Appendix A).

In addition the value of the calibration signal for shunt calibration is also individually determined and recorded in the test certificate. No target value is therefore specified for this. Fig. 7.4 shows an example of a test certificate on a T10F torque flange.

Nr: 0313.002

HBM

V1.6/IB-BM/9.00

Prüfprotokoll

test certificate / protocole d`essai

Typ: type / type	T10F	Auftrag: order no / commission	801008068
Option1, Code: option1, code / option1,code	--	Prüfer: examiner / contrôleur	Breitwieser
Nenndrehmoment: nominaltorque / couple nominal	1kNm	Datum: test date / date d`essai	3.8.2001
IdentNr: serial no / N°-d`ident	XXXXXXX		

Prüfergebnisse:
test results / résultats d`essai

Eingangsgröße des Meßbereichs [%] input quantity / échelle d`essai	Ausgangsgröße [Hz] output quantity / résultats Links anticlockwise / sens anti-horaire	 Rechts clockwise / sens horaire
0	0.0	0.0
50	-2498.7	2499.4
100	-4997.6	4998.2
50	-2498.7	2499.1
0	0.1	0.0

Prüfergebnisse und Kennwert C sind Abweichungen von der Ausgangsfrequenz des Aufnehmers ohne Drehmoment. Sie beträgt etwa 10kHz.
test results and characteristic value C are deviations from the frequency output without torque applied frequency beeing approx 10kHz.
les résultats d`essais et la sensibilité C sont mesurés par rapport à la fréquence du capteur au zéro sans charge (approx. 10kHz).

Aus den Prüfergebnissen berechnete und sonstige meßtechnische Eigenschaften :
metrological characteristics calculated from the measuring results and others
valeurs caractéristiques calculées à partir des résultats d`essai

Kennwert C [Hz] sensitivity / sensibilité	-4997.6	4998.2
Linearitätsabweichung [%vC] linearity deviation / linearité (Abweichung von der bestpassenden Geraden durch das Nullsignal) (deviation from bestfit through zero / écart par rapport à la meilleure droite passant par le zéro)	0.001	0.004
Linearitätsabweichung einschließlich Hysterese [%vC] linearity deviation incl. hysteresis / écart de linearité y compris l'hystérésis	0.002	0.004
Relative Umkehrspanne [%vC] relative hysteresis/hystérésis relatif	-0.002	-0.006
Kalibriersignal [Hz] calibration signal / signal de calibration	2855	
Kalibriersignal [Nm] calibration signal / signal de calibration	571.0	

Allgemeine Zusatzinformationen:
general information / informations complémentaires

Alle weiteren meßtechnischen Eigenschaften des Aufnehmers sind durch Typprüfungen und laufende Produktaudits des Qualitätswesens abgesichert.
All other metrological characteristics of the transducer are verified by type testing and regular product audits of the quality department.
Toutes les autres caractéristiques techniques du capteur sont garanties par le Service Qualité, au moyen d`essais et d`audits suivis sur le produit.

Zertifiziert nach ISO 9001 und ISO14001 (DQS-00001)
ISO 9001 and ISO 14001 certified / Certification selon ISO 9001 et ISO 14001

Akkreditiertes DKD Kalibrierlaboratorium und EMV-Prüflaboratorium
Accredited DKD calibration laboratory and EMC testing laboratory
Laboratoire accrédité par le DKD et laboratoire d`essais CEM
DKD-K-00101; DAT-P-006/012

Hottinger Baldwin Messtechnik GmbH   Im Tiefen See 45   D-64293 Darmstadt

Fig. 7.4 Test certificate for a T10F torque flange

Use for adjusting the measurement chain

If it is intended to rely on the information in the test certificate for adjusting the measurement chain, the two-point adjustment method with the zero signal or initial torque signal acquired by measurement is recommended. In this case the second point is the nominal torque of the torque transducer, using the sensitivity specified in the test certificate as the assigned output signal span. Depending on the application it is necessary to decide whether to use the sensitivity for clockwise or counterclockwise torque. In principle the second point can also be determined with the aid of the shunt calibration signal, but this method is less accurate since in this case the tolerance of the calibration signal itself and the sensitivity tolerance occur as additional uncertainties.

The details of the procedure for measurement chain adjustment are described in chapter 5.

7.4.2 Working standard calibration

A working standard calibration certificate establishes a relationship between the input quantity torque and the output quantity of the torque transducer. No evaluation is carried out for determination of metrological properties such as linearity or hysteresis.

A working standard calibration certificate is the right choice when the qualified statement about the measurement uncertainty which a DKD calibration certificate provides is not necessary, but information is required which goes beyond that given in a test certificate, such as when a larger number of torque steps than is usually found in a test certificate is desired. It is also possible to calibrate in sub-ranges or to calibrate the measurement chain using the measuring amplifier with which the transducer will be operated in the application.

Calibration sequence

At HBM the factory calibration certificate is made out in accordance with DIN ISO 10012 Part 1. HBM offers different variants of the factory calibration certificate which differ in the number of load steps. As in the sequence for the test certificate, a series of load steps is applied in ascending and descending order. Calibration is carried out for clockwise or counterclockwise torque according to the wishes of the customer. Calibration for clockwise and counterclockwise torque consists of two virtually independent calibrations carried out one after the other including the preloading.

As a different option it is also possible to calibrate for alternating torque which changes between clockwise and counterclockwise in a single sequence. The measuring range for which the calibration is carried out can be specified by the customer and may differ from the nominal measuring range of the transducer so that it is also possible to calibrate for measurements in the partial load range.

The measurement sequence includes the following steps:

- Preloading to nominal torque three times
- Measuring the output signal at the load steps specified for the respective calibration with increasing torque (ascending series of tests)
- Measuring the output signal at the load steps specified for the respective calibration with decreasing torque (descending series of tests)

The sequence for a factory calibration with 6 steps is shown as a diagram in Fig. 7.5. The steps are 20 %, 40 %, 50 %, 60 %, 80 %, 100 % of full scale in the measuring range being calibrated.

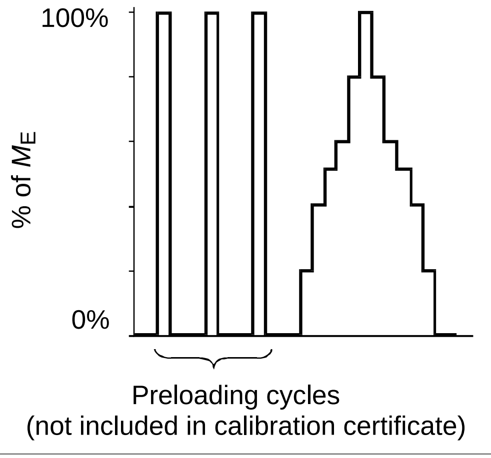


Fig. 7.5 Calibration sequence for working standard calibration

Choice of physical unit of displayed values

When calibrating a torque transducer as such, the physical unit of measure for the displayed values is the unit of electrical output of the transducer, which in most cases is Hz or mV/V.

When calibrating a measurement chain which means that not only the torque transducer but also the associated amplifier electronics are included as the object of calibration, the customer can choose whether the unit of display should be the unit of electrical output of the transducer or the unit N·m for Newton meter. Since the calibration certificate contains comparisons between torque in N·m and the output from the measurement chain in the unit of display, the latter case could cause confusion or lack of clarity. Therefore such tables use the neutral term unit of display (for which the German abbreviation is AE). The choice of the unit of display has an effect on information in the analysis results.

The main problem in choosing N·m for the unit of display is that not only must the measurement chain be adapted to it, but also this adaptation must be made before starting the actual calibration. On the other hand, the adaptation requires that the sensitivity be known. This is determined from the last preloading. The main disadvantage of choosing this unit stems from this method: the sensitivity specified in the calibration certificate for adapting the measurement chain is not optimized on the basis of all measured values, but based on a single measurement. In contrast, after calibration using the unit of output of the transducer the calibration result is available as the optimum data for adapting the measurement chain. However, the adaptation depends on the use (full load range or partial load range). Therefore no generally valid choice of sensitivity can be specified and it has to be worked out by giving accurate consideration to a wealth of information. Using the working standard calibration certificate when adapting the measurement chain will be discussed in further detail below.

Information in the working standard calibration certificate

Information is presented as shown in Fig. 7.7, which illustrates a working standard calibration certificate for a T10F torque flange from HBM. A few points will now be explained.

The object of the calibration is named and the serial number is specified. The name not only gives the type designation but also explicitly shows whether the calibration object is just a torque transducer or an entire measurement chain.

The upper part of page 2 reports on the uncertainty of the calibration device. This should not be confused with a measurement uncertainty which is reported for the object of the calibration and appears only on the DKD calibration certificate.

Page 2 of the calibration certificate contains a detailed description of the devices and accessories used during calibration, even down to the cables and so forth. It must be emphasized at this point that the information about the zero signal under the heading Properties of the Transducer serves only to provide complete documentation about the calibration and its boundary conditions. The specified value is measured in connection with the calibration after removal from the calibration device. Deviations during subsequent use are by and large no cause for concern, but regularly checking the zero signal is an important tool for ongoing monitoring (see chapter 5). The heading Signal Conditioner refers to the actual measuring amplifier. As a rule in the event of modular amplifier systems this is a single module.

The shunt calibration signal from the torque transducer is specified in accordance with the units chosen for the calibration either as a raw signal or in N·m. The display device, in everyday parlance often thought of as part of the measuring amplifier, is listed separately under the heading Indicator. In appropriate cases the Indicator Adaptation field contains the adaptation on which the calibration is based if for example N·m is chosen as the display unit instead of the output unit from the torque transducer. The calibrated measuring range is specified. This is important, since a calibration need not necessarily cover the whole of the transducer's nominal measuring range. Calibration for a sub-range is not uncommon.

The calibration result is simply given in the form of a table and compares the input quantity (torque) with the measured value in the desired display unit, for both the ascending and descending series of tests. The specified measured values are zero-compensated as a rule, that is, in each case the signal span is specified relative to the initial torque signal which is given by the output signal after the last preloading and before the start of the actual measurement cycle (see also Appendix A).

The working standard calibration certificate contains no evaluation of characteristic quantities such as linearity and hysteresis. Nor is any conclusion reached about the optimum adaptation for measuring any torque that does not coincide exactly with the load steps included (interpolation).

HBM

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Tel.: +49 / (0)6151 / 803 - 436
Fax: +49 / (0)6151 / 803 - 590

Zertifiziert nach ISO 9001 u. ISO 14001 (DQS-00001)
Certified acc. to ISO 9001 and ISO 14001 by DQS

Akkreditiert als DKD-Kalibrierlab. (DKD-K-00101)
Accredited as DKD calibration laboratory by PTB

Akkreditiert als EMV-Prüflab. (DAT-P-012 / DAT-P-006)
Accredited as EMC testing laboratory by DATech

Kalibrierschein in Anlehnung an DIN ISO 10012 Teil 1
Calibration Certificate according to ISO 10012 part 1

(Werkskalibrierschein / *Working standard calibration certificate)*

XXXXX
HBM
01-10

Gegenstand *Object*	**Drehmomentaufnehmer**
Hersteller *Manufacturer*	**Hottinger Baldwin Messtechnik, Darmstadt**
Typ *Type*	**T10F/5kN m**
Fabrikate/Serien-Nr. *Serial number*	**XXXXX**
Auftraggeber *Customer*	**XXXXX, XXXXX**
Auftragsnummer *Order No.*	**123456A**

Anzahl der Seiten des Kalibrierscheines 3
Number of pages of the certificate

Die Kalibrierung erfolgte mit Messmitteln, die im Sinne der DIN EN ISO 9001 und DIN ISO 10012, Teil 1, auf Nationale Normale rückführbar sind.

The calibration was performed using calibration equipment traceable to National Standards according to ISO 9001 and ISO 10012 part 1.

Prüfer *Tester*	**Breitwieser**	Abnahme *Release*
Datum *Date*	**2001-10-23**	

HBM Kalibrierdienst

DKD_42 - 02/00

QV1061A1.03 - 01/98

Seite / *page* 1

Fig. 7.6 Working standard calibration certificate for a T10F torque flange

```
+---------+
:  XXXXX  :
+---------+
:   HBM   :
:         :
+---------+
:  01-10  :
+---------+
```

Page 2 of the calibration dated 23.10.2001

The transducer was calibrated on a 20 kN·m torque reference standard measuring unit which produces torque loding by means of weights attached to a double ended lever.

Uncertainty of the calibration machine is (PTB 8.32-021980/93):

from 0N·m to 200N·m 0.04 %
from 250N·m to 20000N·m 0.02 %

The calibration is only valid if the signal conditioner of the same type as described below are used.

Ambient temperature : 21 Grad C

Environmental humidity: 47 %

Transducer data

Type : T10F/5kN·m   Serial. No : XXXXXXXXX

Zero signal: 10021,1 Hz   Mounting parts : HBM-Standard

Calibration range : 5000 N·m   Calibration accessories : AP17

Cable data

Length (fixed) : -   Extension : 3 m

Version : Standard

Connection of amplifier : Standard

Further accessories : -

Signal conditioner data

Amplfier type : HBM-MC60   Serial No : DKD 63

Bridge excitation (volt.,frequ.,form) : 5 VDC   Filter : 0,1 Hz Bessel

Type of connection : Standard   Check signal: 2793,0 Hz

Measuring point : Channel 1   Measuring range : 20 kHz

Indicator data

Indicator type : HBM-AB12   Serial No : 2550/013

Indicator matching : -

The signal conditioner is owned by the calibration laboratory.
The indicator is owned by the calibration laboratory.

Working standard calibration certificate for a T10F torque flange (page 2)

XXXXX

HBM

01-10

Page 3 of the calibration dated 23.10.2001

Measured values : Hz

Load in N·m	reading at clockwise torque		reading at anticlockwise torque	
	increasing load	decreasing load	increasing load	decreasing load
0	0,0	- 1,4	0,0	0,6
1000	1002,0	1000,5	- 1002,3	- 1001,5
2000	2004,1	2002,7	- 2004,5	- 2003,6
2500	2505,1	2503,9	- 2505,6	- 2504,7
3000	3006,1	3004,9	- 3006,4	- 3005,7
4000	4007,9	4007,1	- 4008,3	- 4007,9
5000	5009,3	------	- 5010,1	------

Working standard calibration certificate for a T10F torque flange (page 3)

Use for adjusting the measurement chain

If it is intended to rely on the information in the working standard calibration certificate for adjusting the measurement chain, the two-point adjustment method is recommended. The part of the calibration certificate from which to take the value pairs depends on whether the calibration was carried out with or without adaptation of the display unit.

In the event of a calibration without adaptation, all the torque steps in the working standard calibration certificate may generally be used for each of the two points. Usually, however, the first point is the measured zero signal or initial torque signal. In the case of a working standard calibration certificate it is left to the user to decide whether to take the points from the ascending or descending load series, or whether to take an average.

In the event of a calibration with adaptation of the display device, the adaptation specified on page 2 of the calibration certificate should be used. Since when calibrating with adaptation the calibration signal is also specified in the unit N·m, this can also be used to generate the second point. The advantage is that in this case it is only necessary to enter the value of the input quantity (i.e. the calibration signal in N·m as shown on the calibration certificate), while the assigned electrical signal can be acquired by measurement after activating the shunt resistance.

If there is no calibration for alternating torque, the decision has to be taken whether to carry out adjustment with the calibration results for clockwise or counterclockwise torque. The details of the procedure for measurement chain adjustment are described in chapter 5.

7.4.3 DKD calibration

Range of use of DKD calibration

As explained in section 7.3, calibration within the DKD ensures traceability in a direct line. It includes information on measurement uncertainty as well as certain metrological properties that are individually determined by measurement. Classification is carried out on the basis of the values obtained and reported in the calibration certificate.

DKD calibration is therefore recommended in the following cases:

- For high precision transducers or measurement chains.
- For transducers or measurement chains intended to be used as working standards.

- If the measurement uncertainty and the most important of the metrological properties such as linearity or hysteresis need to be reported individually for the transducer or measurement chain.
- If classification needs to be carried out and reported in the calibration certificate.
- If proof of traceability is necessary that will be accepted by external locations, especially in other countries.

Calibration sequence

At customer request calibrations in a DKD context are carried out at HBM in accordance with German standard DIN 51309 or international Guideline EA-10/14 of the European Co-operation for Accreditation (EAD). In addition DKD Guideline DKD-R 3-5, which is treated as a supplement to DIN 51309, is invoked for the calibration of alternating loads. HBM offers DKD calibrations with 5 and 8 load steps. A large number of load steps is a precondition (but no guarantee) for reporting low measurement uncertainty or a good classification.

The calibration sequence includes several series of tests in ascending and descending order. Calibration is carried out for clockwise or counterclockwise torque according to the wishes of the customer. Calibration for clockwise and counterclockwise torque consists of two virtually independent calibrations carried out one after the other including the preloading. As a different option it is also possible to calibrate for alternating torque which changes between clockwise and counterclockwise in a single sequence. The measuring range for which the calibration is carried out can be specified by the customer and may differ from the nominal measuring range of the transducer so that it is also possible to calibrate for measurements in the partial load range.

The measurement sequence includes the following steps:

- Preloading to nominal torque three times
- Measuring the output signal at the load steps specified for the respective calibration with increasing torque (ascending series of tests)
- Measuring the output signal at the load steps specified for the respective calibration with decreasing torque (descending series of tests)
- Repeated measurement of the output signal with increasing torque (repeatability measurement)
- Re-mounting the transducer in a different situation in the calibration machine
- Preloading in the new situation
- Measuring one ascending and one descending series of tests in the new situation (reproducibility measurement)

- Re-mounting the transducer, preloading it again and carrying out a further series of tests in ascending and descending order in yet a third situation and in still more if necessary (reproducibility measurement)

For preference, measurements are taken in three different mounting positions which are brought about by rotating the transducer through 120° about the longitudinal axis in each case. Different regulations apply to torque transducers with a square drive torque connection.

The above sequence is shown in the form of a diagram in Fig. 7.7. DKD calibration is shown by way of example with five different steps: 20 %, 40 %, 60 %, 80 %, 100 % of full scale in the measuring range being calibrated.

Of special note is the fact that in contrast to factory calibration, more series of tests are carried out. This enables the repeatability and reproducibility to be determined. These concepts are explained in more detail below.

