Project Planning Manual

Drive Engineering - Practical Implementation
Project Planning for Controlled and Non-Controlled Drives

Edition 07/2020

26636735/EN

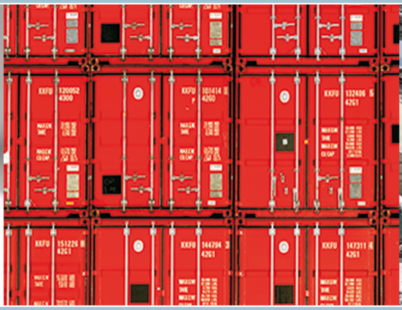


Table of contents

26636735/EN – 07/2020

1 Introduction

SEW-EURODRIVE is one of the leading companies in the world market for electrical drive engineering. An extensive product range and broad spectrum of services make SEW-EURODRIVE the ideal partner when it comes to providing drive systems for demanding applications.

1.1 Target group

This volume from the “Drive Engineering - Practical Implementation” series is aimed at specialists and engineers who intend to configure electric drives. It is also intended for students, apprentices and participants of training courses who would like to learn more about project planning for electric drives.

1.2 Content and structure

This document provides a clear overview of the basics of project planning for electric drives.

The first section discusses applications in drive technology and their directions of movement as a key differentiator.

The following section explains the basics of project planning. The differences in the calculations for controlled and non-controlled drives are described and requirements for project planning as well as specific calculation conventions at SEW-EURODRIVE are presented in addition to criteria for selecting drives.

The chapter “General application-side calculations” focuses on calculations for travel dynamics and forces.

This is followed by the chapters "Project planning for controlled drives" and "Project planning for non-controlled drives," which describe the different processes and calculations.

Thematic structure

This Project Planning Manual covers the following topics:

- Applications in drive technology
- Basics of project planning for electric drives
- General application-side calculations
- Project planning for controlled drives
- Project planning for non-controlled drives
- Table appendix (e.g., friction coefficients of different material combinations, efficiencies, etc.)
- Explanation of abbreviations

Collection of formulas

An extensive, thematically arranged collection of formulas is available as a separate documentation from SEW-EURODRIVE.

26636735/EN – 07/2020

2 Applications in drive technology

2.1 What are applications?

The term "application" refers to anything that is driven by a gear unit output shaft or, in the case of a motor, by a motor shaft. Applications can be highly varied. A primary distinction is made between applications with the following directions of movement:

- Horizontal direction of movement
- Vertical direction of movement
- Rotary direction of movement
- Superimposed direction of movement (simultaneously horizontal/vertical/rotary)

2.1.1 Horizontal direction of movement

Friction determines part of the load in the horizontal direction of movement. For example, friction can arise from rolling friction or the resistance in a ball bearing. In addition, the application must be accelerated. This is called dynamic load. In the horizontal direction of movement, the dynamic load can occupy a significantly greater proportion than the counterforce to friction (friction compensation).

Typical applications:

- Trolley
- Roller conveyor
- Conveyor (belt conveyors, general cargo conveyors, bulk material conveyors, chain conveyors...)
- Industrial trucks
- Electrified monorail system
- Pallet transfer shuttle
- Gantries (X drive)
- Storage/retrieval system (travel drive)

2.1.2 Vertical direction of movement

The load in the vertical direction of movement is determined by the overcoming of gravitational force. As a rule, the proportion of friction is negligible. There is a distinction between upward and downward movement in the vertical direction of movement. During upward movement, the drive operates in motor mode and energy must be supplied. During downward movement, the drive operates in generator mode, and the energy flows from the application to the drive. In order to prevent uncontrolled downward movement in the event of a power failure, the motor must be equipped with a brake in applications with vertical direction of movement.

Typical applications:

- Stationary vertical conveyor
- Ascending conveyor, inclined conveyor
- Press
- Hoists (builders hoists)
- Palletizers
- Gantries (Y drive)

26636735/EN – 07/2020

- Storage/retrieval system (hoist)
- Crane

2.1.3 Rotary direction of movement

The properties of applications with rotary direction of movement are comparable to the properties of applications with horizontal direction of movement. The load is composed of friction and acceleration of the moving parts of the application (dynamic load). With rotary direction of movement, the dynamic load is also significantly higher than the static load in most cases.

Typical applications:

- Rotary table
- Winder
- Rotary kiln
- Carousel
- Rotary table

2.1.4 Superimposed direction of movement

Superimposed movements are a combination of several simultaneous movements in different directions of movement. There are a variety of superimposed directions of movement in robotics. For example, a robot with 6 axes can move freely around a space. A motion controller uses the kinematic model to calculate the travel curve for these complex movements. The loads are calculated by special programs.

Typical applications:

- Robotics (kinematic model)
- Palletizers
- Storage/retrieval system (driving and lifting)

2.2 Ambient conditions of an application

It is important to know the ambient conditions at the installation site when configuring a suitable drive. This includes, among others:

- Ambient temperature
- Installation altitude
- Thermal connection to the environment (heat dissipation, enclosure)
- Additional load due to aggressive or abrasive substances, moisture, etc.
- Potentially explosive atmosphere.
- Stability of the grid
- Legal requirements, application-specific guidelines
- Vibrations

This Project Planning Manual assumes normal ambient conditions without special conditions. Further measures must be taken into account for influencing factors that deviate from these conditions.

26636735/EN – 07/2020

2.3 Worldwide usability

The motors can be used in every country in the world.

Market access is contingent to approvals in many countries. Local laws, regulations and other market-specific requirements must be adhered to. SEW-EURODRIVE provides the latest information on efficiency regulations online via "www.ie-guide.com" as well as via the Online Support under "Engineering & selection – Energy efficiency tools" on the website "www.sew-eurodrive.com".

In many cases, an identification on the motor is required along with the certification. This identification is documented with one or several logos on the nameplate or additional labels on the motor.

The requirements on the condition of asynchronous motors are different all around the world to guarantee safe and efficient operation. A distinction has to be made between statutory provisions (e.g. efficiency regulations) and voluntary measures (e.g. specific certifications for selected markets).

26636735/EN – 07/2020

3 Basics of project planning for electric drives

3.1 Project planning for electric drives for SEW-EURODRIVE

During project planning for an electric drive, it is important to calculate and select the optimum drive for an application with regard to technical suitability and cost-effectiveness on the basis of various requirements.

3.2 Criteria for selecting drives

In addition to basic conditions such as the local energy efficiency guidelines or the climatic conditions at the installation site, factors that affect the sizing of the drive are also of interest to the user. Torque and rotational speed are the influencing factors for an electric drive in a given travel cycle. This corresponds to the required power of the drive. Based on the application data, the following load values for selecting and checking the drive are therefore primarily calculated:

1. The maximum torque to be applied and the associated rotational speed.
2. The thermal requirements of the drive:
   - The thermally equivalent torque averaged over the entire load cycle and the thermal rotational speed of motors in frequency inverter operation.
   - The switching frequency as a thermal parameter of line-powered motors.

The maximum load usually occurs during the acceleration or deceleration of an application with maximum load mass. The drive components to be used must be sufficiently dimensioned in order to withstand this load for a short time.

The thermal strain, however, is regarded as a permanent load. Losses due to energy conversion in the components of the drive train result in heat generation. This can lead to excessive heat buildup in the interior of the component and on the surface of the component. The high temperature can result in faster aging of insulation, lubricants and seals as well as other undesirable effects. Therefore, the continuous thermal load must be considered when a drive is selected.

The load of a drive consists of the following components:

- Move the mass of the load (friction, overcome wind load).
- Accelerate and decelerate the mass of the load.
- Hold the mass of the load or move it in vertical direction (overcome gravitational force).
- Accelerate the rotor (accelerate intrinsic inertia of motor).

The components of the load listed here vary in size depending on the type of application. Traversing drives that move loads at a constant speed in the horizontal direction usually require a smaller amount of static torque compared to the required acceleration torque. For hoists, however, the static holding torque is larger than the acceleration torque.

26636735/EN – 07/2020

3.3 Differences between project planning for controlled and non-controlled drives

A drive is referred to as “controlled” if the rotational speed or torque can be adjusted by a frequency inverter. The frequency inverter is operated from the line power supply system and controls the motor on the output end. The parameterizable acceleration or ramp time allows for constant or variable acceleration during the startup process. Thus, required acceleration torques can be directly influenced.

Non-controlled drives are operated directly from the line power supply system. Their startup behavior is rigid and is based on the existing mass ratios and the electromagnetic properties of the motor being used. There is no way of subsequently changing the operating behavior of a non-controlled drive apart from modifying its design. This applies in particular to the resulting acceleration. The operating speed depends on the applied load and corresponds to the rated speed of the motor at rated load.

There are generally no technical differences between motors for grid or inverter operation. The different operating behavior is caused solely by differences in wiring and control.

The configuration process differs for controlled and non-controlled drive solutions because the operating characteristics are sometimes different.

3.3.1 Procedure for configuring controlled drives

Based on the application, the drive train components are calculated and selected in the order described here. In contrast to project planning for non-controlled drives, the motor is selected after the gear unit.

1. **General application-side calculations**
   - Travel dynamics
   - Output speed and gear ratio requirement
   - Forces and torques
   - Mass moment of inertia
2. **Calculating and selecting the gear unit**
   - Output end torques
   - Selecting the gear unit
   - External forces (overhung loads and axial loads)
3. **Calculating and selecting the motor**
   - Motor torques
   - Motor preselection (type, size)
   - Checking the drive selection
4. **Calculating and selecting the brake**
   - Special requirement for lifting applications
   - Braking work and braking torque
   - Selecting the brake
   - Gear unit load during emergency stop braking
5. **Calculating and selecting the frequency inverter**
   - Assigning the frequency inverter based on the rated motor power
   - Calculating the maximum and effective inverter current

26636735/EN – 07/2020

- Selecting the frequency inverter according to calculated motor currents
- Selecting the frequency inverter for operating modes with current-controlled control
- Derating factors
- Braking resistor (optional)

Basic calculation process

The kinematic target values of the application are calculated first if they are not initially known. This includes:

- Cycle time
- Distances
- Speeds/rotational speeds
- Accelerations

The calculated kinematic target values are used to calculate the dynamic torques. The resulting static torques for the gear unit output are calculated based on specified friction or hoist loads. Existing additional transmissions must be taken into account when the torques and rotational speeds are calculated. A suitable gear unit size is determined based on the maximum occurring torque. In order to select a gear ratio, the type of motor must be specified.

The requirements of the application are converted to the motor side. A motor is selected based on a maximum and an effective operating point.

A motor brake must always be selected for applications with a hoist load. The motor brake is selected as necessary for other applications. Criteria such as required holding reliability, permitted braking work and maximum braking distances are used as the basis for calculation and selection. The mechanical feasibility of the drive train as well as the gear unit load during emergency stop braking is then verified.

Frequency inverter selection is often based on the maximum and effective motor current required. A braking resistor is additionally required if the motor operates as a generator during deceleration. It converts the regenerative energy returned by the application into heat.

3.3.2 Procedure for configuring non-controlled drives

Based on the application, the drive train components are calculated and selected in the order described here. In contrast to project planning for controlled drives, the motor is selected before the gear unit.

1. **General application-side calculations**
   - Travel dynamics
   - Output speed and gear ratio requirement
   - Forces and torques
   - Mass moment of inertia
2. **Calculating power**
3. **Calculating and selecting the motor**
   - Selection criteria
   - Checking motor start-up
   - Switching frequency
4. **Calculating and selecting the brake**

- Special requirements for lifting applications
- Braking work, braking torque and switching frequency
- Selecting the brake
- Gear unit load during braking
- Further selection criteria

5. **Calculating and selecting the gear unit**
   - Preselecting the gear unit
   - Calculating the actual gear unit load
   - Options for reducing the gear unit load

Basic calculation process

The required effective and maximum power demand of the application are calculated based on the intended application speed and the resistance forces to be overcome. A suitable motor is selected after calculating these criteria. In this case, the startup behavior of the grid-powered motor power must be taken into account as a central feature. This can be considered only after the preliminary selection of the motor because of the significant influence of individual motor characteristics.

A line-powered drive always follows its startup characteristic, whereas the torque curve of motors operated with an inverter can be influenced (see chapters "Calculating and selecting the motor" (→ 🗎 93) and "Checking motor start-up" (→ 🗎 97)).

During a grid-powered startup process, the motor current is far above the rated current, regardless of the load condition. Starting at values that can be greater than $8 \times I_N$, the motor current drops as the motor speed increases. Under nominal load, the rated current I_N flows at the rated speed n_N in the motor.

Since the motor current affects motor temperature rise quadratically, grid-powered startup operations lead to disproportionate heat generation. Due to the speed-dependent air flow, the internal cooling of the motor by the integrated fan has a reduced effect during startup. In order not to exceed a permitted temperature increase limit for a large number of starts per time, correspondingly long times with or without reduced torque requirements are necessary in intermittent duty. The permitted temperature increase limit relates to the reference time of 10 minutes described in the standard.

The result is a maximum no-load starting frequency, meaning the permitted number of starts of the motor without load. This permitted switching frequency is reduced if relevant application data, such as load torque and load moment of inertia, are included. To avoid overheating the motor under consideration, the calculated number of starts per hour may not be exceeded.

Non-controlled applications can be decelerated either through sufficient friction in the application or by a mechanical motor brake. Similar to acceleration torque, the deceleration torque of the gear unit output, deceleration times and braking distances are determined from existing inertia and load conditions as well as from the electromechanical properties of the drive. Only a change in inertia, braking torque or motor type has an influence on the braking behavior.

26636735/EN – 07/2020

3.3.3 Project planning notes

- Under certain circumstances, details that determine performance, such as the mass to be moved, friction coefficients and other variables, can only be estimated during drive selection.

  Calculation using preliminary values may be practical for an approximate drive selection, but the designer should verify the selection after all the details become known. If unfavorable values are assumed, the requirements are artificially increased, and a larger drive is required. Conversely, supposedly conservative assumptions such as large coefficients of friction can result in a greater load on the brake than planned.
- During startup, grid-powered motors use their full acceleration torque for overcoming the static load and accelerating all mass moments of inertia.

  The acceleration torque of a motor of energy efficiency class IE3 is approximately 2 to 3 times the rated torque. With a motor twice the size, 4 to 6 times the torque would be applied during startup.

  Only a certain portion of this affects the gear unit and/or the application, depending on the inertia conditions and the static load. Under unfavorable conditions, gear units or downstream mechanisms may be damaged. For travel drives, an oversized motor may also cause slipping of the wheels. For this reason, it is particularly important to know the precise application requirements and parameters for line-powered motors.
- However, various protection mechanisms can be implemented for motors operated with the frequency inverter. On the one hand, the torque is known (calculated) through precise specification of an acceleration value; on the other hand, the motor torque can be additionally limited by parameterizing the current limit in the frequency inverter. Controlled drives can subsequently be better adapted to the application requirements and parameters.

3.3.4 Parameters for line operation

It may be necessary to take an iterative approach to drive selection because the startup and operating behavior depends on application-specific variables as well as the technical characteristics of the drive to be used. For example, a lower braking torque or a larger gear unit must be selected if an analysis of the brake section reveals that the selected gear unit is overloaded. Braking times and distances change with the braking torque. Therefore, the effects of changes on other drive train components must always be verified.

Line-powered drives are often used for the following application requirements and parameters:

- Constant load condition in S1 duty cycle or with low switching frequency (< 10 h^{-1}), e.g. simple conveyor lines in continuous duty.
- Large tolerances for specifications on dynamic behavior and positioning accuracy.
- Crank applications that start from dead center and end at dead center (such as eccentric lifting units).

In addition, various applications can be realized as a line-powered drive. However, given today’s options (technical availability, efficiency) these applications are best realized using other drive concepts. For example, these include:

- Unfavorable mass ratios with high startup frequency (> approx. 200/h). The required gear unit size can be reduced by several degrees through inverter operation.

26636735/EN – 07/2020

- High regenerative power: Inverter operation and DC link coupling/regenerative power supply.
- Keep moving mass small (for example, storage/retrieval system) with 87 Hz operation with frequency inverter.
- Travel cycles with different load conditions. The line-powered motor is selected according to the highest load and is then oversized for lower load conditions.
- Small tolerances for acceleration, deceleration or torque specifications are more easily maintained with a controlled drive.

Due to the rigid dependence on application and motor characteristics, line-powered motors reach their limits when:

- The required switching frequency is greater than the permitted limit switching frequency of the motor and application.
- Slow acceleration / long run-up time with low load moment of inertia.
- Minimal deceleration with low load moment of inertia and low friction in the system.
- High positioning accuracy that cannot be achieved by mechanical braking.

3.4 Formula symbols and indexes

The formula symbols and indexes used are as language-independent as possible and are designed to provide all relevant information in the index. This improves clarity and makes it easier to distinguish between sizes.

The index has a modular structure and is usually composed of several components. It starts with an abbreviation of the component of the drive train being considered and may be followed by up to two additional abbreviations that designate a permitted product characteristic or represent the frame of reference, for example.

The application is an exception; it is not provided with its own abbreviation because it is always in focus throughout the entire project planning process. Only for certain topics is the application indexed to better distinguish it from other variables.

Example: Meaning of the formula symbol $M_{G\_per\_es}$

The formula symbol $M_{G\_per\_es}$ stands for the permitted emergency stop torque of the gear unit output and is structured as follows.

	Meaning
M	Physical quantity, in this case: Torque
G	Component of the drive train, in this case: Gear unit
per	Permitted product characteristic (per = permitted)
es	Operating or reference condition, in this case: Reference to emergency stop braking (es = emergency stop)

3.4.1 Commonly used indexes

The following abbreviations are used in the index:

- The considered component of the drive train.

Index	Meaning (component)
G	Gear unit
Mot	Motor

Index	Meaning (component)
B	Brake
FU	Frequency inverter
BW	Braking resistor

- Different product characteristics.

Index	Meaning
per	Permitted product characteristic
N	Rated value according to nameplate

- Operating or reference condition.

Index	Meaning
es	Reference to emergency stop event
B	Reference to mechanical braking
H	Reference to motor startup
’	Reference to generator mode (e.g., M'_{Mot_stat})
x	Physical quantity, reduced to the motor shaft (J_x = mass moment of inertia of the load reduced to the motor shaft)
gen	Regenerative travel section
n	General travel section. "n" can be replaced by: • "ac" (acceleration) for the "acceleration” travel section • "const" (constant speed) for the "constant travel” section • "dec" (deceleration) for the “deceleration” travel section • A number: 1, ... Numbers are used to sequentially number the travel sections, e.g., for calculating the effective motor torque.

The following symbols deviate from this modular indexing because they are firmly established as concepts:

- M_{a_max} = Continuously permitted output torque of the gear unit
- i_G = Gear unit ratio
- F_R = Overhung load on gear unit output
- F_A = Axial load on gear unit output
- t_2 = Brake application time

26636735/EN – 07/2020

3.4.2 Formulas and units

The units of physical quantities in the formulas are each specified directly below the formula (e.g., [P] = kW). The formulas use standard units in the area of drive technology, which are based on the International System of Units (SI), or units derived from SI units.

The factors resulting from the conversion of the units are integrated into the formula without a unit. The multiplication sign is always represented as "x."

Example: Power

The calculation of the power P from torque M and speed n serves as an example. The power is calculated directly in kilowatts, although the speed in revolutions per minute and the torque in Newton meters are included in the formula. The unitless conversion factor of 9550 (rounded value) in the formula allows the standard units to be used directly.

$$P = \frac{M \times n}{9550}$$

18014418020467467

P = Power		[P] = kW
M = Torque		[M] = Nm
n = Speed		[n] = min^{-1}

3.5 Prerequisites for project planning

This document contains the mathematical principles for drive selection. The formulas are introduced in the order in which they will be applied later.

There can be more than one "correct" result during project planning for a drive. There can be several different drive solutions, all of which are suitable for the application in question. This range of solutions can be optimized according to different characteristics.

For one thing, the input data cannot be precisely known; furthermore, it cannot always be precisely determined. In such cases, estimates are made. Excessive reserves result in oversized, inefficient drives. In the case of line-powered motors, oversizing can place excessive strain on the mechanical components. Incorrect estimates can result in unsuitable drives that damage the drive and the system.

The project planning results are subject to a margin of interpretation within which optimizations can be made according to different criteria:

- Energy efficiency and energy costs
- Operating and process reliability
- Control capability
- Operating costs
- Installation space in the system
- Transmission elements
- Cabling
- Reduced number of variants
- Initial costs

3.5.1 Calculation conventions at SEW-EURODRIVE

Please note the following conventions for calculating an electric drive at SEW-EURODRIVE.

Reference systems and signs

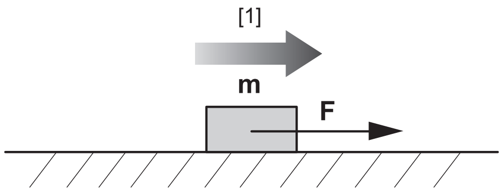
INFORMATION

Signs are used in this Project Planning Manual to provide information about the orientation of a direction-dependent variable in relation to the direction of movement and to distinguish between the motor mode and generator mode of the motor.

The formulas for drive selection for non-controlled drives and especially for the brake (controlled and non-controlled drives) always use the values of the relevant variables.

If the direction of a variable is relevant, it is defined using the sign in relation to the direction of movement, whereby the direction of movement is defined as positive. If a force acts in the direction of movement, the sign of the force is positive. If a force acts against the direction of movement, the sign of the force is negative.

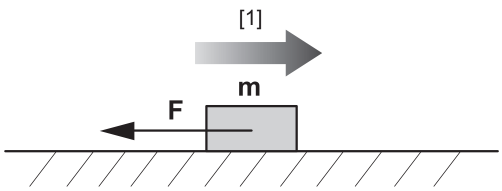
Actual vector quantities in an oblique position relative to the reference system are divided into components parallel to the coordinate directions.



18014418293953931

[1] Direction of movement
F Force
m Mass

The force shown above is directed in the direction of movement and has a positive sign. The force shown below is directed against the direction of movement and has a negative sign.

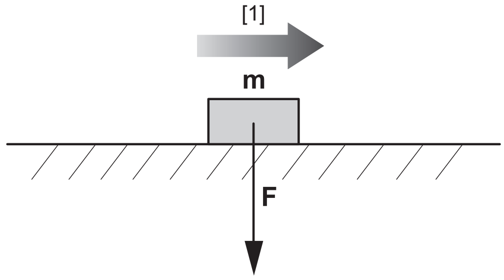


18014418293957515

[1] Direction of movement
F Force
m Mass

26636735/EN – 07/2020

The force component shown in the following figure is not relevant to the sign because it acts perpendicular to the direction of movement. For example, this could be the normal force, which then contributes to a friction force against the direction of movement.



9007219039220107

[1] Direction of movement
F Force
m Mass

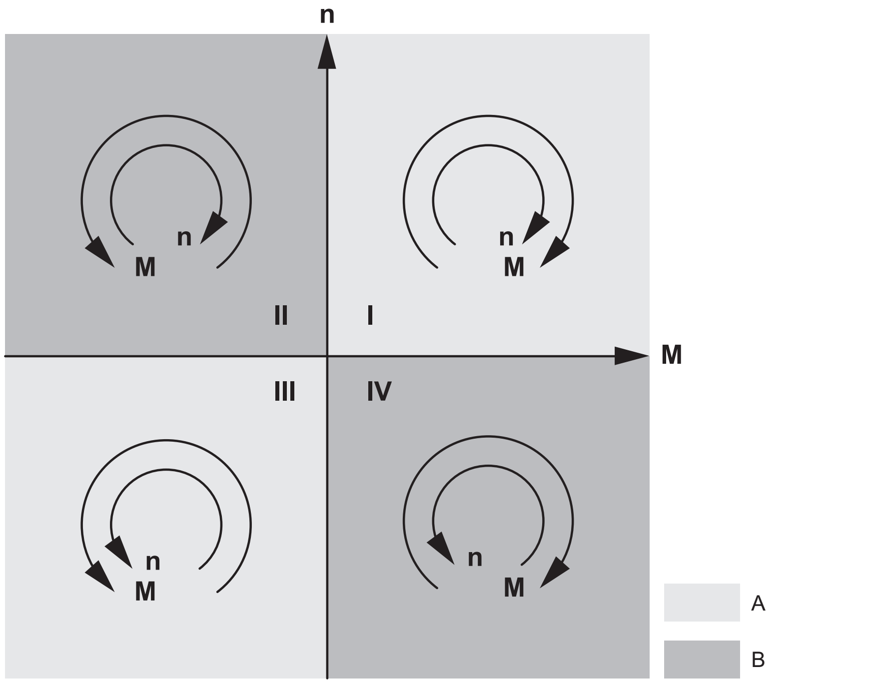
If the original direction of movement changes during the travel cycle, then the reference system rotates. This means that the direction of movement always remains positive. For example, the reference system is directed upwards during upward movement of a hoist and downwards during downward movement. As a result, there is no difference if the signs are correctly interpreted. This means that, similar to a paternoster elevator with a cab, the hoist moves downwards again while the direction of rotation remains the same, instead of having to change the direction of rotation of the drive at the highest point.

In this manner, it is possible to distinguish between motor mode or generator mode simply by the sign of the force or the torque, since the speed is always assumed to be positive. Motor and generator mode are then defined by the sign of the power:

- Motor mode: **P** ~ M × n **> 0**
- Generator mode: **P** ~ M × n **< 0**

26636735/EN – 07/2020

The sign of the power depends both on the sign of the speed and on the sign of the torque. As shown in the following figure, this results in two work envelopes I and III for motor mode and two work envelopes II and IV for generator mode.



28358771467

A	Motor mode
B	Generator mode
n	Speed
M	Torque
I	Positive rotation, positive torque
II	Positive rotation, negative torque
III	Negative rotation, negative torque
IV	Negative rotation, positive torque

These conventions simplify the calculation by distinguishing only between quadrants I and II, although in reality, of course, all 4 quadrants continue to be available as possible work envelopes for the motor. Of course, both positive direction of rotation and negative direction of rotation occur in real applications. The limitation of the quadrant to positive rotational speeds is a purely theoretical tool and has no effect on the actual operating points.

INFORMATION

Signs are used in this Project Planning Manual to provide information about the orientation of a direction-dependent variable in relation to the direction of movement and to distinguish between the motor mode and generator mode of the motor.

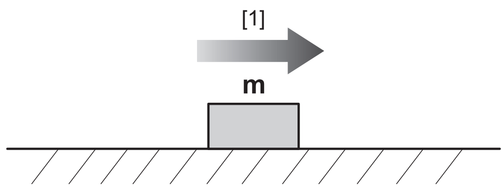
The formulas for drive selection for non-controlled drives and especially for the brake (controlled and non-controlled drives) always use the values of the relevant variables.

3.5.2 Considering basic movements and quantity definitions

The relationship between a linearly moving load and the rotating output shaft of the drive must be established in order to design an electrically operated application. Therefore, it useful to be aware of the relevant quantities for each type of movement.

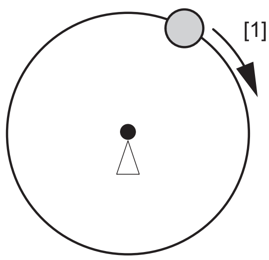
All applications can be divided into 2 basic movements: linear movements and rotary movements around a center of rotation. The following figure shows a comparison of these movements:

- Linear, straight-line movement



18014418020476171

- Rotary motion

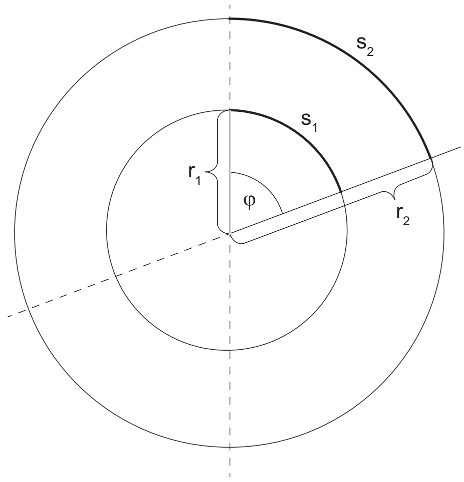


18014418020478347

[1] Direction of movement
m Mass

Linear movement (e.g., travel drive, lifting drive)		Rotary movement (e.g., rotary table)	
Distance	s [s] = m	Angle	φ [φ] = rad or ° The unit sign rad is the angular quantity of the unit circle in radians. [φ] = rad 360° ≙ 2π ≈ 6.28 rad
Speed	v [v] = m s^{-1}	Angular speed	ω [ω] = s^{-1}
Acceleration	a [a] = m s^{-2}	Angular acceleration	α [α] = s^{-2}
Force	F [F] = N	Torque	M [M] = Nm
Mass	m [m] = kg	Mass moment of inertia	J [J] = kg m^2

The radius is the fundamental relationship between the quantities of motion in linear units and in radians (see following figure). All angular quantities are independent of the radius and can be viewed as linear movement according to their distance from the center of rotation. The larger the radius, the further the actual linear distance that must be traveled on the circular arc.



9007219039408267

Relationship between linear quantities and angular quantities:

$$s = \varphi \times r$$
$$v = \omega \times r$$
$$a = \alpha \times r$$

18014418020480523

s = Distance	[s] = m
φ = Angle	[φ] = rad = 1
r = Radius	[r] = m
v = Speed	[v] = m s^{-1}
ω = Angular speed	[ω] = s^{-1}
a = Acceleration	[a] = m s^{-2}
α = Angular acceleration	[α] = s^{-2}

INFORMATION

These equations are in radians.

26636735/EN – 07/2020

4 General application-side calculations

This chapter covers the following topics:

- Travel dynamics
- Output speed and gear ratio requirement
- Forces and torques
- Mass moment of inertia
- Special case of spindle drives

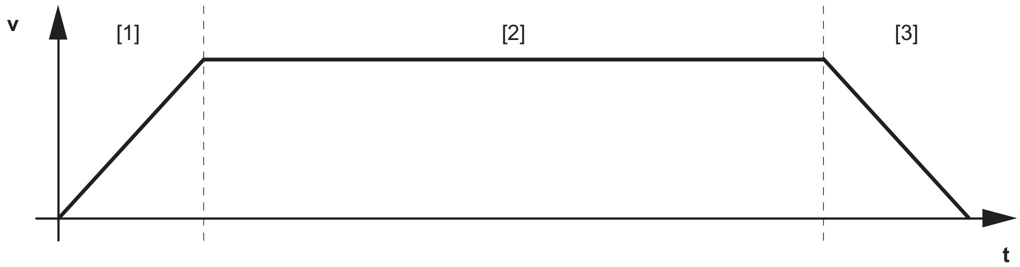
The application is considered as the first step in the project planning process for a drive. Calculations for travel dynamics and forces are the main focus of this step. The technical parameters of the application serve as a further basis for the subsequent selection of a suitable drive.

INFORMATION

Refer to the "Calculation conventions at SEW-EURODRIVE" in the previous chapter. They are the standard procedures for calculations at SEW-EURODRIVE and simplify the explanations in the following chapters.

4.1 Travel dynamics

In considering the travel dynamics, it is necessary to consider the motion profile of the application, which is illustrated in the form of a time-velocity diagram in a travel diagram. A travel diagram is helpful for understanding the planned movement of the application and dividing it into different load sections. The following figure shows a typical trapezoidal travel diagram with 3 travel sections.



9007219039476619

[1] Travel section 1: acceleration
[2] Travel section 2: constant speed
[3] Travel section 3: deceleration

For the sake of simplicity, it is assumed that the application accelerates from an idle state and each travel section is considered individually. The physical relationships are described by the following simplified equations of motion.

26636735/EN – 07/2020

4.1.1 Static equations of motion

Linear movements

In travel section 2, the application travels at a constant speed v (acceleration a = 0 0 m s^{-2}). Static equations of motion apply.

$$v = const.$$
$$s = v \times t$$

18014418020498187

v = Speed	[v] = m s^{-1}
s = Distance	[s] = m
t = Time	[t] = s

Rotary movements

Static equations of motion expressed in angular quantities are applied for static movements of rotary applications.

$$\omega = const.$$
$$\varphi = \omega \times t$$

18014418020503179

φ = Angle	[φ] = rad = 1
ω = Angular speed	[ω] = s^{-1}
t = Time	[t] = s

4.1.2 Dynamic equations of motion

Linear movements

In the travel section 1, the acceleration section, the starting speed v = 0 m s^{-1} is increased to the final speed v with constant acceleration. This is a dynamic motion.

$$a = const.$$
$$v = a \times t$$
$$s = \frac{1}{2} \times a \times t^2 = \frac{1}{2} \times v \times t$$

18014418020494859

a = Acceleration	[a] = m s^{-2}
v = Speed	[v] = m s^{-1}
t = Time	[t] = s
s = Distance	[s] = m

In travel section 3, the deceleration section, the speed is reduced to an idle state with constant deceleration. As in travel section 1, the simplified dynamic equations of motion can be applied here.

Rotary movements

For rotary applications such as rotary tables or corner transfer units, the motion data is often given directly in angular units (for more information on angular units, see chapter "Considering basic movements and quantity definitions" (→ 🗎 21).

26636735/EN – 07/2020

Equations of motion in angular units are applied for dynamic movements of rotary applications.

$$\alpha = const.$$
$$\omega = \alpha \times t$$
$$\varphi = \frac{1}{2} \times \alpha \times t^2 = \frac{1}{2} \times \omega \times t$$

18014418020499851

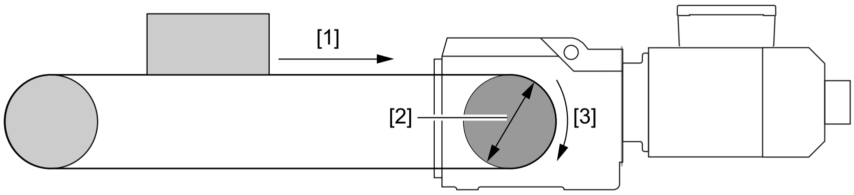
α	= Angular acceleration	$[\alpha] = s^{-2}$
ω	= Angular speed	$[\omega] = s^{-1}$
t	= Time	[t] = s
φ	= Angle	$[\varphi]$ = rad = 1

4.2 Output speed and gear ratio requirement

4.2.1 Output speed

The output speed is calculated in revolutions per minute based on the required maximum travel speed of the application (expressed here in meters per second). The following figure shows the relevant variables using the example of a conveyor belt.

The output speed refers to the application-side output shaft of the gear unit.



18014418020510731

[1] Speed v of the application
[2] Diameter d of the mechanical transmission element
[3] Output speed of the gear unit, n_G

The linear travel speed [1] is converted into the angular speed of the output shaft via the dimension of the mechanical transmission element [2].

