Overload protection on explosion-protected motor gearboxes

Possible techniques and limits for overload protection

by Helmut Greiner

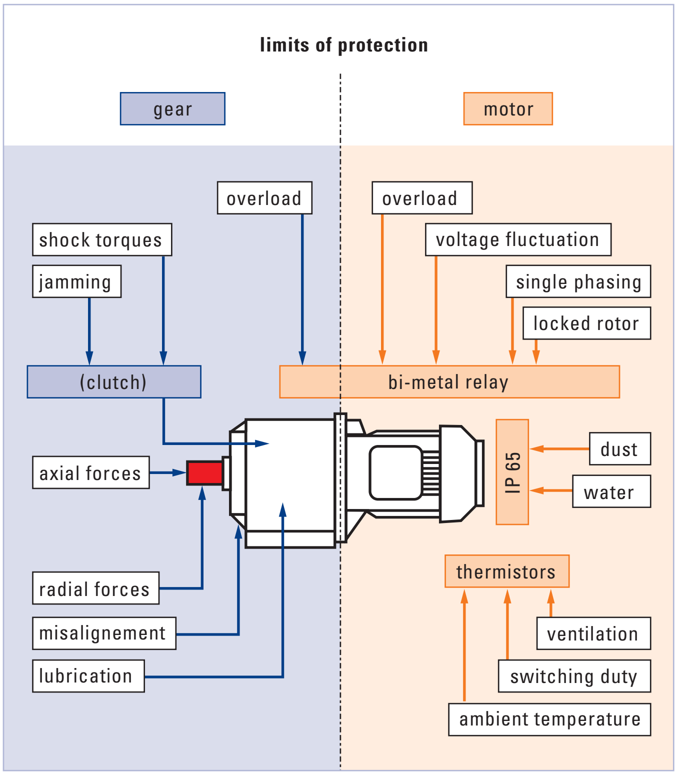


Figure 1: Schematic of the possible techniques for protecting gearbox and motor

On explosion-protected electric motors, the measures for overload protection are a key element of the ignition protection concept. Overload devices for electric motors with type of protection increased safety ›e‹ are, according to section 3.10 of the ATEX Guidelines [1], ›Safety, controlling and regulating devices‹ in accordance with Directive 94/9/EC. These are only allowed to be placed on the market if their functional safety has been confirmed in an EC-type examination [2].

Both current-dependent and therefore thermally-delayed bi-metallic trips and direct temperature monitoring using thermistors detect impermissible overloads during normal operation as well as during ›foreseeable malfunctions‹ due to ›single-phasing‹ or even ›locked rotors‹ [3].

Is the downstream gearbox also protected by the overload protection?

This article will provide information on the possible techniques and limits for electrical and thermal methods for the protection of a mechanical means of transmission against overload.

Basic protection techniques

The protection techniques for the motor as shown in Figure 1 approach ›full protection‹ if combined sensibly. For detailed information on this subject, see [4].

However, of the many possible ways in which a gearbox can be overloaded, only the long duration overload indicated by the motor’s current consumption is detected.

On the gearbox, the possible sources of overload only incompletely represented in the schematic (Figure 1) must be prevented by other means. These include, in particular:

- correct planning of drive and means of transmission (the sections below cover this aspect)
- correct mounting (e.g. avoiding alignment errors)
- periodic inspection (e.g. ensuring reliable lubrication is maintained).

Figure 2: Aerator for increasing the oxygen content of a drinking water tank
Continuous operation: 24 h/d (hours per day)
365 d/a (days per year)

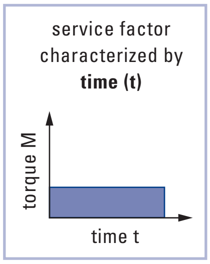
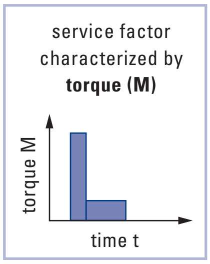


Figure 3: Power station crane for the inspection of turbine and generator
Short operation: A few minutes per year



Service factors for geared motors

Reason for their usage.