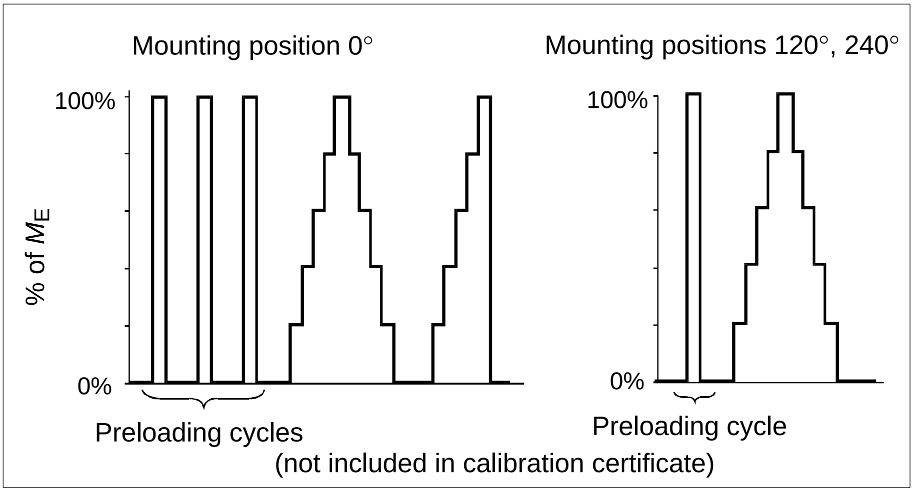

Fig. 7.7 DKD calibration sequence with 5 steps

Choice of physical unit of displayed values

The same aspects as for working standard calibration need to be taken into account when choosing the unit of display.

Information on the DKD calibration certificate

Calibrations in the context of the DKD are regulated by standards and guidelines with regard to not only the calibration sequence but also the analysis of measurements and the contents of the calibration certificate itself, including the information it must contain. The details of the certificate and the information it contains are shown in Fig. 7.8, which illustrates a DKD calibration certificate for a T10F torque flange from HBM. Specially important aspects and points needing further explanation will now be discussed.

The object of the calibration is named and the serial number is specified. The name not only gives the type designation but also explicitly shows whether the calibration object is a torque transducer or an entire measurement chain.

The upper part of page 2 of the DKD calibration certificate reports on the uncertainty of the calibration device. This should not be confused with a measurement uncertainty, which is reported for the calibrated transducer or measurement chain and is discussed below as part of the calibration results. In the most favorable case it can achieve the same uncertainty as the calibration device, if the transducer is sufficiently accurate. As with the working standard calibration certificate, page 2 contains a detailed description of the devices and accessories used during calibration, even down to the cables and so forth. The same comments apply in relation to information on the zero signal, calibration signal and indicator adaptation as for the working standard calibration certificate.

The information on pages 3 to 5 of the calibration certificate is arranged in order of importance for the user. Therefore the most densely compressed illustration comes first. So as to more easily clarify how these analyses come about, the explanations will be given in a different sequence.

Page 5 contains the measured values for all measurements (including the preloading cycles). The specified measured values are zero-compensated as a rule. In each case the signal span is specified relative to the initial torque signal which is given by the output signal after the last preloading and before the start of the first series of tests in the mounting position concerned. However, no further adjustment of the zero signal is made before the second ascending series of tests in the 0° position (repeatability test series).

The calibration result in the strict sense is shown on page 3 of the DKD calibration certificate in the form of a table in which the values of the input torque are assigned to the values of the output signal in the desired display unit. This assignment is unambiguous because it is averaged over the ascending series of tests for all mounting positions. However, the second ascending series of tests in the 0° position is deliberately excluded in order not to weight this mounting position more heavily than the others.

The interpolation polynomials shown on page 4 (they should more accurately be called best fit polynomials) are equations that describe a characteristic curve which intersects all points in the calibration results to the nearest possible approximation. The procedure makes use of the least squares method. The linear interpolation polynomial is an equation of a straight line and therefore the only one that can be implemented even when using measuring amplifiers which only allow a two-point adaptation. Smaller errors are obtained by using the quadratic or cubic polynomials that show a bent characteristic curve.

Separate interpolation polynomials are determined during calibration for clockwise and counterclockwise torque. Additionally a linear interpolation polynomial is determined for clockwise and counterclockwise torque together. This polynomial usually gives rise to a greater degree of error. It is not to be confused with an interpolation polynomial for alternating torque. When calibrating for alternating torque only one linear, one quadratic and one cubic interpolation polynomial each will be specified. These apply to the whole measuring range from negative to positive full scale value.

The next step in the analysis involves characteristic quantities which are quantitative measures for various metrological properties. They are determined by analyzing the measured values. In some cases this analysis includes comparison with the interpolation polynomials. The calibration standards refer to these characteristic quantities as classification criteria. The characteristic quantities include the following:

- Relative repeatability *b'*
- Relative reproducibility *b*
- Relative reversibility *h* as a measure of the hysteresis (see Appendix A)
- Relative zero error f_0
- Resolution *r*
- A separate relative interpolation error f_a for use with each of the linear, quadratic and cubic best fit polynomials

Precise definitions are given in the standards and guidelines referring to calibration. Only the two central terms repeatability and reproducibility will be explained in a little more detail here since they could quite easily be confused. Repeatability describes the property of a transducer to produce the same output signal when measuring the same torque again in the same conditions. In DKD calibration repeatability is determined by comparing the first and second ascending series of tests (repeatability measurement) in the 0° mounting position.

Reproducibility describes the property of a transducer to produce the same output signal when measuring the same torque again in changed conditions. The conditions which are changed may be the mounting conditions, the location, the time of measurement or the personnel carrying out the measurement. In DKD calibration, reproducibility is determined by comparing the series of tests for the various mounting positions.

Page 3 of the DKD calibration certificate shows the measurement uncertainty and the classification according to the chosen guideline (DIN 51309 or EA-10/14). The expression classes is often used when referring to classification. These classes are not to be confused with the expression accuracy class, which is defined by the manufacturers and thus differs between different manufacturers (see Appendix A).

The measurement uncertainty and the classification are the most concise ways of representing the accuracy of a transducer or measurement chain. They are based on an analysis of the characteristic quantities mentioned above. The measurement uncertainty is derived as a summarized quantity. Superimposed within this are the individual influences which affect accuracy and which are quantitatively described in the characteristic quantities. It is calculated according to the laws of statistics. Information on the statistical interpretation of uncertainty can be found in the section devoted to the subject below. The measurement uncertainty is the most suitable quantity for describing the everyday concept of accuracy in a single numerical value.

The classification of a transducer is derived from analyzing the same characteristic quantities, and in this case certain upper limits apply when assigning the transducer to a class in respect of each separate characteristic quantity. It should be noted that different sub-ranges of the total calibrated measuring range can have a different uncertainty and a different classification. In the case of small sub-ranges both of these do not usually reach favorable values due to the fact that the analysis is based on tolerances specified for actual-value related characteristic quantities. The outcome also depends on which best fit polynomial is used.

The last page of the DKD calibration certificate contains graphs. The first graph shows each individual series of tests. The input torque is plotted on the horizontal axis. To ensure that fine differences can actually be seen, the deviation relative to an idealized straight line is chosen for the vertical axis. The straight line passes through the origin and a point defined on the one hand by the full scale torque value and on the other by a reference value for the full scale output signal. The latter is specified at the top of the page. The unit of display for the output signal on the vertical axis is a percentage of this value. It can be seen that all ascending series of tests begin at 0 % since zero-compensated values are plotted in each case. Their ends are linked to the respective descending series of tests.

The second and in some cases third graphs show the interpolation errors, again as a function of the input torque. Unlike the upper graph, the errors are shown here as a percentage of the actual value, for which reason the plotted values are at their greatest when the torque is low. The graphs contain separate curves for each linear, quadratic and cubic best fit polynomial. It can be seen that the greatest errors occur in the case of the linear best fit polynomial.

DEUTSCHER KALIBRIERDIENST **DKD**

Kalibrierlaboratorium für mechanische Messgrößen
Calibration laboratory for mechanical quantities

Akkreditiert durch die / *accredited by the*
Akkreditierungsstelle des DKD bei der
PHYSIKALISCH-TECHNISCHEN BUNDESANSTALT (PTB)

HBM

Hottinger Baldwin Messtechnik GmbH
DKD-Kalibrierlaboratorium
Im Tiefen See 45 DE - 64293 Darmstadt
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Deutscher Akkreditierungs Rat
DAR
DKD-K-00101

Kalibrierschein
Calibration Certificate

Kalibrierzeichen
Calibration label

XXXXX
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Gegenstand *Object*	**Messkette aus Drehmomentaufnehmer und Messverstärker**
Hersteller *Manufacturer*	**Hottinger Baldwin Messtechnik GmbH,**
Typ *Type*	**T10F/5kN m,MGCplus / ML60,ABX22A**
Fabrikat/Serien-Nr. *Serial number*	**XXXXX, XXXXX, XXXXX**
Auftraggeber *Customer*	**XXXXX, XXXXX**
Auftragsnummer *Order No.*	**123456A**
Anzahl der Seiten des Kalibrierscheines *Number of pages of the certificate*	6
Datum der Kalibrierung *Date of calibration*	**2001-10-09**

Dieser Kalibrierschein dokumentiert die Rückführung auf nationale Normale zur Darstellung der Einheiten in Übereinstimmung mit dem Internationalen Einheitensystem (SI).

Der DKD ist Unterzeichner der multilateralen Übereinkommen der European co-operation for Accreditation (EA) und der International Laboratory Accreditation Cooperation (ILAC) zur gegenseitigen Anerkennung der Kalibrierscheine.

Für die Einhaltung einer angemessenen Frist zur Wiederholung der Kalibrierung ist der Benutzer verantwortlich.

This calibration certificate documents the traceability to national standards, which realize the units of measurement according to the International System of Units (SI).

The DKD is signatory to the multilateral agreements of the European co-operation for Accreditation (EA) and of the International Laboratory Accreditation Cooperation (ILAC) for the mutual recognition of calibration certificates.

The user is obliged to have the object recalibrated at appropriate intervals.

Dieser Kalibrierschein darf nur vollständig und unverändert weiterverbreitet werden. Auszüge oder Änderungen bedürfen der Genehmigung sowohl der Akkreditierungsstelle des DKD als auch des ausstellenden Kalibrierlaboratoriums. Kalibrierscheine ohne Unterschrift und Stempel haben keine Gültigkeit.
This calibration certificate may not reproduced other than in full except with the permission of both the Accreditation Body of the DKD and the issuing laboratory. Calibration certificates without signature and seal are not valid.

Stempel *Seal*	Datum *Date*	Stellv. Leiter des Kalibrierlaboratoriums *Deputy Head of the calibration laboratory*	Bearbeiter *Person in charge*
Deutscher DKD-K-00101 Kalibrierdienst	**2001-10-12**	**Harreus**	**Moor**

DKD_T42 - 02/01

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Fig. 7.8 DKD calibration certificate for a T10F torque flange

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In case of doubts the German text of this certificate is valid.

Kalibriereinrichtung *Torque Reference Standard*	**20 kN m - BNME**	PTB-Zeichen: *Official sign*	**PTB-8.32-21088/92**
Anschlussmessunsicherheit: *Best measurement capability*	**<= 0,02 %**	der eingestellten Drehmomentstufe *of the torque applied*	

Angaben zur Kalibrierung
Calibration conditions

Umgebungstemperatur: *Ambient temperature*	**21 ± 1** °C	Umgebungsfeuchte: *Environmental humidity*	**40%** rel.
Umgebungsluftdruck: *Atmospheric pressure*	**1023** hPa		

Die Kalibrierung ist nur gültig bei Verwendung des unten beschriebenen Ausgeber-Typs.
The calibration is only valid if a signal conditioner of the same type as described below is used.

Angaben zum Aufnehmer
Transducer data

Nullsignal: *Zero signal*	**10006,5 Hz**
Kalibrierzubehör: *Calibration accessories*	**DKD-Standard**

Angaben zum Kabel
Cable data

Länge (fest verbunden): *Length (fixed)*	*****	Verlängerung: *Extension*	**6 m**
Ausführung: *Version*	**6 - adrig** *..-core*	Ausgeber-Anschluss: *Connection of amplifier*	**Standard** *..-lead*

Angaben zum Ausgeber
Signal conditioner data

Verstärkertyp: *Amplifier type*	**MGCplus / ML60**	Identifizierung: *Identification*	**835964018**
Messbereich: *Measuring range*	**20 kHz**	Messstelle: *Measuring channel*	**1**
Brückenspeisespannung: *Bridge excitation voltage*	**5VDC**	Filter: *Filter*	**0,1 Hz Bessel**
Kalibriersignal: *Check signal*	**2350,4 Hz**	Anschlussart: *Type of connection*	**Standard** *..-lead*

Angaben zum Anzeiger
Indicator data

Anzeigertyp: *Indicator type*	**ABX22A**	Identifizierung: *Identification*	**ohne**
Anzeigeranpassung: *Indicator matching*	*****		

Sonstiges
Other data

-

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1. **Kalibrierverfahren / *Calibration Procedure* :** DIN 51309 : 1998-02
2. Die im Kalibrierverfahren beschriebenen Vorbelastungen sind vor jeder Benutzung durchzuführen.
   The preloading described in the calibration procedure has to be repeated each time the transducer is used.
3. **Kalibrieranordnung / *Calibration installation* :** Einbaulage horizontal / horizontal position of device
4. Berechnete Werte sind um die jeweilige Nullanzeige reduziert. (Korrektionen lt. Akkreditierung sind berücksichtigt.)
   Calculated values are reduced by the respective zero signal. (Correction factors acc. to accreditation are included.)
5. Die Ergebnisse sind in der letzten Stelle gerundet. *The calculated values are rounded in the last decimal.*
6. **Zusätzliche Angaben / *Additional information* :**

7. **Auswertung / *Analysis***

7.1 **Kalibrierergebnis / *Calibration results***

Drehmoment / *torque* in kN m	Signal / *signal* in Hz	rel. Messunsicherheit / *rel. uncertainty* in %, *k*=2 bei Ausgleichsfkt. / *using interpolation equation* linear *	quadrat.	kubisch
Rechtsdrehmoment / *clockwise torque*				
0	0,00			
0,5	500,98	0,171	0,132	0,132
1	1001,92	0,117	0,081	0,081
1,6	1603,03	0,095	0,062	0,062
2	2003,67	0,078	0,051	0,051
2,5	2504,37	0,069	0,051	0,051
3	3004,90	0,055	0,047	0,048
4	4006,23	0,043	0,042	0,042
5	5006,83	0,059	0,043	0,043
Linksdrehmoment / anti*clockwise torque*				
0	0,00			
-0,5	-501,13	0,178	0,098	0,097
-1	-1002,03	0,114	0,057	0,057
-1,6	-1602,94	0,074	0,037	0,037
-2	-2003,57	0,064	0,033	0,033
-2,5	-2503,97	0,042	0,030	0,030
-3	-3004,47	0,035	0,031	0,031
-4	-4005,70	0,033	0,032	0,032
-5	-5006,40	0,049	0,035	0,035

7.2 **Klasseneinstufung nach DIN 51309 / *Classification according to DIN 51309***

U = erweiterte relative Messunsicherheit gemäß .. / *expanded relative uncertainty according to ..*
Sawla, A. "Guidance for the best measurement capability of force calibration machines and uncertainty of calibration results of force measuring devices", PTB-Mitteilungen 104

Klasse *Class*	**lineare Ausgleichsfkt. *linear interpolation equation*** von/*from* in kN m	bis / *to* in kN m	*U* in %	**quadratische Ausgleichsfkt. *quadratic interpolation equation*** von/*from* in kN m	bis / *to* in kN m	*U* in %	**kubische Ausgleichsfkt. *cubic interpolation equation*** von/*from* in kN m	bis / *to* in kN m	*U* in %
0,05									
0,1	1,6	5	0,10	1,6	5	0,06	1,6	5	0,06
0,2	0,5	5	0,17	0,5	5	0,13	0,5	5	0,13
0,5									
1									
2									
5									
0,05									
0,1	-1,6	-5	0,07	-1	-5	0,06	-1	-5	0,06
0,2	-0,5	-5	0,18	-0,5	-5	0,12	-0,5	-5	0,12
0,5									
1									
2									
5									

* In Abweichung zur DIN 51309 wird die Interpolationsabweichung von einer linearen Ausgleichsfunktion nicht als zufällig betrachtet und analog zu f_q nach Formel (C.5) des Anhanges C der DIN 51309 in der Messunsicherheit verrechnet.

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8. Interpolationsgleichungen / *Interpolation equations* S in Hz M in kN m

8.1 Lineare Interpolationsgleichung / *Linear interpolation equation*

8.1.1 Rechtsdrehmoment / *clockwise torque:*

S_{ai} =	1,001550 E+03	M_i
M_{ai} =	9,984523 E-04	S_i

8.1.2 Linksdrehmoment / *anticlockwise torque:*

S_{ai} =	1,001445 E+03	M_i
M_{ai} =	9,985572 E-04	S_i

8.1.3 Rechts- und Linksdrehmoment / *clockwise and anticlockwise torque:*

S_{ai} =	1,001497 E+03	M_i
M_{ai} =	9,985047 E-04	S_i

(siehe Fußnote 2 / *see footnote 2*)

8.2 Quadratische Interpolationsgleichung / *Quadratic interpolation equation:*

8.2.1 Rechtsdrehmoment / *clockwise torque:*

S_{ai} =	1,002109 E+03	M_i +	-1,462735 E-01	M_i^2
M_{ai} =	9,978948 E-04	S_i +	1,456455 E-10	S_i^2

8.2.2 Linksdrehmoment / *anticlockwise torque:*

S_{ai} =	1,002024 E+03	M_i +	1,514692 E-01	M_i^2
M_{ai} =	9,979798 E-04	S_i +	-1,508690 E-10	S_i^2

8.3 Kubische Interpolationsgleichung / *Cubic interpolation equation:*

8.3.1 Rechtsdrehmoment / *clockwise torque:*

S_{ai} =	1,002036 E+03	M_i +	-9,813972 E-02	M_i^2 +	-6,934177 E-03	M_i^3
M_{ai} =	9,979681 E-04	S_i +	9,741395 E-11	S_i^2 +	6,938535 E-15	S_i^3

8.3.2 Linksdrehmoment / *anticlockwise torque:*

S_{ai} =	1,002332 E+03	M_i +	3,542897 E-01	M_i^2 +	2,921844 E-02	M_i^3
M_{ai} =	9,976732 E-04	S_i +	-3,526275 E-10	S_i^2 +	-2,902739 E-14	S_i^3

9. Kennwerte nach DIN 51309 / *Classification criteria according to DIN 51309*

M_k in kN m	b' in %	b in %	f_0 in %	h in %	r in kN m	f_a (lin.) in %	f_a (quadr.) in %	f_a (kub.) in %
5,0	0,000	0,052	-	-	0,0001	-0,018	-0,001	-0,001
4,0	0,000	0,047	-	0,022	0,0001	0,001	0,003	0,003
3,0	0,003	0,050	-	0,040	0,0001	0,008	-0,004	-0,005
2,5	0,000	0,044	-	0,059	0,0001	0,020	0,000	0,000
2,0	0,005	0,035	-	0,067	0,0001	0,028	0,002	0,002
1,6	0,000	0,037	-	0,089	0,0001	0,034	0,002	0,003
1,0	0,010	0,030	-	0,130	0,0001	0,037	-0,005	-0,001
0,5	0,040	0,000	-	0,220	0,0001	0,040	-0,009	-0,003
0,0	-	-	0,016	-	-	-	-	-
0,0	-	-	0,014	-	-	-	-	-
-0,5	0,020	0,040	-	0,153	0,0001	0,082	0,032	0,011
-1,0	0,000	0,020	-	0,087	0,0001	0,059	0,016	0,003
-1,6	0,000	0,012	-	0,050	0,0001	0,039	0,006	0,000
-2,0	0,010	0,005	-	0,042	0,0001	0,034	0,006	0,004
-2,5	0,004	0,012	-	0,033	0,0001	0,014	-0,006	-0,004
-3,0	0,003	0,023	-	0,026	0,0001	0,004	-0,008	-0,004
-4,0	0,002	0,030	-	0,017	0,0001	-0,002	0,001	0,004
-5,0	0,006	0,038	-	-	0,0001	-0,016	0,001	-0,001

2) Die Bestimmung der linearen Interpolationsgleichung für Rechts- und Linksdrehmoment ist nicht identisch mit einem Kalibrierergebnis für Wechseldrehmoment. Sie ermöglicht es, mit nur einem Kalibrierfaktor das Anzeigegerät optimal für Rechts- und Linksdrehmoment anzupassen.