This relationship between linear quantities and angular quantities is described in chapter "Considering basic movements and quantity definitions" (→ 🗎 21). Instead of the radius, the diameter can also be used as the calculation variable.

$$\omega = \frac{v}{r} = \frac{2v}{d}$$

22898734987

ω	= Angular speed	$[\omega] = s^{-1}$
v	= Speed of the application	$[v] = m\ s^{-1}$
r	= Radius	[r] = m
d	= Diameter of the transmission element	[d] = m

The desired output speed [3] is calculated from the relationship between angular speed and frequency. Since these two variables are given in s^{-1}, the unit is then converted using the factor 60 s min^{-1}.

Conversion of the angular speed into the output speed:

$$\omega = \frac{\varphi}{t} = \frac{2\pi}{T} = 2\pi \times f$$
$$\omega = \frac{2\pi \times n}{60}$$

18014418020507403

ω	= Angular speed	$[\omega] = s^{-1}$
φ	= Angle	$[\varphi]$ = rad = 1
t	= Time	[t] = s
D	= Periodic time	[T] = s
f	= Frequency	$[f] = s^{-1}$ = Hz
n	= Speed	$[n] = min^{-1}$

The previously described relationships between angular speed and linear travel speed result in the desired formula for calculating the output speed.

By equating $\omega = \frac{2v}{d}$ and $\omega = \frac{2\pi \times n}{60}$

And solving for the speed, we obtain:

$$n = \frac{v \times 60000}{\pi \times d}$$

9007222153484811

ω	= Angular speed	$[\omega] = s^{-1}$
v	= Speed	$[v] = m\ s^{-1}$
n	= Speed	$[n] = min^{-1}$
d	= Diameter of the transmission element	[d] = mm

Calculation of the output speed for applications with additional transmissions

Additional transmissions are additional gear ratio designs such as chain or belt drives that are considered part of the application. Note that in applications with additional transmissions, the linear motion is not directly related to the gear unit output, but to the output end of the additional transmission. In this case, the actual output speed is calculated using the gear ratio of the additional transmission.

It is generally assumed that the additional transmission, just like SEW-EURODRIVE gear units, has a gear ratio ≥ 1. As a result, the torque is transmitted and the speed is reduced.

$$n_G = n_V \times i_V$$

18014418020517899

n_G	= Output speed of the gear unit	$[n_G] = min^{-1}$
n_V	= Output speed of the additional transmission	$[n_V] = min^{-1}$
i_V	= Additional transmission ratio	$[i_V] = 1$

4.2.2 Gear ratio requirement

In addition to the output speed, the gear ratio is also of interest when selecting the gear unit afterwards. The gear unit ratio can be estimated from the expected motor speed and the previously calculated output speed of the gear unit.

26636735/EN – 07/2020

Calculating the ideal gear unit ratio:

$$i_{G\_id} = \frac{n_{Mot}}{n_G}$$

18014418020520587

i_{G_id}	= Calculated ideal gear unit ratio	$[i_{G\_id}] = 1$
n_{Mot}	= Motor speed	$[n_{Mot}] = min^{-1}$
n_G	= Output speed of the gear unit	$[n_G] = min^{-1}$

The motor speed to be used here depends on the number of pole pairs of the motor and, in inverter operation, on the underlying voltage frequency characteristic as well (e.g., 50 Hz or 87 Hz). The motor size has not yet been determined at this point in the drive selection process. Therefore, the following table of typical values for various motors can be used as a guide.

Motor	Typical motor speed n_{Mot}
2-pole (50 Hz)	2900 min^{-1}
4-pole (50 Hz)	1450 min^{-1}
4-pole (87 Hz)	2550 min^{-1}
6-pole (50 Hz)	950 min^{-1}

4.3 Forces and torques

In the next step, the forces required for the movement of the application are calculated. The calculation of forces is an essential step in project planning and has a considerable influence on the size of the drive to be selected.

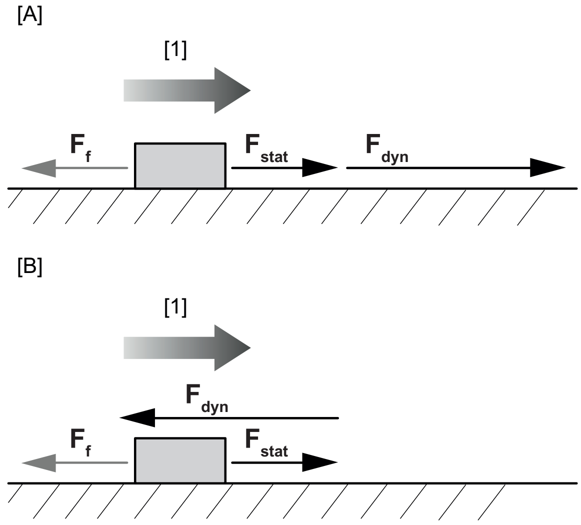
As shown in the travel diagram (see section "Travel dynamics" (→ 🗎 23)), there are 2 motion states: static and dynamic. Each of these states is based on a different force situation. Static forces act in all travel sections, while dynamic forces only act in travel sections with varying speed ($a \neq 0$).

During project planning, a static force F_{sta} is always defined as a force that directly compensates for naturally occurring forces such as friction and gravitation, thus creating a force equilibrium. The dynamic force F_{dyn} is the force component that the drive must apply in addition to the static force in order to accelerate or decelerate the application.

26636735/EN – 07/2020

4.3.1 Forces for horizontal movement

The following figure shows examples of forces for horizontal motion in relation to the direction of movement. The gray arrows represent the application-side load and the black arrows represent the forces to be applied or to be absorbed by the drive.



18014418020523275

[A] Acceleration
[B] Deceleration
[1] Direction of movement
F_f Friction force
F_{stat} Static force:
$F_{stat} > 0$
F_{dyn} Dynamic force:

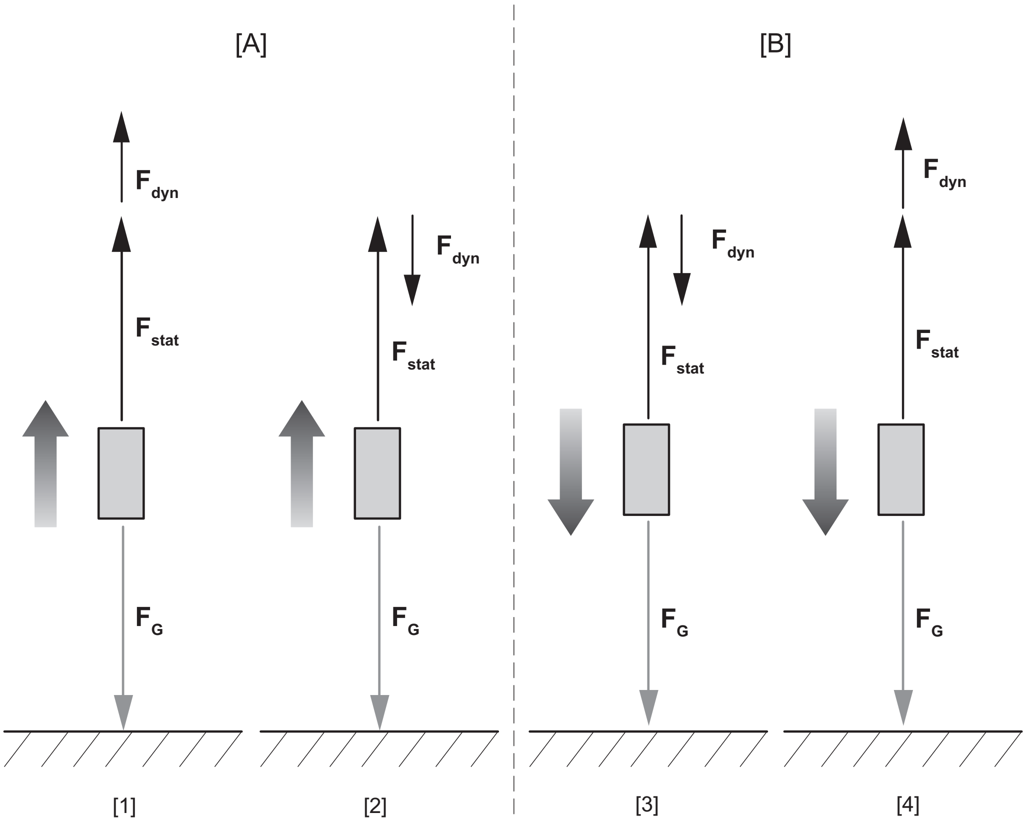
- $F_{dyn} > 0$ (acceleration)
- $F_{dyn} < 0$ (deceleration)

The direction of movement is defined as the reference system for the signs of the occurring forces (see chapter "Considering basic movements and quantity definitions" (→ 🗎 21)). Therefore, in terms of the total amount force to be applied or absorbed by the drive, it is irrelevant whether the application is moved horizontally forward or backward. The static force and the dynamic force are always positive in applications with horizontal direction of movement during acceleration. The dynamic force is always negative during deceleration. The frictional forces always counteract the movement and are negative.

26636735/EN – 07/2020

4.3.2 Forces for vertical movement

The following figure shows examples of forces for vertical motion in relation to the direction of movement. The gray arrows represent the application-side load and the black arrows represent the force to be applied or absorbed by the drive.



23177826315

[A] Direction of movement upward
[B] Direction of movement downward
[1] Upward acceleration (F_{stat} > 0, F_{dyn} > 0)
[2] Downward deceleration (F_{stat} > 0, F_{dyn} < 0)
[3] Downward acceleration (F_{stat} < 0, F_{dyn} > 0)
[4] Downward deceleration (F_{stat} < 0, F_{dyn} < 0)
F_{stat} Static force
F_{dyn} Dynamic force
F_G Gravitational force

In a lifting application, meaning an application with a vertical direction of movement, the gravitational force always acts downwards, regardless of the direction of movement. The static force required for compensation always acts in an upward direction and changes its sign when the direction of movement is reversed from upward to downward. Refer to chapter "Calculation conventions at SEW-EURODRIVE" (→ 🗎 18).

The dynamic force for acceleration of the application remains positive. The dynamic force is negative for deceleration. The friction in guides is usually negligible in lifting applications.

In order to calculate the resulting force to be applied or absorbed by the drive in dynamic travel sections, the dynamic force is added to or subtracted from the static force similarly to the horizontal movement. The signs must be observed.

During upward movement, the static force is:

- Increased by the amount of dynamic force during acceleration:

  $F_{tot} = F_{stat} + F_{dyn}$
- Decreased by the amount of dynamic force during deceleration:

  $F_{tot} = F_{stat} - F_{dyn}$

During downward movement, the static force is:

- Increased by the amount of dynamic force during acceleration:

  $F_{tot} = -F_{stat} + F_{dyn}$
- Decreased by the amount of dynamic force during deceleration:

  $F_{tot} = -F_{stat} - F_{dyn}$

INFORMATION

Note that many applications use 2 motor rotation directions. Therefore, a measured torque may have a different sign than the calculated torque.

4.3.3 Static forces

In general, a distinction is made between 2 types of static forces. First, they are static forces that act to overcome friction and resistance to vehicle motions in travel applications and rotary applications with a horizontal motion component. The second type is the force that acts to compensate for gravitational force during lifting motions.

Friction force/force of resistance to vehicle motion

Any kind of movement of an object is inhibited by friction forces at the points of contact to other objects or media. These are mainly adhesive or sliding friction forces. The sliding friction force is described by a coefficient, whereas bearing friction, rolling friction and track friction can be combined in a formula when they appear at the same time. The different friction forces are added together for this purpose, see example in chapter "Simultaneous occurrence of bearing, rolling and track friction" (→ 🗎 34).

Adhesive and sliding friction force

Adhesive and sliding friction forces occur at the contact surfaces of solids that adhere to or slide along each other.

The static friction force calculates the static force required to move the application from the rest position. The sliding friction force describes the force to be overcome during the movement. Friction coefficients are required for calculating friction forces. These coefficients are determined through experiments, obtained from table collections or derived from user specifications. An overview of several friction coefficients can be found in the table appendix in chapter "Friction coefficients for different material combinations" (→ 🗎 127).

The friction force is dependent not only on the friction coefficient μ but also on the normal force F_N. The normal force F_N is the force with which the solids are pressed together.

26636735/EN – 07/2020

Adhesive or sliding friction force for horizontal movement:

$$F_f = F_N \times \mu = m \times g \times \mu$$

18014418020529547

F_f	= Friction force	$[F_f]$ = N
F_N	= Normal force	$[F_N]$ = N
μ	= Friction coefficient	[μ] = 1
m	= Mass	[m] = kg
g	= Gravitational acceleration (9.81 m s^{-2})	[g] = m s^{-2}

For applications with pronounced sliding friction, it is necessary to consider whether the static friction force at start is so great that it must be included in the calculation.

It is generally difficult to determine exact friction coefficients because they are heavily dependent on lubrication and the degree of contamination of the contact surfaces.

The normal force F_N corresponds to the gravitational force in applications with a horizontal direction of movement. In the case of an inclined plane, the angle of inclination to the horizontal is also considered. This angle divides the gravitational force into normal force and gravity resistance (see chapter "Gravitational force/gravity resistance" (→ 🗎 35)).

Normal force on the horizontal plane:

$$F_N = m \times g$$

18014418020531211

Normal force on the inclined plane:

$$F_N = m \times g \times \cos \beta$$

18014418020532875

F_N	= Normal force	$[F_N]$ = N
m	= Mass	[m] = kg
g	= Gravitational acceleration (9.81 m s^{-2})	[g] = m s^{-2}
β	= Angle of inclination to the horizontal plane	[β] = ° or rad

Bearing friction force

The bearing friction force is applied when the rolling elements roll in a rolling bearing or when the shaft extension and bearing shell slide in a sleeve bearing.

The bearing friction is determined by the bearing friction coefficient μ_{f_b} of the respective bearing type (see table appendix "Bearing friction coefficients" (→ 🗎 127)) and the normal force.

$$F_{f\_b} = F_N \times \mu_{f\_b} = m \times g \times \mu_{f\_b}$$

18014418020539915

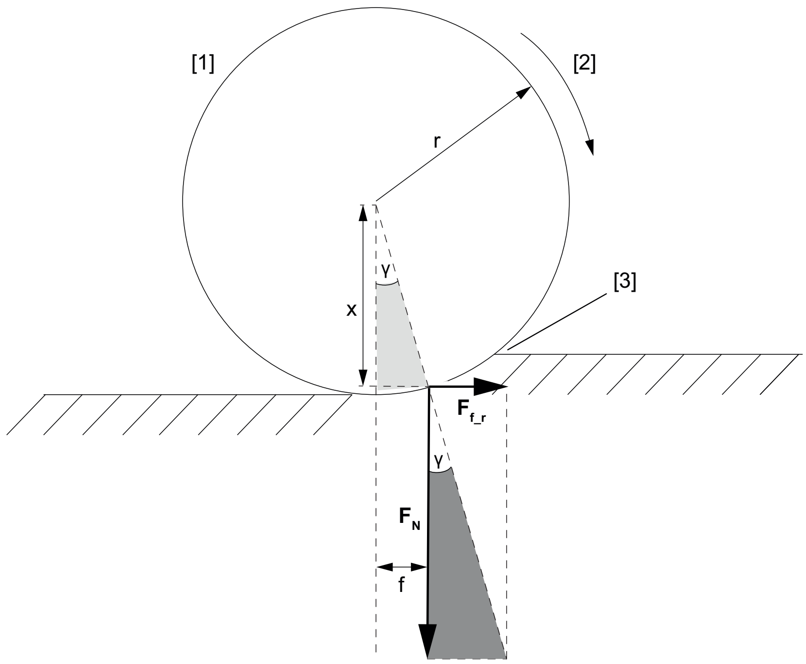
F_{f_b}	= Bearing friction force	$[F_{f\_b}]$ = N
F_N	= Normal force	$[F_N]$ = N
μ_{f_b}	= Bearing friction coefficient	$[\mu_{f\_b}]$ = 1
m	= Mass	[m] = kg
g	= Gravitational acceleration (9.81 m s^{-2})	[g] = m s^{-2}

Rolling friction force

The rolling friction force acts when an object rolls on a surface.

During rolling, the wheel, the ground or both are elastically deformed, and the movement is inhibited by sliding friction and walk components. The type of elastic deformation is dependent on the material combination and is described by an associated geometrical dimension, the lever arm of the rolling friction, f.

The formula for calculating the rolling friction force can be derived with the aid of the geometric relationships from the following figure.



23203685259

[1] Gear
[2] Direction of rotation
[3] Elastic deformation of the ground
F_{f_r} Rolling friction force
F_N Normal force
f Lever arm of the rolling friction
r Radius
x Adjacent leg
γ Angle of friction

The triangle with the dark gray background shows that the ratio of rolling friction force F_{f_r} and normal force F_N is equal to the tangent of the friction angle γ. Similarly, the triangle with the light gray background shows that the tangent of γ for small values of the lever arm of the rolling friction f can be approximated using the ratio of f to the radius r. The approximation used here assumes that the friction angle γ is small, so that the adjacent leg x corresponds approximately to the radius r. The formula for the rolling friction force is obtained by equating the two relationships.

Derivation of rolling friction force:

$$\tan\gamma = \frac{F_{f\_r}}{F_N} = \frac{f}{x}$$

Where	$f \ll r$	The following applies	$x \approx r$

26636735/EN – 07/2020

$$\tan\gamma \approx \frac{f}{r}$$

Equating with x = r and solving for F_{f_r} yields the rolling friction force:

$$F_{f\_r} = F_N \times \frac{f}{r} = F_N \times \mu_{f\_r} = m \times g \times \frac{2f}{d}$$

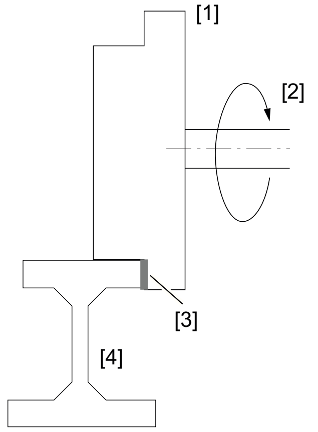
18014418020535563

F_{f_r}	= Rolling friction force	$[F_{f\_r}]$ = N
F_N	= Normal force	$[F_N]$ = N
μ_{f_r}	= Rolling friction coefficient	$[\mu_{f\_r}]$ = 1
f	= Lever arm of the rolling friction	[f] = mm
r	= Radius	[r] = mm
d	= Wheel diameter	[d] = mm
m	= Moved mass	[m] = kg
g	= Gravitational acceleration (9.81 m s^{-2})	[g] = m s^{-2}

If the lever arm of the rolling friction f is calculated in relation to the radius of the wheel, the resulting coefficient corresponds approximately to the friction coefficient $\mu_{f\_}$ for rolling friction. Common values for the lever arm of the rolling friction for different material combinations can be found in the table appendix in chapter "Rolling friction (lever arm of rolling friction)" (→ 🗎 128).

Track friction force

Rollers and wheels are guided by rails in various applications. In these applications, track friction can occur laterally between the gear and the guide rail.



23203439883

[1] Gear
[2] Direction of rotation
[3] Track friction surface
[4] Guide rail

26636735/EN – 07/2020

Track friction force:

$$F_{f\_t} = F_N \times c = m \times g \times c$$

18014418020542603

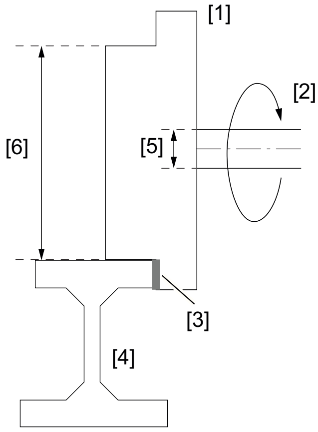
F_{f_t}	= Track friction force	$[F_{f\_t}]$ = N
F_N	= Normal force	$[F_N]$ = N
c	= Track friction coefficient	[c] = 1
m	= Mass	[m] = kg
g	= Gravitational acceleration (9.81 m s^{-2})	[g] = m s^{-2}

The table appendix "Coefficients for track and lateral friction" (→ 🗎 127) contains empirical values for the track friction coefficient c, which are included in calculation similarly to a friction coefficient.

Simultaneous occurrence of bearing, rolling and track friction

If bearing friction, rolling friction and track friction occur simultaneously on the wheel in one travel application, any frictional forces that occur can be added to a resistance to vehicle motion force. All friction forces can be combined in a formula consisting of the normal force and total friction coefficient.

The fact that the occurring friction forces act on different points of the wheel must be taken into account during the calculation. For example, the bearing friction force does not relate to the wheel diameter d but to the bearing diameter d_b.



24554503435

[1] Gear
[2] Direction of rotation
[3] Track friction surface
[4] Guide rail
[5] Bearing diameter d_b
[6] Wheel diameter d

The force of resistance to vehicle motion can be derived by adding together all friction torques. In this case, all forces are related to a common lever arm r.

Calculating the force of resistance to vehicle motion:

$$M_{tr} = M_{f\_b} + M_{f\_r} + M_{f\_t}$$
$$F_{tr} \times r = F_{f\_b} \times r_b + F_{f\_r} \times r + F_{f\_t} \times r$$

23688557835

26636735/EN – 07/2020

Division by the common lever arm r results in:

$$
\begin{aligned}
F_{tr} &= F_{f\_b} \times \frac{r_b}{r} + F_{f\_r} + F_{f\_t} \\
&= F_{f\_b} \times \frac{d_b}{d} + F_{f\_r} + F_{f\_t} \\
&= F_N \times \mu_{f\_b} \times \frac{d_b}{d} + F_N \times \frac{2f}{d} + F_N \times c
\end{aligned}
$$

$$
F_{tr} = F_N \times \left(\frac{2}{d} \times \left(\mu_{f\_b} \times \frac{d_b}{2} + f \right) + c \right) = m \times g \times \mu_{tr}
$$

23203212171

M_{tr}	= Torque of resistance to vehicle motion	$[M_{tr}]$ = Nm
M_{f_b}	= Bearing friction torque	$[M_{f\_b}]$ = Nm
M_{f_r}	= Rolling friction torque	$[M_{f\_r}]$ = Nm
M_{f_t}	= Track friction torque	$[M_{f\_t}]$ = Nm
r_b	= Bearing radius	$[r_b]$ = mm
r	= Radius (here: common lever arm to which all forces relate)	[r] = mm
F_{tr}	= Force of resistance to vehicle motion	$[F_{tr}]$ = N
F_{f_b}	= Bearing friction force	$[F_{f\_b}]$ = N
F_{f_r}	= Rolling friction force	$[F_{f\_r}]$ = N
F_{f_t}	= Track friction force	$[F_{f\_t}]$ = N
F_N	= Normal force	$[F_N]$ = N
μ_{f_b}	= Bearing friction coefficient	$[\mu_{f\_b}]$ = 1
d_b	= Bearing diameter	$[d_b]$ = mm
d	= Wheel diameter	[d] = mm
f	= Lever arm of the rolling friction	[f] = mm
c	= Track friction coefficient	[c] = 1
m	= Mass	[m] = kg
g	= Gravitational acceleration (9.81 m s^{-2})	[g] = m s^{-2}
μ_{tr}	= Total friction coefficient of the resistance to vehicle motion	$[\mu_{tr}]$ = 1

All the friction types listed here are assumed to be independent of the speed for the calculation. There are typical friction forces that occur in standard applications such as materials handling.

Friction types that are dependent on the speed or rotational speed include, e.g. viscous friction in stirrers or mixers, which will not be covered in greater detail here.

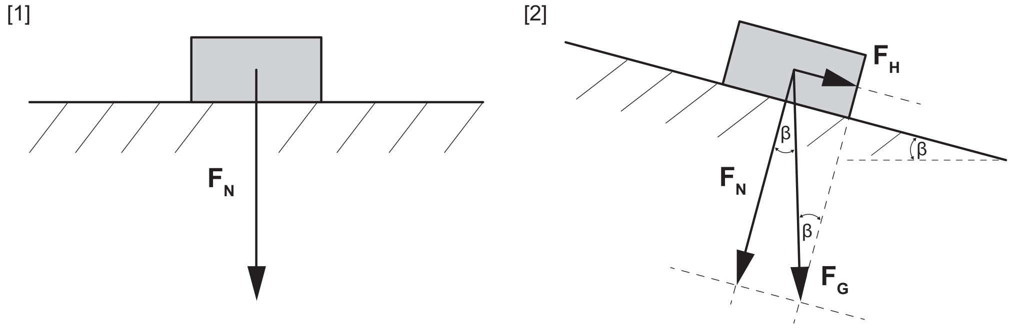
Gravitational force/gravity resistance

If the movement of the application is oblique or vertical, then the gravitational force on the mass to be moved contributes significantly to the forces to be overcome. In the case of a purely vertical lifting movement, the friction forces are usually negligible compared to the gravitational force (see figure in section "Forces for vertical movement" (→ 🗎 29)).

In the case of movements on the inclined plane, the static force required for an equilibrium of forces is calculated from the gravity resistance and the friction force.

26636735/EN – 07/2020

To calculate the gravity resistance, the gravitational force is distributed across the components perpendicular to and along the inclined plane. Normal force and gravity resistance are shown in the following figure.



18014418020551947

[1] Normal force F_N with horizontal movement
[2] Division of gravitational force F_G on the inclined plane into normal force F_N and gravity resistance F_H

Gravitational force:

$$F_G = m \times g$$

18014418020548619

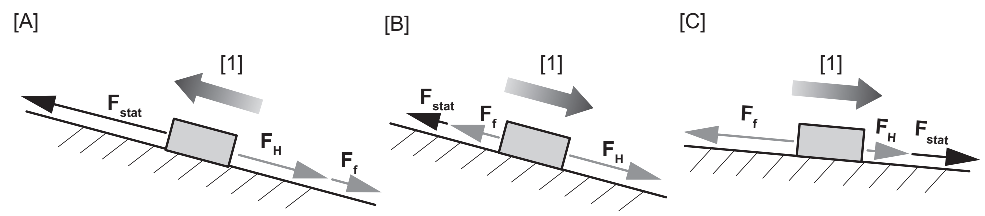
Gravity resistance:

$$F_H = m \times g \times \sin \beta$$

18014418020550283

F_G = Gravitational force	$[F_G]$ = N
m = Mass	[m] = kg
g = Gravitational acceleration	[g] = m s^{-2}
F_H = Gravity resistance	$[F_H]$ = N
β = Angle of inclination to the horizontal plane	[β] = ° or rad

It is important that the direction of gravity resistance, regardless of upward or downward movement, always point downward along the inclined plane, while the friction force always act counter to the direction of movement. The static force to be applied for force equilibrium counteracts the force resulting from the gravity resistance and friction force. The direction of the resulting static force depends on which of the two components is larger. The gray arrows represent the application-side load and the black arrow represents the force to be applied by the drive.



18014418020554123

[A] Static forces for upward movement on the inclined plane
[B] Static forces for downward movement on the inclined plane with $F_H > F_f$
[C] Static forces for downward movement on the inclined plane with $F_H < F_f$
[1] Direction of movement
F_{stat} Static force
F_H Gravity resistance
F_f Friction force

26636735/EN – 07/2020

4.3.4 Dynamic forces

The dynamic force for acceleration of the application is calculated from the mass to be moved and the desired acceleration value. For rotary movements, the required acceleration torque with the corresponding angular quantities is obtained instead of a linear force of acceleration.

Force of acceleration:

$$F_{dyn} = m \times a$$

18014418020557323

Acceleration torque:

$$M_{dyn} = J \times \alpha$$

18014418020558987

F_{dyn}	= Force of acceleration	$[F_{dyn}]$ = N
m	= Mass	[m] = kg
a	= Acceleration	[a] = m s^{-2}
M_{dyn}	= Acceleration torque	$[M_{dyn}]$ = Nm
J	= Mass moment of inertia	[J] = kg m^2
α	= Angular acceleration	[α] = s^{-2}

The sign of the dynamic force is positive during acceleration according to SEW-EURODRIVE conventions (see "Calculation conventions at SEW-EURODRIVE" (→ 🗎 18)) but negative during deceleration, meaning it counteracts the movement.

4.4 Mass moment of inertia

The mass moment of inertia J is required for calculating the dynamic load for rotary applications.

In applications with linear direction of movement, the mass moment of inertia J is calculated from the load to be moved and all moving transmission elements (such as toothed belts, sprockets, drums, etc.).

The acceleration of the load moment of inertia is usually the dominant load variable. The acceleration of additional inertia such as transmission elements is often negligible by comparison. For highly dynamic applications, all inertia of the application may need to be taken into account.

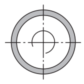
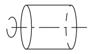
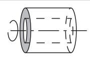
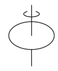
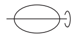
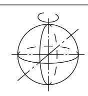
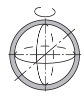
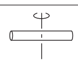
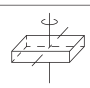
4.4.1 Mass moment of inertia of a rigid object during rotation

Inertia is the tendency of an object to resist change in its motion state. For linear movement, inertia is expressed by the mass m of the object. For rotary movement, the physical quantity is called the mass moment of inertia J. The mass moment of inertia is the resistance of a rigid object to a change in its rotary motion around the rotary axis.

The mass moment of inertia depends on the mass distribution in relation to the rotary axis. The greater the distance between a mass element and the rotary axis, the greater its influence on the mass moment of inertia. The mass itself is linear in the calculation of the mass moment of inertia. The distance is squared.

26636735/EN – 07/2020

For typical geometrical objects, the mass moment of inertia J_{cg} can be calculated using simple formulas. Formulas for commonly used objects with fixed rotary axes are summarized in the following table.

Object	Position of the rotary axis	Symbol	Mass moment of inertia J_{cg}
Ring, thin Hollow cylinder, thick-walled	Perpendicular to the circular ring plane		$J_{cg} = m \times r^2$
Solid cylinder	Longitudinal axis		$J_{cg} = \frac{1}{2} \times m \times r^2$
Hollow cylinder, thick-walled	Longitudinal axis		$J_{cg} = \frac{1}{2} \times m \times \left(r_1^2 + r_2^2\right)$
Circular disk	Perpendicular to the plane of the disk		$J_{cg} = \frac{1}{2} \times m \times r^2$
Circular disk	Axis of symmetry in the plane of the disk		$J_{cg} = \frac{1}{4} \times m \times r^2$
Ball	Through midpoint		$J_{cg} = \frac{2}{5} \times m \times r^2$
Ball sleeve, thin-walled	Through midpoint		$J_{cg} = \frac{2}{3} \times m \times r^2$
Rod, thin, with length l	Perpendicular to the middle of the rod		$J_{cg} = \frac{1}{12} \times m \times l^2$
Block, with width and height edge lengths b, c and d	Parallel to edge c (here: height)		$J_{cg} = \frac{1}{12} \times m \times (b^2 + d^2)$

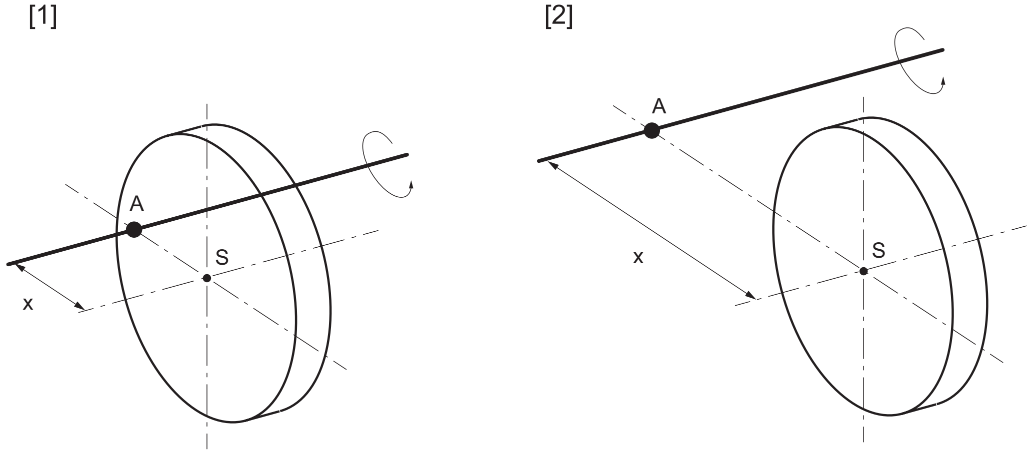
J_{cg}	= Mass moment of inertia of an object with reference to a rotary axis through the center of gravity S	$[J_{cg}]$ = kg m^2
m	= Mass of object	[m] = kg
r	= Radius	[r] = m
$r_{1,2}$	= Inner and outer radius	$[r_{1,2}]$ = m
l	= Rod length	[l] = m
b, c, d	= Edge length	[b, c, d] = m

If the axis of rotation does not pass through the center of gravity of the object, the mass moment of inertia of the object around the rotary axis is determined by applying Steiner's theorem.

The total mass moment of inertia for a parallel-shifted rotary axis is the sum of the intrinsic component J_{cg} (mass moment of inertia of the object around an axis of symmetry through its center of gravity) and the Steiner's theorem component (extension as product of the mass of the object and the distance squared).

The following figure [1] shows a geometrical object whose rotary axis does not go through the center of gravity S, but through the point A. The rotary axis parallel to the center of gravity does not have to lie within the object, see Figure [2].

Steiner's theorem is applicable in both cases.



9007219039599243

Steiner's theorem:

$$J = J_{cg} + m \times x^2$$

18014418020639627

J	= Mass moment of inertia of the object, relative to a rotary axis through A	$[J] = kg\ m^2$
J_{cg}	= Mass moment of inertia of an object with reference to a rotary axis through the center of gravity S	$[J_{cg}] = kg\ m^2$
m	= Mass of object	[m] = kg
x	= Distance between both parallel axes	[x] = m

The following applies: The larger the distance x of the axes, the more the Steiner's theorem component predominates. The intrinsic component of the mass moment of inertia of the object around its own center of gravity is negligible at a sufficiently large distance. The object then behaves like a rotating mass point.

4.4.2 Mass moments of inertia in a drive train

The mass moment of inertia ratio of reduced load inertia J_x to motor inertia J_{Mot} is crucial for selecting the motor. Among other things, it influences the control quality of controlled drives or the startup and braking behavior of line-powered drives. In order to compare the mass moments of inertia, both mass moments of inertia must be related to a common point. In general, the motor shaft is selected as the reference point.

For rotary applications, such as rotary tables or corner transfer units, the total mass moment of inertia of the moving load can be approximately calculated by segmenting it into simple geometric shapes. For example, a rotary table can be approximated as a cylinder, and the workpieces on the table might be approximated as blocks and related to the axis of rotation of the rotary table using Steiner's theorem.

Reducing the inertia of rotary movements

The mass moment of inertia of the load is then converted to the motor shaft by dividing it by the square of the total ratio.