Geared motors are designed and assembled using an appropriate, wide ranging and tightly categorized modular system. During development it is not possible to take into account the subsequent specific application, unlike vehicle gearboxes or large gearboxes. The most important parameter for such a series production gearbox is the rated torque that can be output in continuous operation with an acceptable service life.

Thanks to their compact design, geared motors are increasingly used in drives and for new types of applications.

Due to the widespread usage of geared motors, it is inevitable that the drive must be adapted to a very wide range of load conditions. The examples in Figure 2 and 3 are intended to show this wide range.

Purpose of the service factor

To be able to compare the two drive applications represented in Figures 2 and 3, it is necessary to define and compare a fictive torque. These torques calculated from the related load configuration must be equivalent, i.e. they must result in the same service life on continuous operation with the related gearbox parameters as on loading with the actual torque. ➔

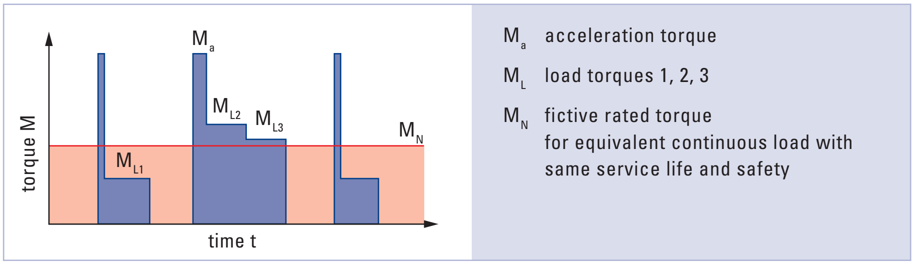


Figure 4: Definition of the service factor

The now withdrawn German code of practice VDI 2151 provided the following corresponding definition for the service factor:

The service factor f_B is the number with which the rated torque M_L for the driven machine must be multiplied to obtain a fictive torque M_N that, when applied constantly over time for any period to the output shaft on the gearbox, will provide the same safety against gearbox damage as the actual torque changing over time (Figure 4).

The gearbox is correctly sized when the continuous load rating is the same as the fictive torque M_N.

The determination of an equivalent rated torque from the load configuration M_a, M_{L1}, M_{L2}, M_{L3} taking into account the related times the loads are applied, and the total running time, requires a significant amount of calculation effort.

In the simplified method, normally used for geared motors, the torques (e. g. M_a, M_{L2} and M_{L3}) beyond the rated torque for the driven machine are covered and taken into consideration by a ›shock classification‹.

What benefits are there for planners and users of geared motors by using service factors?

Drive designed and manufactured for series production, and as a result low cost drive is optimized for a specific drive task.

- driven machine torque shocks due to operation and additional shocks due to an unsuitable means of transmission are evaluated and either reduced by suitable planning or taken into account in the rating of the gearbox.
- gearbox damage is largely prevented.

Normal schematic

The majority of manufacturers of geared motors use a simple schematic for the determination of the service factor with the parameters ›daily operating time‹ and ›shock classification‹. Starting from this basic schematic, in some cases, a further step for the operating time and/or the switching frequency is added (Table 1).

Although systems of this type are widespread, they are not at all standardized. On looking through the catalogues from 42 manufacturers, it was found that very different running times per day were assigned to the service factor 1.0 with shock classification I.

Factor for the daily operating time

For the evaluation of the daily operating time on the definition of the service factor, there are objective, i.e. technically justified and measurable, criteria. The actual running time is defined by the purpose, e.g. for the drive on a shutter or a door, or in case of doubt, is to be determined by using a timer. It will often be necessary to define an estimated or mean value.