2) *The linear interpolation equation for clockwise torque and anticlockwise torque can't be used as a calibration result for alternating torque. It only can be used to adjust the indicator optimally for clockwise torque and anticlockwise torque with a single calibration factor.*

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10. Messdaten / *measuring data* **in Hz**

Rechtsdrehmoment / *clockwise torque*

0	0,0	-1,2	-1,6	0,0	-0,3	-0,3
0,5	-	-	-	500,9	500,7	500,8
1	-	-	-	1001,9	1001,5	1001,7
1,6	-	-	-	1603,2	1602,6	1602,9
2	-	-	-	2003,8	2003,4	2003,6
2,5	-	-	-	2504,9	2504,3	2504,6
3	-	-	-	3005,7	3005,1	3005,3
4	-	-	-	4007,3	4006,9	4007,0
5	5008,4	5007,7	5006,3	5008,3	5008,3	5008,0
kN m	1. Vorbel. *preloading*	2. Vorbel. *preloading*	3. Vorbel. *preloading*	0° /1		0° /2

0	0,0	0,0	-0,6	0,0	0,0	-0,8
0,5	-	500,9	499,6	-	500,9	499,1
1	-	1001,8	1000,2	-	1001,6	999,7
1,6	-	1602,8	1601,2	-	1602,6	1600,5
2	-	2003,5	2001,9	-	2003,1	2001,1
2,5	-	2504,4	2502,6	-	2503,8	2501,8
3	-	3004,8	3003,4	-	3004,2	3002,6
4	-	4006,0	4005,0	-	4005,4	4004,1
5	5007,1	5006,5	5006,5	5003,5	5005,7	5005,7
kN m	Vorbel. *preloading*	120°		Vorbel. *preloading*	240°	

Linksdrehmoment / *anticlockwise torque*

0	0,0	1,0	1,7	0,0	0,6	0,6
-0,5	-	-	-	-500,9	-500,4	-500,4
-1	-	-	-	-1001,7	-1001,3	-1001,1
-1,6	-	-	-	-1602,7	-1602,4	-1602,1
-2	-	-	-	-2003,3	-2003,2	-2002,9
-2,5	-	-	-	-2504,1	-2503,9	-2503,6
-3	-	-	-	-3004,8	-3004,5	-3004,3
-4	-	-	-	-4006,3	-4006,0	-4005,8
-5	-5007,5	-5006,9	-5006,4	-5007,3	-5007,3	-5007,0
kN m	1. Vorbel. *preloading*	2. Vorbel. *preloading*	3. Vorbel. *preloading*	0° /1		0° /2

0	0,0	0,0	0,7	0,0	0,0	0,6
-0,5	-	-501,1	-500,2	-	-501,1	-500,2
-1	-	-1001,9	-1000,7	-	-1001,9	-1000,9
-1,6	-	-1602,6	-1601,5	-	-1602,8	-1601,8
-2	-	-2003,2	-2001,9	-	-2003,3	-2002,2
-2,5	-	-2503,8	-2502,5	-	-2504,0	-2503,0
-3	-	-3004,1	-3002,9	-	-3004,5	-3003,7
-4	-	-4005,1	-4004,1	-	-4005,7	-4005,0
-5	-4999,0	-5005,4	-5005,4	-5007,2	-5006,5	-5006,5
kN m	Vorbel. *preloading*	120°		Vorbel. *preloading*	240°	

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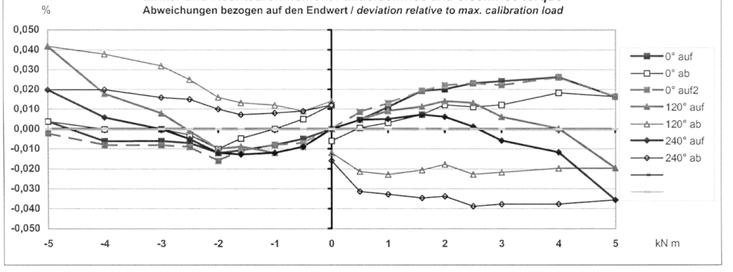
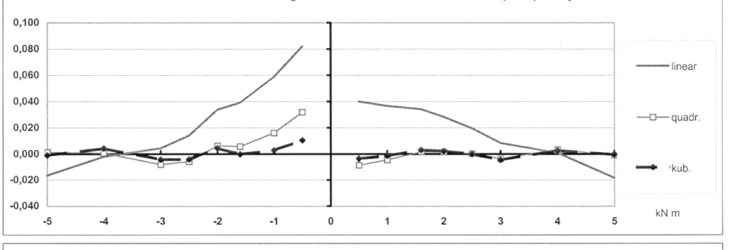
11. **Darstellung der Ergebnisse in Diagrammen /** ***Results in diagrams***

Bezugswert / *Reference:* *5007,5* in Hz

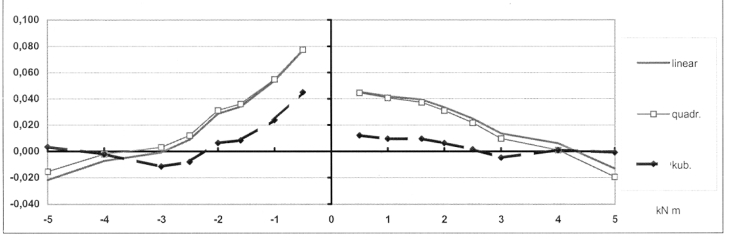
Links- und Rechtsdrehmoment / ***anticlockwise and clockwise torque***
Abweichungen bezogen auf den Endwert / *deviation relative to max. calibration load*



Interpolationsabweichung v.M. / ***Interpolation error relative to actual value***
Links- und Rechtsdrehmoment getrennt / *anticlockwise and clockwise torque separately*



Interpolationsabweichung v.M. / ***Interpolation error relative to actual value***
Links- und Rechtsdrehmoment gemeinsam / *anticlockwise and clockwise torque in common*



Version: 4.2.1

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DKD calibration certificate for a T10F torque flange (page 6)

Mathematical and statistical interpretation of measurement uncertainty and the confidence interval

The measurement uncertainty specifies the extent of any possible deviation between the output value and the input quantity. In concrete terms this means that the actual torque (the ”true value”) lies within a range consisting of the output value plus/minus the measurement uncertainty. The relative measurement uncertainty is specified as a percentage of the measured value.

Since the measurement uncertainty includes statistical errors and is itself determined in accordance with the laws of statistics from a random sample (the measurements taken during calibration), there always exists a residual probability that the true value lies outside the interval. It should be kept in mind that in the known curve of normal distribution the probability density for values well away from the expected value reduces asymptotically toward zero but never reaches it.

The extended measurement uncertainty specified in the course of DKD calibration usually includes a confidence interval of 95 %. In a normal distribution this is reached by using the extension factor k=2, with which the (combined) standard uncertainty of the estimated value is multiplied. A comprehensive discussion of the fundamental concepts for the computation and analysis of measurement uncertainty can be found in [19].

Use for adjusting the measurement chain

If there is a DKD calibration certificate without display unit adaptation, there are very many different ways to adjust the measurement chain that may be put to good use depending on the application and on the functions available in the amplifier electronics. The method recommended for use with the very frequently used two-point adjustment over the whole of the calibrated measuring range is the linear interpolation polynomial. Higher interpolation polynomials with lower interpolation errors can generally only be used with the assistance of a computer link.

Very high quality measuring amplifiers offer facilities for multi-point adjustment, in which case interpolation is carried out automatically over all the points that have been input. The most suitable input for this purpose would be the value pairs under the Calibration Results heading. Since the interpolation polynomials in the calibration certificate are always determined for the whole of the calibrated measuring range, when using partial load ranges it may be preferable to carry out a two-point adjustment on the basis of those value pairs in the calibration results that best fit the partial load range required.

In the event of a calibration with adaptation of the display unit, the two-point adjustment specified on page 2 of the calibration certificate should be used. As already described above for the working standard calibration certificate, the calibration signal can also be used to generate the second point for the adjustment.

Except in the case of a calibration for alternating torque, the decision has to be taken whether to carry out adjustment with the calibration results for clockwise or counterclockwise torque. The details of the procedure for measurement chain adjustment are described in chapter 5.

7.5 Calibration on the test bench

7.5.1 Defining the task and approaches to a solution

Transfer of the measured quantity into the application

The task of calibrating a transducer that is intended to be used in an application such as a power test bench always involves the transfer of the measured quantity torque into the application. By this it is understood that a reference measure that can be traced back to a reference standard must be present, so that the transducer can compare it to the torque to which it is actually being subjected in the application.

Calibration in the laboratory or on site on a test bench

In the simplest case the measured quantity (torque) is transferred directly by the torque transducer in the shaft train. The reference is therefore the torque transducer calibration that has already been carried out by the manufacturer or in a calibration laboratory. Depending on the requirements this can be calibration from any of the different levels in the order of rank from PTB or DKD calibration down to the manufacturer's test certificate.

However, if this torque transducer is then used in its intended test bench application, the boundary conditions prevailing there can crucially influence the uncertainty of the torque measurement in the test bench and lead to further errors. Such errors could be caused by test bench components such as frames and couplings. But factors such as the alignment of components or elastic materials behavior in the adaptations or screwed joints can also have an effect on deformation behavior in the shaft train and therefore also on the torque transducer. All these matters can have an effect on the measurement characteristics of the torque transducer.

An alternative approach is to calibrate the torque transducer on site in its built-in state within the application. This section is devoted to methods of this kind and how to assess them. The most advantageous methods are those in which adverse interactions between the test bench, its environment and the measurement results can be minimized and estimated. The order of rank of on-site calibration is not necessarily worse than that of laboratory calibration, but depends on the order of rank of the reference standard used, as well as on qualification of the device and method.

Calibration on a test bench can have advantages if optimized methods are used. The most important of these advantages are:

- There are no effects that can cause a divergence between the built-in state on the test bench and the built-in state during calibration.
- Time is saved in recalibration due to possibly faster completion, since there is no need for complete removal and transport to the external calibration laboratory.
- Since an application such as a test bench usually represents a measurement device in its entirety which should have its traceability clearly documented, on-site calibration comes closer to the fundamental idea of traceability.

The accuracy requirements expected of calibration

Despite the potential advantages of calibration on the test bench, the question whether it is preferable to calibration in the laboratory cannot be fully answered at this point. For one thing the question cannot be separated from the actual application, and for another the evaluation of calibration devices and methods is still in its infancy compared to evaluation methods and criteria that are already commonplace for calibration in the laboratory.

When choosing a calibration method it is important to be clear about the general requirements from the outset. As already explained on the subject of choosing a torque transducer (see chapter 4), here too the question of the necessary level of accuracy cannot be answered by a single item of numerical information. On the contrary, various clearly defined concepts have to be separately examined.

Particularly when calibrating on the test bench it is common practice simply to ensure the best possible match between the torque transducer being calibrated and the reference torque. At the same time calibration is confined to a single series of tests and a single mounting position for the transducer. The accuracy criterion is then simply taken to be the maximum deviation found.

On the other hand if traceability is an objective, the rules about the order of rank in disseminating the torque should be given due consideration. The first question to ask is whether it is enough to ensure repeatability, or whether reproducibility is needed. If reproducibility is needed, the same conditions as in the calibration equipment must be guaranteed even in different mounting conditions.

For repeatability it is enough to examine just one mounting situation. In this case particularly, calibration on the test bench frequently offers a big advantage since the differences between mounting conditions during the calibration procedure and in actual use are often so minimal that repeatability is enough.

For certain requirements the question also has to be asked whether several subrange calibrations are needed, since this action could well result in improved accuracy in the lower part of the measuring range provided the transducer has the appropriate properties.

7.5.2 Calibration techniques and equipment for use on the test bench

If calibration on the test bench is planned, several methods are available just as in the case of calibration in the laboratory.

Calibration devices based on lever-arm-mass systems

Lever-arm-mass systems are very widely used at the present time. They are calibration systems in which the reference torque is brought about by generating torque at a precisely quantified value. Advantages of this method are that it has proved itself within current accuracy expectations and enjoys full user confidence. In view of the purely mechanical method of transfer, long-term stability is not particularly critical. An important advantage from the standpoint of practicality and design is that a test bench and an associated calibration device on the lever-arm-mass system principle can be designed so that the lever arm can be set up on the test bench without having to undo the shaft train.

There is a practical difficulty in the fact that lever-arm-mass systems are decidedly unwieldy for calibrating the measured quantity, since the masses and lever arms required are relatively large, particularly when rather high torque values are involved.

A major problem is how to exclude or minimize the effects of bearing friction. For in contrast to calibration equipment used in the laboratory, special bearings can hardly ever be used for calibration on a test bench and would usually clash with the requirement that the test bench should emulate normal operation as closely as possible.

The difficulty cannot be overcome if the lever arm is fitted immediately next to the torque transducer, as shown in the diagram in Fig. 7.9. This can best be explained with the aid of the engineering mechanics concept of external and internal moments. The torque introduced via the lever arm acts on the shaft train as an external moment. According to the theorem of moments from statics, equilibrium can only prevail if the sum of all external torque values cancels itself out. By contrast the measured torque acts as an internal moment or intersecting moment, as it is also known. In the consideration of equilibrium it is only apparent when the shaft train is thought of as being sectioned at the position of the torque transducer. Fig. 7.9 illustrates this procedure in its lower part, which is known as a free body diagram. The aim of considering the equilibrium is an equation for defining the torque acting on the torque transducer on the basis of the known reference torque, that is, the external torque from the lever.

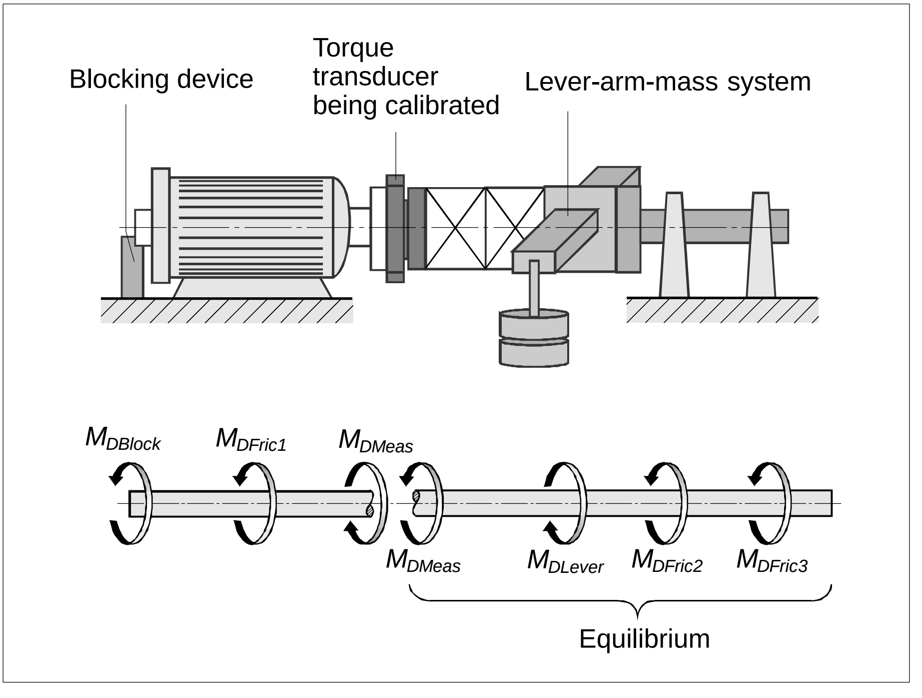


Fig. 7.9 Test bench with calibration device based on the lever-arm-mass system principle, lever arm supported by a separate bearing

In this case the equation obtained by considering the equilibrium in the shaft section identified by the bracket is

$$M_{DMeas} = M_{DLever} - M_{DFric2} - M_{DFric3}$$

From this it can be seen that the moments of friction received via the lever bearing are capable of falsifying the torque.

In present day torque flanges that can support considerable bending moments and radial forces, there are often no bearings at all between the calibration lever arm and the measurement flange, as illustrated in Fig. 7.10. This means, of course, that influences on the calibration results due to bending moments acting as parasitic loads have to be accepted.

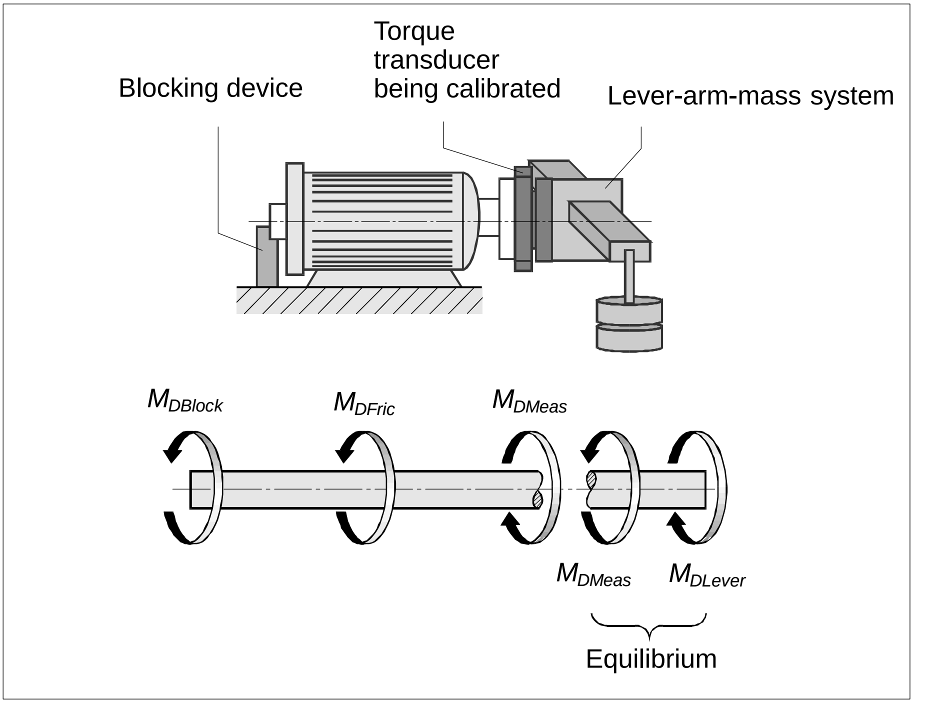


Fig. 7.10 Test bench with calibration device based on the lever-arm-mass system principle, lever arm supported by the torque transducer

The equation obtained from the equilibrium condition set up for the free body diagram for torque (see the lower part of the illustration) for the shaft section identified by the bracket is

$$M_{DMeas} = M_{DLever}$$

This time it can be seen that there is no falsification of the torque due to bearing friction. However, it is now necessary to reckon with falsification due to the influence of parasitic loads, since the weight of the lever arm exerts bending moments and lateral forces on the torque transducer.