Mass moment of inertia of the load reduced to the motor shaft:

$$J_x = \frac{J}{i_{tot}^2} = J \times \left(\frac{n_L}{n_{Mot}} \right)^2$$

18014418020642315

J_x	= Mass moment of inertia of the load reduced to the motor shaft	$[J_x]$ = kg m^2
J	= Mass moment of inertia of the load	$[J]$ = kg m^2
i_{tot}	= Total gear ratio between application and motor	$[i_{tot}]$ = 1
n_L	= Rotational speed of the application	$[n]$ = min^{-1}
n_{Mot}	= Motor speed	$[n_{Mot}]$ = min^{-1}

Deriving the inertia reduction of rotary movement

The inertia reduction can be mathematically derived from the conservation of rotational energy. For the sake of simplicity, it is assumed that the application does not have an additional transmission. If there is an additional transmission, its gear ratio is included in the total gear ratio.

Influence of gear ratio on mass moment of inertia:

$$E_{rot} = E_{rot\_x}$$

$$\frac{1}{2} J \times \omega_G^2 = \frac{1}{2} J_x \times \omega_{Mot}^2$$

$$J_x = J \times \frac{\omega_G^2}{\omega_{Mot}^2} = J \times \left(\frac{\omega_G}{\omega_{Mot}} \right)^2 = J \times \left(\frac{n_G}{n_{Mot}} \right)^2$$

18014418020643979

With $i_G = \frac{n_{Mot}}{n_G}$ or $\frac{n_G}{n_{Mot}} = \frac{1}{i_G}$ the following results:

$$J_x = J \times \left(\frac{1}{i_G} \right)^2$$

$$J_x = \frac{J}{i^2_G}$$

18014418020647307

E_{rot}	= Rotational energy of the load	$[E_{rot}]$ = J
E_{rot_x}	= Rotational energy of the load reduced to the motor shaft	$[E_{rot\_x}]$ = J
J	= Mass moment of inertia of the load	$[J]$ = kg m^2
ω_G	= Angular speed of gear unit output	$[\omega_G]$ = s^{-1}
J_x	= Mass moment of inertia of the load reduced to the motor shaft	$[J_x]$ = kg m^2
ω_{Mot}	= Angular speed of the motor	$[\omega_{Mot}]$ = s^{-1}
n_G	= Output speed of the gear unit	$[n_G]$ = min^{-1}
n_{Mot}	= Motor speed	$[n_{Mot}]$ = min^{-1}
i_G	= Gear unit ratio	$[i_G]$ = 1

26636735/EN – 07/2020

Inertia reduction of linear movements

In applications with linear movement, the linearly moving mass, depending on the radius of the transmission element, acts as the mass moment of inertia J on the output shaft. The linearly moving mass acts like a punctiform mass with distance r to the center of rotation without the influence of its geometry.

A formula for the mass moment of inertia of the linearly moving mass reduced to the motor shaft can be derived from this. This reduced mass moment of inertia depends only on the linear speed, the moving mass and the motor speed.

Linearly moving mass as mass moment of inertia:

with $v = \omega \times r$ or $r = \frac{v}{\omega}$ the following results:

$$J = m \times r^2 = m \times \left(\frac{v}{\omega}\right)^2 = m \times \left(\frac{v}{\frac{2\pi \times n}{60}}\right)^2 = \left(\frac{60}{2\pi}\right)^2 \times m \times \left(\frac{v}{n}\right)^2$$

$$= 91.2 \times m \times \left(\frac{v}{n}\right)^2$$

27021617275398283

Substituting n_{Mot} results in J_x:

$$J_x = 91.2 \times m \times \left(\frac{v}{n_{Mot}}\right)^2$$

27021617275401611

v	= Speed of the application	$[v] = m\ s^{-1}$
ω	= Angular speed	$[\omega] = s^{-1}$
r	= Radius	[r] = mm
J	= Mass moment of inertia of the load	$[J] = kg\ m^2$
m	= Moved mass	[m] = kg
n	= Speed	$[n] = min^{-1}$
J_x	= Mass moment of inertia of the load reduced to the motor shaft	$[J_x] = kg\ m^2$
n_{Mot}	= Motor speed	$[n_{Mot}] = min^{-1}$

Reducing the inertia of the spindle drive

Based on the inertia reduction of linear movements, the mass moment of inertia reduced to the motor shaft can also be specified as a function of the spindle pitch:

$$J_x = m \times \left(\frac{p}{2000 \times \pi}\right)^2$$

9007224523676555

J_x	= Mass moment of inertia of the load reduced to the motor shaft	$[J_x] = kg\ m^2$
m	= Moved mass	[m] = kg
p	= Spindle pitch	[p] = mm

For more information, see chapter "Special case of spindle drives" (→ 🗎 46).

26636735/EN – 07/2020

4.5 Efficiency

The efficiency describes the effectiveness of energy transfer in a driven machine as a dimensionless ratio of the output-side available power P_2 to the input-side supplied power P_1. The efficiency always has a value less than 1.

The difference between these power ratings are losses that occur in any electrical or mechanical system, for example, caused by ohmic resistances of components in a circuit or by friction between mechanical components. The various losses occurring within a system are largely converted to heat and released into the environment.

Definition of efficiency:

$$\eta = \frac{P_2}{P_1}$$

18014418020663307

η	=	Efficiency	$[\eta] = 1$
P_2	=	Output or available power	$[P_2] = W$
P_1	=	Power supplied	$[P_1] = W$

The following efficiencies are considered in the further course of project planning:

- Application and additional transmission
- Gear unit
- Motor
- Frequency inverter

4.5.1 Application and additional transmission

The interaction of the mechanical transmission elements within an application results in losses that enter into the calculation as different efficiencies. The efficiencies of the mechanical transmission elements can be found in the data sheets of the manufacturer, determined empirically or obtained from the customer. Values for different transmission elements can be found in the table appendix "Efficiencies of transmission elements" (→ 🗎 126).

All load-side efficiencies are multiplied and combined into a load efficiency $\eta_L$for subsequent calculations.

4.5.2 Gear unit

There are essentially 2 types of losses for gear units:

- Tooth friction losses
- Churning losses

Tooth friction losses

Tooth friction losses may be approximated depending on the gear unit type for R, F and K series 7 gear units with a 1.5 to 2% reduction in input torque per gear unit stage. Detailed information can be found in the product catalogs or the relevant engineering tools.

The gear unit efficiency must be determined on a case-by-case basis since the efficiency of S and W gear units is particularly dependent on the gear ratio and motor speed. The efficiency η during the warm-up phase can be additionally reduced due to the temperature-dependent viscosity of the gear unit oil.

26636735/EN – 07/2020

The retrodriving efficiency $\eta'_G = 2 - 1/\eta$ applies to retrodriving torques on the output shaft of helical-worm and SPIROPLAN® gear units. If $\eta_G \leq 0.5$, this can lead to self-locking. More detailed information can be found in the relevant product catalogs.

INFORMATION

Consult SEW-EURODRIVE for technical information on using self-locking.

Retrodriving efficiency

Similar to the gear unit efficiency η_G, the retrodriving efficiency η'_G describes the ratio of available power and supplied power during reverse operation of the gear unit. Reverse operation of the gear unit occurs when the energy flows from the application to the motor via the gear unit. This behavior usually occurs during regenerative operation of the motor.

Churning losses

Churning losses are caused by the displacement effect of the gear wheels in the oil and are highly dependent on mounting position and rotational speed. They do not have to be considered in the initial approximation of the drive selection. However, these losses can be considerable in some applications. In this case, the selection of the drive is modified later. High churning losses are expected to occur if at least one of the following criteria applies:

- High input speed (e.g. > 2000 min^{-1}).
- High oil level due to an unfavorable mounting position depending on the gear unit type.
- High oil viscosity.
- Low to extremely low ambient temperatures.

4.5.3 Motor

Depending on the motor design and type, motor losses can be subdivided into stator losses and rotor losses.

Stator losses are primarily divided into:

- Load-dependent ohmic losses.
- Frequency-dependent iron losses.

Rotor losses mainly arise from ohmic losses in the conductor bars.

In general, load-independent and load-dependent losses occur. The higher the relative load-independent losses relative to the rated power, the worse the partial load efficiency of the motor.

Consequently, the overall efficiency of asynchronous AC motors depends on various factors. This includes, among others:

- Energy efficiency class
- Size
- Operating point and capacity utilization during frequency inverter or line operation
- Winding type and condition of laminated cores
- Additional losses (e.g., fan, friction on bearings and seals)

Further losses occur in the motor cable. These are mainly ohmic losses, which depend on the cable length, the cable cross section and the cable type.

4.5.4 Frequency inverter

During frequency inverter operation, the conversion of electrical energy between the grid and the motor causes losses. These are largely determined by the control elements of the power electronics, such as the switching transistors in the DC-AC inverter stage. These losses are influenced by the load of the motor current to be switched and by the switching frequency of the switching transistors. This switching frequency is specified as the frequency of the pulse width modulation (PWM).

Common values for the PWM frequency are 2.5 kHz, 4 kHz, 8 kHz, 12 kHz and 16 kHz. The frequencies are predominantly in the audible range, with the exception of the 12 kHz and 16 kHz frequencies. However, the advantages of quieter operation with high PWM frequency are offset by the disadvantages of larger losses in the frequency inverter. The losses are about twice as high at 16 kHz than at 4 kHz.

The efficiency of frequency inverters can be estimated at approximately 90% for smaller motor outputs (< 550 W) and for operation at 16 kHz. The efficiency increases to more than 95% as the device power increases (> 7.5 kW motor power).

For project planning, the inverter efficiency is only required for creating the energy balance of a drive system (e.g. for hoists with optional regenerative energy supply to the grid). In this case, if no exact information about the power loss of the frequency inverter is available, then 95% efficiency can be assumed for power ratings greater than 1 kW.

4.5.5 Consideration of efficiencies in project planning

This section describes how efficiencies are taken into account during drive selection. In accordance with chapter "Calculation conventions at SEW-EURODRIVE" (→ 🗎 18), the following distinction is made based on the sign of the force or torque.

26636735/EN – 07/2020

Efficiency in motor mode

In motor mode (positive force sign), the force to be applied by the drive increases according to the efficiency.

$$F_\eta = \frac{F}{\eta} \qquad M_\eta = \frac{M}{\eta}$$

F_η	=	Force to be applied as a function of the efficiency (motor mode)	$[F_\eta]$ = N
F	=	Required force	[F] = N
η	=	Efficiency	[η] = 1
M_η	=	Torque to be applied as a function of the efficiency (motor mode)	$[M_\eta]$ = Nm
M	=	Required torque	[M] = Nm

Efficiency in generator mode

In generator mode (negative force sign), the force to be absorbed is reduced as a function of the efficiency. Note that the regenerative efficiency η' in generator mode can differ from the efficiency η in motor mode.

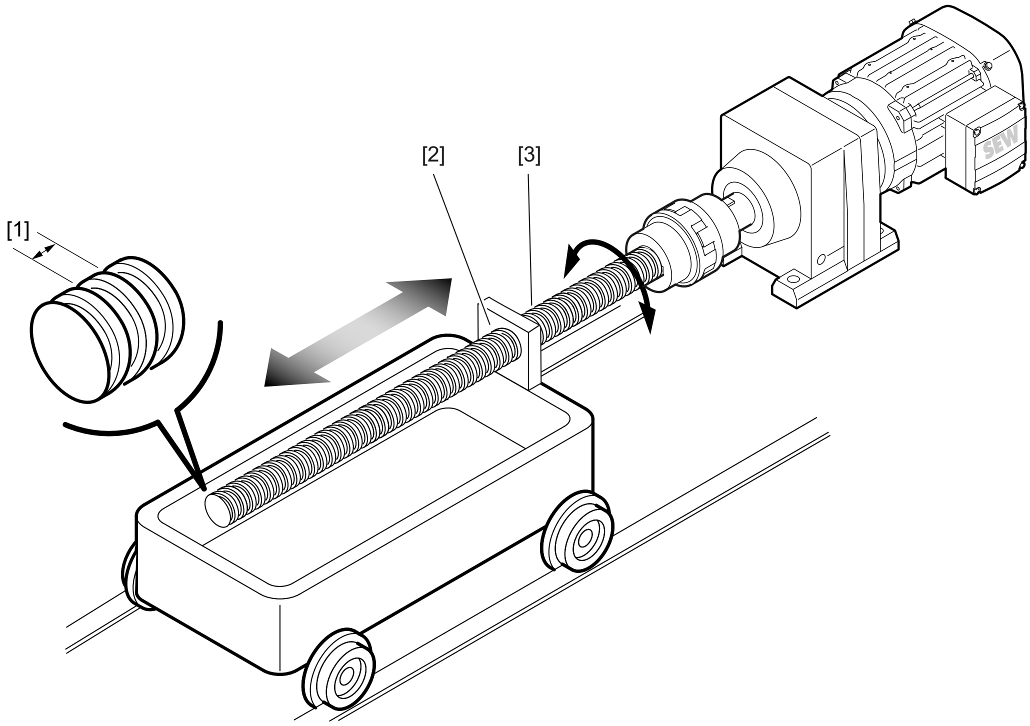
$$F'_\eta = F \times \eta' \qquad M'_\eta = M \times \eta'$$

F'_η	=	Force to be absorbed as a function of the efficiency (regenerative)	$[F'_\eta]$ = N
F	=	Force to be absorbed	[F] = N
η'	=	Retrodriving efficiency	[η'] = 1
M'_η	=	Torque to be absorbed as a function of the efficiency (generator mode)	$[M'_\eta]$ = Nm
M	=	Torque to be absorbed	[M] = Nm

26636735/EN – 07/2020

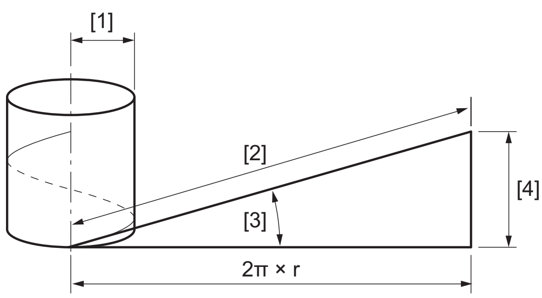
4.6 Special case of spindle drives

Spindle drive are a special type of drive. The following figure shows a spindle drive with a spindle that moves a rail-guided vehicle in a linear direction away from or towards the drive via a spindle nut. Spindle drives are used in applications such as scissor lift tables or for clamps in sheet metal processing. The calculations for spindle drives deviate from the general application-side calculations.



24856985227

[1] Spindle pitch
[2] Spindle nut
[3] Spindle



18014418295410827

[1] Spindle radius r
[2] Unwinding of helical line
[3] Incline angle β
[4] Spindle pitch p

The spindle pitch is calculated as a height difference in the axial direction by unwinding a complete revolution of the spindle, as shown in the figure above. This equals the stroke of a revolution.

26636735/EN – 07/2020

The spindle pitch p is given in millimeters.

$$n = \frac{v \times 60000}{p}$$

18014418020698379

n	= Speed	$[n] = min^{-1}$
v	= Axial speed	$[v] = m\ s^{-1}$
p	= Spindle pitch	[p] = mm

In order to obtain the torque as a function of the spindle pitch, the application-side force is calculated with the spindle pitch and the factor 2π in the denominator.

Torque of a spindle in motor mode:

$$M = \frac{F \times p}{2\pi \times \eta_{Spi} \times 1000}$$

18014418020705035

M	= Torque	[M] = Nm
F	= Application-side force (e.g., friction, hoist load)	[F] = N
p	= Spindle pitch	[p] = mm
η_{Spi}	= Spindle efficiency	$[\eta_{Spi}] = 1$

Torque of a spindle in generator mode:

$$M = \frac{F \times p}{2\pi \times 1000} \times \eta_{spi}$$

23343748875

M	= Torque	[M] = Nm
F	= Application-side force (e.g., friction)	[F] = N
p	= Spindle pitch	[p] = mm
η_{Spi}	= Spindle efficiency	$[\eta_{Spi}] = 1$

Calculating the reduced load moment of inertia on the spindle drive

Since spindle drives do not have a ratio in the sense of a gear unit ratio, the procedure for calculating the load moment of inertia J_x reduced to the motor shaft differs depending on the available values.

1. If the speed and motor speed are known, the previously derived relationship can be used to calculate the reduced load moment of inertia.

$$J_x = 91.2 \times m \times \left(\frac{v}{n_{Mot}} \right)^2$$

27021617275401611

J_x	= Mass moment of inertia of the load reduced to the motor shaft	$[J_x] = kg\ m^2$
m	= Moved mass	[m] = kg
v	= Linear speed of load	$[v] = m\ s^{-1}$
n_{Mot}	= Motor speed	$[n_{Mot}] = min^{-1}$

2. If these values are not available, they can be calculated from the spindle pitch. Starting from the general definition of the mass moment of inertia J = m × r^2, the radius r is substituted as follows:

   With each spindle revolution, the spindle circumference U is theoretically unwound once, while an axial distance equal to the spindle pitch is covered.

$$U = 2\pi r \triangleq p$$

$$r \triangleq \frac{p}{2\pi}$$

25268931851

U	=	Spindle circumference	[U[= mm
r	=	Radius	[r] = mm
p	=	Spindle pitch	[p] = mm

INFORMATION

The radius r does not correspond to a real dimension of the spindle. It is a tool for specifying the reduced load moment of inertia as a function of the spindle pitch. The value of the spindle radius itself must not be used for r.

Substituting r = p / (2π) with p in millimeters results in:

$$J_x = m \times \left(\frac{p}{2000 \times \pi} \right)^2$$

9007224523676555

J_x	= Mass moment of inertia of the load reduced to the motor shaft	[J_x] = kg m^2
m	= Moved mass	[m] = kg
p	= Spindle pitch	[p] = mm

26636735/EN – 07/2020

5 Project planning for controlled drives

During project planning for controlled drives, calculations for selecting a suitable gear unit are carried out once the application requirements have been calculated. Next, the calculations are carried out for selecting a suitable motor, frequency inverter and required options.

INFORMATION

For frequency inverter operation, the motor must be designed with insulation protection class ISO F.

5.1 Calculating and selecting the gear unit

This section covers the following topics:

- Output end torques
- Selecting the gear unit
- External forces (overhung loads and axial loads)

5.1.1 Output end torques

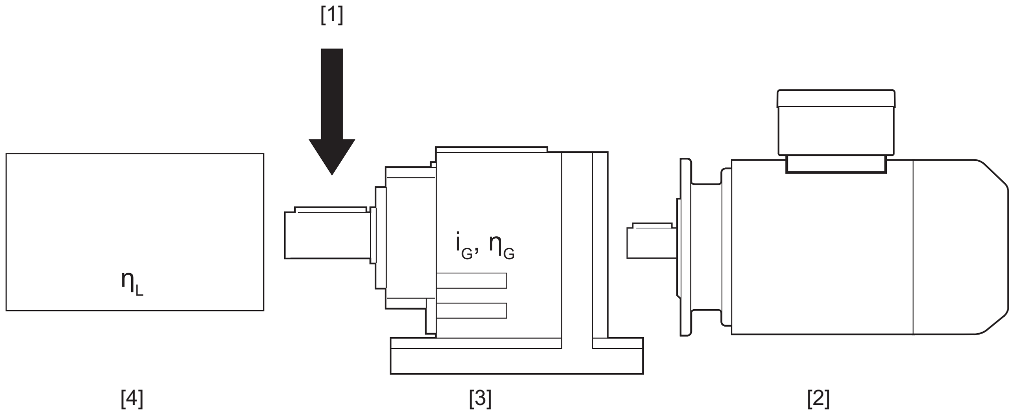
First, the output-side torques M_n for all travel sections are calculated from the previously determined forces, whereby the index n identifies the respective travel section.

As a rule, the maximum load for horizontal movements as well as rotary and vertical upward movements occurs in the "acceleration" travel section. In certain applications, the maximum load may also occur in the "deceleration" travel section. The force situations occurring here must be examined in detail on a case-by-case basis.

Torque equations of different movement types by travel section

Travel section	Horizontal, rotary or upward vertical movement	Vertical downward movement
Acceleration	$M_{ac} = M_{stat} + M_{dyn} = F_{stat} \times r + F_{dyn} \times r$	$M_{ac} = - M_{stat} + M_{dyn} = -F_{stat} \times r + F_{dyn} \times r$
Constant speed	$M_{const} = M_{stat} = F_{stat} \times r$	$M_{const} = - M_{stat} = - F_{stat} \times r$
Deceleration	$M_{dec} = M_{stat} - M_{dyn} = F_{stat} \times r - F_{dyn} \times r$	$M_{dec} = - M_{stat} - M_{dyn} = - F_{stat} \times r - F_{dyn} \times r$

26636735/EN – 07/2020



18014418297494411

[1] Reference point
[2] Motor
[3] Gear unit
[4] Application-side load
η_L Load efficiency
i_G Gear unit ratio
η_G Gear unit efficiency

The position [1] marks the current reference point. All previously calculated data refer to the application-side load [4]. The load efficiency for calculating the required torque of the gear unit output is considered as follows.

$M_n > 0$, motor mode:

$$M_{G\_n} = \frac{M_n}{\eta_L}$$

18014418020717963

Force and torque have a negative sign when they have a regenerative effect (see chapter "Reference systems and signs" (→ 🗎 18)). In this case, the load efficiency is considered as follows.

$M_n < 0$, generator mode:

$$M'_{G\_n} = M_n \times \eta_L$$

18014418020719627

M_{G_n}	= Torque of gear unit output in travel section n (e.g., "acceleration") including load efficiency (motor mode)	$[M_{G\_n}]$ = Nm
M_n	= Application-side torque without load efficiency in the travel section n	$[M_n]$ = Nm
η_L	= Load efficiency	$[\eta_L]$ = 1
M'_{G_n}	= Torque of gear unit output in travel section n (e.g., "deceleration") including load efficiency (generator mode)	$[M'_{G\_n}]$ = Nm

5.1.2 Selecting the gear unit

The gear unit is selected based on the application-side load variables as well as the ambient conditions at the place of use and other user specifications.

Examples of application conditions include:

- Available installation space in the application
- Mechanical gear unit design for connection to the application
  – Axially parallel gear unit

26636735/EN – 07/2020

  - Right-angle gear unit
  - Solid-/hollow-shaft gear unit
- Efficiency/energy efficiency
- Noise characteristics
- Rotational clearance/positioning accuracy
- Special version for agitator, electrified monorail system, precision gear unit, compound gear unit

The following table compares different gear unit types based on a several technical characteristics (relative to n_{Mot} = 1400 min^{-1}). A gear unit type is selected that meets the basic requirements of the application.

	Helical gear units R..	**Parallel-shaft helical gear unit F..**	**Helical-bevel gear unit K..**	**Helical-worm gear units S..**	**SPIROPLAN® gear unit W..**
Axially parallel gear unit	yes	yes	no	no	no
Right-angle gear unit	no	no	yes	yes	yes
Shaft design	Solid shaft only	Solid/hollow shaft	Solid/hollow shaft	Solid/hollow shaft	Solid/hollow shaft
Good efficiency	++	++	++	-	0
Low wear	++	++	++	-	0
Favorable noise behavior	+	+	+	++	0
Low procurement costs	0	0	-	+	++
Gear unit ratio[1)]	> 1.3 – 307	> 3 – 281	> 3 – 197	4 – 288	> 3 – 75
Permitted torque[1)] in Nm	50 – 18000	130 – 18000	80 – 50000	92 – 4000	12 – 180

1) According to size

The gear unit size is selected based on the following key selection criteria in addition to the aforementioned possible selection criteria for the gear unit type.

- Output torque M_{G_max} (maximum value of M_{G_n})
- Ideal gear unit ratio
- Overhung load
- Axial load

Preselection by torque

Torque is the first selection criterion since it is the determining dimensioning variable of the gear unit in the majority of applications. External forces are checked afterwards. A gear unit size is selected with a continuously permitted output torque M_{a_max} greater than the maximum torque occurring at the gear unit output M_{G_max}.across all travel sections.

Torque criterion:

$$M_{G\_max} \leq M_{a\_max}$$

18014418020722315

M_{G_max}	= Maximum torque of the gear unit output, including load efficiency, considered across all travel sections	$[M_{G\_max}]$ = Nm
M_{a_max}	= Continuously permitted output torque of the gear unit	$[M_{a\_max}]$ = Nm

Gear unit ratio

In the following step, a gear unit ratio i_G is selected that is closest to the calculated ideal gear unit ratio i_{G_id}. It does not matter whether the selected value i is higher or lower than the ideal value i_{G_id}.

$$i_G \approx i_{G\_id}$$

18014418020723979

i_G	= Gear unit ratio	$[i_G]$ = 1
i_{G_id}	= Calculated ideal gear unit ratio	$[i_{G\_id}]$ = 1

INFORMATION

Note that the continuously permitted output torque of the gear unit, M_{a_max}, is reduced for small gear unit ratios relative to the torque class of the gear unit size. If the continuously permitted output torque of the gear unit M_{a_max} for the selected gear ratio is smaller than the maximum torque of the application, a larger gear unit or a different gear unit ratio must be selected.

For controlled drives with higher gear unit ratios, the actual motor speed can be above the rated motor speed in the field weakening range (see chapter "Thermal motor utilization" (→ 🗎 61)). The advantage of a higher gear unit ratio is that the torque load of the motor decreases during acceleration. At the same time, ensure that the available motor torque is in the field weakening range and the overload capacity is thereby reduced.

If a lower gear ratio is selected, i.e., if the motor is operated below the rated speed, the motor has more overload reserve available. However, the motor current and therefore the heat buildup increases due to the higher torque load of the motor. This must be taken into account in the case of borderline thermal motor load.

The motor speed n_{Mot} can be calculated based on the output speed required by the application and the selected gear unit ratio. The motor speed can be adjusted as a setpoint on the frequency inverter.

Motor speed (setpoint input):

$$n_{Mot} = n_G \times i_G$$

18014418020725643

n_{Mot}	= Motor speed	$[n_{Mot}]$ = min^{-1}
n_G	= Output speed of the gear unit	$[n_G]$ = min^{-1}
i_G	= Gear unit ratio	$[i_G]$ = 1

Thermal capacity utilization of the gear unit

The thermal capacity utilization of the gear unit must be considered if at least one of the following criteria is met:

- High input speeds ≥ 2500 min^{-1}.

  For example, for 2-pole motors, servo applications or 87 Hz operation.
- Small gear unit ratio in combination with high input speed.

  High circumferential speeds of gearing components lead to increased churning losses.
- Unfavorable mounting positions (e.g., M2 and M4) with high oil levels.

  In these mounting positions, the gearing running in the oil can cause high churning losses and increase the temperature of the gear unit.

For high oil temperatures (e.g., > 80 °C), measures can be taken to counteract premature aging of the lubricant and increased wear. These include the use of synthetic lubricants as well as more thermally resistant materials (e.g., FKM oil seal).

If required, SEW-EURODRIVE can conduct a thermal test of the gear unit.

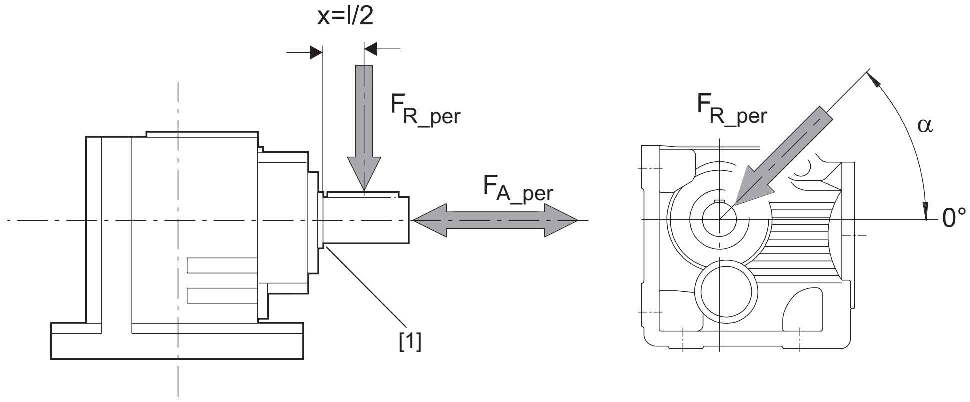
5.1.3 External forces (overhung loads and axial loads)

External forces can be divided into overhung loads and axial loads. Overhung loads act perpendicular to the axis of rotation of the output shaft. Axial loads act coaxially to the output shaft.

INFORMATION

If overhung and axial loads are applied to the gear unit at the same time, please contact SEW-EURODRIVE.

The permitted overhung load F_{R_per} is related to the center of the output shaft x = l/2 and may be applied to the gear unit in the same way as the continuously permitted output torque M_{a_max}. The permitted overhung load F_{R_per} applies to all force application angles, α.

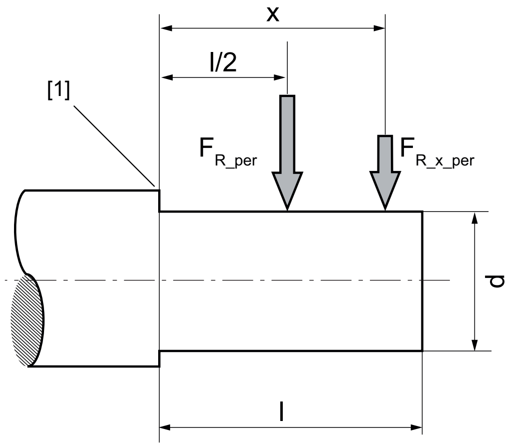


24357112203

[1]	Shaft shoulder	
x	= Distance from shaft shoulder to force application point	[x] = m
l	= Length of output shaft	[l] = mm
F_{R_per}	= Permitted overhung load at distance x = l/2 to shaft shoulder	[F_{R_per}] = N
F_{A_per}	= Permitted axial load (pull or push)	[F_{A_per}] = N
α	= Force application angle	[α] = °

An off-center force application point results in a different permitted overhung load $F_{R\_x\_per}$. If the force application point is closer than x = l/2 to the shaft shoulder, then the permitted overhung load F_{R_per} remains available undiminished. If the force application point is more than x = l/2 away from the shaft collar, the permitted overhung load is reduced depending on the distance x due to the greater bending load of the shaft and the change in the bearing load.

Permitted overhung loads of the gear units and the conversion of these forces to a force application point deviating from x = l/2 are documented in the product catalogs. In the case of an off-center force application point, the value $F_{R\_x\_per}$ must be used instead of F_{R_per} in the subsequent formulas.



25907720331

[1]		Shaft shoulder	
d	=	Shaft diameter	[d] = mm
x	=	Distance from shaft shoulder to force application point	[x] = mm
l	=	Length of output shaft	[l] = mm
F_{R_per}	=	Permitted overhung load at distance x = l/2 to shaft shoulder	$[F_{R\_per}]$ = N
$F_{R\_x\_per}$	=	Permitted overhung load at distance x to shaft shoulder	$[F_{R\_x\_per}]$ = N

5.1.4 Calculating the overhung load

An overhung load F_R occurring at the gear unit output is calculated from the maximum torque M_{G_max} at the gear unit output and the radius r of the mechanical transmission element. Additionally occurring forces, such as the pretensioning force in belt drives or the polygon effect in chain drives, are taken in account using the transmission element factor f_z (see chapter "Table appendix"). If the exact pretensioning force is known, f_z = 1.0 can be set and the pretensioning force can be added to the overhung load resulting from the torque.

F_R = 0 N if the drive is connected to the application without overhung load.

$$F_R = \frac{M_{G\_max} \times 2000}{d} \times f_Z$$

27021617275478795

F_R	=	Overhung load on gear unit output	$[F_R]$ = N
M_{G_max}	=	Maximum torque of the gear unit output including load efficiency from all travel sections	$[M_{G\_max}]$ = Nm
d	=	Diameter of the mechanical transmission element	[d] = mm
f_Z	=	Transmission element factor	$[f_Z]$ = 1

INFORMATION

Note that an emergency stop via steep emergency stop ramps in the frequency inverter or an emergency stop via a mechanical brake will result in a torque load on the output of the gear unit that can be significantly higher than in normal operation. In this case, the resulting increased overhung load must be checked.

5.1.5 Checking the overhung load

The overhung load F_R acting on the output shaft is compared to the permitted overhung load of the gear unit, F_{R_per}. Note that the permitted overhung load F_{R_per} depends on the gear ratio as well as the gear unit type and size.

$$F_R < F_{R\_per}$$

18014418020732939

F_R	= Overhung load on gear unit output	$[F_R]$ = N
F_{R_per}	= Permitted overhung load of gear unit at distance x = l/2 to shaft shoulder	$[F_{R\_per}]$ = N

26636735/EN – 07/2020

5.1.6 Checking the axial load

The applied axial load F_A can only be checked manually if no overhung load is acting on the gear unit output. In this case, the permitted axial load F_{A_per} is defined as half the value of the permitted overhung load (see "gearmotors" catalog). For applications with eccentric axial loads, contact SEW-EURODRIVE.

$$F_{A\_per} = 0.5 \times F_{R\_per}$$

25911812491

The applied axial load is compared to the permitted axial load.

$$F_A < F_{A\_per}$$

28360303755

F_A	= Axial load on gear unit output	$[F_A]$ = N
F_{A_per}	= Permitted axial load on gear unit output	$[F_{A\_per}]$ = N
F_{R_per}	= Permitted overhung load of gear unit at distance x = l/2 to shaft shoulder	$[F_{R\_per}]$ = N

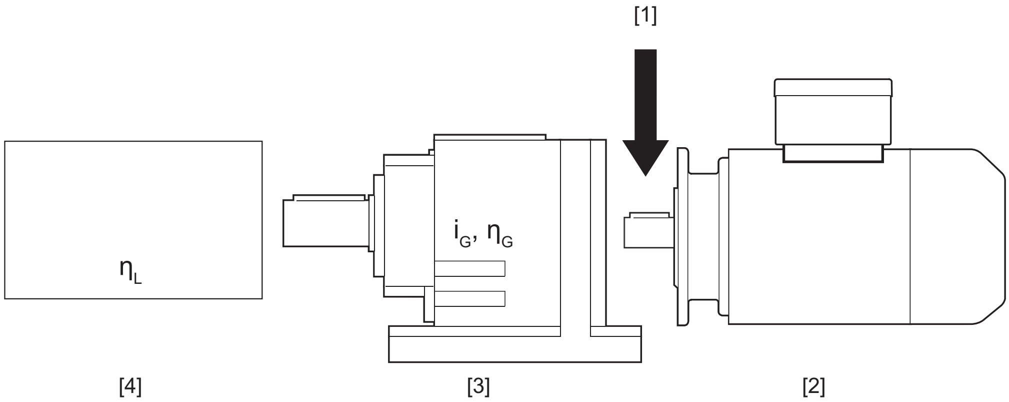
5.2 Calculating and selecting the motor

This section covers the following topics:

- Motor torques
- Motor preselection (type, size)
- Checking the drive selection

5.2.1 Motor torques

After the gear unit is selected (see chapter "Calculating and selecting the gear unit" (→ 🗎 49)), the motor is selected based on the motor torques of the various travel sections. In this case, all output-side torques are first converted to the motor shaft, taking into account the gear ratio and the gear unit efficiency.