If all gearing (except for worm gear sets) is designed for durability, the standardized service life calculation for rolling bearings can be used as a guide. Both in accordance with ISO 281 part 1 as well as in accordance with the modified bearing calculation, the following applies in principle to the nominal service life of rolling bearings:

Type of load		Operating time per day			
Shock classification	**Type of operation of the machine driven**	**3 h/d**	**8 h/d**	**16 h/d**	**24 h/d**
I	Uniform Small masses to be accelerated	0.8	1.0	1.25	1.6
II	Moderate shocks Medium masses to be accelerated	1.0	1.25	1.5	1.8
III	Heavy shocks Larger masses to be accelerated	1.25	1.6	1.8	2.0

Table 1: Practical figures for the service factor as a function of the type of load and operating time

$$L_{10} = \left(\frac{C}{P}\right)^p$$

L_{10} = nominal service life in million revolutions
C = dynamic load rating in N
P = equivalent dynamic bearing load in N
p = exponent for ball bearings: $p = 3$ roller bearings: $p = 10/3$

By using the exponent for ball bearings $p=3$ and for the same life expectancy in years with varying daily running times, the following dependency can be stated for the service factor:

$$f_B \simeq K^{1/3}$$

f_B = service factor
K = factor for the change in the daily running time compared to the definition for $f_B = 1{,}0$

The service factors used in practice vary only slightly from the theoretical values in the diagram (Table 2):

Factor for the shock classification

While objective criteria are available for the parameter ›time‹, the ›shock classification‹ is mostly left to a subjective assessment: Not just in the definitions, but also in the relevant standards, terms as ›moderate‹, ›medium‹ or ›heavy shocks‹ are used.

Objective limits for the permissible torque shocks are only included in the information from a few manufacturers [5].

For categorisation into shock classification a differentiation is made in relation to the shock classification of the production machine.

The torque overload caused by the driven machine

Using ›shock classification‹ it is intended, in particular, to take into account a known or foreseeable torque increase caused by normal operation of the machines driven. Such overloads can be caused, for example, by

- stiffness at low ambient temperatures
- initial resistance of a viscous stirred medium
- occasional transport of an excessively heavy item
- hard points during the machining of inhomogeneous material
- chopping of hard pieces on proper usage of a crusher or mixer.

Based on this definition, the overloads for shock classification III are limited to twice the rated torque – this limit is also given by taking into account the external transmission elements and the rating of the machine driven.

Unlike almost all standardized or common systems for the determination of the service factor, in [5] clear objective limits for shock classification, instead of terms, that can be interpreted subjectively are stated (Table 3):

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Actual running time per day t_d (in h/d)	3	8	16	24
Theoretical factor based on t_d = 8 h/d	0.67	1.0	1.26	1.44
Practical factor for usual schematic	0.8	1.0	1.25	1.6

Table 2: Comparison of theoretically determined service factors and service factors used in practice

Under no circumstance can the extreme overload that can result from improper usage of a driven machine be covered using the shock classification and a corresponding service factor:

- jamming of a crusher due to pieces that are too large or too hard
- impact of a crane carriage against a buffer
- starting a mixer with material baked hard
- jamming of a chain drive by foreign objects.

Torque peaks that result from such an unforeseen and jamming-related process can only be dissipated by using mechanical overload protection (safety clutch, fluid coupling, sliding hub, shear pins).

Protection features based on electrical, electronic or thermal aspects are ineffective because, although they interrupt the supply of power from the mains, they do not remove the damaging rotational energy in the motor's rotor.

You will find detailed information on the subject of ›overload protection‹ in [4].

Shock classifications for driven machines

Shock classifications are assigned to common production machines both in standards and directives, as well as sector-specific or manufacturer-specific documentation. If here, e.g., the shock classification III is assigned to a crusher or a press, this assignment is justified. On the other hand, a conveyor belt can have shock classification I in favourable conditions. However, this situation can rapidly change to shock classification III in intermittent operation, at high speed and transmission using a loose chain.