The bending moment can be reduced by designing the lever arm or an uptake mechanism so that the center of gravity of the lever arm complete with its calibration masses lies as far as possible in the same axial position as the torque transducer. On the other hand the lateral force cannot be reduced by this means.

These two sources of error, bearing friction in the case of a supported lever arm and parasitic loads in the case of an unsupported lever arm, must be quantitatively estimated case by case so that they can be weighed against one another. Weighing up must include design considerations and then the question must be asked whether it is preferable to calibrate with the aid of a transfer transducer or a reference transducer.

Calibration devices using torque reference transducers or transfer transducers

In principle, systems with torque reference transducers or transfer transducers can use any mechanism to generate the torque, which in many cases has considerable advantages for use in on-site calibration on a test bench. Depending on the design of the calibration device the torque can be connected more easily, at a more suitable point and with less parasitic loads. A generally higher level of automation is also possible, since there is no need for any complicated arrangement of weights. For the same reason there are also advantages regarding transportability. There is practically no alternative to this method if a continuous loading procedure is required. However, not a great deal of cumulative experience has been gained with such methods so far, and clarification regarding the most favorable design and the achievable calibration accuracy is strongly dependent on the circumstances of each individual case. In order to estimate the additional uncertainty components, the interactions of the power test bench or calibration device with the transfer transducer need to be known.

Most of the advantages mentioned above can be traced back to the special property of the method, which is that the effect of friction moments arising from the bearing can practically always be excluded. This fact will be discussed in the light of the typical configuration shown by way of example in Fig. 7.11. In this case the deciding factor is that not only is the moment which acts on the transducer being calibrated an internal moment, but so too is the reference moment. To set up a condition of equilibrium which can be used to deduce the former from the latter, the shaft train must therefore be thought of as being sectioned at both positions, as shown in the free body diagram in the lower part of the illustration. The equation obtained by considering the equilibrium in the section identified by the bracket is

$$M_{DMeas} = M_{DRef}$$

It can be seen that it is impossible for friction moments to cause any falsification. In comparison with the examples of calibration devices on the lever-arm-mass system principle described above this is especially remarkable, since in this case the torque connection to input side is surrounded on both sides by bearings.

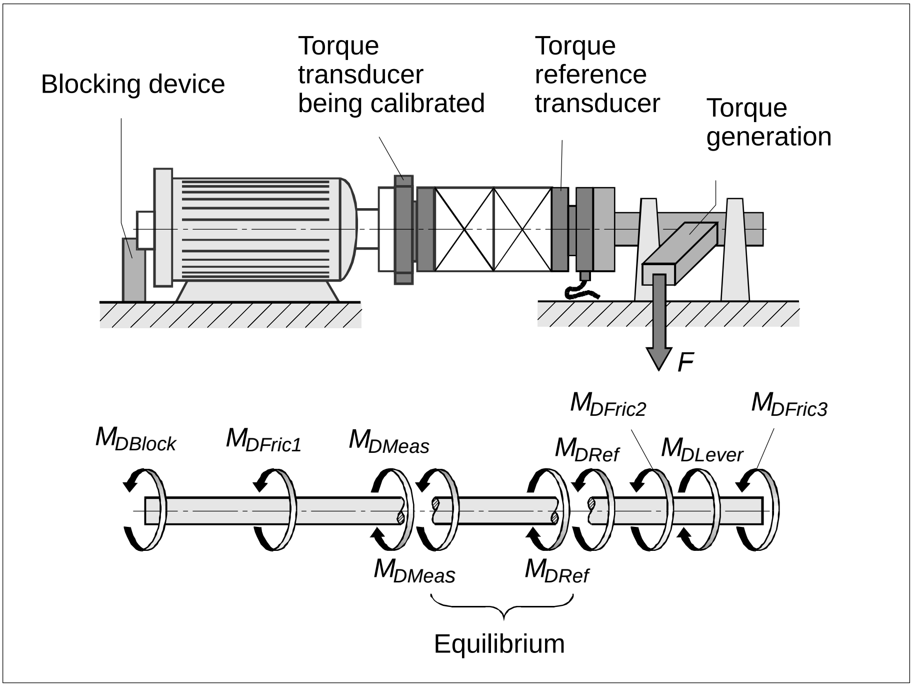


Fig. 7.11 Test bench with calibration device with transfer transducer or reference transducer

Thus practically any manual or motor-driven device can be used as the torque generating mechanism. A simple solution consists of a lever arm loaded with a force. This force can be generated hydraulically or by the distortion of an elastic spring. In any case the bearing is not particularly critical, for the torque generated by the device is measured by the torque transfer transducer or reference transducer. Electrical machinery such as servo motors or stepping motors can also be used to generate the torque.

If the load steps specified for calibration must be delivered exactly, the torque generation unit will need a controller, and this cannot be done with every mechanism.

Calibration devices using force reference transducers or transfer transducers

A final design principle that should be mentioned for on-site calibration devices is again the hybrid form which combines a force reference transducer with a lever arm.

Qualification of on-site calibration devices

Both the calibration device as a whole and the calibration method must as far as possible be qualified (and certified if required). Certification by the manufacturer of the calibration device or by an independent site is recommended. The whole of the test bench and calibration procedure intended for the planned application must be examined in order to assess the measurement results. Again this can be done with the aid of torque transfer transducers, adaptable lever-arm-mass systems or calibration devices using reference transducers.

If formal certification is desired, measurement uncertainty must be determined taking into account the conceivable applications and environmental conditions. Unfortunately at present this is not covered by any generally accepted standard or recommendation such as exists in the case of materials testing machines (e.g. EN ISO 7500-1 which is for static uniaxial testing machines). In any case calibration of the weight sets that will be used or of the transfer transducers and/or reference transducers is insufficient for this purpose.

Evaluation criteria

This section concludes with a summary of the evaluation criteria which can be used when deciding between particular calibration methods once the previously mentioned questions about the accuracy and traceability of the calibration have been answered. Some of these criteria have already been mentioned in connection with certain methods which offer advantages in respect of those questions.

- Keep the variation between the mounting conditions of the torque transducer during calibration and the mounting conditions during standard operation as small as possible.
- Keep the parasitic loading of the torque transducer (i.e. by bending moments and lateral forces) as low as possible.
- Avoid or minimize falsification of the reference torque by bearing friction.
- Mounting conditions and available space in the test bench must be taken into consideration.

7.5.3 Using transfer transducers and reference transducers in an industrial environment

The difference between transfer transducers and reference transducers

The difference between a reference transducer and a transfer transducer is inferred from the stage in its history at which the transducer received its calibration, that is, it depends on the role which the transducer plays when the measured quantity is transferred into the application.

A transfer transducer in fact transfers its calibrated reference torque to another measuring device or application. The measured quantity (torque) is normally disseminated or transferred by means of torque transfer transducers from the national standard (PTB) to a reference standard (DKD), then to working standards such as calibration devices and ultimately to test devices such as power test benches. This process of transferring the reference torque to a calibration device at the next level may also be attained in the form that the transfer transducer stays in that calibration device. This gives rise to a calibration device with a transfer transducer.

According to this definition, however, the torque can also be transferred into the end-user application (test bench) by keeping the transfer transducer as a measuring device. To put it another way, whenever a torque transducer which has been calibrated only in the laboratory and not on the test bench is used, the torque transducer concerned takes on the role of a transfer transducer.

Transfer to a transducer that remains in the test bench, however, can also take place indirectly through a reference transducer. As a rule a calibration device with a reference transducer is provided for this purpose. A reference transducer receives its reference torque by calibration within its planned application and mounting conditions using a suitable transfer method. Its task is to reproduce this reference torque as closely and accurately as possible. On the other hand if the transducer incorporated in the test bench is calibrated directly by the transfer transducer, this transducer itself assumes the function of a reference transducer as defined.

Optimizing the transfer of the measured quantity

A natural basic prerequisite for ensuring an optimum transfer of the measured quantity torque into the application – a test bench – is an unbroken chain of traceability back to the national standard.

In practice it is particularly important that the transducer which performs the transfer from the calibration laboratory into the application in this chain should not have incorrect measurement properties due to incorrect mounting conditions in the calibration equipment. In the absence of any on-site calibration, transfer takes place through the transducer to be retained in the test bench. In this case the mounting instructions for rotating torque transducers must be followed, as described in chapter 5. The guidelines on designing flange connections and minimizing parasitic loads are particularly important in this connection. It should be stressed at this point that it can be very worthwhile to include adaptation parts (such as application-side flanges) in the calibration. Further requirements specific to individual products can be found in the respective product documentation. A detailed study of these influences and effects can be found in [20].

The use of torque transfer transducers in shaft form that are actually optimized for use in calibration laboratories requires special care. Their design is such that sensitive components like the strain gages are not protected by a solid housing. Bending moments cause relatively high deformation compared to torque transducers in flange form. Such transducers are optionally delivered with an additional capability for measuring bending moments so that this influence quantity can be monitored. However, this kind of monitoring is only worthwhile if the mechanical installation allows steps to be taken to minimize bending moments. The same applies to an optional internal temperature measurement point.

Torque reference transducers in flange form, in which the geometry of the measuring body is the same as that of contactless, rotating torque transducers, are available for use in calibration in industrial conditions. The influences on the measurement properties and associated remedial measures correspond to those for the rotating torque transducers described above and in greater detail in chapter 5.

7.6 Dynamic calibration

7.6.1 Definition

Nowadays torque transducers for the test bench engineering field in the widest sense are as a rule calibrated on a purely static basis, even though in practically all cases they are used dynamically, as explained in chapter 4. From the strain gage principle of measurement it is known that by and large they are equally effective for static and dynamic loads, so this procedure is certainly justified to a very close approximation. Nevertheless, in the light of ever-growing demands for greater accuracy and correspondingly high requirements for traceability of the measured quantity, the question of truly dynamic calibration gains in importance. The aim of this section is to point out the fundamental questions that have to be formulated in any discussion about dynamic calibration. At the present time explicit recommendations or ready-defined procedures do not yet exist.

7.6.2 Defining the concepts of dynamic calibration

Dynamic calibration in the narrowest sense will be taken to mean that during calibration the applied torque undergoes rapid variations over time, and these correspond in their dynamics to the variations that may take place over time in the course of normal operation.

Thus it is necessary not only to determine the reference torque but also to measure the output from the transducer being calibrated under the special condition of continuously varying torque. This requires special attention to be paid to the simultaneity of the measurement. When torque is varying rapidly over time, signal delays in the amplifier electronics can also have an effect unless exactly the same type of amplifier is used for both the reference and the calibration object. Different signal delays can also be the result of different filter settings or filter characteristics.

Lever-arm-mass systems have to be ruled out as a possible source for the reference torque. It is usual to work with torque reference transducers. It is also conceivable to compute the torque from the rotary acceleration which torque causes on a body with a known mass moment of inertia.

Another special case of calibration under special conditions is calibration during rotation, which is often mentioned in close connection with or even confused with dynamic calibration, even though rotary motion is not in itself the deciding factor for whether a torque is static or dynamic. Nevertheless the association is to some extent justified, since dynamic torque components can never be completely avoided during rotation (see chapter 4). Within the context of a systematic procedure, however, it is necessary to make the distinction between static and dynamic torque when calibrating under rotation. Even static calibration during rotation represents a task for which there are not yet even the beginnings of a substantiated procedure. Dynamic calibration during rotation presents even more difficulties compared to the static version. The state of the art is merely to determine the effect which the speed of rotation has on the zero signal.

7.6.3 Continuous calibration

As well as actual dynamic calibration, it is also necessary to distinguish continuous calibration, in which the torque is varied continuously rather than in steps. Here too attention must be paid to simultaneity when determining the reference torque and measuring the output from the transducer being calibrated, but the dynamics of the torque variation over time are significantly reduced.

If the torque is generated using mechanically driven means and a certain amount of automation, continuous calibration saves a great deal of time.

A Terms and expressions for specifying torque transducers

A.1 Metrological properties of the torque measuring system

Accuracy class

The accuracy class declared for HBM torque transducers means that the maximum of those deviations specified as percentages is equal to or less than the value declared as the accuracy class. The sensitivity tolerance is not included.

The accuracy class includes the following metrological properties explained in detail below:

- Linearity deviation including hysteresis (d_{lh})
- Relative standard deviation of repeatability (σ_{rel})
- Temperature effect (per 10 K) on the zero signal (TK_0)
- Temperature effect (per 10 K) on the sensitivity (TK_c)

In transducers with two or more electrical outputs (frequency output and voltage output) the output with the highest accuracy is the deciding factor in determining the accuracy class. The accuracy class must not be mistaken for classification according to DIN 51309 or EA-10/14.

The accuracy class provides practical guidance on the respective type series' grouping within the HBM range of products. It must not be mistaken for the overall accuracy in practical use, with different individual influences acting at the same time.
Example:
We look at two versions of the T10F torque flange: on the one hand, option "S" (standard version) and on the other hand, option "G" (that is, reduced linearity deviation incl. hysteresis) in each case for the measuring ranges from 100 N·m to 10 kN·m.
In the data sheet the following maximum values are specified for the "S" version: 0.05 % for the temperature effect on the zero signal (TK_0), 0.1 % for the temperature effect on sensitivity (TK_C), and ±0.1 % for the linearity deviation incl. hysteresis (d_{lh}). Due to the two values given last, the accuracy class is specified as 0.1. Version "G", however, offers an improved linearity deviation incl. hysteresis (d_{lh}) amounting to 0.05 % only.

The temperature effect on sensitivity (TK_C) is still 0.1 % and thus is the maximum deviation among those given as percentages. Therefore, the accuracy class for version “G” still has to be specified as 0.1. Apparently, version “G” does not provide any increased benefit. However, it shows the relatively biggest effect only with regard to one characteristic value, the TK_C. In addition, this characteristic value is the only measure for deviations related to the actual value. As a consequence, its influence is considerably smaller, for example, with measurements in the partial load range (see below).

Sensitivity *C*

The span between the output signal values at nominal torque and at zero torque. Usually two separate sensitivities are specified for HBM torque transducers, one for clockwise torque and one for counterclockwise torque.

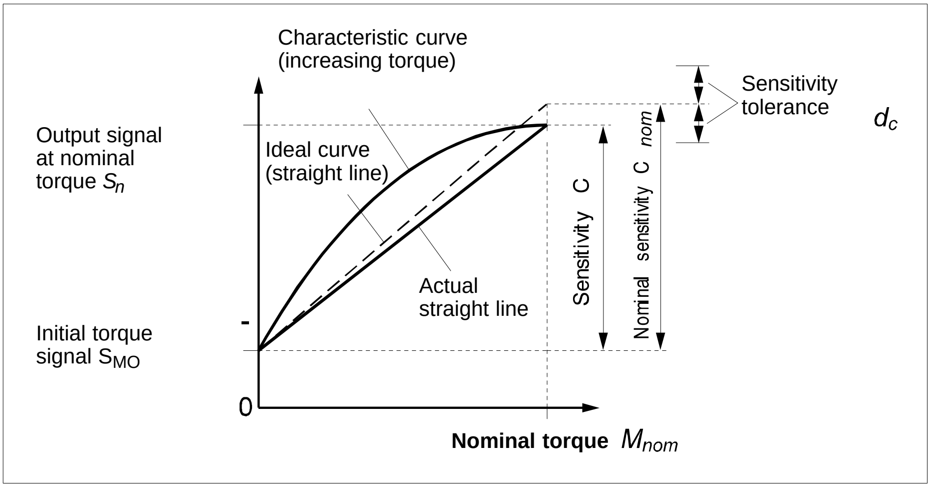


Fig. A.1 Sensitivity and nominal torque

The sensitivity C characterizes the slope of the characteristic curve. The characteristic curve is chosen as the straight line connecting the output signal S_{M0} determined with mounted but unloaded torque transducer (initial torque signal) and the output signal S_n at nominal torque determined at increasing torque. This gives the simple equation

$$C = S_n - S_{M0}$$

The sensitivity and the nominal torque form a known pair of values combining a given torque and the respective span of the output signal. If two such pairs of values are given they can be used for setting the amplifier. Usually, the second pair of values is zero torque and zero output signal span (i.e. output signal = initial torque signal).

Nominal sensitivity C_{nom}

The nominal value characterizing the transducer's sensitivity. Usually it is equal for clockwise and counterclockwise torque.

The nominal sensitivity is a value characterizing the respective transducer's type and measuring range. However, the actual sensitivity of the individual is equal to the nominal sensitivity only within specified tolerances.

Sensitivity tolerance d_C

The permissible deviation of the actual sensitivity from the nominal sensitivity. It is given as a percentage with respect to the nominal sensitivity.

For HBM torque transducers the actual sensitivity of the individual is determined before delivery. The value is documented in the test certificate or calibration certificate. For this reason the sensitivity tolerance is not taken into consideration when determining the accuracy class.

Temperature effect on the sensitivity TK_C

The temperature effect on the sensitivity is the variation of the actual output signal due to a 10 K change in temperature determined at nominal torque and related to the sensitivity. The specified value is the maximum occurring in the nominal temperature range.

The temperature effect on the sensitivity (also called the temperature coefficient of sensitivity) is a measure of the temperature effect on the output signal with a load applied to the transducer. This value is determined by measuring the transducer's actual output signal with a constant torque being applied. The value for the temperature effect is the variation in the output signal due to a 10-K change in temperature after re-establishment of a stationary temperature state.

The significant temperature is the transducer temperature. A stationary temperature state as defined at HBM means that the maximum temperature variation in a 15-minute period does not exceed 0.1 K. The amount of the deviation is given as a percentage of the actual span of the output signal with the respective torque applied (in the event of loading with the nominal torque this is the sensitivity).