18014418020861195

[1] Reference point
[2] Motor
[3] Gear unit
[4] Application-side load
η_L Load efficiency
i_G Gear unit ratio
η_G Gear unit efficiency

26636735/EN – 07/2020

Depending on the direction of force flow, a distinction must be made between motor and generator mode, resulting in different calculation methods.

$M_{Mot\_n} > 0$, motor mode:

$$M_{Mot\_n} = \frac{M_{G\_n}}{i_G \times \eta_G}$$

18014418020747147

$M'_{Mot\_n} < 0$, generator mode:

$$M'_{Mot\_n} = \frac{M'_{G\_n}}{i_G} \times \eta'_G$$

18014418020748811

M_{Mot_n}	=	Torque of the application as a requirement of the motor in travel section n, including load efficiency (motor mode)	$[M_{Mot\_n}]$ = Nm
M_{G_n}	=	Torque of the application as a requirement of the gear unit in travel section n, including load efficiency (motor mode)	$[M_{G\_n}]$ = Nm
i_G	=	Gear unit ratio	$[i_G]$ = 1
η_G	=	Gear unit efficiency	$[\eta_G]$ = 1
M'_{Mot_n}	=	Torque of the application as a requirement of the motor in travel section n, including efficiencies (generator mode)	$[M'_{Mot\_n}]$ = Nm
M'_{G_n}	=	Torque of gear unit output in travel section n, including load efficiency (generator mode)	$[M'_{G\_n}]$ = Nm
η'_G	=	Retrodriving gear unit efficiency	$[\eta'_G]$ = 1

- The following applies to helical-worm and SPIROPLAN® gear units:
  $\eta'_G = 2 - 1/\eta_G$
- The following applies to all other gear units:
  $\eta'_G = \eta_G$

5.2.2 Motor preselection

When the motor is selected, a distinction is made between applications that continuously operate near the nominal operation point (S1 duty cycle) and those that operate in intermittent or partial load duty.

Note that the motor must be operated at full thermal capacity in continuous duty. In intermittent duty, the motor can be briefly operated at 150% capacity if there are sections in the travel cycle that reduce the thermal load on the motor. The adequacy of this reduction in thermal load in intermittent duty must be mathematically verified (see chapter "Thermal motor utilization" (→ 🗎 61)).

Preselection for continuous duty

The selection criterion for S1 continuous duty assumes that the motor torque required for acceleration is within the 150% overload capacity and the duration of the "acceleration" travel section is negligible compared to the "constant speed" travel section.

$$M_{Mot\_const} \leq M_N$$

18014418020751499

M_{Mot_const}	=	Torque of the application in the “constant speed” travel section as a requirement of the motor, including efficiencies (motor mode)	$[M_{Mot\_const}]$ = Nm
M_N	=	Rated torque of the motor	$[M_N]$ = Nm

Example:

The following motor torques are required for an application to be calculated:

- M_{Mot_ac} = 10.5 Nm for acceleration
- M_{Mot_const} = 9 Nm continuous load for at least 10 minutes (equivalent to S1 duty cycle)

A motor with rated torque M_N = 9.8 Nm (*DRN90L4*) greater than the continuous load is selected. If a motor with a lower rated torque is selected (*DRN90S4* with M_N = 7.2 Nm), it will result in thermal overloading of the DRN90S4 motor.

Preselection for intermittent duty

The selection criterion for intermittent duty assumes a change in load and pause times. It is assumed that the required motor torque in the "acceleration" travel section is much larger than the motor torque in the "constant speed" travel section. As a result, the motor torque in the "acceleration" travel section is crucial for motor selection. In this case, the aforementioned overload capacity of 150% (factor of 1.5) of the rated motor torque is used as the comparative value.

$$M_{Mot\_ac} \leq 1.5 \times M_N$$

18014418020753163

M_{Mot_ac}	= Torque of the application in the “acceleration” travel section as a requirement of the motor, including efficiencies (motor mode).	$[M_{Mot\_ac}]$ = Nm
M_N	= Rated torque of the motor	$[M_N]$ = Nm

Example:

If the required acceleration torque is M_{Mot_ac} = 10.5 Nm, but the load in the "constant speed” travel section is only M_{Mot_const} = 4 Nm, then a motor with rated torque M_N = 7.2 Nm (*DRN90S4*) can be provisionally selected, taking into account the overload capacity of 150%. A final thermal analysis is required for motor selection for intermittent duty (see chapter "Thermal motor utilization" (→ 🗎 61)).

Checking the drive selection

After preselection of the motor, additional selection criteria and parameters must be checked.

- Maximum motor utilization
- Thermal motor utilization
- Consideration of the mass moment of inertia ratio
- Feasibility of the drive combination.

This ensures that the selected drive meets all requirements.

5.2.3 Maximum motor utilization

During acceleration, the motor must apply the torque required by the application as well as accelerate its own inertia (rotor). This additional acceleration torque can only be calculated after the motor has been selected and must be taken into account in all dynamic travel sections. The maximum motor utilization is then checked again.

Calculating the dynamic torque for the intrinsic acceleration of the motor

The dynamic torque for the intrinsic acceleration of the motor M_{Mot_iac} is calculated from the mass moment of inertia of the motor J_{Mot} and the angular acceleration α. Note that during project planning for motors with additional options (e.g., flywheel fan /Z), the mass moments of inertia of all additional components must also be added to the mass moment of inertia of the motor. The relevant values can be found in the appropriate product catalogs.

In the following calculation, the mass moment of inertia of the motor J_{Mot} without additional components is used as an example.

The angular acceleration α can be represented as a function of the rotational speed n and the acceleration time t_{ac}, which are known from the previous calculations. The motor is to be accelerated to the motor speed n_{Mot} (setpoint input).

Dynamic torque for intrinsic acceleration of the motor:

$$M_{Mot\_iac} = J_{Mot} \times \alpha = J_{Mot} \times \frac{\omega}{t_{ac}}$$

18014418020762123

Substituting $\omega = \frac{2\pi \times n}{60} \approx \frac{n}{9.55}$ and n = n_{Mot} results in:

$$M_{Mot\_iac} = J_{Mot} \times \frac{n_{Mot}}{9.55 \times t_{ac}}$$

18014418020767115

M_{Mot_iac}	= Dynamic torque for intrinsic acceleration of the motor	$[M_{Mot\_iac}]$ = Nm
J_{mot}	= Mass moment of inertia of the motor	$[J_{Mot}]$ = kg m^2
α	= Angular acceleration	[α] = s^{-2}
ω	= Angular speed	[ω] = s^{-1}
t_{ac}	= Acceleration time in the “acceleration” travel section	$[t_{ac}]$ = s
n_{Mot}	= Motor speed (setpoint input)	$[n_{Mot}]$ = min^{-1}
n	= Speed	[n] = min^{-1}

26636735/EN – 07/2020

Calculating motor torques

The dynamic torque for the intrinsic acceleration of the motor in the "acceleration" and "deceleration" travel sections must be taken into account because the inertia of an object only affects dynamic movements.

“Acceleration” travel section:

The dynamic torque for intrinsic acceleration of the motor M_{Mot_iac} is added to the required motor torque M_{Mot_ac} in the "acceleration” travel section.

$$M_{Mot\_ac\_tot} = M_{Mot\_ac} + M_{Mot\_iac}$$

18014418020769803

“Deceleration” travel section:

If $t_{dec} = t_{ac}$, the required motor torques in the "acceleration" and "deceleration" travel sections have the same value. The dynamic torque for intrinsic acceleration of the motor M_{Mot_iac} is subtracted from the required motor torque M_{Mot_dec} in the "deceleration” travel section. If $t_{dec} \neq t_{ac}$, the required motor torque must be calculated from the deceleration time t_{dec} for the deceleration process. Both motor operation and regenerative mode of the motor are possible in the "deceleration" travel section.

- Motor mode:

$$M_{Mot\_dec\_tot} = M_{Mot\_dec} - M_{Mot\_iac}$$

18014418020773131

- Generator mode:

$$M'_{Mot\_dec\_tot} = M'_{Mot\_dec} - M_{Mot\_iac}$$

28902785675

$M_{Mot\_ac\_tot}$	= Total torque of the application including the intrinsic acceleration of the motor in the "acceleration" travel section as a requirement of the motor, including efficiencies (motor mode)	$[M_{Mot\_ac\_tot}]$ = Nm
M_{Mot_ac}	= Torque of the application in the “acceleration” travel section as a requirement of the motor, including efficiencies (motor mode)	$[M_{Mot\_ac}]$ = Nm
M_{Mot_iac}	= Dynamic torque for intrinsic acceleration or deceleration of the motor	$[M_{Mot\_iac}]$ = Nm
$M_{Mot\_dec\_tot}$	= Total torque of the application including the intrinsic deceleration of the motor in the "deceleration" travel section as a requirement of the motor, including efficiencies (motor mode)	$[M_{Mot\_dec\_tot}]$ = Nm
M_{Mot_dec}	= Torque of the application in the “deceleration” travel section as a requirement of the motor, including efficiencies (motor mode)	$[M_{Mot\_dec}]$ = Nm
$M'_{Mot\_dec\_tot}$	= Total torque of the application including the intrinsic deceleration of the motor in the "deceleration" travel section as a requirement of the motor, including efficiencies (generator mode)	$[M'_{Mot\_dec\_tot}]$ = Nm
M'_{Mot_dec}	= Torque of the application in the “deceleration” travel section as a requirement of the motor, including efficiencies (generator mode)	$[M'_{Mot\_dec}]$ = Nm

26636735/EN – 07/2020

Checking the maximum motor utilization

Based on the calculated values of the individual travel sections, it is possible to check whether the maximum motor utilization is less than 150% of the rated motor torque, even after considering the dynamic torque for the intrinsic acceleration of the motor. In general, it can be assumed that the maximum motor utilization is reached in the "acceleration" section.

Maximum motor utilization:

$$M_{Mot\_ac\_\mathrm{tot}} \leq 1.5 \times M_N$$

18014418020775819

$M_{Mot\_ac\_tot}$	=	Total torque of the application including the intrinsic acceleration of the motor in the "acceleration" travel section as a requirement of the motor, including efficiencies (motor mode)	$[M_{Mot\_ac\_tot}]$ = Nm
M_N	=	Rated torque of the motor	$[M_N]$ = Nm

If the condition is not met, a larger motor must be selected and the adjusted motor selection rechecked. A frequency inverter with greater overload capacity may be sufficient.

5.2.4 Thermal motor utilization

A motor can be operated continuously at rated torque and rated speed without thermal overload. At other rotational speeds, the thermal working capacity may be limited. For example, operation at low speeds can limit the thermal working capacity due to the reduced self-cooling of the motor.

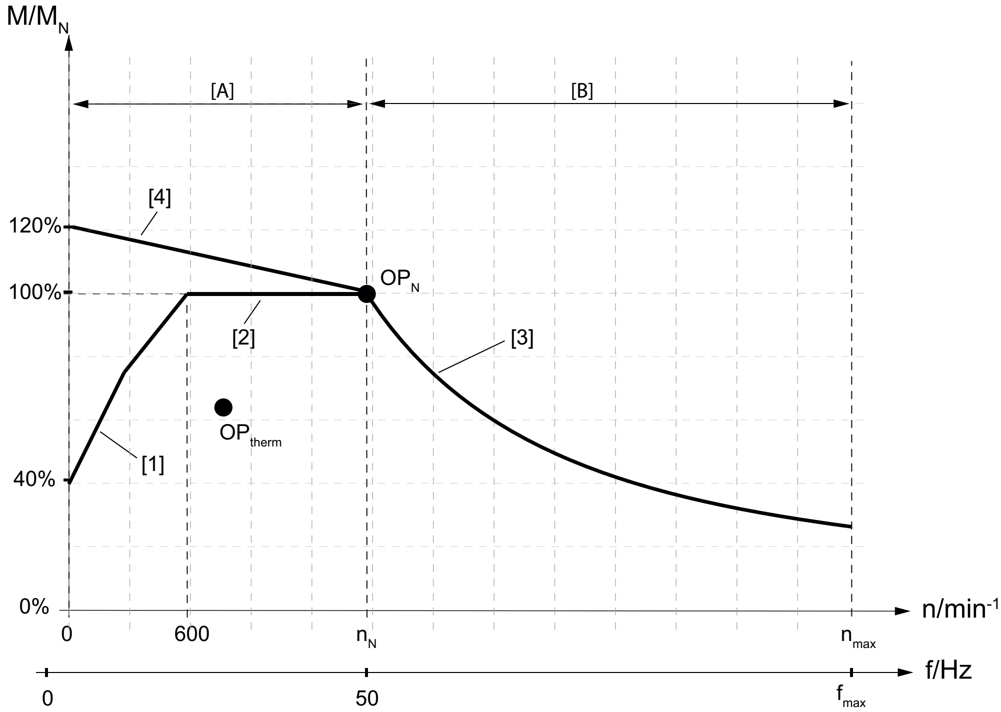
The thermal capacity of the motor must be tested based on the thermal operating point OP_{therm}, which is described by the effective torque M_{Mot_eff} and the mean speed $\overline{n}_{Mot}$. The effective torque and the mean speed are calculated across all travel sections and represent an equivalent continuous thermal load for the motor. The thermal operating point of the motor, as the theoretical continuous operating point OP_{therm}, is compared with the thermal limit characteristic curve of the motor.

26636735/EN – 07/2020

Thermal limit characteristic curve

The thermal limit characteristic curve is a speed-torque characteristic curve and represents a characteristic parameter of the motor.

The following figure explains in more detail individual areas of a universal thermal limit characteristic curve.



24635011467

M/M_N	Ratio of motor torque to rated torque of the motor
[A]	Basic control range
[B]	Field weakening range
[1]	Reduced torque due to reduced cooling
[2]	Constant torque
[3]	Constant power
[4]	Increased torque due to external cooling
OP_{therm}	Thermal operating point (example shown here)
OP_N	Rated operating point
n_N	Rated speed
n_{max}	Maximum speed
f_{max}	Maximum frequency (here: rotating field frequency)

In order to show the thermal load capacity of an asynchronous motor based on a universal thermal limit characteristic curve, the motor torque M is shown on the vertical axis with reference to the rated torque M_N. The motor speed n and the corresponding rotating field frequency f are shown on the horizontal axis.

The thermal limit characteristic curve (characteristic curve sections [1] – [3]), which is based on a base frequency of 50 Hz, describes the thermal working capacity of a fan-cooled motor in S1 continuous duty. The thermal limit characteristic curve can be subdivided into the basic control range [A] and the field weakening range [B] at the rated operating point OP_N. The basic control range applies for rotating field frequencies less than 50 Hz, the field weakening range applies for rotating field frequencies greater than 50 Hz.

The full rated torque of the motor is continuously available in the characteristic curve section [2] from approximately 600 min^{-1} up to the rated speed n_N.

In the characteristic curve section [1] below 600 min^{-1}, the thermal working capacity is continuously reduced to low speeds due to the speed-dependent self-cooling of the motor. This means that the self-cooling by the built-in motor fan is not sufficient to continuously deliver the rated torque at low speeds.

When a forced cooling fan is used, the thermal working capacity of the motor in the basic control range [A] can be increased. This results in the replacement characteristic curve section [4] for external cooling. If the rotational speeds are greater than the rated speed, the characteristic curve section [3] always applies.

The motor can be operated with constant mechanical power in the characteristic curve section [3] for rotational speeds above the rated speed. The rotational speed increases with the rotating field frequency. The permitted thermal torque decreases in the same ratio. As shown in the figure, characteristic curve section [3] of the thermal limit characteristic lies in the field weakening range [B].

Technical background: Field weakening range

For rotating field frequencies above the base frequency, the motor voltage cannot be further increased in constant relation to the frequency to the extent necessary for the constant magnetization of the motor. The maximum voltage applied to the motor is limited by the line voltage (e.g., AC 400 V) and corresponds to the rated motor voltage starting from the base frequency. To counteract the decreasing magnetization, the motor current required to reach the rated torque would have to be increased disproportionately. This high current consumption would lead to impermissible heating of the motor and therefore cause the falling thermal limit characteristic. Further options for operating a motor above the rated speed are described in chapter "Extended motor load above the rated speed in 87 Hz operation" (→ 🗎 85).

26636735/EN – 07/2020

Calculating the mean speed

In addition to the effective torque, the mean speed describes the equivalent continuous thermal load OP_{therm} and is calculated as a time-weighted mean value across the entire travel cycle (according to EN 60034: maximum 10 min). In an intermediate step, the mean speed is calculated using the total track s_{tot} and the total time t_{tot} (travel time + pause time).

Mean speed:

$$\bar{v} = \frac{v_1 \times t_1 + \ldots + v_n \times t_n}{t_1 + \ldots + t_n} = \frac{s_1 + \ldots + s_n}{t_1 + \ldots + t_n} = \frac{s_{tot}}{t_{tot}}$$

18014418020787851

The mean speed $\bar{v}$ is converted into the mean output speed of the gear unit, $\bar{n}_G$. The mean motor speed $\bar{n}_{Mot}$ is then calculated by multiplying the value $\bar{n}_G$ by the gear unit ratio i.

Mean output speed:

$$\bar{n}_G = \frac{\bar{v} \times 60000}{\pi \times d}$$

18014418020789515

Mean motor speed:

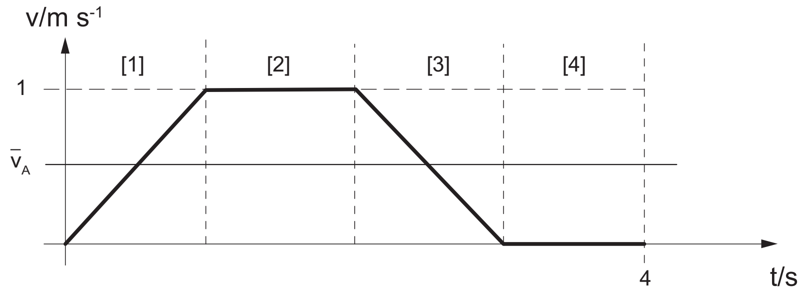
$$\bar{n}_{Mot} = \bar{n}_G \times i_G$$

18014418020791179

$\bar{v}$	= Mean speed	$[\bar{v}] = m\ s^{-1}$
$\bar{n}_G$	= Mean output speed	$[\bar{n}_G] = min^{-1}$
$\bar{n}_{Mot}$	= Mean motor speed	$[\bar{n}_{Mot}] = min^{-1}$
s_{tot}	= Total track	$[s_{tot}] = m$
t_{tot}	= Total time (travel time + pause time)	$[t_{tot}] = s$
d	= Diameter of the mechanical transmission element	[d] = mm
i_G	= Gear unit ratio	$[i_G] = 1$

Example A:

- $v = 1\ m\ s^{-1}$
- $s_{tot} = 2\ m$
- $t_{tot} = 4\ s$



24642827531

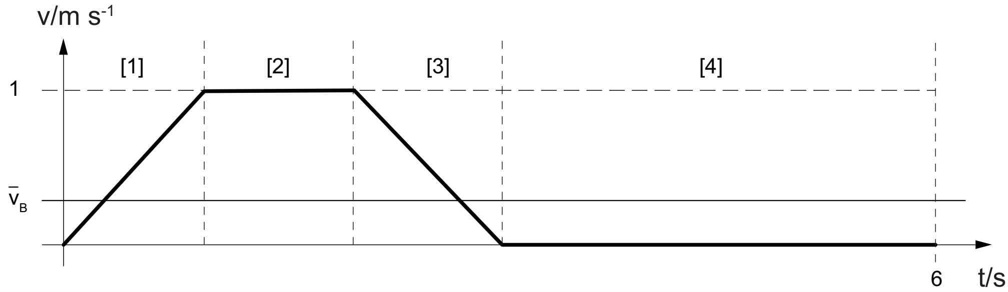
[1] Acceleration travel section $t_1 = 1\ s$
[2] Constant speed travel section $t_2 = 1\ s$
[3] Deceleration travel section $t_3 = 1\ s$
[4] Pause time $t_4 = 1\ s$

Result: Mean speed $\bar{v} = 0.5\ m\ s^{-1}$

26636735/EN – 07/2020

Example B:

- $v = 1\ m\ s^{-1}$
- $s_{tot} = 2\ m$
- $t_{tot} = 6\ s$



24647503883

[1] Acceleration travel section $t_1 = 1\ s$
[2] Constant speed travel section $t_2 = 1\ s$
[3] Deceleration travel section $t_3 = 1\ s$
[4] Pause time $t_4 = 3\ s$

Result: Mean speed $\bar{v} = 0.33\ m\ s^{-1}$

Calculating the effective motor torque

The effective motor torque is the second variable that describes the thermal operating point OP_{therm} of the motor.

In order to calculate the effective motor torque, the motor torques of the individual travel sections are squared, multiplied by the duration of the respective travel sections and then totaled. This sum in the numerator is divided by the total time of the cycle to calculate the square root (see the following formula, "effective motor torque"). Pause times, especially for hoists, must be taken into account.

Example:

If a hoist motor is to hold the load during pause times (position control), these travel sections do not reduce the load and are included in the calculation of the effective motor torque. If the load is held by a brake during pause times, these time periods are included in the total time in the denominator of the following formula (effective motor torque).

Torques in both motor and generator mode are taken into account because they both contribute equally to motor temperature increase.

Effective motor torque:

$$M_{Mot\_eff} = \sqrt{\frac{M_{Mot\_1\_tot}^2 \times t_1 + M_{Mot\_2\_tot}^2 \times t_2 + \ldots + M_{Mot\_n\_tot}^2 \times t_n}{t_1 + t_2 + \ldots + t_n}}$$

18014418020795531

M_{Mot_eff}	= Effective motor torque	$[M_{Mot\_eff}]$ = Nm
$M_{Mot\_n\_tot}$	= Total torque of the application including the intrinsic acceleration of the motor in the travel section n as a requirement of the motor	$[M_{Mot\_n\_tot}]$ = Nm
t_n	= Duration of travel section n	$[t_n]$ = s

Checking the thermal motor utilization

To complete the thermal test, the calculated thermal operating point of the motor, which includes all application information, is compared to the thermal limit characteristic curve.

The thermal operating point OP_{therm} of the motor is:

$$\left(\bar{n}_{Mot}, \frac{M_{Mot\_eff}}{M_N} \right)$$

18014418020797195

If the thermal operating point OP_{therm} is below the thermal limit characteristic curve, the preselected motor is thermally suitable for the present application.

If this is not the case, the thermal limit characteristic curve can be changed by using the “external fan /V” option if the mean speed is lower than the rated speed. This increases the thermal range of application of the motor. This action is sufficient if the thermal operating point OP_{therm} is below the thermal limit characteristic curve for forced air cooling.

Otherwise, a more powerful motor with a larger cooling surface and therefore a higher thermal limit characteristic curve must be selected. In any case, the adjusted motor selection must be rechecked.

5.2.5 Consideration of the mass moment of inertia ratio

The mass moment of inertia ratio influences the speed control of the frequency inverter, among other things. The aim of a favorable mass moment of inertia ratio is to optimize the speed controller so that the setpoint / actual value deviations required for the control process do not interfere with the application.

A distinction is made between the mass moment of inertia on the load end, J, and the moment of inertia of the motor, J_{Mot}. The crucial factor to be considered is the mass moment of inertia on the load end reduced to the motor shaft, J_x, (for calculation, see section "Mass moments of inertia in a drive train" (→ 🖹 39)) compared to the mass moment of inertia of the motor, J_{Mot}.

The mass moment of inertia ratio is considered so that the quality of the speed control of the frequency inverter can already be influenced during the project planning stage. This ensures that the required dynamics and stable control conditions are achieved. The following general principle applies: The higher the dynamics of an application (fast speed changes, start and stop ramps), the lower the ratio of the mass moments of inertia of the motor and the application should be.

Unfavorable mass moment of inertia ratios and the simultaneously necessary high dynamics result in special requirements for the parameterization of the speed controller. If the parameterization of the speed controller is not sufficiently adapted, this can lead to slow reactions in the event of rotational speed deviations and even oscillations in the rotational speed. In extreme cases, the rotational speed will oscillate so much that the frequency inverters will shut down with an error message.

This extreme case can also negatively affect the load on the mechanical components of the drive train:

Even before the drive shuts down, the oscillation of the rotational speed may result in torque impulses that are above the mechanical limit values of the drive mechanism or the application. This can cause damage and failure of the drive train.

Experience has shown that an average mass moment of inertia ratio of less than 50 is therefore recommended during configuration of standard drives.

26636735/EN – 07/2020

Checking the mass inertia ratio:

$$\frac{J_x}{J_{Mot}} \leq 50$$

18014418020799883

J_x = Mass moment of inertia of the load reduced to the motor shaft [J_x] = kg m^2
J_{mot} = Mass moment of inertia of the motor [J_{Mot}] = kg m^2

To reduce the mass moment of inertia ratio, either a more powerful motor with higher intrinsic inertia or the "additional oscillating weight (flywheel fan)" option must be selected for the motor.

Another way to reduce this ratio is to further reduce the load moment of inertia. A larger gear ratio or transmission element gear ratio decreases the mass moment of inertia of the load reduced to the motor shaft, J_x. This increases the motor speed with the same application speed.

Note that as the motor speed increases, the motor must still have sufficient torque reserve, even if it is operated in the field weakening range. Otherwise, the required torque can be achieved through 87 Hz operation or by selecting a more powerful motor. Higher motor speeds place a greater thermal load on the gear unit, which should be checked if necessary.

Practical examples

In typical applications for travel drives, small motors are usually sufficient as drives due to low friction forces. This often results in a mass moment of inertia ratio between 10 and > 50. This mass moment of inertia ratio can be reduced only by taking unfavorable measures such as oversizing the motor.

Hoist applications, on the other hand, are usually equipped with powerful and therefore sluggish motors due to their high static load. The dynamics are usually less important. As a result, highly favorable mass moment of inertia ratios of less than 1 can occur. Mass moment of inertia ratios < 5 are common.

Other applications such as stationary conveyor systems and rotary applications are usually between these two extreme values.

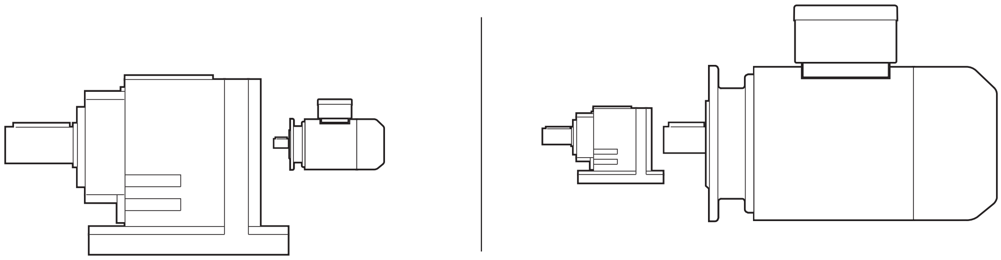
Mass moment of inertia ratio in the selection of servo drives

When selecting servo drives, a mass moment of inertia ratio of 15 should generally not be exceeded; smaller values are preferable for dynamic applications.

26636735/EN – 07/2020

5.2.6 Feasibility of the drive combination

First of all, check whether the selected motor can be combined with the selected gear unit by referring to the possible geometrical combinations (combination overviews) in the product documentation. For extreme size differences, some drive combinations are not feasible for reasons such as component strength. The following figures show examples of extreme size differences that are not feasible.



25221669003

If the desired combination is not feasible, either the gear ratio, the gear unit size or the motor size must be adjusted.

5.2.7 Evaluating starting behavior

For certain applications, a close examination of the starting behavior may be required. Excessively high acceleration can cause damage to mechanical components such as transmission elements, additional transmissions or gear units, or lead to slippage of wheels or belts in travel applications, for example. This can result in a deviating, uncontrolled travel cycle and greater material wear.

In this case, it may be necessary to determine the maximum possible acceleration of the load and check it against the configured acceleration.

The procedure is the same as "Evaluating starting behavior" (→ 🗎 99) for non-controlled drives. In the case of controlled drives, however, the acceleration is not checked against startup acceleration a_H, but against the configured acceleration or deceleration.

5.3 Calculating and selecting the brake

This section covers the following topics:

- Special requirement for lifting applications
- Braking work and brake application speed
- Feasibility of the brakemotor
- Service life until inspection
- Gear unit load during emergency stop braking
- Calculating the overhung load to be absorbed during emergency stop braking
- Calculating the permitted emergency stop characteristic values
- Further selection criteria

INFORMATION

Note that the value of the each variable must be included in the brake design formulas.

Controlled drives are usually decelerated electrically via an adjustable speed ramp. A mechanical brake can serve as a holding brake at an idle state or as a working brake in the event of an emergency stop. The relationships of mechanical deceleration in emergency stop situations can be mathematically considered as similar to a braking operation of a line-powered drive (see chapter "Calculating and selecting the brake" (→ 🗎 107) for non-controlled drives).

The brake has different requirements depending on the application. In considering these requirements, a distinction is made between applications in the horizontal and vertical direction of movement. Lifting applications with a vertical or oblique direction of movement have a special requirement that must be observed in addition to the general specifications for brake configuration. The same requirement applies to applications with external loads on the drive at an idle state, such as additional process forces or wind load.

INFORMATION

In this chapter, all brake calculations refer to the "Standard” load range as well as the nominal value of the braking torque. For detailed information, overload ranges as well as further calculation options, see the "Project Planning for BE.. Brakes” manual.

5.3.1 Special requirement for lifting applications

For lifting applications, the following must be observed when dimensioning the brake and the braking torque:

- The brake must reliably hold the application at an idle state (static load on the brake).
- In the event of an emergency stop, the application must be reliably braked to an idle state (dynamic load on the brake).

In order to meet these requirements, the following criterion must be met:

The braking torque must be at least 250% of the required static motor torque during downward movement.

$$M_B \geq 2.5 \times M'_{Mot\_stat}$$

18014418020804107

In this case, the static motor torque M'_{Mot_stat} corresponds to the motor torque for downward movement at a constant speed M'_{Mot_const} (generator mode). This torque is already known from the calculation process for controlled drives.

$$M'_{Mot\_stat} = M'_{Mot\_const}$$

28895873803

M_B	= Braking torque	$[M_B]$ = Nm
M'_{Mot_stat}	= Static torque of the application as a requirement of the motor, including efficiencies (generator mode)	$[M'_{Mot\_stat}]$ = Nm
M'_{Mot_const}	= Torque of the application in the “constant speed” travel section as a requirement of the motor, including efficiencies (generator mode)	$[M'_{Mot\_const}]$ = Nm

For further calculation steps, a preliminary braking torque is selected according to this requirement. This torque may need to be increased later in the calculation process.

Technical background: Factor 2.5

The factor 2.5 (250%) can be explained as follows:

In order to decelerate the application mechanically in event of an emergency stop, the braking torque must at least be greater than the static motor torque during downward movement. If both torques were equal, this would result only in downward movement with constant speed and would not result in mechanical deceleration.

In addition, the braking torque is subject to various physical influences, such as friction speed, temperature and other environmental influences that reduce the braking effect. The braking effect may be further reduced if the brake is used exclusively as a holding brake, since the brake lining can age prematurely due to long periods of non-use. To account for these effects, the static motor torque of the downward movement is applied with the above-mentioned factor of 2.5. This is the minimum requirement for the braking torque to be selected.

5.3.2 Braking work

The permitted braking work is a characteristic that describes the thermal working capacity of the brake for a single braking operation. The permitted braking work is fundamentally dependent on the brake size as well as the switching frequency and the brake application speed.

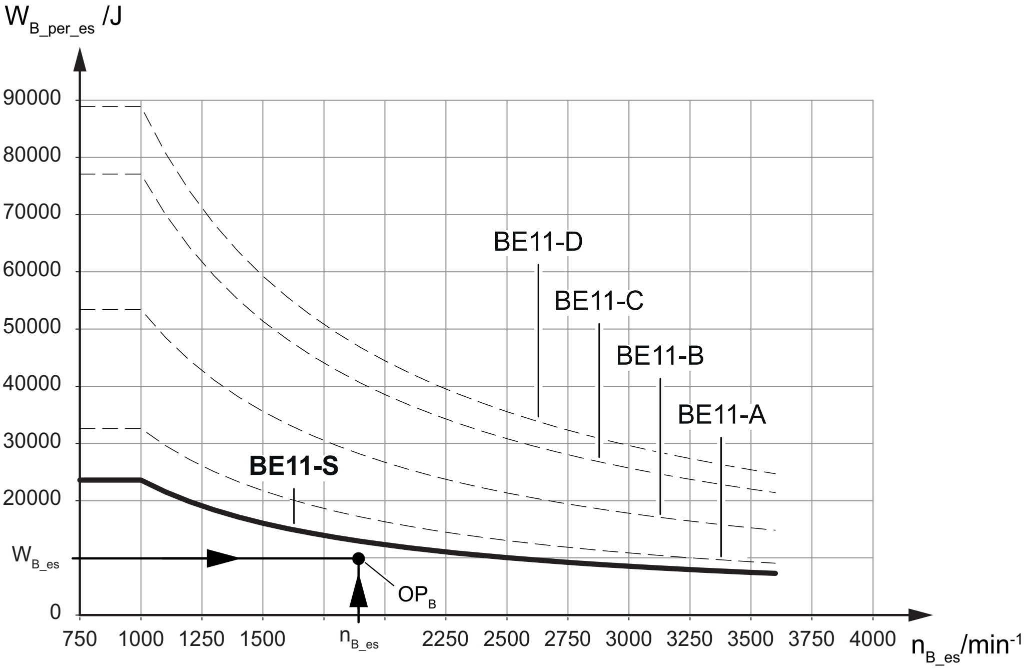
The permitted braking work per brake size depends only on the brake application speed in the event of an emergency stop because mechanical braking does not occur during normal operation of controlled drives. The calculation of the brake application speed is described in chapter "Brake application speed" (→ 🗎 73).

An operating point of the brake, OP_B, is calculated from the braking work to be done in the event of an emergency stop, W_{B_es}, and the required switching frequency Z_{req}. In the next step, a brake is selected with a characteristic curve that runs above this operating point.

The corresponding characteristic curves for the permitted braking work in the event of an emergency stop, $W_{B\_per\_es}$, as a function of the brake application speed n_{B_es}, are listed in the "Project Planning for BE.. Brakes” manual.