Categorisation using such tables should therefore not be taken at face value. It provides a rough estimate; for the final assignment of the shock classification, objective criteria, particularly the inertia factor, switching frequency and type of transmission, must be taken into account.

Mass moment of inertia

The acceleration torque M_a developed by the motor is distributed linearly over the masses: This law is important for the load on the downstream gearbox.

The mass inertia present in the system is given, as per the standard, as the ›factor of inertia FI‹:

$$FI = \frac{J_{ext} + J_{rot}}{J_{rot}}$$

FI = factor of inertia
J_{ext} = external mass moment of inertia
J_{rot} = rotor-mass moment of inertia

The portion of the acceleration torque flowing to the exterior is calculated from

$$\frac{M_{ext}}{M_a} = \frac{J_{ext}}{\sum J} = \frac{J_{rot} \cdot (FI - 1)}{J_{rot} \cdot FI} = \frac{FI - 1}{FI}$$

These considerations make it clear as to why the factor of inertia FI has an important function in the determination of the ›shock classification‹.

Shock classification	Description	Permissible brief overload
I	Uniform without shocks	$M/M_N \leq 1$
II	Moderate shocks	$1.0 < M/M_N \leq 1.6$
III	Heavy shocks	$1.6 < M/M_N \leq 2$

Table 3: Shock classification as a function of the overload limit for the rated torque after [5]

Switching frequency

For a configured load, along with the magnitude of the acceleration torque flowing through the gearbox, the frequency of the actions – the switching frequency – is important.

At 1,000 c/h (switching cycles per hour) with a run-up time in each case of 1 s, the gearbox is subject to increased torque for 1,000 seconds an hour – that is almost 30 % of the time. However, apparently more important than this mathematically determined load time, is that each speed change can result in dynamic torque peaks that cannot be determined by calculation.

The effect of the switching frequency on the service factor is therefore mostly assessed empirically in a different manner from manufacturer to manufacturer, if at all. Figure 5 shows how the service factor changes for 100 and 1,000 switching operations per hour for ten different manufacturers.

The relatively high evaluation of small switching frequencies and the only moderate additional increase at very high switching frequencies by manufacturer 6 is due to a good reason:

Incorrectly selected means of transmission with play (chains, claw couplings) can produce significant peak torques in intermittent operation (see following sections). These loads may approach the magnitude of the tensile strength. It is therefore necessary for safety sake to significantly increase the service factor, even at low switching frequency. With this safe basis, it is then not necessary to do much more for the fatigue life at extremely high load changing figures.

However, it does appear more appropriate to ›reduce the severity‹ of the intermittent operation by selecting suitable means of transmission (see following section), instead of drastically increasing the service factor (that is the size of the gearbox).

A gearbox increased in size to suit the shock classification does not also need the full size increase factor due to the switching frequency. With this assessment method, there is a natural upper limit for the service factor of around 2. The application of all factors one after the other, as practiced by a few manufacturers, with an end result of up to 6 may appear mathematically logical, but is technically inappropriate.

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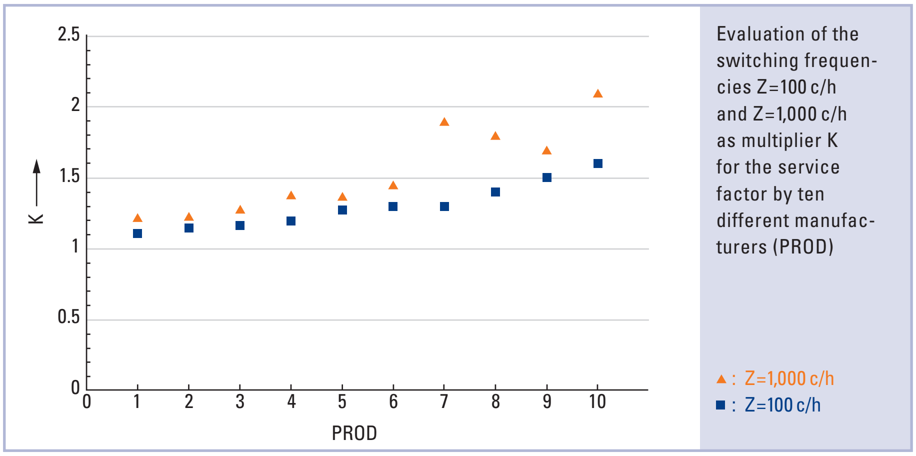
Switching frequency Z (c/h)	1 < Z ≤ 100	100 < Z ≤ 1,000	1,000 < Z
Service factor f_B	1.3	1.45	1.5