The temperature effect on sensitivity results in a change of slope of the characteristic curve (see Fig. A.2). It is of particular importance when a transducer is operated at a temperature differing significantly from the reference temperature. For partial load ranges, however, it has very little effect because the resulting deviation acts always as a percentage of the actual output signal span.
Please note that normally the temperature effect on sensitivity and the temperature effect on the zero point (TK_0) are superimposed on each other.
<u>Example:</u>
Consider a torque transducer with 1 kN·m nominal torque, let the temperature effect on the sensitivity be specified as $TK_C \leq 0.1$ %, the reference temperature as 23 °C and the nominal temperature range from +10 °C to +60 °C.
If the transducer is operated at a temperature of 33 °C (or 13 °C), the sensitivity deviation due to the temperature variation may amount to up to 0.1 %. For a torque of 1 kN·m (nominal torque) this amounts to a deviation in the displayed value of 1 N·m. For a torque of 200 N·m, however, the deviation amounts to 0.2 N·m only, since the TK_C is always a percentage deviation referring to the actual output signal span. This is due to the fact that the sensitivity is referred to as the measure of the slope of the straight line. Using the same transducer at 43 °C (20 K deviation from the nominal temperature) may result in a maximum deviation of up to 0.2 % in the worst case. This does not apply to usage at 3 °C, since this temperature is not within the nominal temperature range.

Temperature effect on the zero signal TK_0

The temperature effect on the zero signal is the variation, due to a 10-K change in temperature, in the unloaded transducer's output signal related to the nominal sensitivity. The specified value is the maximum occurring in the nominal temperature range.

The temperature effect on the zero signal (also called the temperature coefficient of the zero signal) is determined by measuring the variation due to a 10–K change in temperature in the unloaded transducer's actual output signal at zero torque after re-establishment of a stationary temperature state. The significant temperature is the transducer temperature. A stationary temperature state as defined at HBM means that the maximum temperature variation in a 15-minute period does not exceed 0.1 K.

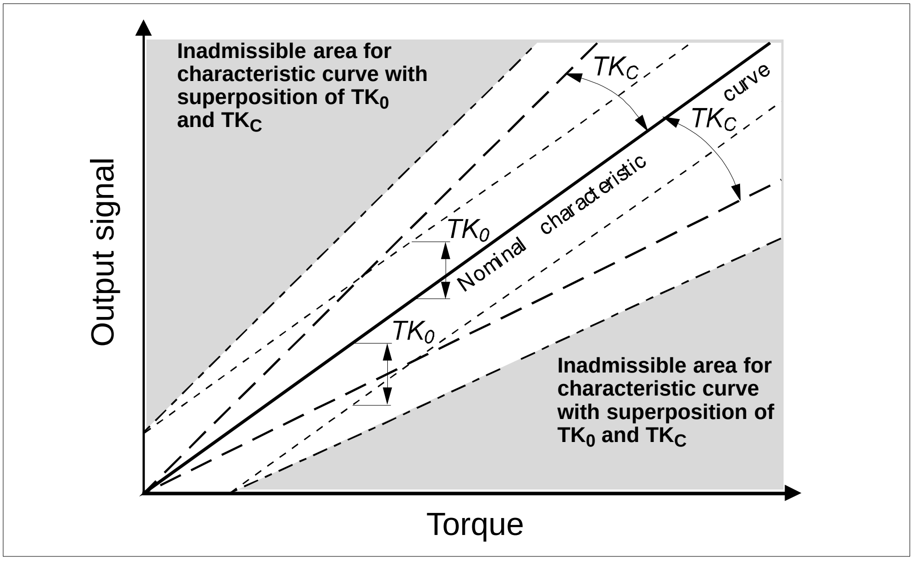


Fig. A.2 Temperature effect on the sensitivity TK_C and on the zero point TK_0.

The temperature effect on the zero signal results in a parallel shift in the characteristic curve (see Fig. A.2). It is of particular importance when a transducer is operated at a temperature differing significantly from the reference temperature. By taring or zero balancing at operating temperature, the measurement error due to the temperature effect on the zero signal can be eliminated.
Please note that normally the temperature effect on the zero point and the temperature effect on the sensitivity (TK_C) are superimposed on each other.
<u>Example:</u>
Consider a torque transducer with 1 kN·m nominal torque, let the temperature effect on the zero signal be specified as $TK_0 \leq 0.05$ %, the reference temperature as 23 °C and the nominal temperature range from +10 °C to +60 °C.
If the transducer is operated at a temperature of 33 °C (or 13 °C), the zero signal deviation may amount to up to 0.05 % of the nominal sensitivity.
This corresponds to a deviation in the displayed value of 0.5 N·m. This deviation is independent of the torque with which the transducer is loaded.
Using the transducer at 43 °C may result in a maximum deviation of up to 0.1 % in the worst case. This does not apply to usage at 3 °C, since this temperature is not within the nominal temperature range.

Linearity deviation d_{lin}

Absolute value of the maximum deviation of a torque transducer's characteristic curve determined with increasing load from the reference straight line which approximates the characteristic curve as a straight line. The specified value is expressed as a percentage of the sensitivity *C*.

For determining the linearity deviation, a series of measurements is taken with the load increasing from zero to the nominal torque. The reference straight line is the best-fit straight line through the initial point, such that the maximum deviations (upward/downward) from the measurement signal have the same amount (see Fig. A.3). The specified linearity deviation is the maximum deviation of the actual output signal from the reference straight line. It can also be described as half the width of the tolerance band that is symmetrical about the reference straight line.

The linearity deviation has to be taken into consideration because usually when adjusting the measurement chain, a characteristic curve in the form of a straight line is assumed. It takes maximum effect when a transducer is used for a wide measuring range, in the most extreme case from zero torque up to nominal torque.

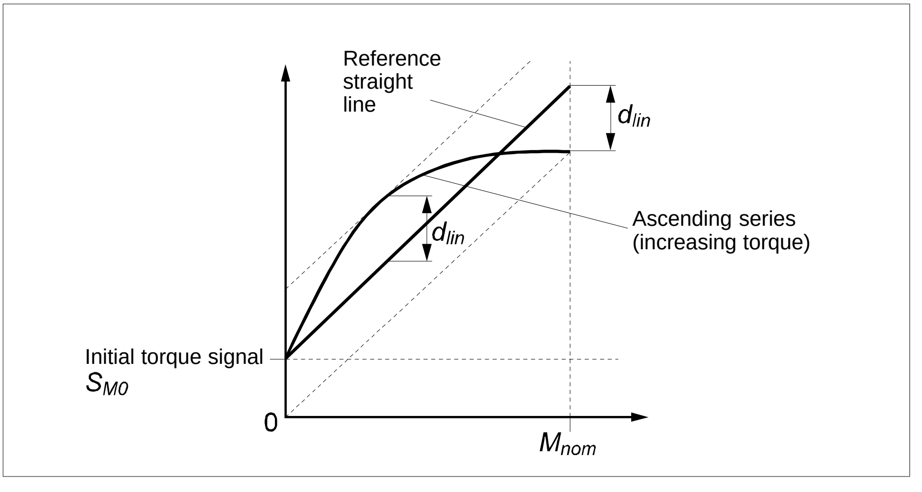


Fig. A.3 Determination of the linearity deviation d_{lin}

Relative reversibility error d_{hy}

The relative reversibility error is the difference of the output signals when measuring the same torque applied in increasing and decreasing steps (see Fig. A.4). The specified value is the maximum deviation (according to absolute value) in the measuring range. It is specified as a percentage of the sensitivity *C*.

The relative reversibility error is a measure of hysteresis, that is, the difference between the characteristic curves determined with increasing and decreasing torque. For determining the relative reversibility error, a load cycle from zero torque through nominal torque and back is recorded. The practical calculation is based on measurements at a number of predefined points in the load cycle (e.g. 0 %, 50 %, 100 % of M_{nom}).

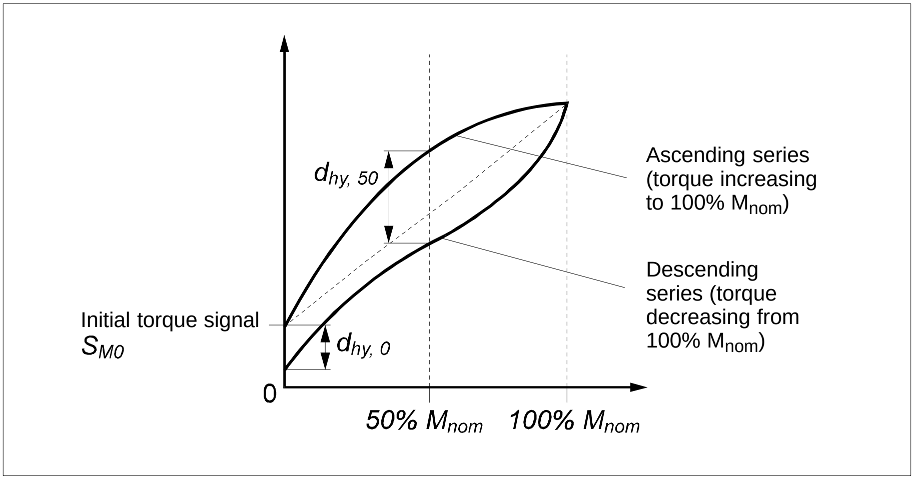


Fig. A.4 Determination of the relative reversibility error d_{hy} from a load-relieve cycle (here based on the load steps 0 %, 50 %, 100 % M_{nom}). The value to be specified is the maximum reversibility error of the given load steps (here $d_{hy,0}$ and $d_{hy,50}$)

Hysteresis describes the dependency of the measuring signal on the transducer's loading history. It is of particular importance if a transducer is used for a wide measuring range and no unloading takes place between acquiring two relevant measurement points. The most extreme case is the use from zero torque up to nominal torque. The effect of hysteresis occurring during a partial load cycle is usually significantly smaller than the hysteresis during a load cycle covering the entire nominal torque range.

Linearity deviation including hysteresis d_{lh}

The linearity deviation including hysteresis specifies the maximum deviation (according to absolute value) of the output signal value from the reference straight line. The reference straight line is the best-fit straight line through the starting point (see Fig. A.5). Thus, both linearity deviation and hysteresis are taken into consideration. The specified value is expressed as a percentage of the sensitivity *C*.

The load cycle for determining the linearity deviation including hysteresis covers the loading of the transducer from zero up to nominal torque and the subsequent relieving down to zero torque (see Fig. A.5). The reference straight line is the best-fit straight line through the initial point, such that the maximum deviations (upward/downward) from the measurement signal have the same amount.

The linearity deviation including hysteresis can also be interpreted as half the width of the tolerance band that is symmetrical about the reference straight line (see Fig. A.5). The only difference to the procedure for the determination of the linearity deviation d_{lin} is that the load cycle here includes also the measurements at decreasing torque. This difference takes effect on both the calculation of the reference straight line and the deviations from the reference straight line.

To determine this value, HBM proceeds as described below:

- The transducer is preloaded with counterclockwise torque in three load cycles from zero to 100 % of the nominal torque and back to zero torque. The purpose of this preloading is to eliminate the influence of mounting like settling of bolts and smoothing of contact surfaces
- One load cycle with counterclockwise torque and recording of the respective values for the measurement signal at the predefined load steps (when testing during production at HBM these steps are at torque 0 %, 50 %, 100 %, 50 % and 0 % of M_{nom})
- The transducer is preloaded with clockwise torque in three load cycles from zero to 100 % of the nominal torque and back to zero torque
- One load cycle with clockwise torque and recording of the respective values for the measurement signal at the predefined torque steps
- The best-fit straight line is calculated according to the above definition, separately for clockwise torque and counterclockwise torque
- The amount of the maximum deviation from the best-fit straight line is determined separately for clockwise torque and counterclockwise torque

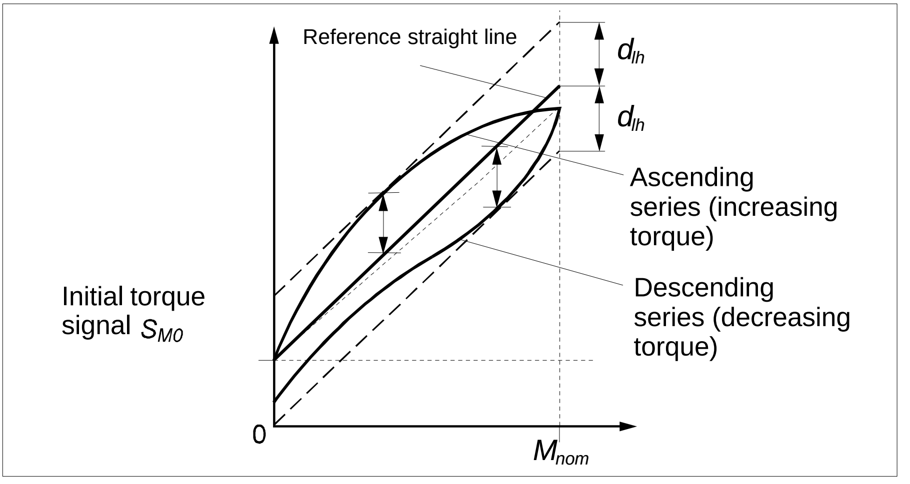


Fig. A.5 Determination of the linearity deviation including hysteresis d_{lh} from a load-relieve cycle

The linearity deviation including hysteresis is of importance because normally when adjusting the measurement chain, a characteristic curve in the form of a straight line is assumed. It takes maximum effect if a transducer is used for a wide measuring range and it is not relieved between two relevant measurements. The most extreme case is the use from zero torque up to nominal torque.

Example:

Consider a T10FS torque flange of which the maximum permissible linearity deviation including hysteresis is specified as $d_{lh} \leq 0.05$ % and the nominal sensitivity amounts to 5 kHz. Provided that the measurement chain has been adjusted optimally, the output signal error due to the linearity deviation and hysteresis may amount to 2.5 Hz at most.

Relative standard deviation of repeatability σ_{rel}

Repeatability describes the property that the output signal is the same for all measurements of the same torque when measured several times. During the measurements the mounting position of the torque transducer shall remain unchanged and the torque transducer shall not be mounted and dismounted (repeat conditions). The standard deviation indicates the average deviation between all measurements of the same torque when measured several times.

The relative standard deviation of repeatability is a measure of the repeatability according to DIN 1319. It is defined as the standard deviation of repeatability according to DIN 1319 and is expressed as a percentage of the range of the signal span covered during the testing procedure. It is a statistic measure of random measurement deviations. For this reason, all those conditions are kept constant that, if changed, might cause variations in systematic measurement deviations (repeat conditions to DIN 1319).

The determination of the relative standard deviation of repeatability is a type test carried out on a static calibration system as follows:

- The torque transducer is preloaded up to nominal torque. The measurement signal $S_{1,100\%}$ is taken at nominal torque
- The load is reduced to 50 % of the nominal load. The measurement signal $S_{1,50\%}$ is taken at half the nominal torque
- Alternation between 50 % and 100 % of nominal torque. The measurement signals $S_{i,50\%}$ and $S_{i,100\%}$ are taken until 10 measured values each are available for each torque
- The equation below is used to calculate the relative standard deviation (in mathematical terminology, the empirical standard deviation of a random sample) for each torque and relate it to the output signal span:

$$\sigma_{rel,50\%} = \frac{1}{\bar{S}_{100\%} - \bar{S}_{50\%}} \sqrt{\frac{1}{n-1} \sum_{i=1}^{n} \left(S_{i,50\%} - \bar{S}_{50\%}\right)^2}$$

and

$$\sigma_{rel,100\%} = \frac{1}{\bar{S}_{100\%} - \bar{S}_{50\%}} \sqrt{\frac{1}{n-1} \sum_{i=1}^{n} \left(S_{i,50\%} - \bar{S}_{50\%}\right)^2}$$

with n=10 being the number of measurements for the respective torque applied (50 % or 100 %) and the arithmetic mean of the measurement signals according to

$$\bar{S}_{50\%} = \frac{1}{n} \sum_{i=1}^{n} S_{i,50\%} \qquad \text{and} \qquad \bar{S}_{100\%} = \frac{1}{n} \sum_{i=1}^{n} S_{i,100\%}$$

- The technical data specifies the inferior of the two values $\sigma_{\mathrm{rel},50\%}$ and $\sigma_{\mathrm{rel},100\%}$.

Example:
Consider a T10F torque flange with 1 kN·m nominal measuring range which is used in a test bench for combustion engines. Different settings are applied to the engine's control electronics, and the measurements taken are compared. The torque transducer assembly and ambient conditions remain unchanged. HBM specifies the standard deviation of repeatability for the T10F torque flange as $\sigma_{rel} \leq 0.03$ %. This value refers to the output signal span between each torque applied. Here this corresponds to a span of displayed values of 500 N·m. Thus, the maximum standard deviation of repeatability amounts to 0.15 N·m.
This specifies the permissible change due to statistic variation. Further torque signal differences result from different instances of torque, statistical variations in the application, or systematic (as opposed to random) deviations.

A.2 Ambient conditions and load limits

Nominal rotation speed

The nominal rotation speed is the upper limit of the speed range starting from zero. It applies to both clockwise and counterclockwise rotation.

Nominal torque M_{nom}

The nominal torque is the torque defining the upper limit of the range in which specified tolerances of the transducer properties are not exceeded.

Maximum service torque

The maximum service torque is the upper limit of the range in which there is an unambiguous relation between output signal and torque. If within this range torque is increased above the nominal torque the limit values declared in the specifications may be exceeded.

If the transducer has been used between nominal torque and maximum service torque, the limit values given in the specifications will be kept by the transducer when it is used again for torque values up to nominal torque. A slight shift of the zero signal may occur, but this is not considered to be a violation of the specifications.

The torque transducer can be used for measurements up to the maximum service torque, though the measurement properties may prove less favorable.
The limitation on maximum service torque may be given by electronic properties (such as the modulation range of the internal amplifier electronics) or by mechanical properties (such as an overload stop). In the case of transducers that have neither internal electronics nor mechanical overload protection, the maximum service torque and the limit torque are frequently identical.

Limit torque

The limit torque is the torque up to which the transducer's measuring capability will not suffer permanent damage.

If the transducer has been used between nominal torque and limit torque, the limit values given in the specifications will be kept by the transducer when it is used again for torque values up to nominal torque. A slight shift of the zero signal may occur, but this is not considered to be a violation of the specifications.

In the event of a continuous vibrating load, the limits discussed below in the section on permissible oscillation bandwidth have priority over the limit torque.

Breaking torque

The breaking torque is the torque which when exceeded may lead to mechanical destruction of the transducer.

In the event of torque values between limit torque and breaking torque no mechanical destruction will occur, but the transducer may be damaged to such an extent that it will be permanently unusable.