26636735/EN – 07/2020

Example: Emergency stop characteristic BE11-S and overload characteristics BE11-A to BE11-D



25405291403

$W_{B\_per\_es}$	= Permitted braking work in the event of an emergency stop	$[W_{B\_per\_es}]$ = J
W_{B_es}	= Braking work to be done in the event of an emergency stop	$[W_{B\_es}]$ = J
n_{B_es}	= Brake application speed in the event of an emergency stop	$[n_{B\_es}] = \text{min}^{-1}$
OP_B	= Operating point of the brake	

INFORMATION

The maximum brake application speeds must be observed to avoid a thermal overload of the brake. The curves may not be extrapolated.

The BE..- S characteristic curve of the Standard load range is generally used. The overload characteristics apply only under certain conditions. For further information, see the "Project Planning for BE.. Brakes" manual

In addition to various application data, the braking torque M_B enters into the calculation of the braking work to be done in the event of an emergency stop, W_{B_es}. In the case of lifting applications, the provisionally selected braking torque corresponding to the criterion in chapter "Special requirement for lifting applications" (→ 🗎 69) is applied. For all other applications without external loads acting on the drive at an idle state, the standard brake torque of the respective brake size is used in accordance with the “Project Planning for BE.. Brakes” manual.

Different calculation formulas are used for the braking work to be done in the event of an emergency stop, depending on whether the static torque of the application supports or impedes (places a load on) the braking process.

Calculating the braking work to be done in the event of an emergency stop

Horizontal or rotary movement:

$$W_{B\_es} = \frac{M_B}{M_B + M'_{Mot\_stat}} \times \frac{(J_{Mot} + J_x \times \eta_L \times \eta'_G) \times n^2_{B\_es}}{182.5}$$

26637426315

The braking work in the vertically upward direction of movement is smaller than in the vertically downward direction of movement. As a result, it is not calculated here.

Vertically downward movement:

$$W_{B\_es} = \frac{M_B}{M_B - M'_{Mot\_stat}} \times \frac{(J_{Mot} + J_x \times \eta_L \times \eta'_G) \times n^2_{B\_es}}{182.5}$$

26637431947

W_{B_es}	= Braking work to be done in the event of an emergency stop	$[W_{B\_es}]$ = J
M_B	= Braking torque	$[M_B]$ = Nm
M'_{Mot_stat}	= Static torque of the application as a requirement of the motor, including efficiencies (generator mode)	$[M'_{Mot\_stat}]$ = Nm
J_{mot}	= Mass moment of inertia of the motor	$[J_{Mot}]$ = kg m^2
J_x	= Mass moment of inertia of the load relative to the motor shaft	$[J_x]$ = kg m^2
η_L	= Load efficiency	$[\eta_L]$ = 1
η'_G	= Retrodriving gear unit efficiency • The following applies to helical-worm and SPIROPLAN® gear units: $\eta'_G = 2-1/\eta_G$ • The following applies to all other gear units: $\eta'_G = \eta_G$	$[\eta'_G]$ = 1
n_{B_es}	= Brake application speed in the event of an emergency stop	$[n_{B\_es}]$ = min^{-1}

Based on the emergency stop characteristic curves of the various brakes, a brake is selected whose permitted braking work in the event of an emergency stop, $W_{B\_per\_es}$, for the brake application speed in the event of an emergency stop, n_{B_es}, is greater than the braking work to be done in the event of an emergency stop, W_{B_es}.

$$W_{B\_es} \leq W_{B\_per\_es}$$

25424959499

W_{B_es}	= Braking work to be done in the event of an emergency stop	$[W_{B\_es}]$ = J
$W_{B\_per\_es}$	= Permitted braking work in the event of an emergency stop	$[W_{B\_per\_es}]$ = J

An overload characteristic can be used under certain conditions if the emergency stop characteristic curve of a brake does not meet the application requirements with regard to brake application speed n_{B_es} or braking work W_{B_es}. These requirements, as well as any restrictions, can be found in the “Project Planning for BE.. Brakes” manual.

If no overload characteristic can be used, another brake must be selected and rechecked. The braking work can be divided across several brakes by using multi-motor operation if no brake is available that can do the braking work at a given switching frequency and brake application speed. In order to sufficiently reduce the braking work, the switching frequency or the brake application speed, alternative measures can be taken so that an available brake size can be used. These measures can be found in the "Project Planning for BE.. Brakes" manual.

5.3.3 Brake application speed

The brake application speed is defined as the motor speed at which the mechanical braking process begins in the event of an emergency stop.

The brake application speed corresponds in most cases to the operational motor speed of the application under consideration. If external forces such as gravitational force or process forces are applied, the load during the brake application time t_2 is additionally accelerated. The motor speed can increase significantly in this case. As a result, the brake application speed is greater than the operational motor speed. To determine the resulting brake application speed, the speed difference is calculated and added to the operational motor speed.

Example: Deceleration of a hoist without counterweight in the downward direction of movement.

Calculation of the speed difference during brake application:

$$n_{dif} = \frac{9.55 \times M'_{Mot\_stat} \times t_2}{J_{Mot} + J_x \times \eta_L \times \eta'_G}$$

25358078219

n_{dif}	=	Speed difference during brake application	$[n_{dif}] = \text{min}^{-1}$
M'_{Mot_stat}	=	Static torque of the application as a requirement of the motor, including efficiencies (generator mode)	$[M'_{Mot\_stat}] = \text{Nm}$
t_2	=	Brake application time depending on the wiring of the brake: • $t_{2,I}$ = Brake application time for cut-off in the AC circuit • $t_{2,II}$ = Brake application time for cut-off in the DC and AC circuit	$[t_2] = \text{s}$
J_{mot}	=	Mass moment of inertia of the motor	$[J_{Mot}] = \text{kg m}^2$
J_x	=	Mass moment of inertia of the load reduced to the motor shaft	$[J_x] = \text{kg m}^2$
η_L	=	Load efficiency	$[\eta_L] = 1$
η'_G	=	Retrodriving gear unit efficiency • The following applies to helical-worm and SPIROPLAN® gear units: $\eta'_G = 2\text{-}1/\eta_G$ • The following applies to all other gear units: $\eta'_G = \eta_G$	$[\eta'_G] = 1$

Calculating the brake application speed in the event of an emergency stop:

$$n_{B\_es} = n_{Mot} + n_{dif}$$

26638773899

n_{B_es}	=	Brake application speed in the event of an emergency stop	$[n_{B\_es}] = \text{min}^{-1}$
n_{Mot}	=	Operational motor speed	$[n_{Mot}] = \text{min}^{-1}$
n_{dif}	=	Speed difference during brake application	$[n_{dif}] = \text{min}^{-1}$

The calculation steps in the "Braking work" (→ 🗎 70) chapter can be performed using the calculated brake application time. The configuration of the brake can then be continued.

5.3.4 Feasibility of the brakemotor

After a brake size and the braking torque are selected, the feasibility of the motor and brake must be verified. In special cases, it may be necessary to adjust the brake or motor size later.

Example:

For a vertical punch application, the motor is sized for the maximum torque required during the punching operation. The motor brake, however, only has to hold the dead weight of the moving tool. The required braking torque is lower than the required motor torque. The resulting brake size would be too small to be installed on the selected motor. This means that a larger brake size with oversized braking torque must be selected so that the brake can be installed on the motor.

5.3.5 Service life until inspection

The number of permitted emergency stop braking operations until brake inspection can be calculated using the braking work to be done in the event of an emergency stop, W_{B_es} , taking into account the permitted braking work until brake inspection, W_{B_insp}, and the calculation coefficient f_W.

$$N_{B\_insp} = \frac{W_{B\_insp}}{W_{B\_es} \times f_W}$$

26638796299

N_{B_insp}	=	Number of permitted emergency stop braking operations until brake inspection Note the "Project planning notes" (→ 🗎 14).	$[N_{B\_insp}] = 1$
W_{B_insp}	=	Permitted braking work until brake inspection	$[W_{B\_insp}] = J$
W_{B_es}	=	Braking work to be done in the event of an emergency stop	$[W_{B\_es}] = J$
f_W	=	Wear factor; determination in relation to the used load range for braking work	$[f_W] = 1$

Due to the increased gear unit load, the maximum number of permitted emergency stop braking operations is limited to 1000 based on permitted emergency stop characteristic values (see chapter "Calculating the permitted emergency stop characteristic values" (→ 🗎 77))

The product characteristics W_{B_insp} and f_W can be found in the "Project Planning for BE.. Brakes" manual.

5.3.6 Gear unit load during emergency stop braking

During mechanical braking, torques and overhung loads act on the gear unit and the application. These forces depend exclusively on the interaction between the application, drive and brake and cannot be influenced by the inverter or controller.

These loads occur in controlled drives during emergency stop braking and can be significantly greater than the operational load. For this reason, the loads must be calculated after brake selection.

Calculating the output torque during emergency stop braking

Horizontal or rotary movement:

$$M_{G\_es} = \frac{i_G}{\eta'_G}\left(\left(M_B + M'_{Mot\_stat}\right)\times\frac{\frac{J_x \times \eta_L \times \eta'_G}{J_{Mot}}}{\frac{J_x \times \eta_L \times \eta'_G}{J_{Mot}}+1} - M'_{Mot\_stat}\right)$$

9007224678613259

The torque load during mechanical braking in the vertically upward direction of movement is smaller than in the vertically downward direction of movement. As a result, it is not calculated here.

Vertically downward movement:

$$M_{G\_es} = \frac{i_G}{\eta'_G}\left(\left(M_B - M'_{Mot\_stat}\right)\times\frac{\frac{J_x \times \eta_L \times \eta'_G}{J_{Mot}}}{\frac{J_x \times \eta_L \times \eta'_G}{J_{Mot}}+1} + M'_{Mot\_stat}\right)$$

9007224613013259

M_{G_es}	= Output torque during emergency stop braking	$[M_{G\_es}]$ = Nm
i_G	= Gear unit ratio	$[i_G]$ = 1
η'_G	= Retrodriving gear unit efficiency • The following applies to helical-worm and SPIROPLAN® gear units: $\eta'_G = 2\text{-}1/\eta_G$ • The following applies to all other gear units: $\eta'_G = \eta_G$	$[\eta'_G]$ = 1
M_B	= Braking torque	$[M_B]$ = Nm
M'_{Mot_stat}	= Static torque of the application as a requirement of the motor, including efficiencies (generator mode)	$[M'_{Mot\_stat}]$ = Nm
J_x	= Mass moment of inertia of the load reduced to the motor shaft	$[J_x]$ = kg m²
η_L	= Load efficiency	$[\eta_L]$ = 1
η_G	= Gear unit efficiency	$[\eta_G]$ = 1
J_{mot}	= Mass moment of inertia of the motor	$[J_{Mot}]$ = kg m²

Checking the output torque load during emergency stop braking

The calculated output torque during emergency stop braking M_{G_es} is compared to the permitted output torque during emergency stop braking $M_{G\_per\_es}$. For the calculation of the permitted emergency stop characteristic values, see chapter "Calculating the permitted emergency stop characteristic values" (→ 🗎 77).

$$M_{G\_es} \leq M_{G\_per\_es}$$

25423875979

M_{G_es}	= Output torque during emergency stop braking	$[M_{G\_es}]$ = Nm
$M_{G\_per\_es}$	= Permitted output torque during emergency stop braking	$[M_{G\_per\_es}]$ = Nm

26636735/EN – 07/2020

5.3.7 Calculating the overhung load to be absorbed during emergency stop braking

For applications that require an operational overhung load on the drive, the overhung load absorbed during emergency stop braking must also be checked. The overhung load to be absorbed is calculated from the output torque during emergency stop braking, the diameter of the transmission element and the transmission element factor (e.g., for initial belt tension).

$$F_{R\_es} = \frac{M_{G\_es} \times 2000}{d} \times f_z$$

25425632011

F_{R_es}	=	Overhung load absorbed on gear unit output during emergency stop braking	$[F_{R\_es}]$ = N
M_{G_es}	=	Output torque during emergency stop braking	$[M_{G\_es}]$ = Nm
d	=	Diameter of the mechanical transmission element	[d] = mm
f_z	=	Transmission element factor	$[f_Z]$ = 1

Checking the overhung load during emergency stop braking

The overhung load absorbed during emergency stop braking, F_{R_es} , is compared to the permitted overhung load of the gear unit during emergency stop braking, $F_{R\_per\_es}$. For the calculation of the permitted emergency stop characteristic values, see chapter "Calculating the permitted emergency stop characteristic values" (→ 🗎 77).

$$F_{R\_es} \leq F_{R\_per\_es}$$

25424954891

F_{R_es}	=	Overhung load absorbed on gear unit output during emergency stop braking	$[F_{R\_es}]$ = N
$F_{R\_per\_es}$	=	Permitted overhung load on gear unit output during emergency stop braking	$[F_{R\_per\_es}]$ = N

Actions must be taken if the calculated gear unit load during emergency stop braking is greater than the product characteristics of the gear unit. These actions may include:

- Selecting a smaller braking torque.
- Selecting a larger gear unit.
- Connecting the gear unit to the application without overhung load by using a separate bearing.

In travel applications, the braking distance increases as the braking torque decreases. For lifting applications, limits are imposed here by the required stop function (see chapter "Special requirement for lifting applications" (→ 🗎 69)). If a larger gear unit is selected, all checks such as feasibility, etc., must be carried out again. If a larger gear unit cannot be built, different actions can be taken. These actions are documented in the "Project Planning for BE.. Brakes" manual.

26636735/EN – 07/2020

5.3.8 Calculating the permitted emergency stop characteristic values

This section describes the calculation of permitted emergency stop characteristic values.

Calculation of the permitted output torque for emergency stop braking for different service factors, f_B

- For $f_B \leq 2.5$:

$$M_{G\_per\_es} = 1.7 \times M_{a\_\max}$$

25897099275

- For $f_B > 2.5$:

$$M_{G\_per\_es} = 1.7 \times i_G \times 2.5 \times M_N$$

25897412235

Calculation of the permitted overhung load on the gear unit output during emergency stop braking:

$$F_{R\_per\_es} = 1.7 \times F_{R\_per}$$

25917224587

$M_{G\_per\_es}$	= Permitted output torque during emergency stop braking	$[M_{G\_per\_es}]$ = Nm
M_{a_max}	= Continuously permitted output torque of the gear unit	$[M_{a\_max}]$ = Nm
i_G	= Gear unit ratio	$[i_G]$ = 1
M_N	= Rated torque of the motor	$[M_N]$ = Nm
$F_{R\_per\_es}$	= Permitted overhung load on gear unit output during emergency stop braking	$[F_{R\_per\_es}]$ = N
F_{R_per}	= Permitted overhung load on gear unit output	$[F_{R\_per}]$ = N

INFORMATION

Due to the increased gear unit load, the maximum number of permitted emergency stop braking operations is limited to 1000 based on permitted emergency stop characteristic values.

5.3.9 Further selection criteria

Additional application requirements can influence the selection of the brake in addition to the dimensioning-related configuration criteria described above. These can result from safety considerations, a normative basis or the requirements of the operator. Additional selection criteria include the maximum stopping distance, the braking time or the stopping accuracy. The required calculation steps are described in more detail in the "Project Planning for BE.. Brakes" manual.

26636735/EN – 07/2020

5.4 Calculating and selecting the frequency inverter

This section covers the following topics:

- Assigning the frequency inverter based on the rated motor power
- Calculating the maximum and effective inverter current
- Selecting the frequency inverter according to calculated motor currents
- Selecting the frequency inverter for operating modes with current-controlled control
- Derating factors
- Braking resistor (optional)

5.4.1 Assigning the frequency inverter based on the rated motor power

In many cases, it is sufficient to assign the frequency inverter based on the rated motor power of the previously selected motor. The specified recommended motor power at constant load is commonly referred to as inverter power.

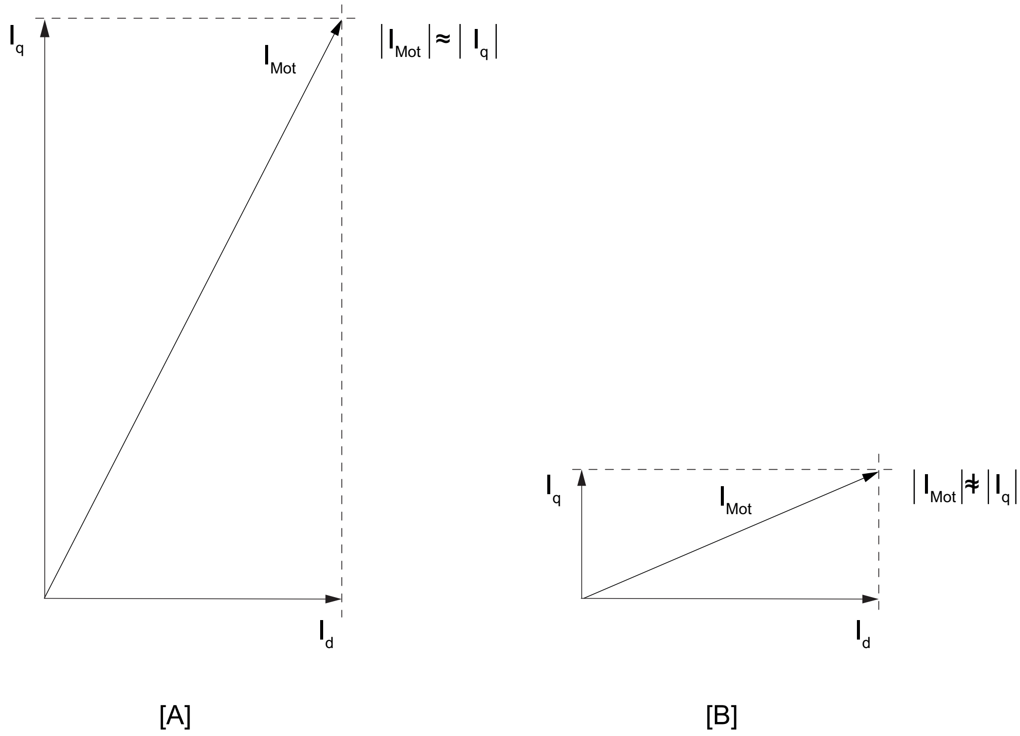
Dimensioning the frequency inverter according to the required maximum and effective current output and therefore the actual capacity utilization of the motor is preferred as a universal procedure for power assignment. Both procedures apply to operating modes such as V/f control and voltage-controlled control modes.

26636735/EN – 07/2020

5.4.2 Calculating the maximum and effective inverter current

The motor current to be applied by the frequency inverter is approximately comprised of the magnetizing current and the torque-generating current.

These relationships are shown in the following current vector diagrams.



24878517003

[A] Motor utilization near the rated operating point
[B] Motor utilization in partial load duty
I_q Torque-generating current
I_{Mot} Motor current
I_d Magnetizing current

The motor current is calculated using the Pythagorean theorem:

$$I_{Mot} = \sqrt{I_q^2 + I_d^2}$$

24878463243

I_{Mot}	= Motor current	$[I_{Mot}]$ = A
I_q	= Torque-generating current	$[I_q]$ = A
I_d	= Magnetizing current	$[I_d]$ = A

All proportional currents at the rated operating point are already known from the technical data of the motor. These values show that the rated motor current is approximately equal to the torque-generating current at the rated operating point. This also approximately applies to motor utilization rates (> 75%) near the rated operating point [A].

This comparison is not permitted for partial load duty of the motor [B].

26636735/EN – 07/2020

For any operating point, the torque-generating current I_q can be calculated from the rated value I_{q_N} using the actual torque load of the motor. The above formula for the motor current is adjusted accordingly:

$$I_{Mot} = \sqrt{\left(\frac{M_{Mot}}{M_N} \times I_{q\_N}\right)^2 + I_d^2}$$

24878470411

I_{Mot}	=	Motor current	$[I_{Mot}]$ = A
M_{Mot}	=	Motor torque required by the application	$[M_{Mot}]$ = Nm
M_N	=	Rated torque of the motor	$[M_N]$ = Nm
I_{q_N}	=	Rated value of the torque-generating current	$[I_{q\_N}]$ = A
I_d	=	Magnetizing current	$[I_d]$ = A

For capacity utilizations > 75%, the magnetizing current can be ignored in the initial approximation of the calculation. This results in the following simplified relationship for calculating the motor current to be applied by the frequency inverter. While the motor current is precisely calculated for any operating point, inaccuracies averaging from 8% to 15% have to be considered for DRN motors.

$$I_{Mot} = I_N \times \frac{M_{Mot}}{M_N}$$

24878466827

I_{Mot}	=	Motor current	$[I_{Mot}]$ = A
I_N	=	Rated current of the motor	$[I_N]$ = A
M_{Mot}	=	Motor torque required by the application	$[M_{Mot}]$ = Nm
M_N	=	Rated torque of the motor	$[M_N]$ = Nm

Similar to the motor, the frequency inverter is selected based on the maximum and effective load of the application.

Maximum required motor current for selection of the frequency inverter:

$$I_{max} = I_N \times \frac{M_{Mot\_ac\_tot}}{M_N}$$

18014418020810891

Effectively required motor current for selection of the frequency inverter:

$$I_{eff} = I_N \times \frac{M_{Mot\_eff}}{M_N}$$

18014418020812555

I_{max}	=	Maximum required motor current	$[I_{max}]$ = A
I_N	=	Rated current of the motor	$[I_N]$ = A
$M_{Mot\_ac\_tot}$	=	Total torque of the application including the intrinsic acceleration of the motor in the "acceleration" travel section as a requirement of the motor, including efficiencies (motor mode)	$[M_{Mot\_ac\_tot}]$ = Nm
M_N	=	Rated torque of the motor	$[M_N]$ = Nm
I_{eff}	=	Effectively required motor current	$[I_{eff}]$ = A
M_{Mot_eff}	=	Motor rms torque	$[M_{Mot\_eff}]$ = Nm

5.4.3 Selecting the frequency inverter according to calculated motor currents

The frequency inverter for V/f and voltage-controlled control modes is selected based on the calculated motor currents and the rated output current I_{N_FU} of the frequency inverter specified in the relevant product catalog. The rated output current refers to a specific PWM frequency depending on the type and size of frequency inverter. It may be necessary to consider a higher PWM frequency using the aforementioned derating factors.

In general, frequency inverters provide a maximum permitted overload capacity of 150%, for example. The overload capacity and its permitted duration varies according to type and size and is described by the overload factor f_{ol} in the following selection criterion.

The inverter size is selected according to the criteria below.

- Maximum required motor current (maximum utilization):

$$I_{max} < f_{ol} \times I_{N\_FU}$$

18014418020816907

- Effectively required motor current (continuous utilization):

$$I_{eff} < I_{N\_FU}$$

18014418020818571

I_{max}	= Maximum required motor current	$[I_{max}]$ = A
I_{eff}	= Effectively required motor current	$[I_{eff}]$ = A
I_{N_FU}	= Rated output current of the frequency inverter	$[I_{N\_FU}]$ = A
f_{ol}	= Overload factor of the frequency inverter (e.g., 1.5 with overload capacity of 150%)	$[f_{ol}]$ = 1

For product-specific information on these values, see the relevant product catalog.

5.4.4 Selecting the frequency inverter for operating modes with current-controlled control

The required current for torque generation cannot be calculated manually for current-controlled control modes (e.g., CFC = Current Mode Flux Control operating mode). Instead, the frequency inverter is selected according to the combination overviews (assignment tables) listed in the catalog. The combination overview (assignment table) specifies the motor/frequency inverter combination in which a specific torque M_{Mot_max} is available up to a certain speed limit.

In the overview, a required frequency inverter size can be directly allocated to the maximum required motor torque, in this case $M_{Mot\_ac\_tot}$, via the comparison value M_{Mot_max}.

INFORMATION

The combination overviews (assignment tables) in the product documentation differ according to motor type, motor connection type, line voltage and line frequency (e.g., delta or star at 230 V AC / 400 V and 50 Hz, double star at 460 V AC and 60 Hz). Make sure to use the correct combination overview (assignment table).

Refer to the project planning notes and recommendations for each frequency inverter type in the appropriate product catalog. These notes can also be relevant for operation in other control modes (e.g., V/f, VFC, ...).

5.4.5 Derating factors

For various reasons, it may be necessary to select the frequency inverter with a higher power rating than the rated motor power. Possible influencing factors include:

- High overload requirements
- Special ambient conditions (installation altitude, line voltage, temperature)
- Continuous duty with increased PWM frequency
- 87 Hz operation and delta connection of the motor (see chapter "Extended motor load above the rated speed in 87 Hz operation" (→ 🗎 85))

These influences, which reduce the maximum or effective output current of the frequency inverter, can be taken into account using different derating factors. These derating factors can be found in tables and characteristic curves in the relevant product catalogs and are not taken into account in the calculations.

5.4.6 Braking resistor (optional)

Once the size of the drive has been determined, all necessary options for operating the application are selected. If an application is in generator mode for only a short time, a braking resistor must be configured due to the limited capacity of the DC link memory. The regenerative energy from the DC link is converted into heat with the aid of the braking resistor.

Calculating and selecting the braking resistor according to thermal capacity utilization

The braking resistor is classified by its ohmic resistance and continuous braking power (= 100% cyclic duration factor). Note that the ohmic resistance of the selected braking resistor must not fall below a certain value. The smaller the resistance, the higher the current and the associated load on the brake chopper. The minimum resistance value of the braking resistor R_{BW_min} is documented for each frequency inverter size in the product catalog. Note:

$$R_{BW} \geq R_{BW\_min}$$

18014418020837131

R_{BW} = Resistance value of the braking resistor (to be selected) $[R_{BW}] = \Omega$
R_{BW_min} = Minimum resistance value of the braking resistor according to frequency inverter size $[R_{BW\_min}] = \Omega$

To select the braking resistor with regard to its thermal capacity utilization, the average regenerative braking power that the drive consumes during the course of a travel cycle is first calculated. In this calculation, only travel sections in generator mode are considered, i.e., those that result in a negative torque value (see chapter "Basics of project planning for electric drives" (→ 🗎 10)).

The mean rotational speed in the individual travel sections is calculated as the mean value of the final and initial rotational speeds. For acceleration or deceleration ramps, this corresponds to half the maximum speed, for example for the "deceleration" travel section:

$$\bar{n}_{Mot\_dec} = \frac{n_{Mot\_max}}{2}$$

18014418020822795

In the static travel sections, the mean speed corresponds exactly to the maximum speed reached, for example for the "constant speed" travel section:

$$\bar{n}_{Mot\_const} = n_{Mot\_max}$$

18014418020824459

The average braking power is calculated from the motor torque and mean speed values of a regenerative travel section.

$$\bar{P}_{gen\_n} = \frac{M'_{Mot\_n\_tot} \times \bar{n}_{Mot\_n}}{9550}$$

18014418020826123

$\bar{P}_{gen\_n}$	=	Mean braking power in the regenerative travel section n	$[\bar{P}_{gen\_n}]$ = kW
$M'_{Mot\_n\_tot}$	=	Total torque of the application, including intrinsic acceleration in travel section n, as a requirement of the motor, including efficiencies (generator mode)	$[M'_{Mot\_n\_tot}]$ = Nm
$\bar{n}_{Mot\_n}$	=	Mean motor speed in travel section n	$[\bar{n}_{Mot\_n}] = min^{-1}$

In order to obtain a mean braking power for multiple regenerative travel sections, a time-weighted mean value is formed from the mean braking power of the individual regenerative travel sections.

$$\bar{P}_{gen} = \frac{\sum_{n=gen} \bar{P}_{gen\_n} \times t_n}{\sum_{n=gen} t_n} = \frac{\bar{P}_{gen\_1} \times t_1 + \bar{P}_{gen\_2} \times t_2 + ..\bar{P}_{gen\_n} \times t_n}{t_1 + t_2 + ..t_n}$$

18014418020829451

$\bar{P}_{gen}$	=	Mean regenerative braking power	$[\bar{P}_{gen}]$ = kW
$\bar{P}_{gen\_n}$	=	Mean braking power in the regenerative travel section n	$[\bar{P}_{gen\_n}]$ = kW
t_n	=	Time in the regenerative travel section n	$[t_n]$ = s

In addition, the regenerative cyclic duration factor is required to select the braking resistor. This is the ratio of the regenerative time sections to the total time of the travel cycle.

$$ED_{BW} = \frac{\sum_{n=gen} t_n}{t_{tot}} \times 100$$

18014418020832779

ED_{BW}	=	Regenerative cyclic duration factor	$[ED_{BW}]$ = %
t_n	=	Time in the regenerative travel section n	$[t_n]$ = s
t_{tot}	=	Total time of the travel cycle	$[t_{tot}]$ = s

INFORMATION

i

The regenerative cyclic duration factor ED_{BW} may refer to a maximum total travel cycle time of t_{tot} = 120 s because the maximum permitted limit temperature of the resistor is considered reached at that point. ED_{BW} = 100% generally applies for a regenerative cyclic duration factor $ED_{BW} \geq$ 120 s.

The performance data of the braking resistor is specified in the relevant product catalog in graduations of 6%, 12%, 25%, 50% and 100% cyclic duration factor ED_{BW}. The comparison value for the average total regenerative power is the power value that belongs to the next higher cyclic duration factor than the calculated one. For example, if the result of the calculation is ED_{BW} = 17%, the calculated value of the averaged total regenerative power must be less than the catalog value at 25% ED_{BW}. For more precise selection of the cyclic duration factors between catalog increments, refer to the product manual of the corresponding frequency inverter for diagrams for converting the calculated power values to 100% ED_{BW} depending on the type of resistor.

Checking the selected braking resistor with regard to peak braking power

A certain braking resistance value must not be exceeded; otherwise, the short-term power peaks cannot be reduced. This limit value depends on the peak braking power applied and the voltage threshold in the DC link at which the brake chopper is activated. For example, the voltage threshold for control cabinet inverters (3 × 400 V AC) is 970 V DC (see product documentation).

The peak braking power must be briefly consumed in the regenerative travel section when the application generates the highest torque at maximum motor speed. The aforementioned load due to the peak braking power usually occurs at the beginning of the deceleration section.

$$P_{gen\_pk} = \frac{M'_{Mot\_dec\_tot} \times n_{Mot\_max}}{9550}$$

18014418020838795

P_{gen_pk}	=	Peak braking power	$[P_{gen\_pk}]$ = kW
$M'_{Mot\_dec\_tot}$	=	Total torque of the application including the intrinsic deceleration of the motor in the "deceleration" travel section as a requirement of the motor, including efficiencies (generator mode)	$[M'_{Mot\_dec\_tot}]$ = Nm
n_{Mot_max}	=	Maximum motor speed	$[n_{Mot\_max}]$ = min^{-1}

Using the above values, the maximum permitted braking resistance value for the application can be calculated and compared with the previously selected braking resistor:

$$R_{BW\_max} = \frac{U_{DCL}^2}{P_{gen\_pk} \times f_{BW}}$$

36028816530322443

$$R_{BW} < R_{BW\_max}$$

18014418020842123

R_{BW_max}	=	Maximum resistance value of the braking resistor depending on the application	$[R_{BW\_max}]$ = Ω
U_{DCL}	=	Voltage threshold in the DC link at which the brake chopper is activated	$[U_{DCL}]$ = V
P_{gen_pk}	=	Peak braking power	$[P_{gen\_pk}]$ = W
f_{BW}	=	Additional factor of the braking resistor due to tolerances and controller settings (f_{BW} = 1.4 for products of the MOVI-C® modular automation system)	$[f_{BW}]$ = 1
R_{BW}	=	Resistance value of the (selected) braking resistor	$[R_{BW}]$ = Ω

26636735/EN – 07/2020

5.4.7 Extended motor load above the rated speed in 87 Hz operation

In principle, the torque or the rotational speed can be increased beyond the respective rated value so that a motor can be continuously utilized above its mechanical rated power. The relationship and procedure for increasing power cannot be derived without examining the electrical parameters. In order to deliver increased mechanical power, the electrical power consumption must be increased.

$$P_{act} = \sqrt{3} \times U_{Mot} \times I_{Mot} \times \cos\varphi$$

25739721995

P_{act}	=	Active power of motor	$[P_{act}]$ = kW
U_{Mot}	=	Motor voltage (phase-to-phase)	$[U_{Mot}]$ = V
I_{Mot}	=	Motor current (outer conductor current)	$[I_{Mot}]$ = A

In order to avoid thermal overloading of the motor, the motor current I_{Mot} should generally not be increased beyond the rated motor current I_N. As a result, the electrical power can only be increased via the voltage as a manipulated value.

In order to increase the voltage without a change in motor current, the frequency must be increased in proportion to the voltage. The ratio of voltage to frequency (V/f) describes the magnetization within the motor. If the V/f ratio can be kept the same, constant magnetization of the motor is ensured and the transition to field weakening (transition point) is shifted to a higher frequency (see chapter "Thermal motor utilization" (→ 🗎 61)).

For comparison, as the V/f ratio decreases, the magnetization decreases. To counteract the decrease in magnetization, the motor current needed to reach the rated torque would have to be increased disproportionately, which could cause thermal overloading of the motor.

With respect to the goal of increasing the mechanical rated power, increased frequency results in a higher motor speed. Conversely, unchanged motor current means constant torque.

The motor voltage can generally be raised beyond the rated value for the output of the frequency inverter by taking the following actions:

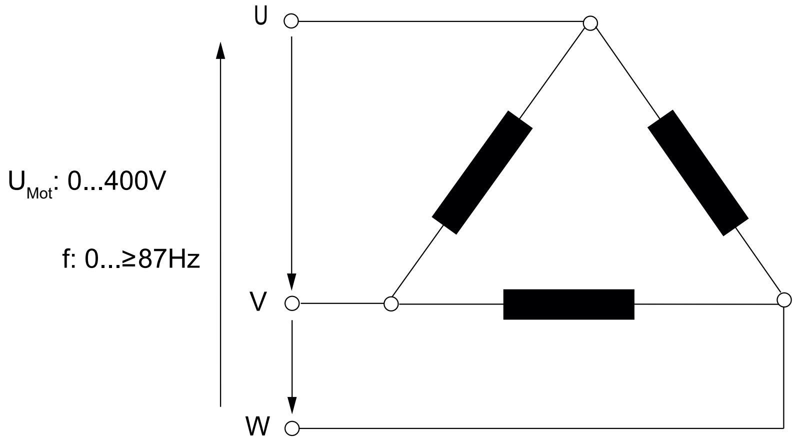
- Higher line voltage at the frequency inverter supply
- Higher DC link voltage (e.g., sinusoidal regenerative power supply)
- Use of a motor with a lower nominal voltage than the line voltage, e.g. inverter motors or standard motors in a delta connection

Motor connected in delta

In order to provide a voltage reserve between the line voltage and rated motor voltage, a motor is selected whose nominal voltage in a star connection corresponds to the line voltage. This motor is connected in delta, which makes the line voltage √3 times higher than the motor rated voltage in a delta connection.