Table 4: Effect of the switching frequency on the service factor after [5]



Evaluation of the switching frequencies Z=100 c/h and Z=1,000 c/h as multiplier K for the service factor by ten different manufacturers (PROD)

Figure 5: Evaluation of the switching frequencies

Means of transmission

In comparison to the known systems for the determination of the service factor, the effect of the means of transmission has an important role in the approach described in [5]. Of the systems with ›shock absorbing‹, ›shock neutral‹ or ›shock amplifying‹ characteristics described there in detail, here only the preferred, recommended variant is covered for reasons of space.

Shock absorbing couplings

Highly elastic shaft couplings can dissipate dynamically generated torque peaks. If, due to the nature of their design, they also do not have any play, they are an ideal prerequisite for categorisation as ›shock classification I‹ – particularly for intermittent operation.

If the elastic element permits an extremely large amount of twisting, it can absorb the stored energy in the rotor and even replace the action of a safety clutch. Figure 6 shows the torque peaks measured with a torque measuring hub and recorded with a plotter on steel jamming against steel. If, in conditions that otherwise remain unchanged, a highly elastic shaft tyre coupling is included in the torque path, the torque peak will be significantly reduced as per Figure 7.

The diagram in Figure 8 is representative for this type of coupling and shows that very large twisting φ can occur before the rubber tyre tears. The result is the prerequisite for the dissipation of torque peaks. If the coupling has the function of shear pins or a safety clutch in an emergency, the loss of the shaft tyre is minor compared to the damage that could otherwise occur on the gearbox or on the driven machine.

It is easy to replace the shaft tyre without removing the drive and driven machine. Examples of several coupling systems are represented in Figure 9.

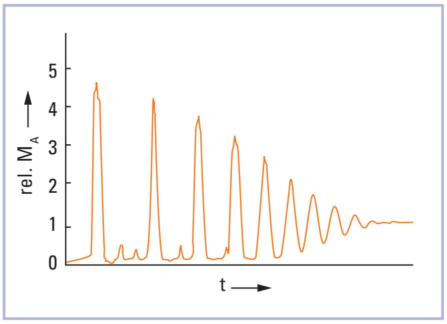


Figure 6: Torque peaks with hard jamming of steel against steel

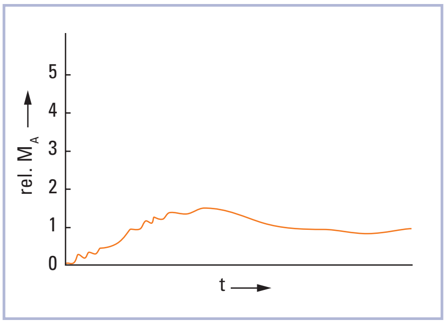


Figure 7: Torque damping on soft jamming by inserting a highly elastic shaft coupling

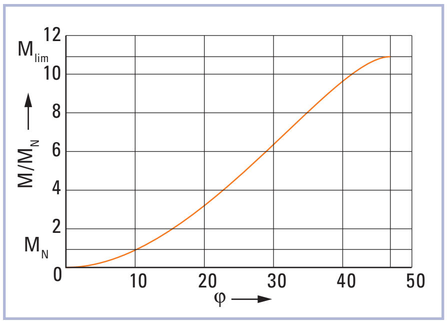


Figure 8: Typical twist characteristic for a highly elastic coupling with shaft tyre (PERIFLEX system, manufactured by STROMAG), Angle of twist φ as a function of the torque M up to fracture M_{lim}

Summary

The gearboxes for explosion-protected geared motors are mostly designed in the types of protection constructional safety ›c‹ and liquid immersion ›k‹.