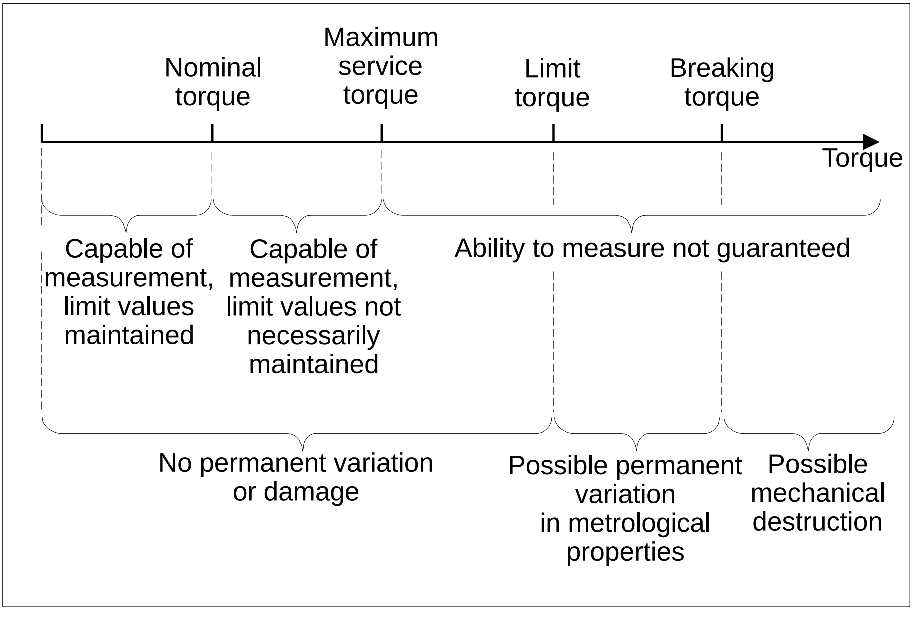


Fig. A.6 Load limits

Permissible oscillation bandwidth

The permissible oscillation bandwidth is the oscillation amplitude of a sinusoidally varying torque with which the transducer can be stressed for $10 \cdot 10^6$ vibration cycles without causing any significant variations in its metrological properties.

The amplitude is specified as a peak-to-peak value, that is, as the difference between maximum and minimum torque. See also Fig. A.7.

As well as the permissible vibration bandwidth it is also necessary to define a permissible upper limit for the torque which occurs. This upper limit usually coincides with the nominal torque (in both the positive and negative direction). Values which differ from this are explicitly declared in the specifications.

The concept has been taken from standard DIN 50100, which deals with continuous vibration testing (fatigue testing) within the context of materials testing, and has been transferred from mechanical stress to torque.
The deciding factor for fatigue strength is the number of vibration cycles alone. The frequency is not significant within the frequency range that is relevant to mechanical processes. See [21]. According to DIN 50100, it can be assumed to a close approximation for the case of steel materials that a mechanical component is fatigue proof under a given load if it endures the number of $10 \cdot 10^6$ load cycles under the respective load.
The upper limit for torque in the case of vibrational loading replaces explicit information about the mean vibrational loading. Within the range defined by the positive and negative limits, both pulsating torque and alternating torque are permissible (see Fig. A.7).

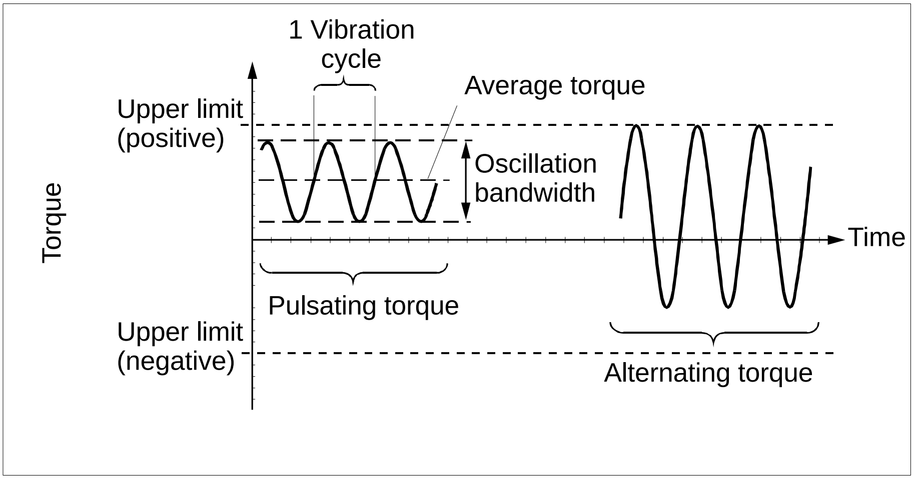


Fig. A.7   Terms used in connection with the oscillation bandwidth

Axial limit force

The axial limit force is the maximum permissible longitudinal force (or axial force), shown as F_a in Fig. A.8. If the axial limit force is exceeded, the ability of the transducer to measure may be permanently damaged.

- In HBM torque transducers the axial limit force sets an upper limit to the service range. Torque transducers can be used for measurement if the axial forces do not exceed the axial limit force. However, there may be some effect on the measurement signal. The upper limit for this influence is separately notified in the specifications.

- The permissible axial force gets smaller than the specified axial limit force if another irregular stress occurs (such as a bending moment, lateral force or exceeding the nominal torque). Otherwise the limit values have to be reduced. If for example 30 % of both the bending limit moment and the lateral limit force occur, only 40 % of the axial limit force is allowed in the case that the nominal torque is not exceeded. If parasitic loads occur as continuous vibrating loads, the respective permissible vibration bandwidths may differ from the respective limit loads.

Lateral limit force

The lateral limit force is the maximum permissible lateral force (in the case of radial force), shown as F_r in Fig. A.8. If the lateral limit force is exceeded, the ability of the transducer to measure may be permanently damaged.

In HBM torque transducers the lateral limit force sets an upper limit to the service range. Torque transducers can be used for measurement if the lateral forces do not exceed the lateral limit force. However, there may be some effect on the measurement signal. The upper limit for this influence is separately notified in the specifications.

The permissible lateral force gets smaller than the specified lateral limit force if another irregular stress occurs (such as an axial force, bending moment or exceeding the nominal torque). Otherwise the limit values have to be reduced. If for example 30 % of both the axial limit force and the bending limit moment occur, only 40 % of the lateral limit force is allowed in the case that the nominal torque is not exceeded. If parasitic loads occur as continuous vibrating loads, the respective permissible vibration bandwidths may differ from the respective limit loads.

Bending limit moment

The bending limit moment is the maximum permissible bending moment, shown as M_b in Fig. A.8. If the bending limit moment is exceeded, the ability of the transducer to measure may be permanently damaged.

In HBM torque transducers the bending limit moment sets an upper limit to the service range. Torque transducers can be used for measurement if the bending moments do not exceed the bending limit moment. However, there may be some effect on the measurement signal. The upper limit for this influence is separately notified in the specifications.

The permissible bending moment gets smaller than the specified bending limit moment if another irregular stress occurs (such as an axial force, lateral force or exceeding the nominal torque). Otherwise the limit values have to be reduced. If for example 30 % of both the axial limit force and the lateral force limit occur, only 40 % of the bending limit moment is allowed in the case that the nominal torque is not exceeded. If parasitic loads occur as continuous vibrating loads, the respective permissible vibration bandwidths may differ from the respective limit loads.

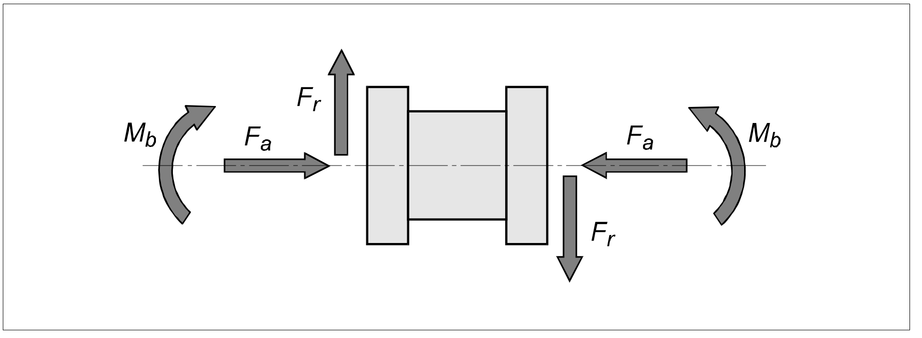


Fig. A.8 Parasitic loads: axial force F_a, lateral force F_r, bending moment M_b

Reference temperature

The reference temperature is the ambient temperature at which the transducer specifications apply as far as there are no temperature ranges specifically defined in which the specifications concerned apply.

Nominal temperature range

The nominal temperature range is the ambient temperature range within which the transducer can be operated for all practical purposes and within which it maintains the limit values for the metrological properties declared in the specifications.

Service temperature range

The service temperature range is the ambient temperature range within which the transducer can be operated without permanent variations occurring in its metrological properties.

At temperatures within the service temperature range but outside the nominal temperature range, there is no guarantee that the limit values declared for the metrological properties in the specifications will be maintained.

Storage temperature range

The storage temperature range is the ambient temperature range within which the transducer can be stored with no mechanical or electrical load without permanent variations occurring in its metrological properties.

B Brief summary of vibration engineering

This appendix gives a concise introduction to the fundamental concepts of mechanical vibration theory. It is therefore no substitute for a textbook on engineering mechanics or the theory of vibrations. Instead it presents the concepts and basic results needed in order to follow the discussions about the vibrational responses in rotating machines. Readers already familiar with these basic concepts will find explanations of the notation used throughout the book.

B.1 Examples of vibrating systems

This section first of all presents vibrating systems with one degree of freedom in order to explain the basic effects.

The standard example from mechanical vibration theory is the single mass oscillator drawn in Fig. B.1, consisting of a rigid body with mass *m* which can make translational movements in one direction (displacement coordinate *x*).

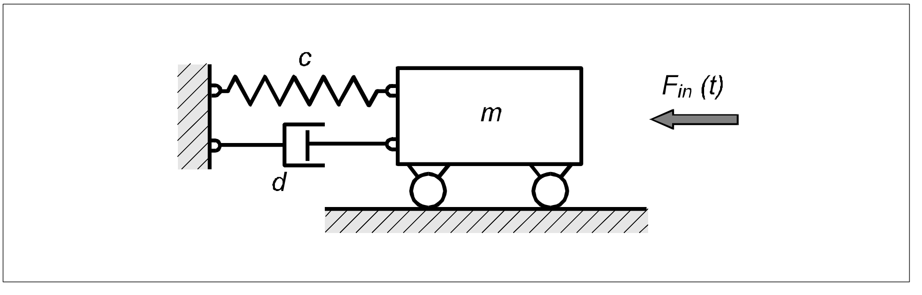


Fig. B.1 Single mass oscillator consisting of a body with mass, a linear spring and a linear viscous damping device

This body is held by an elastic spring with a restoring force which is defined with the aid of spring constant *c* and the linear spring law

$$F_c = -c\,x$$

and is idealized as massless. The negative sign in the law expresses the fact that the restoring force of the spring always acts against the direction of the displacement.

In addition to the elastic spring, a damping effect also acts on the body. This is represented as an idealized linear viscous damper with a braking force which is proportional to velocity and defined with the aid of the damping constant *d* and the equation

$$F_d = -d\dot{x}$$

which expresses the damping law. The dot superscript is short notation for the time derivative, whereby $\dot{x}$ represents the velocity. The negative sign in the law expresses the fact that the damping force always acts against the direction of movement.

In addition to these internal system forces, an external excitation force F_{in} is also included for the general case and is a function of time *t*. By using the sum of all the forces described above in the Newtonian law of motion (force equals mass times acceleration)

$$F = m\,a$$

we obtain the equation describing the motion of the vibrating system

$$m\,\ddot{x} + d\,\dot{x} + c\,x = F_{in}(t)$$

where the acceleration is expressed as the second time derivative of the displacement coordinate

x, again symbolized by means of the convention of the dot superscript. In terms of mathematics this equation describing the motion of the mechanical system is a differential equation of the second order dependent on time. For the sake of clarification this book also often refers to this concept as a differential equation of vibration. This equation is used here by way of example as the starting point for a discussion on the various effects and phenomena.

Equations with precisely the same structure can also occur for systems with various other mechanical arrangements and for electrical oscillating circuits.

The simplest example of this type is the electrical oscillating circuit illustrated in Fig. B.2, consisting of a coil (inductance *L*), capacitor (capacitance *C*) and ohmic resistor (resistance *R*). In this instance an external voltage *U(t)* acts as an excitation.

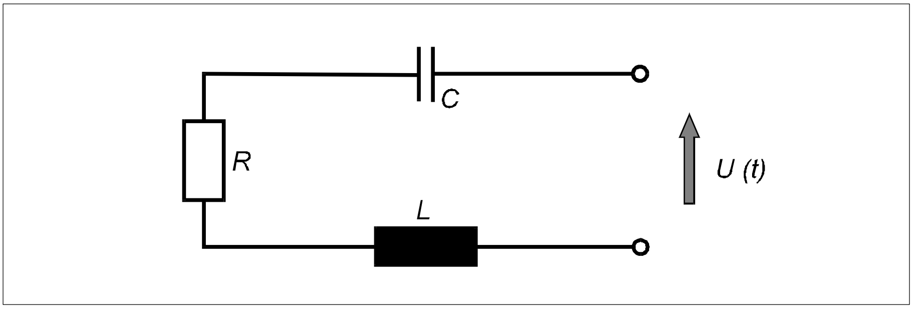


Fig. B.2 Electrical oscillating circuit

A load q produces the following differential equation of vibration:

$$L\,\ddot{q} + R\,\dot{q} + \frac{1}{C}q = U(t)$$

The analogy with the equation for the single mass oscillator can clearly be seen. Consequently the general propositions on vibration phenomena explained below in connection with the example of a single mass oscillator (for example resonance behavior) are also valid for electrical RLC oscillating circuits.

B.2 Free vibration

Free vibration refers to vibration that occurs without the excitation being continuously present. It is therefore necessary to find a mathematical solution to the differential equation of vibration for the case $F_{in}(t) \equiv 0$. This is known as solving the homogeneous differential equation, for which the index h is used for the purpose of identification. This motion occurs in the form of vibrations continuing after an initial displacement or velocity. In practice for instance, this can occur in the system shown in Fig. B.1, if the wagon in the drawing is first displaced by an external force and then released.

The solution is specified without further derivation. The reader is referred to the literature on engineering mechanics [23] and technical vibration theory [15].

B.2.1 Undamped free vibration

First of all consider the undamped free vibration that occurs in systems where the damping constant d is zero. In this case the displacement is described by

$$x_h(t) = \hat{x}_h \cos(\omega_0 t + \beta)$$

with the aid of constants $\hat{x}_h$, ω_0 and β. It can be seen that this solution always has the form of sinusoidal movements, also known as harmonic vibrations, as shown in Fig. B.3.

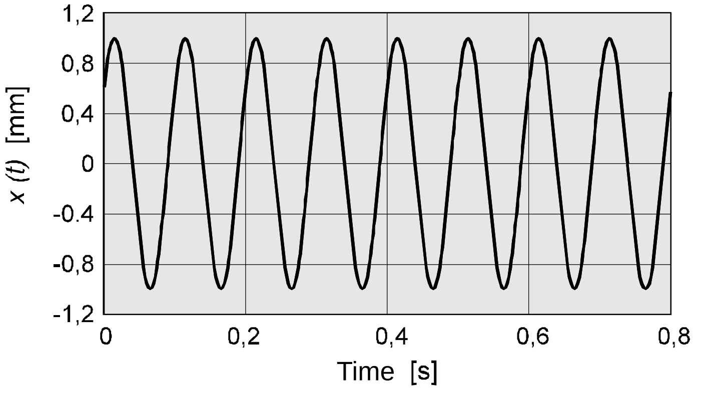


Fig . B.3 Undamped free vibration

Among the constants, the natural angular frequency ω_0 is especially important. For the mechanical system in the example it is given by

$$\omega_0 = \sqrt{\frac{c}{m}}$$

It is expressed in the unit s^{-1} or rad/s and is converted via the equation

$$f_0 = \frac{\omega_0}{2\pi}$$

into the natural frequency f_0, which is in the unit Hz. The natural frequency or natural angular frequency is a decisive characterizing quantity of every vibrating mechanical system. Since it depends only on the system parameters stiffness and mass, it too is a system parameter which makes it substantially different than the constants amplitude $\hat{x}_h$ and phase angle ψ. The latter are integration constants that depend on the initial conditions. They therefore have to be redefined for every case of vibration within the same system. The reader should refer to the literature for details of the computations involved.

B.2.2 Damped free vibration

In the case of free vibration in systems with damping one has to distinguish between a number of different cases. Of these, only the case relevant to vibrations in the strict sense will be considered, and this is the case known as “weak damping”.

In this case vibrational displacements $x_h(t)$ obey the equation

$$x_h(t) = K\, e^{-\omega_0 D t} \cos\left(\sqrt{1-D^2} \cdot \omega_0\, t + \gamma\right)$$

and their curve over time is sketched in Fig. B.4.

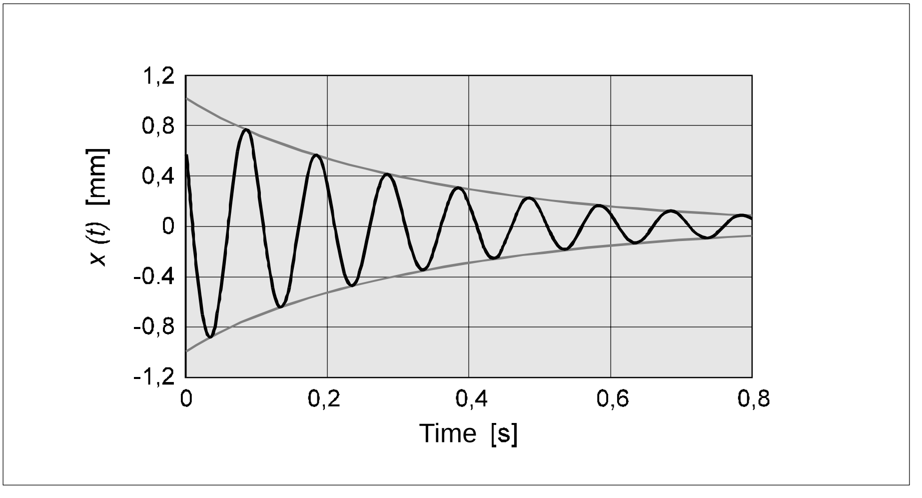


Fig. B.4 Weakly damped free vibration

The damping factor D occurs as a supplementary system parameter. For the simple mechanical oscillator in the example, this takes the form

$$D = \frac{d}{2\sqrt{c\,m}} = \frac{d}{2\,m\,\omega_0}$$

The restriction to weak damping mentioned above means that the damping factor obeys the expression $0 < D < 1$.

As both the equation for $x_h(t)$ and Fig. B.4 show, the movements can be represented by an envelope curve in which the sinusoidal vibration dies out exponentially. Thus the amplitude never reaches a value of nil in finite time.