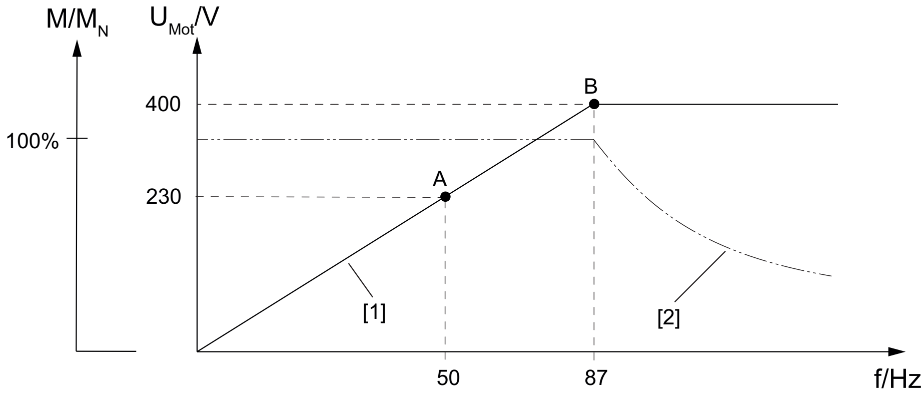
26636735/EN – 07/2020

Example: Motor according to nameplate with nominal voltage of 230 V△/400 V⅄. The motor is connected in delta and connected to a frequency inverter. The line voltage at the input of the frequency inverter is 400 V. Due to the delta connection, the rated point of the electrical values is 230 V/50 Hz.



24901356939

V/f characteristic and curve of the motor torque as a function of the frequency



28362493707

M/M_N	Ratio of motor torque to rated torque of the motor
U_{Mot}	Motor voltage
f	Frequency
[A]	Rated point at 230 V/50 Hz in a delta connection
[B]	Transition point at 400 V/87 Hz in a delta connection
[1]	Voltage/frequency characteristic
[2]	Curve of the motor torque as a function of the frequency

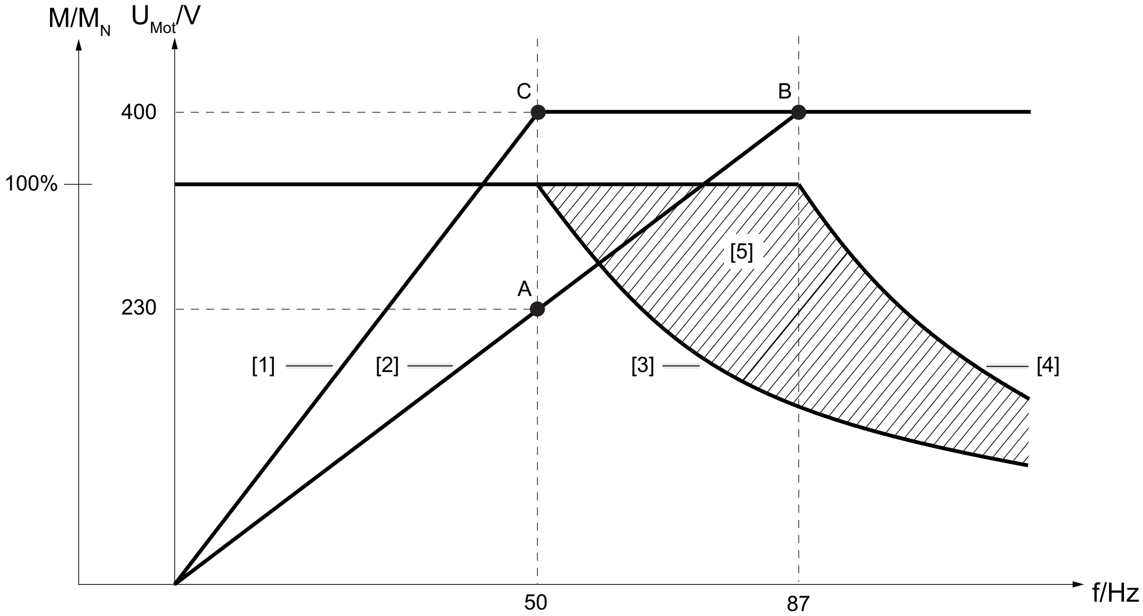
The difference between the frequency inverter input voltage and the rated motor voltage results in a voltage reserve and the ability to further increase the voltage and frequency in the same ratio above the rated point. At a frequency of 87 Hz, the motor voltage at a constant V/f ratio corresponds to the line voltage and cannot be further increased. 87 Hz operation starts when the transition point moves from 50 Hz to 87 Hz. The field weakening range begins after the 87 Hz transition point. In the field weakening range, the available motor torque decreases in proportion to the frequency.

In 87 Hz operation, a higher voltage than the rated value is applied to the winding. Its function is not impaired because the dielectric strength of the winding insulation is greater than 1000 V. Despite higher voltage than the rated value, 87 Hz operation does not result in significantly higher current and therefore does not cause thermal overloading. There is almost no change in the motor current because the torque required by the application remains constant and the magnetization (V/f ratio) remains unchanged. The desired increase in mechanical power is achieved exclusively by increasing the rotational speed (equal to increased frequency) and not by increasing the torque. This results in a power rating that is √3 higher than the rated power of the motor.

V/f characteristic and power gain

In the following diagram, the circuit-dependent V/f characteristics are compared with the corresponding transition and rated points in star and delta connections. The transition points do not necessarily represent the rated points.

The gain in mechanical power is shown by the motor torque curves in the shaded area in the following figure.



25894556555

M/M_N	Ratio of motor torque to rated torque of the motor
U_{Mot}	Motor voltage
[A]	Rated point at 230 V/50 Hz in a delta connection
[B]	Transition point at 400 V/87 Hz in a delta connection
[C]	Transition point (= rated point) at 400 V/50 Hz in a star connection
[1]	V/f characteristic in a star connection
[2]	V/f characteristic in a delta connection
[3]	Curve of the motor torque as a function of the frequency in relation to [1]
[4]	Curve of the motor torque as a function of the frequency in relation to [2]
[5]	Gain in mechanical power

26636735/EN – 07/2020

Motivation and possible applications

The motivation for configuring drives for 87 Hz operation is:

- Expanding the speed setting range with constant motor torque, enabling higher application speeds.
- Using a smaller motor size in conjunction with a higher gear unit ratio reduces
  – The initial costs of the motor
  – The installation space required for the motor
  – The mass of the motor

Please consider:

87 Hz operation should take place in a voltage-controlled control mode. Current-controlled control modes are also possible with limited gains in the speed setting range.

When the field weakening range is used, it is necessary to observe the breakdown torque (proportional to $1/f^2$), which decreases quadratically as the frequency increases and further restricts the available motor torque. For 87 Hz operation, the inverter is selected based on the required motor current in a delta connection, which is $\sqrt{3}$ times higher than in a star connection. On the gear unit side, a higher input speed causes larger churning losses, which can increase the gear unit temperature. This needs to be checked during further project planning.

26636735/EN – 07/2020

6 Project planning for non-controlled drives

Drives that are operated directly from the grid are referred to as non-controlled drives in this documentation. The procedure for configuring non-controlled drives differs from the procedure for configuring controlled drives. The starting and operating behavior is strongly load-dependent and follows the speed-torque characteristic curve due to the motor type and size (see chapter "Basics of project planning for electric drives" (→ 🗎 10)).

The following chapters describe motor selection based on power data as well as gear unit selection based on the service factor:

- Calculating power
- Calculating and selecting the motor
- Calculating and selecting the brake
- Calculating and selecting the gear unit

6.1 Calculating power

During project planning for line-powered motors, it is customary to select the motor based on performance data.

The mechanical power is constant across the entire drive train, except for the losses attributed to efficiencies (see chapter "Efficiency" (→ 🗎 42)). The mechanical power is calculated as the product of force and speed or torque and angular speed.

As a result, the required motor power can already be calculated from the variables of the application. The power can be divided by the factor of 1000 and specified directly in kilowatts.

Power (linear movement):

$$P = \frac{F \times v}{1000}$$

18014418020849547

For rotary movements, the power can be described using the already derived relationship between angular speed and rotational speed as a function of the rotational speed (see chapter "Output speed and gear ratio requirement" (→ 🗎 25)).

Power (rotary movement):

$$P = \frac{M \times \omega}{1000} = \frac{M \times \frac{2\pi \times n}{60}}{1000} = \frac{M \times n}{9550}$$

18014418020851211

P = Power	[P] = kW
F = Force	[F] = N
v = Speed	[v] = m s^{-1}
M = Torque	[M] = Nm
ω = Angular speed	[ω] = s^{-1}
n = Speed	[n] = min^{-1}

Since the motor is the determining variable of the non-controlled drive train, it is selected after the required power as the first component of the drive train, even before the gear unit.

26636735/EN – 07/2020

The main criterion for selecting the motor is the continuous static power to be applied, which is calculated from the static force used to overcome friction or gravity. This ensures that the motor is not thermally overloaded during continuous duty. It may also be necessary to calculate and check the short-term maximum load, i.e., the sum of static and dynamic power, especially during cycle mode.

Static power (linear movement):

$$P_{stat} = \frac{F_{stat} \times v}{1000}$$

18014418020852875

Static power (rotary movement):

$$P_{stat} = \frac{M_{stat} \times n_L}{9550}$$

18014418020854539

Dynamic power (linear movement):

$$P_{dyn} = \frac{F_{dyn} \times v}{1000}$$

18014418020856203

Dynamic power (rotary movement):

$$P_{dyn} = \frac{M_{dyn} \times n_L}{9550}$$

18014418020857867

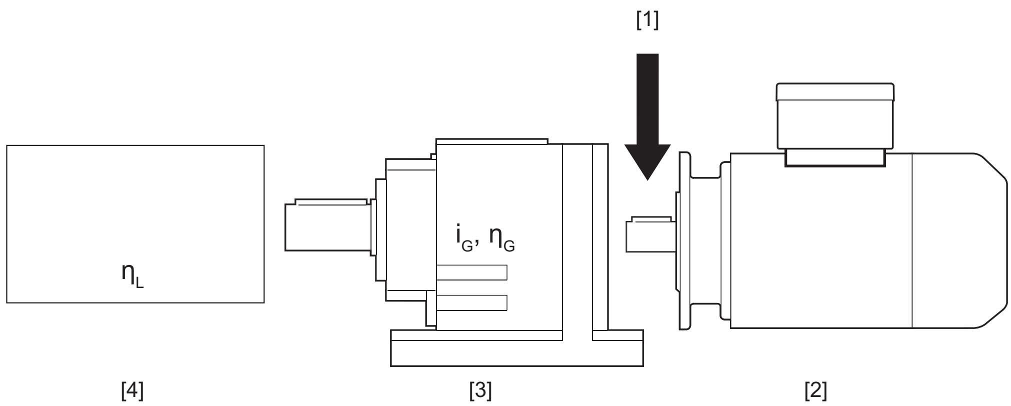
Maximum required power of the application in the "acceleration" travel section:

$$P_{max} = P_{stat} + P_{dyn}$$

18014418020859531

P_{stat}	= Static power of the application	$[P_{stat}]$ = kW
F_{stat}	= Static force of the application	$[F_{stat}]$ = N
v	= Speed of the application	[v] = m s^{-1}
M_{stat}	= Static torque of the application	$[M_{stat}]$ = Nm
n_L	= Rotational speed of the application	$[n_L]$ = min^{-1}
P_{dyn}	= Dynamic power of the application	$[P_{dyn}]$ = kW
F_{dyn}	= Dynamic force of the application	$[F_{dyn}]$ = N
M_{dyn}	= Dynamic torque of the application	$[M_{dyn}]$ = Nm
P_{max}	= Maximum required power of the application in the "acceleration" travel section	$[P_{max}]$ = kW

26636735/EN – 07/2020



18014418020861195

[1] Reference point on motor shaft
[2] Motor
[3] Gear unit
[4] Application
η_L Load efficiency
i_G Gear unit ratio
η_G Gear unit efficiency

The calculated power values of the application are applied to the overall efficiency as the product of load efficiency η_L and gear unit efficiency η_G (see chapter "Efficiency" (→ 🗎 42)). This results in the power requirements for the motor.

Static power, including load and gear unit efficiency, for the “constant speed” travel section:

$$P_{Mot\_stat} = \frac{P_{stat}}{\eta_L \times \eta_G}$$

18014418020863371

Maximum power of the application, including load and gear unit efficiency, for the "acceleration" travel section:

$$P_{Mot\_max} = \frac{P_{max}}{\eta_L \times \eta_G}$$

18014418020865035

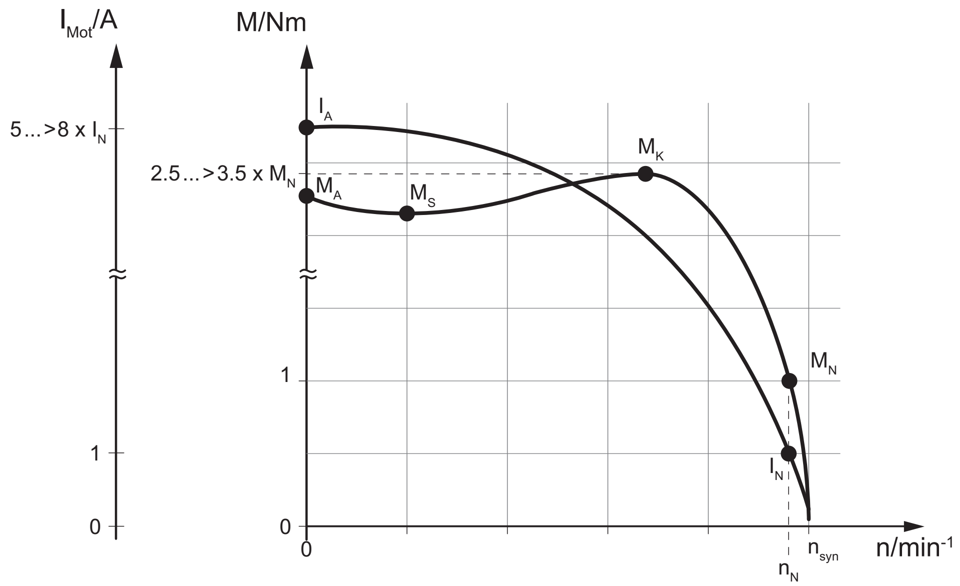
P_{Mot_stat}	= Static power of the application as a requirement of the motor, including efficiencies (motor mode)	$[P_{Mot\_stat}]$ = kW
P_{stat}	= Static power of the application	$[P_{stat}]$ = kW
η_L	= Load efficiency	$[\eta_L]$ = 1
η_G	= Gear unit efficiency	$[\eta_G]$ = 1
P_{Mot_max}	= Maximum power of the application as a requirement of the motor, including efficiencies (motor mode)	$[P_{Mot\_max}]$ = kW
P_{max}	= Power of the application in the "acceleration" travel section	$[P_{max}]$ = kW

26636735/EN – 07/2020

6.2 Calculating and selecting the motor

6.2.1 Speed-torque characteristic of the asynchronous motor

The speed-torque characteristic (see figure below) describes the behavior of asynchronous AC motors during startup from the line power supply system. Each motor type and motor size is characterized by its own speed-torque characteristic. The following values apply to IE3 DRN series motors. In addition, the associated speed-current characteristic is shown as an additional parameter in the same diagram.



18014418020869259

I_{Mot}	Motor current as a function of the rotational speed
M	Motor torque as a function of the rotational speed
I_A	Starting current
M_A	Starting torque
M_S	Pull-up torque
M_K	Breakdown torque
M_N	Rated torque
I_N	Rated motor current
n_N	Rated speed
n_{syn}	Synchronous speed
n	Speed

The starting torque M_A is independent of the applied motor utilization as soon as the motor is switched on from the line power supply system. The motor passes through its speed-torque characteristic curve during the subsequent startup phase. It passes through the characteristic points for pull-up torque M_S and breakdown torque M_K until the load-dependent operating point is reached. When loaded with rated torque, the motor reaches its rated speed after the acceleration phase. The exact course of the speed-torque characteristic is dependent on design factors, such as the shape of the conductor bars in the rotor for skin effect, or the number of poles of the motor.

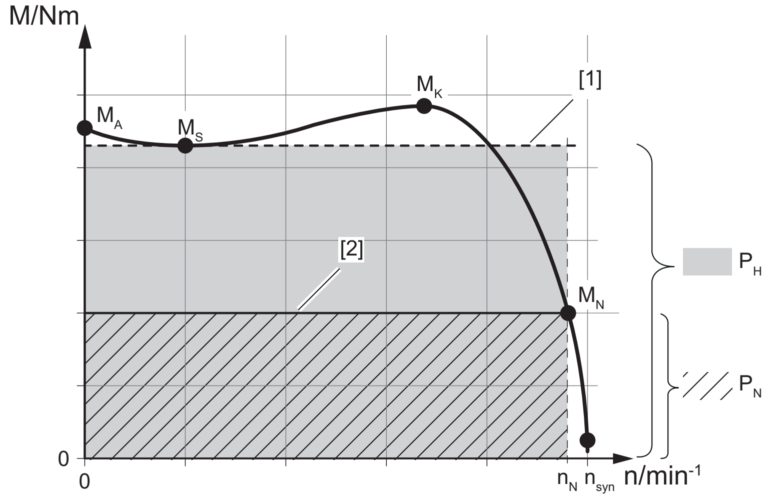
Depending on the motor type and design, the breakdown torque is 2 to 4 times greater than the rated torque. If the motor is loaded so heavily during operation that the required torque could exceed the breakdown torque, the motor speed drops back to the value 0 (motor idle state). This is commonly referred to as the stall torque of the motor. If no preventive measures are taken, the motor can be destroyed from overheating due to the high current load under these conditions.

26636735/EN – 07/2020

The maximum motor current flows at the moment of startup from the grid power source (speed = 0 min^{-1}). The maximum current for IE3 motors is 5 to 9 times the rated motor current. As the speed increases, the motor current decreases until the rated motor current flows once the rated torque is reached.

The rated operating point is the thermal rated point of the motor. If the motor is loaded with nominal load, it can be operated continuously without thermal overloading. Under higher loads, the motor may be operated only briefly in intermittent duty.

To simplify the speed-torque characteristic, a constant acceleration torque M_H is defined as a substitute characteristic quantity. This acceleration torque is approximately available during the entire acceleration phase.



18014418301184651

[1] Acceleration torque M_H
[2] Load characteristic at constant load ($M_{Mot\_stat} = M_N$)
M Torque
M_A Starting torque
M_S Pull-up torque
M_K Breakdown torque
M_N Rated torque
n_N Rated speed
n_{Syn} Synchronous speed
n Speed
P_H Available motor power during startup
P_N Rated power of the motor

When selecting the line-powered motor, the operating point should be below the nominal operation point. This results in a motor speed close to the rated speed, between the rated speed and synchronous speed. Due to the minimal rotational speed deviation, the motor can be selected based on the required power (see chapter "Calculating power" (→ 🗎 90)). In the figure, the power is simplified as a product of torque and speed and indicated by the marked areas.

6.2.2 Selection criteria

The line-powered motor is selected based on 2 criteria:

First selection criterion: Static power

Selecting the motor according to static power ensures that the motor will not be thermally overloaded in S1 duty. The required continuous power P_{Mot_stat} must be less than or equal to the rated power P_N of the motor.

26636735/EN – 07/2020

The influence of possible switching operations on the thermal capacity utilization of the motor in intermittent duty is explained in detail in the section "Switching frequency" (→ 🗎 102).

$$P_{Mot\_stat} \leq P_N$$

18014418020876683

P_{Mot_stat}	=	Static power of the application as a requirement of the motor, including efficiencies (motor mode)	$[P_{Mot\_stat}]$ = kW
P_N	=	Rated power of the motor (catalog value)	$[P_N]$ = kW

Second selection criterion: Maximum required power

Selecting the motor according to the maximum required power ensures that in addition to S1 duty, the required minimum acceleration of the application can be maintained. The maximum required power P_{Mot_max} must be less than the power P_H of the motor converted during startup. The available startup power can be calculated from the rated power and the startup factor (M_H/M_N). These values can be found in the relevant product catalog.

$$P_H = P_N \times \frac{M_H}{M_N}$$

28421257867

$$P_{Mot\_max} \leq P_H$$

18014418020878347

P_H	=	Available motor power during startup	$[P_H]$ = kW
P_N	=	Rated power of the motor (catalog value)	$[P_N]$ = kW
M_H/M_N	=	Startup factor (catalog value)	$[M_H/M_N]$ = 1
P_{Mot_max}	=	Maximum required power of the application as a requirement of the motor, including efficiencies (motor mode)	$[P_{Mot\_max}]$ = kW

Of the two selection criteria, the first criterion (selection according static power) has priority because it prevents thermal overloading of the drive. Failure to meet the first criterion may result in damage to the motor.

If the selection criterion for maximum required power is not met, the motor will not be damaged, but the application will be accelerated more slowly. Possible requirements for cycle times cannot be met.

Intermittent duty

If the motor does not run in S1 mode, there is less heat development due to the change of load sections and cooling during break sections. These additional thermal reserves allow for greater capacity utilization of the motor in the load sections than at rated power. The possible higher motor load is expressed by the power increase factor K.

$$P_{Mot\_stat} \leq P_N \times K$$

25008551179

P_{Mot_stat}	=	Static power of the application as a requirement of the motor, including efficiencies (motor mode)	$[P_{Mot\_stat}]$ = kW
P_N	=	Rated power of the motor (catalog value)	$[P_N]$ = kW
K	=	Power increase factor	[K] = 1

26636735/EN – 07/2020

The table below shows the power increase factor K. The respective operating modes are defined in IEC 60034.

Power increase factor K for the different operating modes

Operating mode			Power increase factor K
S2	Operating time	60 min	1.1
		30 min	1.2
		10 min	1.4
S3	Cyclic duration factor ED relative to 10 min cycle duration.	60%/75%	1.1
		40%	1.15
		25%	1.3
		15%	1.4
S4 – S10	The following information must be specified to determine the rated power and the operating mode: number and type of cycles per hour, run-up time, time under load, braking type, braking time, idling time, cycle duration, period at rest and power demand.		On request

6.2.3 Checking motor start-up

The ratio of startup power to rated power can be within a value range of 1.6 to 2.9 due to characteristic features of AC asynchronous motors. This range of values applies equally to the ratio of acceleration torque to rated torque. As a result, a significantly higher acceleration torque may be available than is necessary for the application (see chapter "Evaluating starting behavior" (→ 🗎 68)).

Excessively high acceleration can cause damage to mechanical components such as transmission elements, additional transmissions or gear units, or lead to slippage of wheels or belts in travel applications, for example. This can result in a deviating travel cycle or greater material wear. It therefore makes sense to recalculate the actual starting behavior of the drive and to compare it with the required parameters before further components of the drive train are selected.

The selected motor develops its acceleration torque during each startup process, regardless of the load. Part of this acceleration torque is used to overcome the static resistance forces (friction or gravity). This static load torque M_{Mot_stat} on the motor shaft results from the already calculated static power P_{Mot_stat}.

$$M_{Mot\_stat} = \frac{P_{Mot\_stat} \times 9550}{n_{Mot}}$$

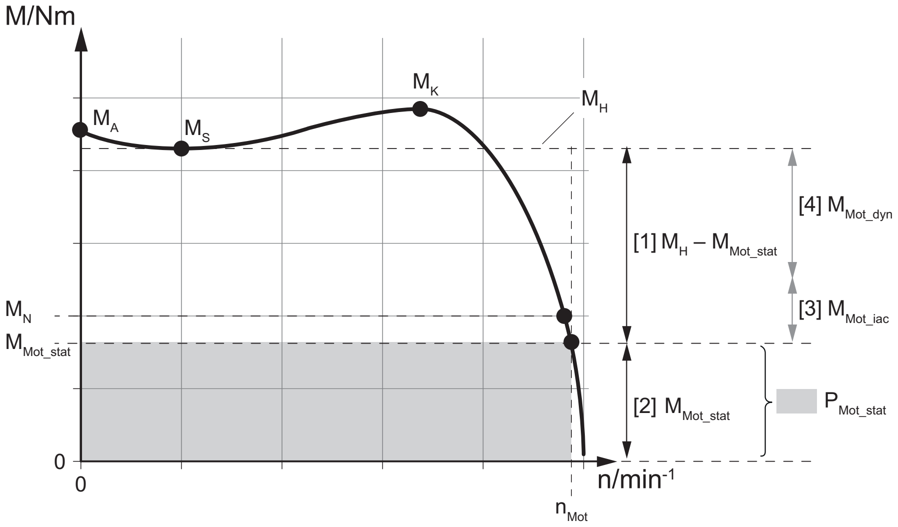
25047385867

M_{Mot_stat}	=	Static torque of the application as a requirement of the motor, including efficiencies (motor mode)	$[M_{Mot\_stat}]$ = Nm
P_{Mot_stat}	=	Static power of the application as a requirement of the motor, including efficiencies (motor mode)	$[P_{Mot\_stat}]$ = kW
n_{Mot}	=	Motor speed	$[n_{Mot}]$ = min^{-1}

The remaining part of the acceleration torque ($M_H - M_{Mot\_stat}$) is used to accelerate all existing mass moments of inertia. These mass moments of inertia are primarily the intrinsic inertia of the motor J_{Mot} and the options as well as the load inertia reduced to the motor shaft. Depending on the mass moment of inertia ratio (J_x/J_{Mot}), the available acceleration torque (M_H - M_{Mot_stat}) is divided proportionally across both mass moments of inertia.

26636735/EN – 07/2020

The figure below shows examples of the relevant torques. All application parameters (M_{Mot_stat}, J_x) are considered on the motor side in this context so that the values can be directly compared to the technical data of the motor (M_N, M_H, J_{Mot}).



18014418301458827

[1]	Acceleration torque (M_H - M_{Mot_stat})
[2]	Static load torque on the motor shaft M_{Mot_stat}
[3]	Acceleration torque, proportional for motor (In this example, 1:2 where J_x/J_{Mot} = 2)
[4]	Acceleration torque, proportional for load (In this example 2:1 where J_x/J_{Mot} = 2)
M	Torque
M_N	Rated torque
M_{Mot_stat}	Static torque of the application as a requirement of the motor, including efficiencies (motor mode)
M_{Mot_dyn}	Dynamic torque
M_{Mot_iac}	Dynamic torque for intrinsic acceleration or deceleration of the motor
P_{Mot_stat}	Static power of the application as a requirement of the motor, including efficiencies (motor mode)
M_A	Starting torque
M_S	Pull-up torque
M_K	Breakdown torque
M_H	Acceleration torque
n_{Mot}	Motor speed
n	Speed

The previously adapted calculation formula (see section "Motor utilization") for the dynamic torque as a function of the rotational speed and the acceleration time is used to calculate the actual run-up time t_H. This formula is solved using the acceleration time t_{ac}.

Dynamic torque:

$$M_{dyn} = J \times \alpha = \frac{J \times n}{9.55 \times t_{ac}}$$

18014418020884619

26636735/EN – 07/2020

Acceleration time:

$$t_{ac} = \frac{J \times n}{9.55 \times M_{dyn}}$$

18014418020886283

M_{dyn}	= Dynamic torque	$[M_{dyn}]$ = Nm
J	= Mass moment of inertia	[J] = kg m^2
α	= Angular acceleration	[α] = s^{-2}
n	= Speed	[n] = min^{-1}
t_{ac}	= Acceleration time	$[t_{ac}]$ = s

The run-up time t_H of the drive is obtained by substituting the following variables in this formula:

- Acceleration time $t_{ac} = t_H$
- Mass moment of inertia $J = J_{Mot} + J_x$
- Rotational speed $n = n_{Mot}$
- Acceleration torque $M_{dyn} = M_H - M_{Mot\_stat}$

The relationship ($M_H - M_{Mot\_stat}$) applies to vertical upward movements, horizontal and rotary movements. The variables reduced to the motor shaft, such as load moment of inertia and static load torque, take into account the relevant efficiencies.

Run-up time (for upward vertical, horizontal and rotary movements):

$$t_H = \frac{\left(J_{Mot} + \frac{J_x}{\eta_L \times \eta_G}\right) \times n_{Mot}}{9.55 \times \left(M_H - M_{Mot\_stat}\right)}$$

18014418020887947

t_H	= Run-up time	$[t_H]$ = s
J_{mot}	= Mass moment of inertia of the motor	$[J_{Mot}]$ = kg m^2
J_x	= Mass moment of inertia of the load reduced to the motor shaft	$[J_x]$ = kg m^2
η_L	= Load efficiency	$[\eta_L]$ = 1
η_G	= Gear unit efficiency	$[\eta_G]$ = 1
n_{Mot}	= Motor speed	$[n_{Mot}]$ = min^{-1}
M_H	= Acceleration torque	$[M_H]$ = Nm
M_{Mot_stat}	= Static torque of the application as a requirement of the motor, including efficiencies (motor mode)	$[M_{Mot\_stat}]$ = Nm

The actual operating speed of the motor, n_{Mot}, is usually greater than the rated motor speed and less than the synchronous speed. If the load torque is known, the operating speed can be precisely determined only with the aid of the motor characteristics. The rated motor speed is used to simplify the calculation due to the small deviation between the operating and rated motor speed.

Evaluating starting behavior

For certain applications, a close examination of the starting behavior may be required. As mentioned above, excessively high acceleration can cause damage to mechanical components such as transmission elements, additional transmissions or gear units, or lead to slippage of wheels or belts in travel applications, for example. This can result in a deviating, uncontrolled travel cycle and greater material wear.

It is therefore necessary to calculate the acceleration of the application during motor startup and to compare it with the required parameters before further components of the drive train are selected.

Acceleration of the application during motor startup:

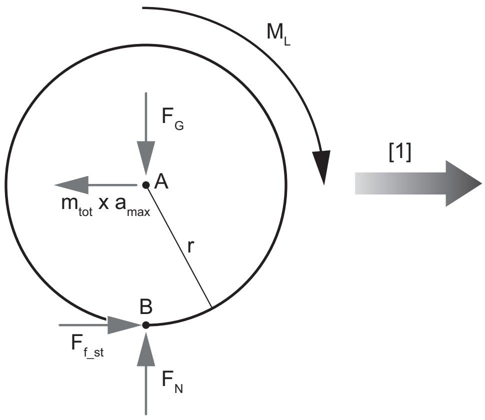
$$a_H \approx \frac{v}{t_H}$$

18014418020889611

a_H	=	Acceleration of the application during motor startup	$[a_H] = m\ s^{-2}$
v	=	Speed of the application	$[v] = m\ s^{-1}$
t_H	=	Run-up time	$[t_H] = s$

In general, the calculated startup acceleration can be checked against the maximum permitted acceleration of the application. For travel applications, the maximum transferable acceleration can be calculated from the application data itself:

In the case of travel applications, the maximum permitted acceleration can be deduced by considering the slippage of the wheels. After a free body diagram of a wheel is created, equilibra of forces and moments are established and a term is derived for calculating the maximum possible acceleration.



26877721867

[1] Direction of movement
A Center of rotation of wheel
B Contact point between wheel and floor (no relative movement)
M_L Input torque of the application
F_G Gravitational force
m_{tot} Total mass of the application
a_{max} Maximum possible acceleration
r Radius
F_{f_st} Static friction force
F_N Normal force

Force equilibrium in horizontal direction of movement:

$$F_{f\_st} = m_{tot} \times a_{max}$$

27505964939

The moment equilibrium around the reference point B results in the following relationship:

$$M_L = m_{tot} \times a_{max} \times r$$

27505807755

26636735/EN – 07/2020

The maximum transferable input torque M_L, which does not yet cause slippage of the wheels, corresponds to the counter-torque resulting from the static friction force and is used instead of the maximum transferable input torque, M_L:

$$F_{f\_st} \times r = m_{tot} \times a_{max} \times r$$

27505804043

The static friction force F_{f_st} is replaced by the product of normal force F_N and friction coefficient μ_{f_st}, where the normal force F_N corresponds to the gravitational force F_G. In multi-wheel travel applications, it is assumed that the total mass is evenly distributed across all the wheels.

For travel applications with multiple drives, the following design still applies if the total input torque is evenly distributed across all driven wheels.

$$m_1 \times g \times \mu_{f\_st} \times r = m_{tot} \times a_{max} \times r$$

27505799947

This results in the maximum permitted acceleration as a function of the mass distribution or the number of driven wheels:

$$a_{max} = \frac{m_1}{m_{tot}} \times g \times \mu_{f\_st}$$
$$a_{max} = \frac{N_1}{N_{tot}} \times g \times \mu_{f\_st}$$

27505458699

For a travel application in which all wheels are driven ($m_1 = m_{tot}$), the following relationship applies:

$$a_{max} = g \times \mu_{f\_st}$$

27505454987

F_{f_st}	= Static friction force	$[F_{f\_st}]$ = N
m_{tot}	= Linearly moving total mass	$[m_{tot}]$ = kg
a_{max}	= Maximum possible acceleration	$[a_{max}]$ = m s^{-2}
M_L	= Input torque of the application	$[M_L]$ = Nm
r	= Radius of driven wheels	[r] = m
m_1	= Proportion of the total mass applied to the driven wheels	$[m_1]$ = kg
g	= Gravitational acceleration (9.81 m s^{-2})	[g] = m s^{-2}
μ_{f_st}	= Static friction coefficient	$[\mu_{f\_st}]$ = 1
N_1	= Number of driven wheels	$[N_1]$ = 1
N_{tot}	= Total number of wheels	$[N_{tot}]$ = 1

The startup acceleration of the application resulting from motor startup is compared with the calculated limit acceleration. The following condition must be met to ensure that wheels do not slip during travel applications or to ensure that any generally undesirable behavior of the application during motor startup is avoided.

$$a_H < a_{max}$$

28272983179

a_H	= Startup acceleration	$[a_H]$ = m s^{-2}
a_{max}	= Maximum possible acceleration	$[a_{max}]$ = m s^{-2}

If this condition is met, the wheels will not slip, even when the motor starts up. Otherwise, the following actions can be taken:

- Increase proportion of driven wheels (maximum 100% of the wheels driven).
- Increase static friction coefficient using a suitable combination of materials.

- Reduce the available acceleration torque for the application by varying the inertia ratio.
- Use controlled drives.

After one or more actions are taken, it is necessary to verify whether the condition is met.

6.2.4 Switching frequency

If a line-powered motor specified for S1 operating mode is to be used in any form of intermittent duty, it must be thermally tested separately. This is done using the permitted switching frequency, which specifies the maximum number of starts per hour and must be greater than the required switching frequency of the application.

$$Z_{per} \geq Z_{req}$$

18014418020918539

Z_{per}	= Permitted switching frequency	$[Z_{per}] = h^{-1}$
Z_{req}	= Required switching frequency	$[Z_{req}] = h^{-1}$

The permitted switching frequency Z_{per} is calculated from the no-load starting frequency, Z_0, of the brakemotor and load-dependent calculation factors that have a reductive effect. This takes into account the high inrush current of the motor, which can significantly influence the thermal load. In addition, relevant application data are included in the calculation.