Definition for ›c‹ [6]: A type of ignition protection in which constructional measures are applied so as to protect against the possibility of ignition from hot surfaces, sparks and adiabatic compression generated by moving parts.

The ›physical measures‹ for type of protection ›c‹ can only provide the required safety within limited loads. As the possible techniques for providing overload protection on gearboxes are very limited, the permissible operating parameters must be defined as accurately as possible to suit practical use, and must be documented.

For geared motors the manufacturer-specific systems for ›service factors‹ have also proven useful. These should be included in the marking in an abbreviated form (Figure 10) and in the operating instructions. Installers and operators should heed these requirements, as unlike the drive motor, no overload protection device is effective for the gearbox.

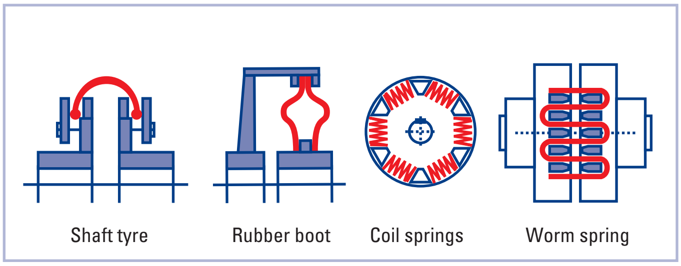
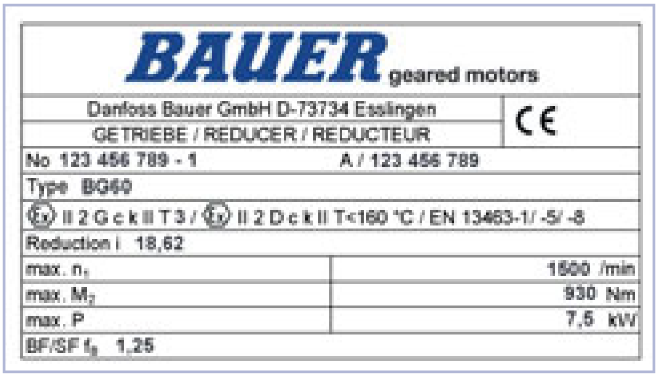


Figure 9: Examples of coupling systems with relatively high facility for elastic deformation



BAUER geared motors

Danfoss Bauer GmbH D-73734 Esslingen

GETRIEBE / REDUCER / REDUCTEUR

CE

No 123 456 789 - 1 A / 123 456 789

Type BG60

Ex II 2 G c k II T 3 / Ex II 2 D c k II T<160 °C / EN 13463-1/ -5/ -8

Reduction i 18,62

max. n_1	1500 /min
max. M_2	930 Nm
max. P	7,5 kW

BF/SF f_B 1,25

Figure 10: Marking on an explosion-protected gearbox
max. n_1: Maximum permissible input speed
max. M_2: Maximum permissible rated torque at the output shaft
max. P: Maximum permissible rated power at the output shaft (transmitted power)
BF/SF f_B: Service factor
II 2 G c k T_: Suitable for Zone 1, Temperature Class T_
(in specific case note T3 or T4)
II 2 D c k T<160°C: Suitable for Zone 21, housing temperature < 160°C

References

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Non-electrical equipment intended for use in potentially explosive atmospheres
Part 5: Protection by constructional safety ›c‹