The equation shows that the frequency of the free vibrations

$$\omega_d = \omega_0 \sqrt{1 - D^2}$$

in a damped system is the natural frequency of the damped vibrations, which in a strict sense differs from that of the undamped vibrations. Nevertheless for characterizing the system, natural frequency ω_0 is often used even in damped systems, which means that the damping is disregarded. To distinguish it from the mathematically exact natural frequency ω_d it is known in the context of damped systems as the characteristic frequency.

Since the figures involved in damping are generally quite small, the difference between characteristic frequency and natural frequency in a damped system is often negligible. Typically in many applications a damping factor of D = 10 % is very heavy damping, but the effect on the natural frequency is a mere 1 %.

Two integration constants also occur in damped free vibration just as in undamped free vibration, and are also dependent on the initial conditions. One is characteristic of the phase position, the other of amplitude. Since amplitude changes over time when damped vibration is dying out, the integration constant itself cannot be directly equated with amplitude.

B.3 Forced vibration

Forced vibration is unlike free vibration in that it occurs as a result of an excitation. In the mechanical system shown by way of example in Fig. B.1, it is represented by the excitation force $F_{in}(t)$. This means that in this case it is necessary to solve an inhomogeneous differential equation. Specialized solutions for the inhomogeneous differential equation are known as particular solutions, for which the index p is used for the purpose of identification. As before the solutions are given without further derivation. The reader is referred to the literature on engineering mechanics [22] and technical vibration theory .

To begin with it is helpful to limit consideration to harmonic excitation functions. Taking into account this limitation of the differential equation which describes the motion of the system shown as an example in Fig. B.1 can always be written in the form

$$m\,\ddot{x} + d\,\dot{x} + c\,x = \hat{F}_{in}\cos(\Omega t)$$

by choosing a suitable zero point for counting the time. The harmonic excitation is characterized here by the amplitude of the excitation force $\hat{F}_{in}$ and by the angular frequency Ω of the excitation. Both of these are parameters of the excitation and not properties of the vibrating system.

The task is now to find the mathematical relation between an excitation and the displacement it generates in the vibrating system. The term used in this connection is the vibrational response. Due to the linearity of the equation, it is always possible for a free vibration component to be superimposed on the solutions outlined here. Only the sum represents the complete solution to the inhomogeneous differential equation. This solution is only unambiguous when the initial conditions are taken into account, since these have an influence on system motion just as in the case of free vibration.

But specializing the general solution with respect to the initial conditions is usually of little relevance to the practical concepts during evaluation of design concepts and vibration analysis and will not be discussed further here. One reason for this is that in actual technical systems a certain amount of damping is always present, so that the free vibration component ultimately dies down. Among all conceivable solutions for the equation, therefore, discussion will center on the stationary state which describes the system response after a sufficiently long phase of transient vibration.

It can be shown that vibrational motion has the form

$$x_p(t) = \hat{x}_p \cos(\Omega t - \psi)$$

in the stationary state. The vibrational response in the case of harmonic excitation is therefore also a harmonic motion. Its angular frequency matches the excitation frequency Ω (and not the natural frequency ω_0 of the system, for instance). The two constants are $\hat{x}_p$, the displacement amplitude, and ψ, the angle of phase difference. By substituting the above expression into the initial equation and performing a few mathematical operations these constants can be determined as

$$\hat{x}_p = \hat{F}_{in} \frac{1}{\sqrt{(\omega_0^2 - \Omega^2)^2 + (2D\omega_0\Omega)^2}}$$

and

$$\psi(\Omega) = \arctan\left(\frac{2D\omega_0\Omega}{\omega_0^2 - \Omega^2}\right)$$

It can be seen that the amplitude of the excitation affects only the amplitude of the response, and not the phase angle. The effect on the amplitude is strictly linear, so that doubling the excitation amplitude leads to a doubling of the response amplitude. It is therefore worthwhile giving some thought to the ratio of the response amplitude to the excitation amplitude. Multiplying by the stiffness c gives a dimensionless ratio number known as the magnification function $V_A(\Omega)$. This function can be physically interpreted in two ways: first as the ratio of the displacement amplitude $\hat{x}_p$ to the hypothetical displacement $\hat{F}_{in}/c$ which a static force of the same value would cause, and secondly as the ratio of the spring force amplitude $c\,\hat{x}_p$ to the excitation force $\hat{F}_{in}$. The magnification function for the example is as follows:

$$V_A(\Omega) = c\,\frac{\hat{x}_p}{\hat{F}_{in}} = \frac{\omega_0^2}{\sqrt{\left(\omega_0^2 - \Omega^2\right)^2 + \left(2\,D\,\omega_0\,\Omega\right)^2}}$$

For a given vibrating system it is only a function of the excitation frequency, just like the phase shift function $\psi(\Omega)$. If both the functions for a vibrating system are known, the response amplitude can be specified for every harmonic excitation of any frequency and amplitude.

In the complex notation which will not be further explained here, the magnification function and the phase function can be combined into one complex function which has the form of a quotient between a complex response amplitude and a complex excitation amplitude. This is known as the transfer function. For clear and practical assessment it is very useful to display the magnification function and the phase shift function in the form of graphs, as shown in Fig. B.5 for the standard example from Fig. B.1.

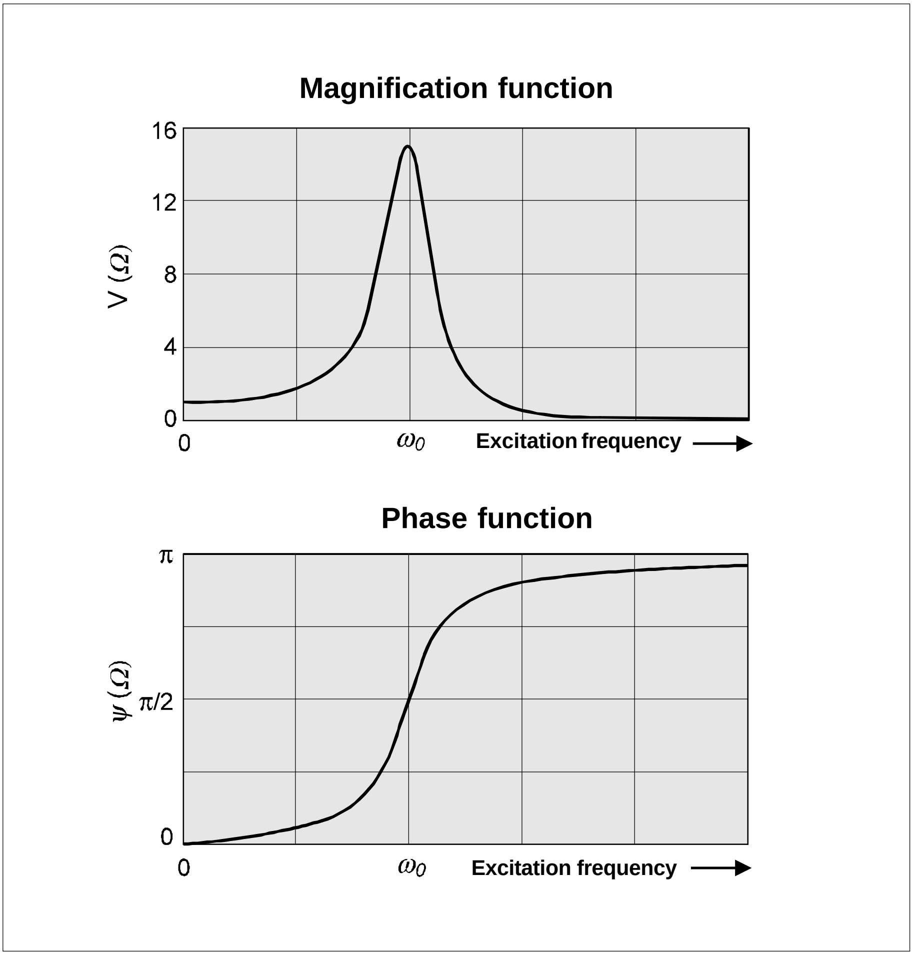


Fig. B.5 Magnification function and phase function

The known phenomenon of resonance can clearly be seen from the figure. When the excitation frequency approaches the natural frequency of the system, vibration amplitude increases sharply. The size of the increase in resonance depends on the system damping. At zero damping the amplitude of the vibrational response at resonance frequency approaches infinity.

The dependency of the increase in resonance on the damping factor D is shown in Fig. B.6 for selected values. In damped systems the frequency at which the peak in the amplitude curve occurs is to a small extent dependent on the damping, but in practice the characteristic frequency ω_0 is almost always a close approximation of the resonance frequency.

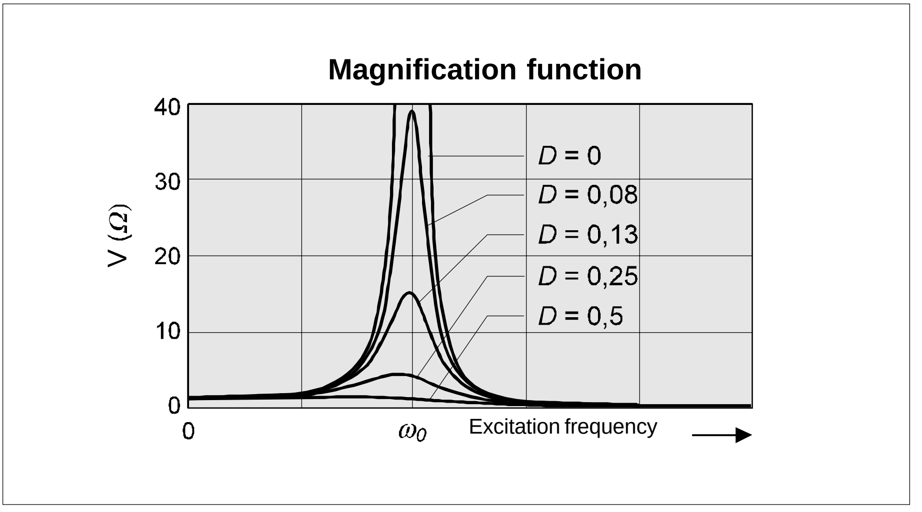


Fig. B.6 Magnification function for various damping constants

At excitation frequencies well below the resonance frequency, the magnification function approaches one and the phase shift approaches zero. The system behaves like a direct, non-vibrating link between the excitation and the response. The term used in this connection is the subcritical range. At excitation frequencies well above the resonance frequency, the magnification function approaches zero and the phase shift approaches π. The system vibrates out of phase with the excitation. At very high excitation frequencies therefore, the excitation causes no movement in the system due to system inertia. The term used in this connection is the supercritical range.

Without giving a detailed derivation it needs to be added that similar relationships also apply if the excitation type (input quantity) is not a force but, say, a movement in the foot of the spring or in the foot shared by the spring and the damper. By the same token, similar relationships also apply if the quantity being considered as the output quantity is not the displacement amplitude but, say, the amplitude of the velocity or the holding force of the spring and damper on the end wall. Of special note at this point is the case in the system shown by way of example in Fig. B.1, where both the input quantity and the output quantity are a force, as is the holding force on the end wall.

This case is of special practical interest for torque measurement. Simply by transferring this system to the corresponding system with rotary motion, it becomes an excitation by torque and an output quantity which is also torque. Excitation by torque corresponds to the usual case in practical rotating machines and the output quantity torque is of prime interest in practice, since it is not only the measured quantity but also the quantity that is important for estimating the loading on the transducer and on the whole shaft train. The curves for the magnification function $V_A(\Omega)$ and the phase function $\psi(\Omega)$ correspond qualitatively to those shown in Fig. B.5. The torque will also undergo a sharp increase in the resonance range, and this can overload the torque transducer or other components in the shaft train. In the subcritical range, where the magnification function is roughly one, the system acts like a connection through a rigid shaft section so that the excitation torque comes through to the other end of the vibrating system almost unaltered. In the supercritical range, where the magnification function is roughly zero, only a minimal part of the excitation torque comes through to the other end of the vibrating system.

In the case of non-harmonic excitation the system response can be determined by dividing the excitation function into harmonic components. The system response is then derived by summing the system responses on the individual harmonic components of the excitation function. This method is mathematically correct due to the linearity of the equations that describe the system response. The excitation function is divided into harmonic components with the aid of the Fourier transform, which is explained in chapter 6.

B.4 Systems with multiple degrees of freedom

All the matters considered and discussed so far were devoted to vibrating systems with one degree of freedom. They made it possible to clarify many of the basic concepts and phenomena connected with vibration engineering. A quantitative computation of an actual technical system is also often possible if it can be meaningfully mapped onto a simple mechanical substitution model with only one degree of freedom.

In many cases, however, the formulation of a meaningful mechanical substitution model requires a number of degrees of freedom to be taken into account. The vibrational displacements are then considered at several reference points, where the movements of each individual reference point cannot be directly expressed by the movements of the others.

The dual mass oscillator in Fig. B.7 illustrates the principle of a vibrating system with more than one degree of freedom. It can be seen that to describe the motion of the system it is necessary to specify the two displacement coordinates x_1 and x_2.

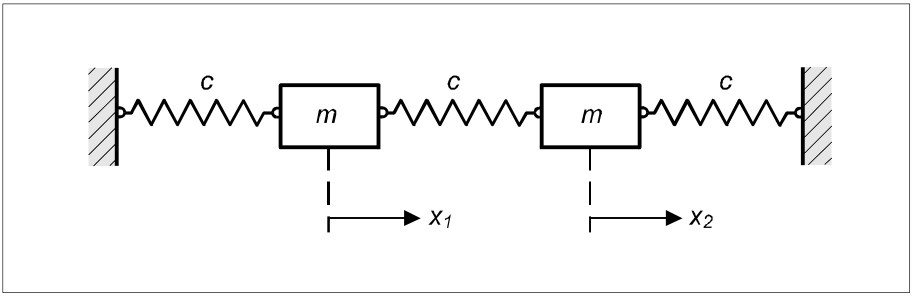


Fig. B.7 Dual mass oscillator

Two differential equations of motion are therefore also required, but these cannot be solved independently of one another. The term used in this connection is a system of coupled differential equations. The reader should refer to the literature for details of the mathematical treatment involved. The simple aim of this section is to briefly present the main additional phenomena which occur in systems with multiple degrees of freedom.

Of special importance to torque measurement are shafts with multiple disks, which can generate torsional vibration as in the system illustrated in Fig. B.8. The degrees of freedom of torsional vibration relate to the torsion angles φ_i. The same system can also generate bending vibration, for which the degrees of freedom are given by the deflection values w_i of the individual disks.

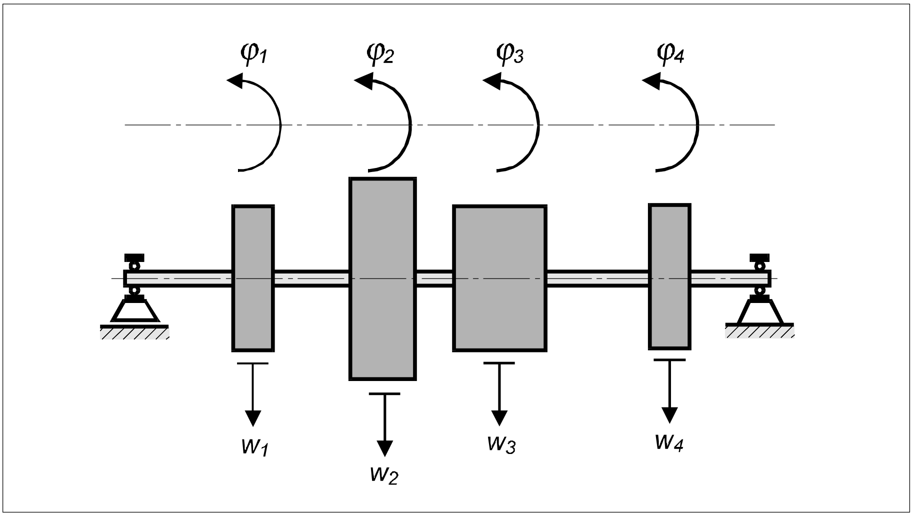


Fig. B.8 Shaft with multiple disks

Lastly, a further modeling refinement leads to the model with an infinite number of degrees of freedom, in which vibrational displacement is not considered at a finite number of reference points only, but rather as a function across a range. An example is the elastic shaft shown in Fig. B.9.

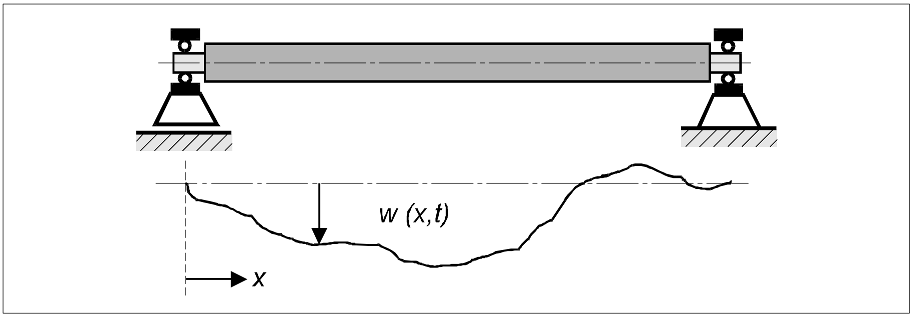


Fig. B.9 Shaft with uniformly distributed mass and stiffness

In order to model such systems mathematically it is necessary to specify not only the motion coordinates but also parameters such as mass and stiffness as a function of location. The example shows the simplest special case: a uniform shaft with mass and bending stiffness distributed uniformly over its length.

B.4.1 Free vibration

The free vibrations in a system with multiple degrees of freedom consist of vibrations with multiple natural frequencies. These individual components are known as natural vibrations. Each of these components is characterized by its natural frequency and mode shape. The latter specifies a fixed relationship between the various displacements of the individual degrees of freedom.

The mode shapes of the symmetrically configured dual mass oscillator as shown in Fig. B.7 are particularly simple. The first mode shape is given by:

$x_2(t) = x_1(t)$ with the natural frequency being $\omega_1 = \sqrt{\frac{c}{m}}$.

The above refers to an in-phase motion of both bodies. The second mode shape represents an out-of-phase motion. For this the following applies:

$x_2(t) = -x_1(t)$ with the natural frequency being $\omega_2 = \sqrt{\frac{3c}{m}}$.

The number of mode shapes or natural frequencies usually corresponds to the number of degrees of freedom, which in the above example is two. A more important special case exists when movements are possible in which no deformation of the system occurs. The term used in this connection is rigid-body modes. An example of such a motion is the even rotation of the shaft shown in Fig. B.8. Under the enumeration rule such a form of motion also counts as a mode shape, although it exhibits no vibration in the true sense. The associated natural frequency is formally assigned a value of zero.