The effect of the respective influencing factor is considered separately for each calculation factor. In order to map the entirety of the application, the no-load starting frequency of the brakemotor is multiplied by all the calculation factors.

$$Z_{per} = Z_0 \times K_J \times K_M \times K_P$$

18014418020901643

Z_{per}	= Permitted switching frequency	$[Z_{per}] = h^{-1}$
Z_0	= No-load starting frequency of the brakemotor	$[Z_0] = h^{-1}$
K_J	= Calculation factor for mass moment of inertia f (J_x, J_Z, J_{Mot})	$[K_J] = 1$
K_M	= Calculation factor for static motor torque f (M_{Mot_stat}, M_H)	$[K_M] = 1$
K_P	= Calculation factor for static power and cyclic duration factor f (P_{stat}, P_N, ED)	$[K_P] = 1$

No-load starting frequency Z_0

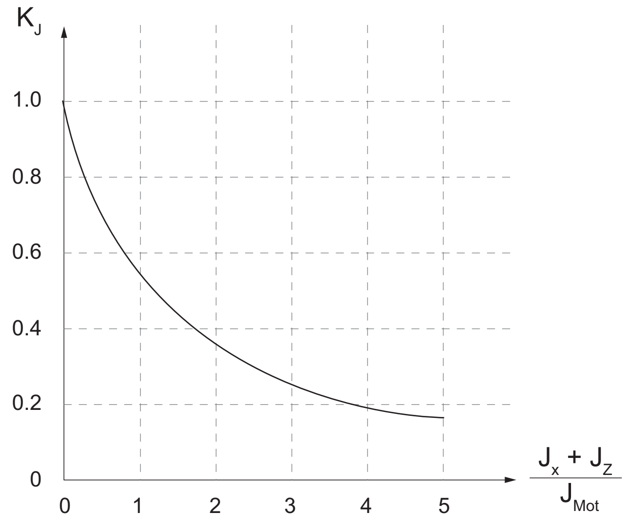
The no-load starting frequency describes the permitted number of starts of a brakemotor per hour without external loads such as static load torque or mass moment of inertia of the load. The motor current causes a rise in temperature by cyclically accelerating the intrinsic inertia. The thermal load is within a permitted range if the number of starts is smaller than the no-load starting frequency. The no-load starting frequency Z_0 of each brakemotor is dependent on the brake control used, each of which has a different brake application time and can be found in the product catalogs.

Different brake application times influence the no-load starting frequency because the brakemotor briefly runs against the applied brake when the brake is released. Reduced no-load starting frequencies generally apply for longer brake application times. Brake rectifiers with a short brake application time (e.g., BGE brake rectifier) are usually used. For line-powered motors without a brake, cycle mode is often not possible, since coasting the motor to a stop is time consuming and prevents a short-pulsed voltage supply.

Calculation factor K_J (mass moment of inertia)

Calculation factor K_J is used to take into account all other inertia factors that must be accelerated in addition to the intrinsic inertia of the motor.

With a constant acceleration torque, the higher total inertia leads to a longer acceleration time. This means that the motor passes through the speed-torque-current characteristic more slowly. The resulting increase in effective current during the run-up time causes a stronger increase in the temperature of the motor. To prevent thermal overloading of the motor, the permitted number of acceleration processes must be reduced by the factor K_J.



21490572811

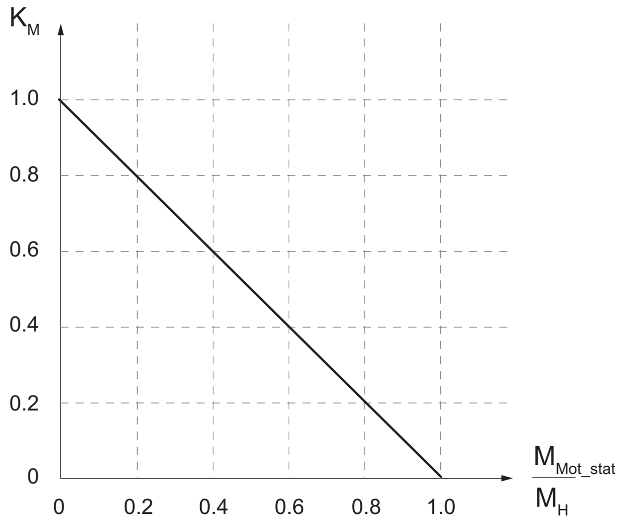
K_J Calculation factor for mass moment of inertia
J_x Load moment of inertia, reduced to motor shaft
J_Z Mass moment of inertia of heavy additional mass/Z
J_{Mot} Motor's mass moment of inertia

26636735/EN – 07/2020

Calculation factor K_M (static motor torque)

The calculation factor K_M is used for considering the part of the acceleration torque that is available minus the static torque for acceleration of the motor inertia.

This reduced acceleration torque means that the motor inertia is accelerated more slowly. In other words, the motor passes through the characteristic speed-torque-current characteristic more slowly. The resulting increase in effective current during the run-up time causes a stronger increase in the temperature of the motor. To prevent thermal overloading of the motor, the permitted number of acceleration processes must be reduced by the factor K_M.



21490577163

K_M	Calculation factor for static torque of the motor
M_{Mot_stat}	Static torque of the application as a requirement of the motor, including efficiencies (motor mode)
M_H	Acceleration torque of the motor

Calculation factor K_P (static power and cyclic duration factor)

The calculation factor K_P is used for considering the temperature increase of the motor that is independent of the static capacity utilization and the cyclic duration factor.

This temperature increase limits the permitted number of acceleration processes because the motor temperature continues to increase each time the motor passes through the speed-torque-current characteristic.

Based on the available static capacity utilization, the assigned curve is selected from the series of curves to determine the K_P factor as a function of the cyclic duration factor.

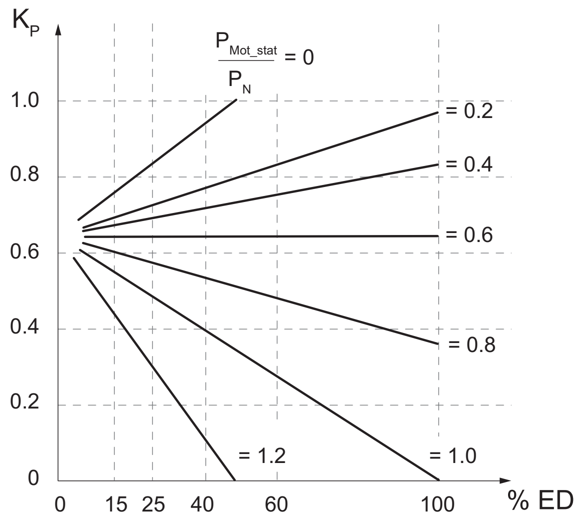
Example 1:

If the static capacity utilization of the motor is already 100%, the assigned curve $P_{Mot\_stat} / P_N = 1.0$ from the series of curves must be used. If the cyclic duration factor is 100%, the K_P factor = 0. This means that no additional acceleration is possible.

26636735/EN – 07/2020

Example 2:

The K_P factor = 0.4 with same capacity utilization P_{Mot_stat} /P_N = 1.0 and 40% cyclic duration factor. This means that 40% of the permitted empty no-load starting frequency Z_0 is available.



21491016715

K_P	Calculation factor for static power and cyclic duration factor
P_{Mot_stat}	Static power of the application as a requirement of the motor, including efficiencies (motor mode)
P_N	Rated power of the motor
ED	Cyclic duration factor

26636735/EN – 07/2020

Calculating the permitted switching frequency

To calculate the permitted switching frequency Z_{per}, the calculation factors K_J and K_M can be expressed using the already known parameters of the motor and application and summarized in a formula. The calculation factor K_P cannot be calculated in a practical manner due to multiple influencing factors, and is read from the series of curves.

Permitted switching frequency (upward vertical, horizontal or rotary direction of movement):

$$Z_{per} = Z_0 \times \frac{1 - \frac{M_{Mot\_stat}}{M_H}}{\frac{J_{Mot} + \frac{J_x}{\eta_L \times \eta_G}}{J_{Mot}}} \times K_P$$

18014418020916875

Z_{per}	= Permitted switching frequency	$[Z_{per}] = h^{-1}$
Z_0	= No-load starting frequency with 50% cyclic duration factor	$[Z_0] = h^{-1}$
M_{Mot_stat}	= Static torque of the application as a requirement of the motor, including efficiencies (motor mode)	$[M_{Mot\_stat}]$ = Nm
M_H	= Acceleration torque	$[M_H]$ = Nm
J_{Mot}	= Mass moment of inertia of the motor	$[J_{Mot}] = kgm^2$
J_x	= Mass moment of inertia of the load reduced to the motor shaft	$[J_x] = kgm^2$
η_L	= Load efficiency	$[\eta_L] = 1$
η_G	= Gear unit efficiency	$[\eta_G] = 1$
K_P	= Calculation factor: static power and cyclic duration factor f (P_{Mot_stat}, P_N, ED)	$[K_P] = 1$

The permitted switching frequency in the vertically downward direction of movement cannot be calculated using a compact formula because of the ambiguous direction of the force flow.

26636735/EN – 07/2020

6.3 Calculating and selecting the brake

This section covers the following topics:

- Special requirement for lifting applications
- Braking work and brake application speed
- Service life until inspection
- Effects on the gear unit
- Further selection criteria

INFORMATION

Note that the value of the each variable must be included in the brake design formulas.

Non-controlled drives can always be brought to an idle state in two different ways.

- Coasting to a stop: Stopped by friction in the application (exception: lifting applications).
- Active braking: Stopped by a mechanical brake (working brake).

The brake has different requirements depending on the application. In considering these requirements, a distinction is made between applications in the horizontal and vertical direction of movement. For lifting applications with a vertical or oblique direction of movement, this results in a special requirement with respect to the ratio of braking torque to load torque. This requirement must be observed in addition to the general specifications for brake configuration. This requirement generally applies to all applications with external loads on the drive at an idle state, such as additional process forces or wind load.

The following chapters consider only those braking processes in which the mechanical brake is primarily responsible for stopping the application. This means that braking torque generated by friction in the application is significantly lower than the braking torque generated by the mechanical brake. In all other cases, contact SEW-EURODRIVE.

INFORMATION

In this chapter, all brake calculations refer to the "Standard” load range as well as the nominal value of the braking torque. For detailed information, overload ranges as well as further calculation options, see the "Project Planning for BE.. Brakes” manual.

26636735/EN – 07/2020

6.3.1 Special requirement for lifting applications

For lifting applications, the following must be observed when dimensioning the brake and the braking torque:

- The brake must reliably hold the application at an idle state (static load on the brake).
- The application must be reliably stoppable to an idle state (dynamic load on the brake).

In order to meet these requirements, the following criterion must be met:

The braking torque must be at least 200% of the required static motor torque during downward movement.

$$M_B \geq 2 \times M'_{Mot\_stat}$$

26546341643

INFORMATION

The efficiencies must be multiplied due to the regenerative operation of the motor during downward movement.

$$M'_{Mot\_stat} = \frac{P_{stat} \times 9550}{n_{Mot}} \times \eta_L \times \eta'_G$$

28885791883

M_B	= Braking torque	$[M_B]$ = Nm
M'_{Mot_stat}	= Static torque of the application as a requirement of the motor, including efficiencies (generator mode)	$[M'_{Mot\_stat}]$ = Nm
P_{stat}	= Static power	$[P_{stat}]$ = kW
n_{Mot}	= Motor speed	$[n_{Mot}]$ = min^{-1}
η_L	= Load efficiency	$[\eta_L]$ = 1
η'_G	= Retrodriving gear unit efficiency	$[\eta'_G]$ = 1

- The following applies to helical-worm and SPIROPLAN® gear units:
  $\eta'_G = 2-1/\eta_G$
- The following applies to all other gear units:
  $\eta'_G = \eta_G$

For further calculation steps, a preliminary braking torque is selected according to this requirement. This torque may need to be increased later in the calculation process.

Technical background: Factor 2

The factor 2 (200%) can be explained as follows:

In order to decelerate the application mechanically, the braking torque must at least be greater than the static motor torque during downward movement. If both torques were equal, this would result only in downward movement with constant speed and would not result in mechanical deceleration.

In addition, the braking torque is subject to various physical influences, such as friction speed, temperature and other environmental influences that reduce the braking effect. To account for these effects, the static motor torque of the downward movement is applied with the above-mentioned factor of 2. This is the minimum requirement for the braking torque to be selected.

26636735/EN – 07/2020

6.3.2 Braking work

In contrast to brake selection for controlled drives, the switching frequency must be considered because mechanical braking occurs during normal operation of non-controlled drives. The switching frequency of the brake corresponds to the required and already known switching frequency of the motor, Z_{req} (see chapter "Switching frequency" (→ 🗎 102)).

The permitted braking work is a characteristic that describes the thermal working capacity of the brake for a single braking operation. The permitted braking work is fundamentally dependent on the brake size as well as the switching frequency and the brake application speed. The calculation of these influencing factors is described in the following subsections.

Characteristic curves exist for all brakes at typical brake application speeds (close to the rated speed of the respective motor). These characteristics are summarized in a diagram and show the dependence of the permitted braking work W_{B_per} on the permitted switching frequency Z_{per}. The diagram shows a typical brake application speed above the operational brake application speed. The operating point OP_B of the brake to be selected is obtained using the diagram for the typical brake application speed as well as the maximum braking work to be calculated and the required switching frequency.

In the next step, a brake is selected with a characteristic curve that runs above this operating point.

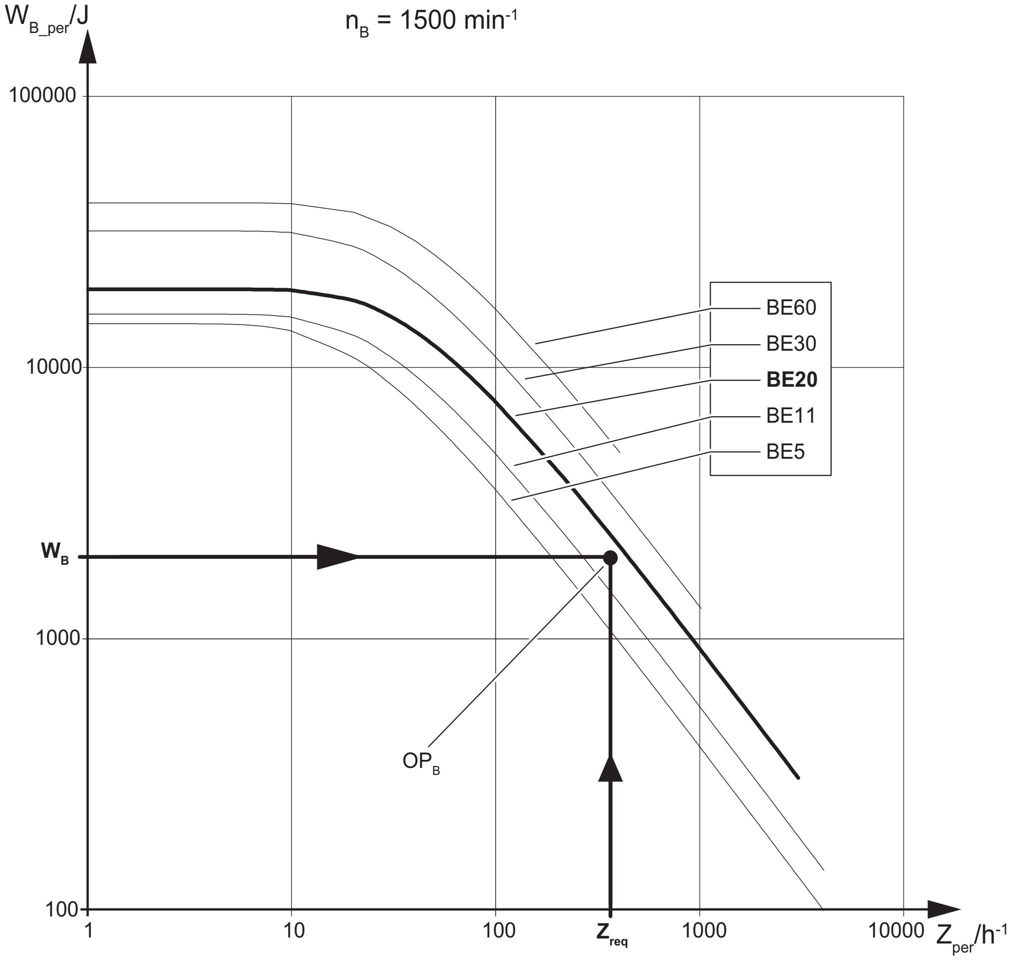
The characteristic curves for the permitted braking work are listed in the "Project Planning for BE.. Brakes" manual and the product documentation.

Example:

- Operational brake application speed n_B = 1450 min^{-1}: diagram to be selected with typical brake application speed n_B = 1500 min^{-1}
- Braking work to be done W_B = 2000 J
- Required switching frequency Z_{req} = 400 h^{-1}

26636735/EN – 07/2020

Selection of brake BE20 based on operating point OP_B (Z_{req}/W_B).



26546154507

W_{B_per}	= Permitted braking work	$[W_{B\_per}]$ = J
W_B	= Braking work to be done	$[W_B]$ = J
Z_{per}	= Permitted switching frequency	$[Z_{per}] = h^{-1}$
Z_{req}	= Required switching frequency	$[Z_{req}] = h^{-1}$
OP_B	= Operating point of the brake	
n_B	= Brake application speed	$[n_B] = min^{-1}$

INFORMATION

To avoid thermal overloading of the brake, the maximum brake application speed for each size and the maximum switching frequency must be observed. The curves may not be extrapolated.

In addition to various application data, the braking torque M_B enters into the calculation of the braking work to be done, W_B. In the case of lifting applications, the provisionally selected braking torque corresponding to the criterion (see chapter "Special requirement for lifting applications" (→ 🗎 108)) is applied. For all other applications without external loads acting on the drive at an idle state, the standard brake torque of the respective brake size is used in accordance with the "Project Planning for BE.. Brakes" manual.

Different calculation formulas are used for the absorbed braking work W_B depending on whether the static torque of the application M_{stat} supports or impedes the braking process.

Calculating the braking work to be done

- Horizontal or rotary direction of movement:

$$W_B = \frac{M_B}{M_B + M'_{Mot\_stat}} \times \frac{(J_{Mot} + J_x \times \eta_L \times \eta'_G) \times n_B^2}{182.5}$$

25358018699

The braking work in the vertically upward direction of movement is smaller than in the vertically downward direction of movement. As a result, it is not calculated here.

- Vertically downward direction of movement:

$$W_B = \frac{M_B}{M_B - M'_{Mot\_stat}} \times \frac{(J_{Mot} + J_x \times \eta_L \times \eta'_G) \times n_B^2}{182.5}$$

25358022411

W_B	=	Braking work to be done	$[W_B]$ = J
M_B	=	Braking torque	$[M_B]$ = Nm
M'_{Mot_stat}	=	Static torque of the application as a requirement of the motor, including efficiencies (generator mode)	$[M'_{Mot\_stat}]$ = Nm
J_{mot}	=	Mass moment of inertia of the motor	$[J_{Mot}]$ = kg m^2
J_x	=	Mass moment of inertia of the load reduced to the motor shaft	$[J_x]$ = kg m^2
η_L	=	Load efficiency	$[\eta_L]$ = 1
η'_G	=	Retrodriving gear unit efficiency • The following applies to helical-worm and SPIROPLAN® gear units: $\eta'_G = 2\text{-}1/\eta_G$ • The following applies to all other gear units: $\eta'_G = \eta_G$	$[\eta'_G]$ = 1
n_B	=	Brake application speed	$[n_B]$ = min^{-1}

Based on the characteristics of the various brakes, a brake is selected whose permitted braking work W_{B_per} for the brake application speed n_B is greater than the brake work to be done, W_B.

$$W_B \leq W_{B\_per}$$

26666293643

W_B	=	Braking work to be done	$[W_B]$ = J
W_{B_per}	=	Permitted braking work as a function of the brake size and application speed	$[W_{B\_per}]$ = J

The braking work can be divided across several brakes using multi-motor operation if no brake is available that can do the braking work at a given switching frequency and brake application speed. In order to sufficiently reduce the braking work, the switching frequency or the brake application speed, alternative measures can be taken so that an available brake size can be used. For example, the braking work to be done for a lifting application can be reduced by selecting a larger braking torque. This causes an increase in gear unit load, which must be mathematically verified.

26636735/EN – 07/2020

6.3.3 Brake application speed

The brake application speed is defined as the motor speed at which the mechanical braking process begins.

The brake application speed corresponds in most cases to the operational motor speed of the application under consideration. If external forces such as gravitational force or process forces are applied, the load during the brake application time t_2 is additionally accelerated. The motor speed can increase significantly in this case. As a result, the brake application speed is greater than the operational motor speed. To determine the resulting brake application speed, the speed difference is calculated and added to the operational motor speed.

Example: Deceleration of a hoist without counterweight in the downward direction of movement.

Calculation of the speed difference during brake application:

$$n_{dif} = \frac{9.55 \times M'_{Mot\_stat} \times t_2}{J_{Mot} + J_x \times \eta_L \times \eta'_G}$$

25358078219

n_{dif}	=	Speed difference during brake application	$[n_{dif}] = min^{-1}$
M'_{Mot_stat}	=	Static torque of the application as a requirement of the motor, including efficiencies (generator mode)	$[M'_{Mot\_stat}] = Nm$
t_2	=	Brake application time depending on the wiring of the brake: • $t_{2,I}$ = Brake application time for cut-off in the AC circuit • $t_{2,II}$ = Brake application time for cut-off in the DC and AC circuit	$[t_2] = s$
J_{mot}	=	Mass moment of inertia of the motor	$[J_{Mot}] = kg\ m^2$
J_x	=	Mass moment of inertia of the load reduced to the motor shaft	$[J_x] = kg\ m^2$
η_L	=	Load efficiency	$[\eta_L] = 1$
η'_G	=	Retrodriving gear unit efficiency • The following applies to helical-worm and SPIROPLAN® gear units: $\eta'_G = 2\text{-}1/\eta_G$ • The following applies to all other gear units: $\eta'_G = \eta_G$	$[\eta'_G] = 1$

Calculating the brake application speed:

$$n_B = n_{Mot} + n_{dif}$$

25358141067

n_B	= Brake application speed	$[n_B] = min^{-1}$
n_{Mot}	= Operational motor speed	$[n_{Mot}] = min^{-1}$
n_{dif}	= Speed difference during brake application	$[n_{dif}] = min^{-1}$

The calculation steps in the "Braking work" (→ 🗎 109) chapter can be performed using the calculated brake application time. The configuration of the brake can then be continued.

26636735/EN – 07/2020

6.3.4 Service life until inspection

The number of permitted braking operations can be calculated using the previously calculated braking work to be done, W_B, taking into account the permitted braking work until inspection, W_{B_insp}.

$$N_{B\_insp} = \frac{W_{B\_insp}}{W_B}$$

25358144779

N_{B_insp}	=	Number of permitted braking operations until brake inspection. Refer to the information in chapter "Project planning notes" (→ 🗎 14)	$[N_{B\_insp}] = 1$
W_{B_insp}	=	Permitted braking work until brake inspection	$[W_{B\_insp}] = J$
W_B	=	Braking work to be done	$[W_B] = J$

The product characteristic W_{B_insp} can be found in the "Project Planning for BE.. Brakes" manual.

6.3.5 Effects on the gear unit

During mechanical braking, torques and overhung loads act on the gear unit and the application. These forces depend exclusively on the interaction between the application, drive and brake and cannot be influenced by the inverter or controller. These loads can be significantly higher than the operational load and must be calculated after selecting the brake (see chapter "Gear unit load during mechanical braking" (→ 🗎 122)).

6.3.6 Further selection criteria

Additional application requirements can affect the selection of the brake in addition to the dimensioning-related configuration criteria described above. These additional requirements can result from safety considerations, a normative basis or the requirements of the operator. These include the stopping time, the braking speed, the stopping distance and deceleration. Further selection criteria with the required calculation steps are described in more detail in the "Project Planning for BE.. Brakes" manual.

Stopping time

The stopping time t_s is the time from the activation of the mechanical brake to when the drive reaches the idle state. The stopping time t_s is the sum of the brake application time t_2 and the braking time t_B. The duration of the brake application time depends on the type of brake control. Values for the brake application time t_2 can be found in the product documentation.

$$t_s = t_2 + t_B$$

19511459211

t_s	=	Stopping time	$[t_s] = s$
t_2	=	Brake application time	$[t_2] = s$
t_B	=	Braking time	$[t_B] = s$

The braking time t_B is calculated using the equation for dynamic torque. After substituting the relevant application and drive parameters, the braking time can be calculated using the following formulas.

26636735/EN – 07/2020

Braking time (horizontal or rotary direction of movement):

$$t_B = \frac{\left(J_{Mot} + J_x \times \eta_L \times \eta'_G\right) \times n_B}{9.55 \times \left(M_B + M'_{Mot\_stat}\right)}$$

18014418020927883

The braking time in the vertically upward direction of movement cannot be calculated using a compact formula because of the ambiguous direction of the force flow. This time is less than the braking time in the vertically downward direction of movement and is therefore not relevant for the dimensioning of the brake.

Braking time (vertically downward direction of movement):

$$t_B = \frac{\left(J_{Mot} + J_x \times \eta_L \times \eta'_G\right) \times n_B}{9.55 \times \left(M_B - M'_{Mot\_stat}\right)}$$

18014418020929547

t_B	= Braking time	$[t_B]$ = s
J_{mot}	= Mass moment of inertia of the motor	$[J_{Mot}]$ = kg m^2
J_x	= Mass moment of inertia of the load reduced to the motor shaft	$[J_x]$ = kg m^2
η_L	= Load efficiency	$[\eta_L]$ = 1
η'_G	= Retrodriving gear unit efficiency • The following applies to helical-worm and SPIROPLAN® gear units: $\eta'_G = 2-1/\eta_G$ • The following applies to all other gear units: $\eta'_G = \eta_G$	$[\eta'_G]$ = 1
n_B	= Brake application speed	$[n_B]$ = min^{-1}
M_B	= Braking torque	$[M_B]$ = Nm
M'_{Mot_stat}	= Static torque of the application as a requirement of the motor, including efficiencies (generator mode)	$[M'_{Mot\_stat}]$ = Nm

Braking speed

The speed during brake application is calculated from the brake application speed and is required for the subsequent calculation of the stopping distance.

$$v_B = \frac{n_B \times d \times \pi}{i_G \times 60}$$

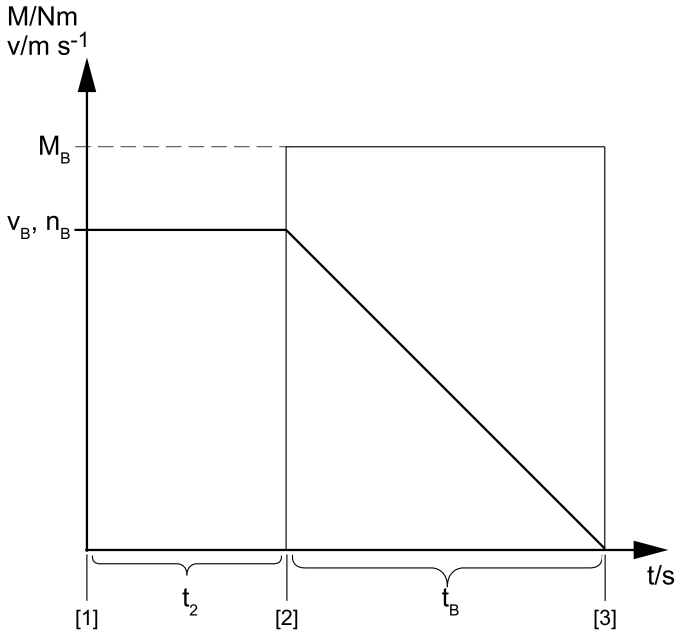
26723404811

v_B	= Speed of application during brake application	$[v_B]$ = m s^{-1}
n_B	= Brake application speed	$[n_B]$ = min^{-1}
d	= Diameter of the mechanical transmission element	[d] = m
i_G	= Gear unit ratio	$[i_G]$ = 1

26636735/EN – 07/2020

Stopping distance

The stopping distance is the distance traveled from the activation of the mechanical brake until the drive reaches the idle state. The stopping distance is the sum of the distance during the brake application time and the actual braking distance.



19511494155

[1] Activation of the brake
[2] Start of braking
[3] Idle state
M Torque
v Speed
M_B Braking torque
v_B Speed of application during brake application
n_B Brake application speed
t_2 Brake application time
t_B Braking time
t Time

The drive continuously moves without restriction during the brake application time. The type of movement depends on the application. For applications in the horizontal direction of movement, this means the drive moves at a constant speed, not taking into account friction. The travel speed of the application is equal to the speed at the time of brake application.

Applications in the vertical direction of movement, such as hoists, continue to move dynamically during the brake application time due to gravitational acceleration. Depending on the direction of movement, the speed increases or decreases at the time of brake application relative to the travel speed of the application.

To summarize all applications, the calculation of the stopping distance during the brake application time is generally simplified using the speed at the time of brake application.

With respect to the actual braking distance, the braking torque immediately after brake application time is constantly effective. This means that the application moves with constant deceleration until reaching the idle state.

26636735/EN – 07/2020

Stopping distance (horizontal direction of movement):

$$s_s = v_B \times t_2 + s_B = v_B \times \left(t_2 + \frac{1}{2} \times t_B \right)$$

32323533579

s_s	= Stopping distance	$[s_s]$ = m
v_B	= Speed of application during brake application	$[v_B]$ = m s^{-1}
t_2	= Brake application time (catalog value)	$[t_2]$ = s
s_B	= Braking distance	$[s_B]$ = m
t_B	= Braking time	$[t_B]$ = s

Deceleration

If required, the deceleration of the application can be calculated from the parameters of braking speed and braking time. In this manner, the load on mechanical transmission elements can be checked. The maximum permitted deceleration may be a requirement for drive selection.

$$a_B = \frac{v_B}{t_B}$$

19511457547

a_B	= Deceleration of the application	$[a_B]$ = m s^{-2}
v_B	= Speed of application during brake application	$[v_B]$ = m s^{-1}
t_B	= Braking time	$[t_B]$ = s

6.4 Calculating and selecting the gear unit

This section covers the following topics:

- Preselecting the gear unit
- Calculating the actual gear unit load
- Options for reducing the gear unit load

6.4.1 Preselecting the gear unit

Service factor

The service factor f_B is a characteristic of the gearmotor and describes the ratio of the continuously permitted output torque of the gear unit to the transmitted rated motor torque. A service factor of f_B = 1.0 means that the gear unit is at 100% capacity when the motor is operated at the rated motor torque. If the service factor is f_B > 1.0, the gear unit has reserves during rated motor operation.

The service factor is calculated as follows:

$$f_B = \frac{M_{a\_max}}{M_N \times i_G}$$

26723515915

f_B	= Service factor	$[f_B]$ = 1
M_{a_max}	= Continuously permitted output torque of the gear unit	$[M_{a\_max}]$ = Nm
M_N	= Rated torque of the motor	$[M_N]$ = Nm
i_G	= Gear unit ratio	$[i_G]$ = 1

Application and motor parameters are used to calculate the minimum requirement for the service factor. This is called the minimum service factor, f_{B_req}. The minimum service factor can be used to estimate the suitability of a gear unit for the application and preselect the gear unit. The minimum service factor is calculated based on the load classification, the switching frequency and the duration of use per day. There are additional influencing factors for helical-worm gear units.

INFORMATION

The determination of service factors is manufacturer-specific rather than standardized and is therefore not comparable.

After preselecting the gear unit based on the service factor, the next step is to calculate the actual gear unit load resulting from the motor acceleration torque or braking torque and the ratio of the application inertia to the motor inertia.

Load classification

The load classification is an auxiliary variable that describes the mass moment of inertia ratio between the application and the motor and takes into account the proportion of motor acceleration torque that places a mechanical load on the gear unit. The mass moment of inertia ratios related to the motor shaft are summarized in 3 different load classifications:

- (I) Uniform load on the gear unit:

  Permitted mass moment of inertia ratio ≤ 0.2
- (II) Non-uniform load on the gear unit:

  Permitted mass moment of inertia ratio ≤ 3
- (III) Highly non-uniform load on the gear unit:

  Permitted mass moment of inertia ratio ≤ 10

The load classification is selected as a function of the mass moment of inertia ratio f_a (also referred to as the mass acceleration factor), which is calculated from the ratio of the application inertia to the motor inertia:

$$f_a = \frac{J_x}{J_{Mot}}$$

18014418020970379

f_a	= Mass moment of inertia ratio	$[f_a] = 1$
J_x	= Mass moment of inertia of the load reduced to the motor shaft	$[J_x]$ = kg m^2
J_{mot}	= Mass moment of inertia of the motor	$[J_{Mot}]$ = kg m^2

Minimum service factor

The minimum service factor f_{B_req} is calculated from the minimum service factor of the application f_{B_L} and the following possible additional factors:

- f_{B1} = Additional factor for ambient temperature (only for helical-worm gear units)
- f_{B2} = Additional factor for cyclic duration factor (only for helical-worm gear units)
- f_{B3} = Additional factor for low temperature range < -30 °C

Calculating the minimum service factor:

$$f_{B\_req} = f_{B\_L} \times f_{B1} \times f_{B2} \times f_{B3}$$

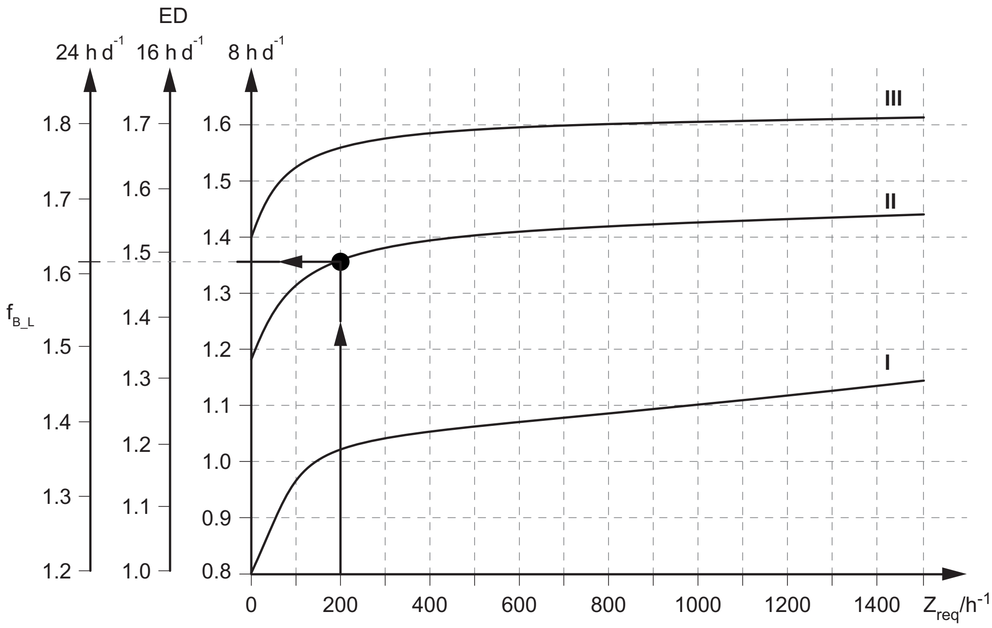
26859984651

f_{B_req}	= Minimum service factor	$[f_{B\_req}] = 1$
f_{B_L}	= Minimum service factor of the application	$[f_{B\_L}] = 1$
f_{B1}	= Additional factor for ambient temperature (only for helical-worm gear units)	$[f_{B1}] = 1$
f_{B2}	= Additional factor for cyclic duration factor (only for helical-worm gear units)	$[f_{B2}] = 1$
f_{B3}	= Additional factor for low temperature range	$[f_{B3}] = 1$

If the conditions described above are met, the additional factors f_{B1} to f_{B3} are taken into account.