Rather than using coordinates representing the physical degrees of freedom it is advantageous to use coordinates in a generalized sense which represent the amplitude of one individual mode of vibration each. This is known as modal description. Which natural vibrations are represented, and at which amplitude and phase position, will depend on the initial conditions as in the case of the free vibrations of systems with a single degree of freedom.

In the case of uniform, continuous systems there is an infinite number of mode shapes and natural frequencies. However, usually only the lower natural frequencies are of practical significance. Fig. B.10 shows by way of example the first two mode shapes for the elastic, uniform shaft from Fig. B.9. Here it should be noted that the natural frequency for the second mode shape is the quadruple of the first natural frequency. Generally the natural frequency increases with the number of vibration nodes at which displacements remain at zero. For the mathematical treatment and other details regarding vibrations in continuous systems the reader is referred to the specialized literature, for example [23].

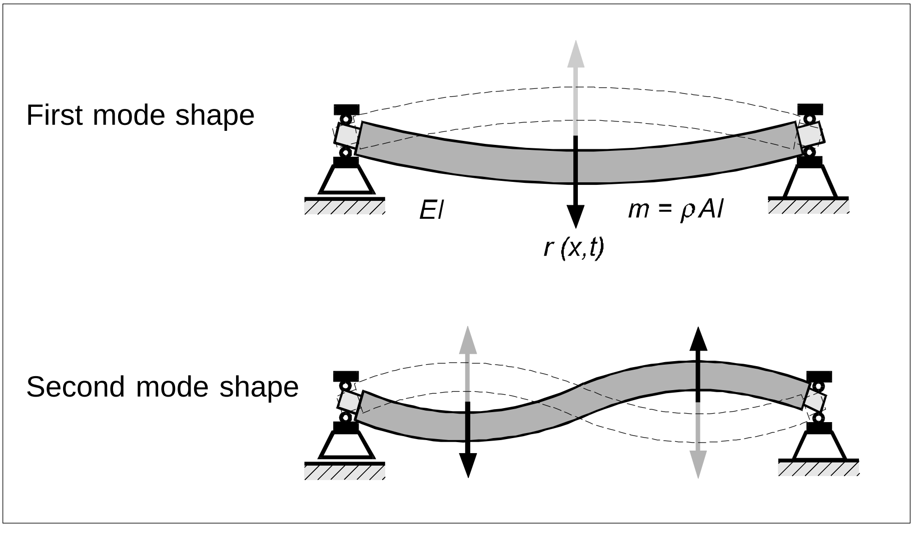


Fig. B.10 Mode shapes for an elastic, uniform shaft

B.4.2 Forced vibration

Generally speaking, every natural frequency in a system constitutes a potential resonance frequency. Excitation frequencies close to natural frequencies should therefore be avoided. However in special cases it is possible for an excitation at or near a natural frequency not to give rise to an increase in resonance. This is the case if the motion which the excitation imposes upon the system when divided into the system's individual mode shapes is found to have no component in a particular mode shape. For example if the dual mass oscillator from Fig. B.7 is excited by imparting the same forces to both bodies in phase, the second mode shape is not stimulated. In this case no resonance occurs at the second natural frequency.

In systems with several degrees of freedom there is generally a higher number of excitation frequencies that give rise to critical increases in amplitude. In everyday engineering practice the number of degrees of freedom to be taken into consideration in mechanical modeling (and thus the number of natural frequencies) is often decided on the basis of the highest excitation frequency that can be expected in the application. It is then often possible to dispense with a refined model the mere use of which would be to determine the higher natural frequencies.

B.5 Further excitation mechanisms for vibration

The discussion so far has dealt with linear vibrating systems that do not change over time and have non-negative damping devices. Additional excitation mechanisms for vibration are made possible by removing one or more of these restrictions. These mechanisms will be discussed here only from the point of view of their phenomenology. A comprehensive discussion including the mathematical treatment can be found in textbooks about non-linear vibration such as [24].

B.5.1 Vibrations in non-linear systems

Vibrations are said to be non-linear if the differential equation of vibration describing the system motions is non-linear.

Free vibration

The general proposition that free vibrations always represent harmonic functions over time does not apply to non-linear vibrations. A particular identifying feature of non-linear vibrations is that the frequency of the free vibrations is dependent on their amplitude.

In the event of strong non-linearity the motional response is so complex that the familiar concepts of linear vibration cease to have any validity. Free vibrations cannot be said to be periodic in such cases, up to and including the oft-quoted "chaotic vibrations". Such extreme instances are not normally found in sensibly designed technical systems, however.

Forced vibration

Subharmonic and superharmonic vibrations are a special feature of forced vibration in non-linear systems. The frequency of a vibrational response is said to be subharmonic if it is a fraction (1/2, 1/3, …) of the excitation frequency. The frequency of a response is said to be superharmonic if it is a multiple of the excitation frequency.

However, superharmonics should not be confused with the effect that excitation mechanisms often contain a non-harmonic excitation which, when divided into harmonics in a Fourier series, always includes components in multiples of the fundamental frequency. If such multiples of the fundamental frequency are already present in the excitation, they will naturally be proportionally present in the vibrational response, and this is also true of linear systems. If a multiple of the fundamental frequency of the excitation exactly coincides with a resonance frequency of the system, this multiple appears as a dominant in the vibrational response, giving the impression of a superharmonic vibration.

B.5.2 Parametrically excited vibrations

Parametrically excited vibrations can occur within systems which have parameters (such as stiffness) that vary over time, typically in a periodic manner. Parametrically excited vibrations can generally occur in both linear and non-linear systems. At this point only a few of the fundamental relationships will be mentioned, to the extent that they also occur in linear systems. The relationships which occur in non-linear systems can be even more complex.

An example of parametrically excited vibrations which is important for rotor dynamics is the non-circular shaft, for example if the shaft cross-section is a rectangle (i.e. with sides of different lengths). In the case of a non-circular shaft the stiffness against radial forces is dependent on the direction of the force. Relative to the weight force, which always acts in the same direction, the stiffness depends on the angular position of the rotating shaft, and therefore indirectly on time. This may lead to what are known as weight excited vibrations. Typical of this is the fact that their frequency is equivalent to roughly twice the rotation speed. Resonance occurs when the speed is equivalent to half the natural frequency of the system. This is known as the weight critical speed.

Similar phenomena can also be caused by rotationally non-symmetrical couplings. Directionally constant loading can be caused not only by weight force but also by spatially fixed prestressing as a result of alignment errors during mounting (see [13]).

A whole group of typical parametrically excited vibrations is described by the Mathieu equation. The common features of this group are that they can only occur when the ratio between the excitation frequency Ω and the natural frequency ω_0 is close to the values 2/1, 2/2, 2/3 etc., the vibrations occurring at the first mentioned ratio number being the strongest.

B.5.3 Self-excited vibrations

Similarly, self-excited vibrations do not belong to the group of forced vibrations but instead refer to a system behavior where free vibrations increase rather than decrease without external stimulus. From this follows one of the important features of self-excited vibrations, which is that the frequency is always a natural frequency, and therefore does not vary with the parameters which usually determine the excitation frequencies, such as speed. A simplified explanation of the mathematical cause is the existence of negative damping terms in the differential equation of vibration.

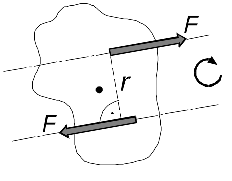
Practical technical examples of self-excited vibrations are aerodynamic fluttering (such as in aircraft wings) and the bending vibration of rotating shafts with internal damping. This is a form of damping that reacts on the deformation velocity in a jointly rotating system. Although locally it actually dissipates kinetic energy, mathematically it has the effect of negative damping. The energy for the increase in vibrations comes from the shaft rotation drive.

In linear systems, self-excited vibrations build up very quickly and their amplitude is extremely high. According to mathematical theory the amplitude can even be infinite. In non-linear systems there is usually an effect which opposes an unlimited increase in amplitude. The build-up stops on reaching a particular form of motion typical of the system, and there the system remains. The term used for this state is limit cycle.

C Equations and tables

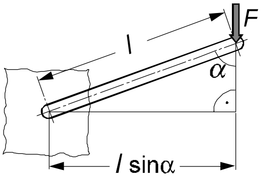
C.1 Torque and moment

If we try to summarize all the forces acting on a rigid body into a single resultant force, we find that this does not succeed in the general case. Two equal forces acting in opposite directions on different but parallel lines of action cannot be summarized into a single resultant force. Such a force couple constitutes a basic element in mechanics known as a moment, which is a basic element in its own right just as force is. The purpose of a moment is to rotate the body upon which it is acting, and it does so about an axis which is perpendicular to the plane of forces. In the general spatial case the torque is a vector. If we keep to two-dimensional problems, the magnitude of the moment results from the product of force F and perpendicular separation r. In this book the sign is positive when the direction of rotation is clockwise. A moment is called a torque if its axis corresponds to the axis of rotation defined by the design of the machine concerned.



$$M = F\,r$$

Moment and force couple - two equivalent forms of representation



$$M = F\,l\,\sin\alpha$$

Moment of a force about a defined axis of rotation

In the case of the moment that results from the action of a single force when the axis of rotation is defined, the computation includes not only the length of the lever arm but also the angle α between the lever arm and the force.

C.2 Power of a rotating shaft

The power P of a rotating shaft is obtained from the product of torque and angular velocity:

$$P = M_D \,\Omega$$

C.3 Mechanical efficiency coefficient

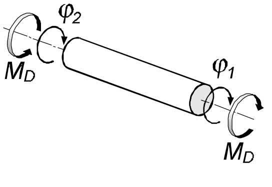
Every machine consumes more power than it delivers. This is due to losses such as friction, atmospheric resistance and heating. The ratio of the delivered power to the applied power is known as efficiency coefficient η

$$\eta = \frac{P_{out}}{P_{in}}$$

C.4 Torsional stiffness of elastic shafts or shaft sections

Definition of torsional stiffness

Torsional stiffness c_T is the quotient of a torque M_D with which the shaft section is loaded and the torsion $\Delta\varphi$ resulting from this torque.



$$c_T = \frac{M_D}{\Delta\varphi} \quad \text{where} \quad \Delta\varphi = \varphi_1 - \varphi_2$$

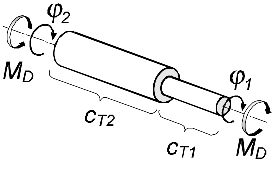
Computing the torsional stiffness for straight shaft sections with uniform cross-section

The torsional stiffness is dependent on the length *l* of the shaft and the torsional moment of inertia I_T (see C.6) that describes the influence of the shaft cross-section, and a material constant known as a shear modulus *G* (see C.9).

$$c_T = \frac{G\,I_T}{l}$$

Computing the torsional stiffness for shaft sections composed of several sub-sections

The torsional stiffness for all shaft sections composed of several sub-sections is calculated on the basis of the torsional stiffness from the equation for elastic springs connected in series.

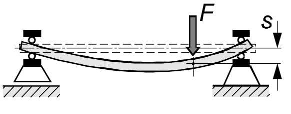


$$\frac{1}{c_{Tges}} = \frac{1}{c_{T1}} + \frac{1}{c_{T2}} + \ ...$$

C.5 Bending stiffness of elastic shafts

Definition of bending stiffness

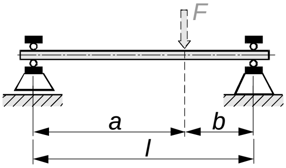
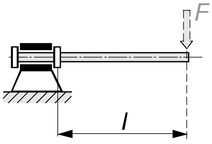
Bending stiffness c_b is the quotient of a lateral force F_r with which the shaft section is loaded and the lateral deflection *s* resulting from this lateral force at the axial position upon which the lateral force is acting.



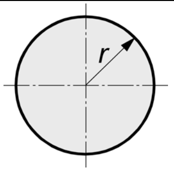
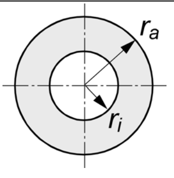
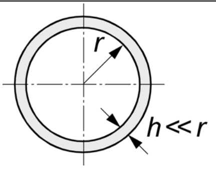
$$c_B = \frac{F_r}{s}$$

Table of bending stiffness depending on shaft geometry and the position of the lateral force

The bending stiffness is dependent on the geometry of the shaft including the bearing configuration, on a material constant known as the modulus of elasticity (or Young's modulus) E (see C.9), and on the position of the application of force.

Geometry	Bending stiffness c_B
	$\dfrac{6EIl}{ab(l^2-a^2-b^2)}$ special case $a=b=\dfrac{l}{2} \Rightarrow \dfrac{48EI}{l^3}$
	$\dfrac{3EI}{l^3}$

C.6 Area moments of inertia and torsional moments of inertia

Shaft cross-section	Area moment of inertia I	Torsional moment of inertia I_T
 Solid, circular cross-section	$\frac{\pi}{4} r^4$	$\frac{\pi}{2} r^4$
 Thick-walled, hollow circular cross-section	$\frac{\pi}{4}(r_a^4 - r_i^4)$	$\frac{\pi}{2}(r_a^4 - r_i^4)$
 Thin-walled, hollow circular cross-section	$\pi r^3 h$	$2\pi r^3 h$

C.7 Mass moments of inertia

Significance and definition of mass moment of inertia

The mass moments of inertia of a body are a measure of the inertia effect with which the body hinders a rotary acceleration, that is, the variation in its angular velocity. Mass moments of inertia can be thought of in the same way as the mass of a body, which is a measure of the inertia effect with which the body hinders a rotary acceleration, that is, the variation in its translational velocity. In the former case the cause of the acceleration is not a force but a moment. A quantitative description of the relationship is expressed by the theorem of moments, which is analogous to Newton's first law of motion for translational movements.

The rather complex relationships of general spatial problems being acted upon by various moments and with rotary movements in any direction are of no importance for most practical applications connected with torque measurement technology, because in the latter case the axis of rotation is given to be the shaft axis, and of all the possible moments the only one of interest is torque, which is the moment acting in precisely this axial direction.

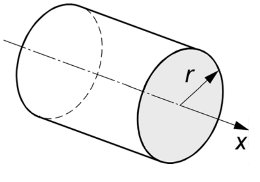
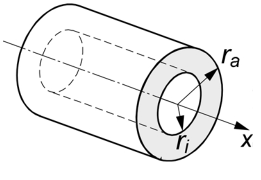
The mass moment of inertia J of a body about any predefined axis of rotation is computed from the integral

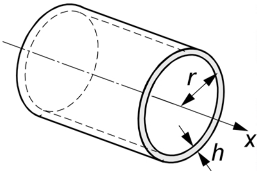
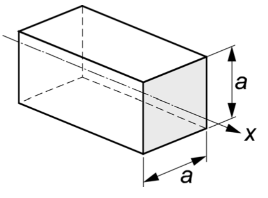
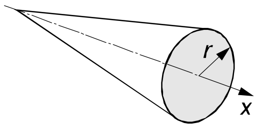
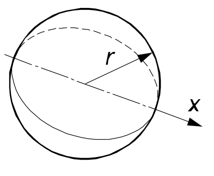
$$J = \int_{(m)} r^2 dm$$

where r represents the distance of the incremental particle of mass from the axis of rotation.

Table of the mass moments of inertia of uniform symmetrical bodies

The mass moments of inertia specified in the table are valid for rotary movements about the x axis (see drawings). Provided there is uniform distribution of mass, the dependency upon the mass distribution can be expressed by the geometry and total mass of the body.

Geometry	Mass moment of inertia J
 Solid cylinder/circular disk	$\frac{1}{2} m r^2$
 Thick-walled, hollow cylinder	$\frac{1}{2} m (r_a^2 + r_i^2)$

Geometry	Mass moment of inertia J
 Thin-walled, hollow cylinder/circular ring	mr^2
 Prism with square base surface	$\frac{1}{6}ma^2$
 Circular cone	$\frac{3}{10}mr^2$
 Sphere	$\frac{2}{5}mr^2$

Computing the mass moments of inertia of composite bodies

From the definition of a mass moment of inertia it follows that the moments of inertia of a body composed of several smaller bodies are calculated from the sum of the moments of inertia of the smaller bodies. This approach can also be used for calculating the mass moments of inertia of hollow bodies, when a negative mass is used for mathematically representing the hollow space.

$$J_{Tot} = J_1 + J_2 + \ldots$$

However, a prerequisite for this simple addition is the fact that the given mass moments of inertia for smaller bodies must all refer to the same axis of rotation.

C.8 Physical quantities and their units

Quantity and math. symbol	Unit	Conversion
Length	m (meter) in (inch) ft (foot)	 1 in = 0.0254 m 1 ft = 0.3048 m
Force F	N (Newton = kg·m/s^2) lbf (pound-force) ozf (ounce-force) kp (kilogram force)	 1 lbf = 4.448 N 1 ozf = 0.2780 N 1 kp = 9.807 N
Torque or bending moment M_D or M_b	N·m (Newtonmeter=N·m) ozf·ft (ounce-force foot) ozf·in (ounce-force inch) lbf·in (pound-force inch) lbf·ft (pound-force foot)	 1 ozf·ft = 0.08474 N·m 1 ozf·in = 0.7062·10^{-2} N·m 1 lbf·in = 0.1130 N·m 1 lbf·ft = 1.356 N·m
Speed of rotation n Angular velocity Ω	min^{-1} (revs. per minute) rpm (revolutions per minute) s^{-1}, also rad/s	 1 rpm = 1 min^{-1} $\Omega[\mathrm{s}^{-1}] = \frac{2\pi}{60} n[\mathrm{min}^{-1}]$
Frequency f Angular frequency Ω	Hz (Hertz = s^{-1}) s^{-1}, also rad/s	 $\Omega[\mathrm{s}^{-1}] = 2\pi\, f\, [Hz]$
Power P	W (Watt = N·m/s) PS (metric horsepower) hp (horsepower, electr.)	 1 PS = 735.5 W 1 hp = 745.7 W
Torsional stiffness c_T	N·m/rad lbf·in/rad (pound-force inch per radian)	 1 lbf·in/rad = 0.1130 N·m/rad

Quantity and math. symbol	Unit	Conversion
Spring stiffness or bending stiffness c or c_B	N/m lbf/in (pound-force per inch)	 1 lbf/in = 175.1 N/m
Mass moment of inertia J	kg·m^2 (= N·m·s^2) lbf·in·s^2	 1 lbf·in·s^2= 0.1130 kg·m^2
Voltage ratio (voltage change relative to excitation)	mV/V (millivolt per volt)	

C.9 Material constants of common materials

Material	Density ρ [kg/dm^3]	Modulus of elasticity E [N/mm^2]	Shear modulus G [N/mm^2]
Steel	7.85	2.11·10^5	0.83·10^5
Gray cast iron	7.25	0.9·10^5…1.55·10^5	0.36·10^5…0.63·10^5
Brass	8.3	1.04·10^5	0.40·10^5
Aluminum	2.7	0.72·10^5	0.27·10^5
Titanium	4.5	1.05·10^5	0.39·10^5

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