Minimum service factor of the application

The following diagram shows the minimum service factor f_{B_L} based on the load classification, switching frequency and the duration of use per day.



18014418020982155

ED	= Relative cyclic duration factor per day	$[ED] = h\ d^{-1}$
f_{B_L}	= Minimum service factor of the application	$[f_{B\_L}] = 1$
Z_{req}	= Required switching frequency	$[Z_{req}] = h^{-1}$

Example:

An application with switching frequency of $Z_{req} = 200\ h^{-1}$ and load classification II requires:

- A minimum service factor of the application $f_{B\_L} \geq 1.35$ when the cyclic duration factor ED = 8 h d^{-1}
- A minimum service factor of the application $f_{B\_L} \geq 1.62$ when the cyclic duration factor ED = 24 h d^{-1}

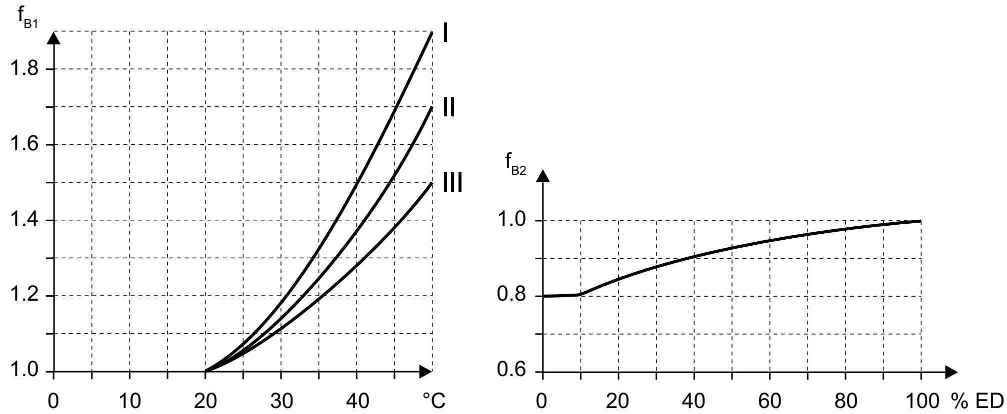
26636735/EN – 07/2020

Additional factor for low temperature range

At ambient temperatures <-30 °C, an additional factor f_{B3} = 1.2 must be observed in order to consider the low-temperature properties of the materials used for the housing and flange.

Additional factors for helical-worm gear units

The additional factors f_{B1} and f_{B2} are determined for helical-worm gear units in order to consider the properties of the gearing and lubricant



28273047307

For temperatures outside the temperature range shown here, consult SEW-EURODRIVE.

The relative cyclic duration factor in the diagram on the right is calculated from the ratio of time under load (all travel sections where motor speed ≠ 0) to total time. If the time under load ≥ 1 hour, the cyclic duration factor ED_G is 100%.

Relative cyclic duration factor for helical-worm gear unit:

$$ED_G = \frac{\sum_{n=gen} t_n}{t_{tot}} \times 100$$

29008688651

ED_G	= Relative cyclic duration factor for helical-worm gear unit	$[ED_G]$ = %
t_n	= Time in travel section n where motor speed ≠ 0	$[t_n]$ = s
t_{tot}	= Total time of the travel cycle	$[t_{tot}]$ = s

Selecting the gear unit

A suitable gear unit can be selected as follows from the "Gearmotors" catalog.

1. Choose selection table according to rated motor power.
2. Select output speed or gear unit ratio close to the calculated ideal value.
3. Read associated service factor f_B.
4. Check service factor f_B against minimum service factor f_{B_req}:
   - $f_B \geq f_{B\_req}$: Permitted
   - $f_B < f_{B\_req}$: Not permitted
5. Select the gear unit size associated with the service factor:
   - $f_B \geq f_{B\_req}$: Smallest possible gear unit size
   - $f_B >> f_{B\_req}$: Selected gear unit size can be reduced.

With this procedure, several gear unit sizes are possible for a rated motor power and a minimum service factor.

Example:

[1] $P_m = 3.0\,kW$

n_a min^{-1}	M_a Nm	i	F_{Ra} 1) N	SEW f_B					m kg	
62	460	23.44	8660	1.20						
73	390	19.89	8350	1.55						
81	350	17.95	8150	1.65	R	67	DRN	100L4	58	342
[2] 92	310	15.79	7900	1.80	RF	67	DRN	100L4	61	343
98	290	14.91	7790	1.85	RM	67	DRN	100L4	77	343
115	245	12.70	7470	2.1						
126	225	11.54	7290	2.2						
146	197	10.00	7010	2.4						

28273258123

[1] Rated motor power: 3 kW
[2] Required output speed: n_a = 90 min^{-1} or ideal gear unit ratio: i_{G_id} = 15
[3] Minimum service factor: f_{B_req} = 1.6
[5] Selected gear unit size: R67

6.4.2 Calculating the actual gear unit load

As an alternative to gear unit selection according to the service factor, the actual gear unit load in the individual travel sections can be calculated. The procedure is similar to calculating and selecting the gear unit for controlled drives. The decisive factor in gear unit selection is usually the gear unit load during motor startup or mechanical braking.

Gear unit load during motor startup

The gear unit load for a motor connected to the supply system is the result of the acceleration torque minus the dynamic torque for the intrinsic acceleration of the motor (see figure in chapter "Checking motor start-up" (→ 🗎 97)).

Calculating the output torque during motor startup

The torque load on the gear unit output during motor startup is comprised of the static and dynamic component of the application. The intrinsic acceleration torque of the motor does not contribute to the gear unit load.

Upward vertical, horizontal or rotary direction of movement:

$$M_{G\_H} = \left(M_{Mot\_stat} + \left(M_H - M_{Mot\_stat} \right) \times \frac{\frac{J_x}{\eta_L \times \eta_G}}{J_{Mot} + \frac{J_x}{\eta_L \times \eta_G}} \right) \times i_G \times \eta_G$$

18014418020998539

M_{G_H}	=	Output torque during motor startup	$[M_{G\_H}]$ = Nm
M_{Mot_stat}	=	Static torque of the application as a requirement of the motor, including efficiencies (motor mode)	$[M_{Mot\_stat}]$ = Nm
M_H	=	Acceleration torque of the motor	$[M_H]$ = Nm
J_x	=	Mass moment of inertia of the load reduced to the motor shaft	$[J_x]$ = kg m^2
η_L	=	Load efficiency	$[\eta_L]$ = 1
η_G	=	Gear unit efficiency	$[\eta_G]$ = 1
J_{mot}	=	Mass moment of inertia of the motor	$[J_{Mot}]$ = kg m^2
i_G	=	Gear unit ratio	$[i_G]$ = 1

The gear unit load during motor startup for applications in the vertically downward direction of movement in the "acceleration" travel section is smaller than in the vertically upward direction of movement in the "acceleration" travel section. This load condition is therefore not calculated.

Checking the output torque load at motor startup

Subsequently, a gear unit is selected whose continuously permitted output torque of the gear unit, M_{a_max}, is above the previously calculated output torque at motor startup.

$$M_{G\_H} \leq M_{a\_max}$$

18014418021003531

M_{G_H}	=	Output torque during motor startup	$[M_{G\_H}]$ = Nm
M_{a_max}	=	Continuously permitted output torque of the gear unit	$[M_{a\_max}]$ = Nm

Calculating the overhung load to be absorbed during motor startup

The overhung load to be absorbed during motor startup must be additionally checked for applications with an overhung load acting on the gear unit. This is calculated from the torque, the diameter of the transmission element and the transmission element factor (e.g., for initial belt tension).

$$F_{R\_H} = \frac{M_{G\_H} \times 2000}{d} \times f_z$$

26666400267

F_{R_H}	=	Overhung load to be absorbed on gear unit output during motor startup	$[F_{R\_H}]$ = N
M_{G_H}	=	Output torque during motor startup	$[M_{G\_H}]$ = Nm
d	=	Diameter of the mechanical transmission element	[d] = mm
f_z	=	Transmission element factor	$[f_Z]$ = 1

Checking the overhung load during motor startup

The overhung load F_{R_H} to be absorbed during motor startup is compared with the permitted overhung load of the gear unit F_{R_per}.

$$F_{R\_H} \leq F_{R\_per}$$

26666560395

F_{R_H}	=	Overhung load to be absorbed on gear unit output during motor startup	$[F_{R\_H}]$ = N
F_{R_per}	=	Permitted gear unit overhung load	$[F_{R\_per}]$ = N

Actions must be taken if the mathematically calculated gear unit load during motor start up is greater than the product characteristics of the gear unit. For example, if a larger gear unit is selected, all tests such as feasibility, etc., must be carried out again. If a larger gear unit cannot be built, different actions can be taken. These actions are documented in chapter "Options for reducing the gear unit load" (→ 🗎 124).

Gear unit load during mechanical braking

During mechanical braking, torques and overhung loads act on the gear unit and the application. These forces depend exclusively on the interaction between the application, drive and brake.

Calculating the output torque during mechanical braking

The torque load during mechanical braking is calculated from the braking torque minus the dynamic torque for the intrinsic acceleration of the motor and can be derived from the formula for the dynamic torque. Substituting all relevant application and drive variables results in the following relationship results.

Horizontal or rotary direction of movement:

$$M_{G\_B} = \frac{i_G}{\eta'_G} \left(\left(M_B + M'_{Mot\_stat}\right) \times \frac{\frac{J_x \times \eta_L \times \eta'_G}{J_{Mot}}}{\frac{J_x \times \eta_L \times \eta'_G}{J_{Mot}} + 1} - M'_{Mot\_stat} \right)$$

26664747531

The torque load during mechanical braking in the vertically upward direction of movement is smaller than in the vertically downward direction of movement. As a result, it is not calculated here.

Vertically downward direction of movement:

$$M_{G\_B} = \frac{i_G}{\eta'_G}\left(\left(M_B - M'_{Mot\_stat}\right) \times \frac{\frac{J_x \times \eta_L \times \eta'_G}{J_{Mot}}}{\frac{J_x \times \eta_L \times \eta'_G}{J_{Mot}} + 1} + M'_{Mot\_stat} \right)$$

26666392459

M_{G_B}	=	Output torque during mechanical braking	$[M_{G\_B}]$ = Nm
i_G	=	Gear unit ratio	$[i_G]$ = 1
η'_G	=	Retrodriving gear unit efficiency • The following applies to helical-worm and SPIROPLAN® gear units: $\eta'_G = 2-1/\eta_G$ • The following applies to all other gear units: $\eta'_G = \eta_G$	$[\eta'_G]$ = 1
M_B	=	Braking torque	$[M_B]$ = Nm
M'_{Mot_stat}	=	Static torque of the application as a requirement of the motor, including efficiencies (generator mode)	$[M'_{Mot\_stat}]$ = Nm
J_x	=	Mass moment of inertia of the load reduced to the motor shaft	$[J_x]$ = kg m^2
η_L	=	Load efficiency	$[\eta_L]$ = 1
J_{mot}	=	Mass moment of inertia of the motor	$[J_{Mot}]$ = kg m^2

Checking the output torque during mechanical braking

The output torque during mechanical braking M_{G_B} is compared to the continuously permitted output torque of the gear unit M_{a_max}. In contrast to project planning for controlled drives, no increased characteristic values are permitted for the gear units of non-controlled drives because the drive is always mechanically braked during operation.

$$M_{G\_B} \le M_{a\_max}$$

26645352331

M_{G_B}	=	Output torque during mechanical braking	$[M_{G\_B}]$ = Nm
M_{a_max}	=	Continuously permitted output torque of the gear unit	$[M_{a\_max}]$ = Nm

Calculating the overhung load to be absorbed during mechanical braking

The overhung load to be absorbed during braking must be additionally checked for applications with an overhung load acting on the gear unit. This is calculated from the torque, the diameter of the transmission element and the transmission element factor (e.g., for initial belt tension).

$$F_{R\_B} = \frac{M_{G\_B} \times 2000}{d} \times f_z$$

26874760203

F_{R_B}	=	Overhung load to be absorbed on gear unit output during mechanical braking	$[F_{R\_B}]$ = N
M_{G_B}	=	Output torque during mechanical braking	$[M_{G\_B}]$ = Nm
d	=	Diameter of the mechanical transmission element	[d] = mm
f_z	=	Transmission element factor	$[f_Z]$ = 1

Checking the overhung load during mechanical braking

The overhung load F_{R_B} to be absorbed during mechanical braking is compared with the permitted overhung load of the gear unit F_{R_per}. In contrast to project planning for controlled drives, no increased characteristic values are permitted for the gear units of non-controlled drives because the drive is always mechanically braked during operation.

$$F_{R\_B} \leq F_{R\_per}$$

26874763787

F_{R_B}	=	Overhung load to be absorbed on gear unit output during mechanical braking	$[F_{R\_B}]$ = N
F_{R_per}	=	Permitted overhung load on gear unit output	$[F_{R\_per}]$ = N

If the mathematically calculated gear unit load is greater than the product characteristics of the gear unit during normal braking, actions must be taken, such as selecting a lower braking torque or a larger gear unit.

In travel applications, the braking distance increases as the braking torque decreases. For lifting applications, limits are imposed here by the required stop function (see "Special requirement for lifting applications" (→ 🗎 108)). If a larger gear unit is selected, all tests such as feasibility, etc., must be carried out again. If a larger gear unit cannot be built, different actions can be taken. These actions are documented in the "Project Planning for BE.. Brakes" manual.

6.4.3 Options for reducing the gear unit load

If the gear unit load is too high for a desired gear unit size during motor startup or mechanical braking, measures to reduce the gear unit load may be taken.

- Increase the mass moment of inertia of the motor (e.g., flywheel fan).
- Select a motor with different number of pole pairs and adjust the gear unit ratio.
- Startup in a 人/△ connection (observe rated motor voltage!).
- Soft-start units.
- Select a controlled drive.

After taking one or more actions, the gear unit load must be rechecked.

26636735/EN – 07/2020

6.5 Information about pole-changing motors

A pole-changing motor is a special form of AC asynchronous motor. The motor can be operated at 2 or more different rated speeds by choosing suitable electrical configuration and design of the stator winding.

The motor can be switching between 2 rated speeds in idle state or during operation. An external switching device (e.g. polarity-changing switch) switches the line voltage to the appropriate winding phases.

If the motor is switched to a different rated speed during operation, the motor and thus the application accelerates or decelerates.

The motor acts as a motor during acceleration and as a generator during deceleration. The resulting switching torques are divided across the motor and application, depending on the mass moment of inertia. The proportion of the switching torque that accelerates or decelerates the application must be transmitted by the mechanical components of the drive train and must be taken into account during dimensioning.

Non-controlled drives can be positioned more accurately through polarity changing during operation.

The motor is initially operated at a high rated speed (rapid speed). When the required stop position is approached, the motor switches to a lower rated speed (creep speed) at which the motor is mechanically braked. This utilizes the effect of the motor speed on the stopping accuracy after mechanical braking. The lower the motor speed at which a mechanical braking operation begins, the shorter the braking distance and the less the stopping position reached deviates from the required stopping position.

The procedure for configuring pole-changing motors differs from the procedure for non-controlled drives. If necessary, contact SEW-EURODRIVE.

26636735/EN – 07/2020

7 Table appendix

7.1 Efficiencies of transmission elements

Transmission element	Conditions	Efficiency
Wire rope	Per complete loop of the rope roll (with friction or plain bearing)	0.91 – 0.95
V-belt	Per complete loop of the V-belt pulley (normal initial belt tension)	0.88 – 0.93
Plastic bands	Per complete loop / rollers with friction bearings (normal tension of the band)	0.81 – 0.85
Rubber bands	Per complete loop / rollers with friction bearings (normal tension of the band)	0.81 – 0.85
Toothed belt	Per complete loop / rollers with friction bearings (normal tension of the band)	0.90 – 0.96
Chains	Per complete loop / wheels with friction bearing (depending on chain size)	0.90 – 0.96
Gear unit	Oil lubrication: • 3-stage (helical gears), depending on gear unit quality • For helical-worm and bevel-helical gear units	 0.94 – 0.97 According to manufacturer specifications

7.2 Transmission element factor f_Z of various transmission elements for calculating the overhung load

Transmission element	Transmission element factor f_Z	Comments
Gear wheel	1.15	< 17 teeth
Sprocket	1.40	< 13 teeth
Sprocket	1.25	< 20 teeth
Narrow V-belt pulley washer	1.75	Influence of pretensioning
Flat belt pulley	2.50	Influence of pretensioning
Toothed belt pulley	1.50	Influence of pretensioning
Gear rack pinion, not pretensioned	1.15	< 17 teeth
Gear rack pinion, pretensioned	2.00	Influence of pretensioning

26636735/EN – 07/2020

7.3 Spindle efficiencies

Spindle	Efficiency
Trapezoidal thread depending on pitch and lubrication	0.3 – 0.5
Recirculating ball screw	0.8 – 0.9

7.4 Friction coefficients for different material combinations

Material combination	Type of friction	Friction coefficient
Steel on steel	Static friction (dry)	μ_{f_st} = 0.12 – 0.60
	Sliding friction (dry)	μ = 0.08 – 0.50
	Static friction (lubricated)	μ_{f_st} = 0.12 – 0.35
	Sliding friction (lubricated)	μ = 0.04 – 0.25
Wood on steel	Static friction (dry)	μ_{f_st} = 0.45 – 0.75
	Sliding friction (dry)	μ = 0.30 – 0.60
Wood on wood	Static friction (dry)	μ_{f_st} = 0.40 – 0.75
	Sliding friction (dry)	μ = 0.30 – 0.50
Plastic belts on steel	Static friction (dry)	μ_{f_st} = 0.25 – 0.45
	Sliding friction (dry)	μ = 0.25
Steel on plastic	Static friction (dry)	μ_{f_st} = 0.20 – 0.45
	Sliding friction (lubricated)	μ = 0.18 – 0.35

7.5 Bearing friction coefficients

Bearing	Friction coefficient
Rolling bearing	μ_{f_b} = 0.005
Sleeve bearing	μ_{f_b}= 0.09

7.6 Coefficients for track and lateral friction

Track and side friction	Coefficient
Wheels with friction bearing	c = 0.003
Wheels with sleeve bearing	c = 0.005
Lateral guide rollers	c = 0.002

26636735/EN – 07/2020

7.7 Rolling friction (lever arm of rolling friction)

<table>
<tr><th colspan="2">Combination</th><th colspan="2">Lever arm</th></tr>
<tr><td colspan="2">Steel on steel</td><td colspan="2">f ≈ 0.5 mm</td></tr>
<tr><td colspan="2">Wood on steel (roller conveyor)</td><td colspan="2">f ≈ 1.2 mm</td></tr>
<tr><td colspan="2">Plastic on steel</td><td colspan="2">f ≈ 2 mm</td></tr>
<tr><td colspan="2">Hard rubber on steel</td><td colspan="2">f ≈ 7 mm</td></tr>
<tr><td colspan="2">Plastic on concrete</td><td colspan="2">f ≈ 5 mm</td></tr>
<tr><td colspan="2">Hard rubber on concrete</td><td colspan="2">f ≈ 10 – 20 mm</td></tr>
<tr><td colspan="2">Medium-hard rubber on concrete</td><td colspan="2">f ≈ 15 – 35 mm</td></tr>
<tr><td rowspan="4">Vulkollan on steel</td><td>Ø 100 mm</td><td>f ≈ 0.75 mm</td><td rowspan="4">Notice! Lever arm of rolling friction is heavily dependent on manufacturer, geometry and temperature.</td></tr>
<tr><td>Ø 125 mm</td><td>f ≈ 0.9 mm</td></tr>
<tr><td>Ø 200 mm</td><td>f ≈ 1.5 mm</td></tr>
<tr><td>Ø 415 mm</td><td>f ≈ 3.1 mm</td></tr>
</table>

8 Explanation of abbreviations

The meaning of the formula symbols is listed in alphabetical order in the legend.

Abb.	Meaning	Unit
a	Acceleration	m s^{-2}
a_H	Startup acceleration	m s^{-2}
a_{max}	Maximum possible acceleration	m s^{-2}
c	Track friction coefficient	1
d	Diameter of the transmission element	m
d_b	Bearing diameter	mm
ED_{BW}	Regenerative cyclic duration factor	%
f	Frequency	Hz
f	Lever arm of the rolling friction	mm
f_a	Mass moment of inertia ratio	1
f_B	Service factor	1
f_{B1}	Additional factor for ambient temperature (only for helical-worm gear units)	1
f_{B2}	Additional factor for cyclic duration factor (only for helical-worm gear units)	1
f_{B3}	Additional factor for low temperature range	1
f_{B_L}	Minimum service factor of the application	1
f_{B_req}	Minimum service factor	1
f_{ol}	Overload factor of the frequency inverter (e.g., 1.5 with overload capacity of 150%)	1
f_W	Wear factor; determination in relation to the used load range for braking work	1
f_Z	Transmission element factor	1
F	Force	N
F_A	Axial load on gear unit output	N
F_{A_per}	Permitted axial load (pull or push)	N
F_{dyn}	Dynamic force (acceleration or deceleration force)	N
F_f	Friction force	N
F_{f_b}	Bearing friction force	N
F_{f_r}	Rolling friction force	N
F_{f_st}	Static friction force	N
F_{f_t}	Track friction force	N
F_G	Gravitational force	N
F_H	Gravity resistance	N
F_N	Normal force	N
F_R	Overhung load on gear unit output	N

26636735/EN – 07/2020

Abb.	Meaning	Unit
F_{R_B}	Overhung load to be absorbed on gear unit output during mechanical braking	N
F_{R_es}	Overhung load absorbed on gear unit output during emergency stop braking	N
F_{R_H}	Overhung load to be absorbed on gear unit output during motor startup	N
F_{R_per}	Permitted overhung load on gear unit output at distance l/2 to shaft shoulder	N
$F_{R\_per\_es}$	Permitted overhung load on gear unit output during emergency stop braking	N
$F_{R\_x\_per}$	Permitted overhung load at distance x to shaft shoulder	N
F_{stat}	Static force	N
F_{tot}	Total force	N
F_{tr}	Force of resistance to vehicle motion	N
F_{η}	Force to be applied as a function of the efficiency (motor mode)	N
F'_{η}	Force to be absorbed as a function of the efficiency (regenerative)	N
g	Gravitational acceleration	$m\ s^{-2}$
I_A	Starting current	A
I_d	Magnetizing current	A
I_{eff}	Effectively required motor current	A
i_G	Gear unit ratio	1
i_{G_id}	Calculated ideal gear unit ratio	1
I_{max}	Maximum required motor current	A
I_{Mot}	Motor current (outer conductor current)	A
I_N	Rated current of the motor	A
I_{N_FU}	Rated output current of the frequency inverter	A
I_q	Torque-generating current	A
I_{q_N}	Rated value of the torque-generating current	A
i_{tot}	Total gear ratio between application and motor	1
i_V	Additional transmission ratio	1
J	Mass moment of inertia	$kg\ m^2$
J_{BMot}	Massenträgheitsmoment des Bremsmotors	$kg\ m^2$
J_{cg}	Mass moment of inertia of an object with reference to a rotary axis through the center of gravity S	$kg\ m^2$
J_{Mot}	Mass moment of inertia of the motor	$kg\ m^2$
J_x	Mass moment of inertia of the load reduced to the motor shaft	$kg\ m^2$
J_Z	Mass moment of inertia of heavy additional mass/Z	$kg\ m^2$

26636735/EN – 07/2020

Abb.	Meaning	Unit
K	Power increase factor for the different operating modes	1
K_J	Calculation factor for mass moment of inertia	h^{-1}
K_M	Calculation factor for static motor torque	h^{-1}
K_P	Calculation factor for static power and cyclic duration factor	h^{-1}
l	Length	m
m	Mass	kg
m_{tot}	Total mass of the application	kg
M	Torque	Nm
M_{a_max}	Continuously permitted output torque of the gear unit	Nm
M_A	Starting torque	Nm
M_B	Braking torque	Nm
M_{dyn}	Dynamic torque/acceleration torque	Nm
M_{f_b}	Bearing friction torque	Nm
M_{f_r}	Rolling friction torque	Nm
M_{f_t}	Track friction torque	Nm
M_{G_B}	Output torque during mechanical braking	Nm
M_{G_es}	Output torque during emergency stop braking	Nm
M_{G_H}	Output torque during motor startup	Nm
M_{G_max}	Maximum torque of the gear unit output, including load efficiency, across all travel sections	Nm
M_{G_n}	Torque of gear unit output in travel section n (e.g., acceleration) including load efficiency (motor mode)	Nm
$M_{G\_per\_es}$	Permitted output torque during emergency stop braking	Nm
M_H	Acceleration torque	Nm
M_K	Breakdown torque	Nm
M_L	Input torque of the application	Nm
M_{Mot}	Motor torque required by the application	Nm
M_{Mot_ac}	Torque of the application in the “acceleration” travel section as a requirement of the motor, including efficiencies (motor mode)	Nm
$M_{Mot\_ac\_tot}$	Total torque of the application including the intrinsic acceleration of the motor in the "acceleration" travel section as a requirement of the motor, including efficiencies (motor mode)	Nm
M_{Mot_const}	Torque of the application in the “constant speed” travel section as a requirement of the motor, including efficiencies (motor mode)	Nm
M'_{Mot_const}	Torque of the application in the “constant speed” travel section as a requirement of the motor, including efficiencies (generator mode)	Nm

Abb.	Meaning	Unit
M_{Mot_dec}	Torque of the application in the “deceleration” travel section as a requirement of the motor, including efficiencies (motor mode)	Nm
$M_{Mot\_dec\_tot}$	Total torque of the application including the intrinsic deceleration of the motor in the "deceleration" travel section as a requirement of the motor, including efficiencies (motor mode)	Nm
M_{Mot_eff}	Motor rms torque	Nm
M_{Mot_iac}	Dynamic torque for intrinsic acceleration or deceleration of the motor	Nm
M_{Mot_n}	Torque of the application as a requirement of the motor in travel section n, including load efficiency (motor mode)	Nm
$M_{Mot\_n\_tot}$	Total torque of the application including the intrinsic acceleration of the motor in the travel section n as a requirement of the motor	Nm
M_{Mot_stat}	Static torque of the application as a requirement of the motor, including efficiencies (motor mode)	Nm
M_n	Application-side torque without load efficiency in the travel section n	Nm
M_N	Rated torque of the motor	Nm
M_S	Pull-up torque	Nm
M_{stat}	Static torque of the application	Nm
M_{tr}	Torque of resistance to vehicle motion	Nm
M_η	Torque to be applied as a function of the efficiency (motor mode)	Nm
M'_{G_n}	Torque of gear unit output in travel section n (e.g., deceleration) including load efficiency (generator mode)	Nm
M'_{Mot_dec}	Torque of the application in the “deceleration” travel section as a requirement of the motor, including efficiencies (generator mode)	Nm
$M'_{Mot\_dec\_tot}$	Total torque of the application including the intrinsic deceleration of the motor in the "deceleration" travel section as a requirement of the motor, including efficiencies (generator mode)	Nm
M'_{Mot_n}	Torque of the application as a requirement of the motor in travel section n, including load efficiency (generator mode)	Nm
M'_{Mot_stat}	Static torque of the application as a requirement of the motor, including efficiencies (generator mode)	Nm
M'_η	Torque to be absorbed as a function of the efficiency (generator mode)	Nm
n	Speed	min^{-1}
n_a	Output speed	min^{-1}
n_B	Brake application speed	min^{-1}

26636735/EN – 07/2020

Abb.	Meaning	Unit
n_{B_es}	Brake application speed in the event of an emergency stop	min^{-1}
N_{B_insp}	Number of permitted emergency stop braking operations until brake inspection	1
n_{dif}	Speed difference during brake application	min^{-1}
n_G	Output speed of the gear unit	min^{-1}
n_L	Rotational speed of the application	min^{-1}
n_{max}	Maximum speed	min^{-1}
n_{Mot}	Motor speed	min^{-1}
n_{Mot_max}	Maximum motor speed	min^{-1}
$\overline{n}_{Mot\_n}$	Mean motor speed in travel section n	min^{-1}
n_N	Rated speed	min^{-1}
n_{syn}	Synchronous speed	min^{-1}
N_{tot}	Total number of wheels	1
n_V	Output speed of the additional transmission	min^{-1}
p	Spindle pitch	mm
P	Power	kW
P_1	Power supplied	W
P_2	Output or available power	W
P_{dyn}	Dynamic power	kW
$\overline{P}_{gen}$	Mean regenerative braking power	kW
$\overline{P}_{gen\_n}$	Mean braking power in the regenerative travel section n	kW
P_{gen_pk}	Peak braking power	kW
P_H	Available motor power during startup	kW
P_{max}	Maximum power	kW
P_{Mot_max}	Maximum power of the application as a requirement of the motor, including efficiencies (motor mode)	kW
P_{Mot_stat}	Static power of the application as a requirement of the motor, including efficiencies (motor mode)	kW
P_N	Rated power of the motor	kW
P_{stat}	Static power	kW
r	Radius	m
$r_{1,2}$	Inner and outer radius	m
r_b	Bearing radius	mm
R_{BW}	Resistance value of the braking resistor	Ω
R_{BW_max}	Maximum resistance value of the braking resistor depending on the application	Ω
s	Distance	m
s_B	Braking distance	m

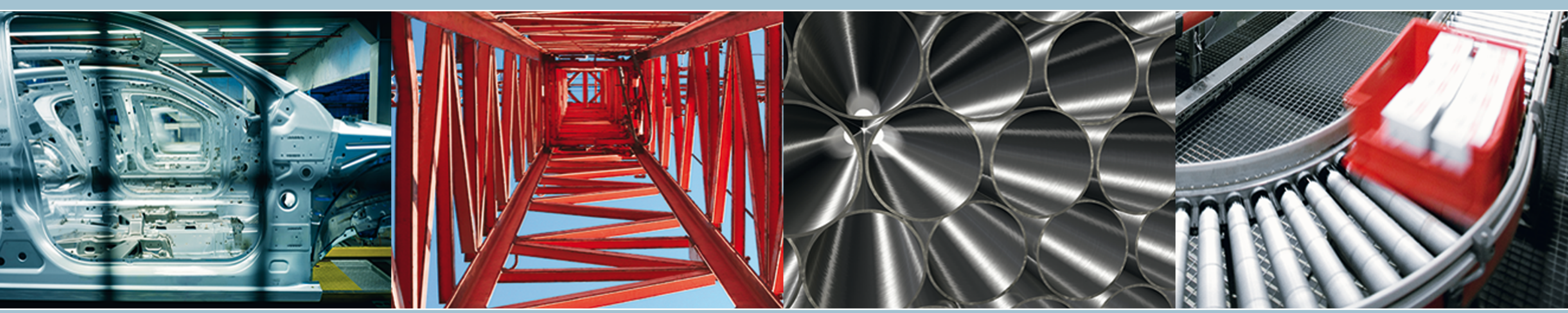
26636735/EN – 07/2020

Abb.	Meaning	Unit
s_c	Anhalteweg ohne Bremse	m
s_{tot}	Total track	m
t	Time	s
T	Periodic time	s
$t_{2,I}$	Brake application time for cut-off in the AC circuit	s
$t_{2,II}$	Brake application time for cut-off in the DC and AC circuit	s
t_{ac}	Acceleration time in the “acceleration” travel section	s
t_B	Braking time	s
t_C	Anhaltezeit ohne Bremse	s
t_H	Run-up time	s
t_n	Duration of travel section n	s
t_s	Stopping time	s
t_{tot}	Total time	s
U	Spindle circumference	mm
U_{DCL}	Voltage threshold in the DC link at which the brake chopper is activated	V
U_{Mot}	Motor voltage (phase-to-phase)	V
v	Speed	$m\ s^{-1}$
v_B	Speed of application during brake application	$m\ s^{-1}$
W_B	Braking work to be done	J
W_{B_es}	Braking work to be done in the event of an emergency stop	J
W_{B_insp}	Permitted braking work until brake inspection	J
W_{B_per}	Permitted braking work as a function of the brake size and application speed	J
$W_{B\_per\_es}$	Permitted braking work in the event of an emergency stop	J
x	Distance	m
Z_0	No-load starting frequency	h^{-1}
Z_{per}	Permitted switching frequency	h^{-1}
Z_{req}	Required switching frequency	h^{-1}
α	Angular acceleration	s^{-2}
β	Angle of inclination to the horizontal plane	° oder rad
γ	Angle of friction	° oder rad
η	Efficiency	1
η_L	Load efficiency	1
η_{spi}	Spindle efficiency	1
η_G	Getriebewirkungsgrad	1
η'_G	Retrodriving gear unit efficiency	1

Abb.	Meaning	Unit
μ	Friction coefficient	1
μ_{f_b}	Bearing friction coefficient	1
μ_{f_r}	Rollreibungskoeffizient	1
μ_{f_st}	Static friction coefficient	1
μ_{tr}	Total friction coefficient of the resistance to vehicle motion	1
φ	Angle	rad oder °
ω	Angular speed	s^{-1}
ω_G	Angular speed of the gear unit output	s^{-1}
ω_{Mot}	Angular speed of the motor	s^{-1}

26636735/EN – 07/2020

Index



SEW-EURODRIVE
Driving the world

SEW-EURODRIVE GmbH & Co KG
Ernst-Blickle-Str. 42
76646 BRUCHSAL
GERMANY
Tel. +49 7251 75-0
Fax +49 7251 75-1970
sew@sew-eurodrive.com
→ www.sew-eurodrive.com