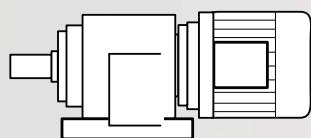
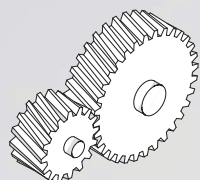


Standard Drives Engineering Manual



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We do not provide a warranty for any of the information contained in this document.

The examples contained herein are non-binding and do not in any way claim to be complete. They do not represent any specific solutions and are only intended to offer assistance for typical tasks.

Siemens cannot assume liability for recommendations that appear or are implied in this manual either. The relevant technical descriptions must always be observed in order to ensure that the products are used correctly and as prescribed.

Standard Drives Engineering Manual

A5E02252473B AB

10/2008 Edition

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Safety information

This manual contains information that you should observe in order to ensure your own personal safety as well as to protect the product and connected equipment from material damage. The information referring to your personal safety is highlighted in the manual by a safety alert symbol; information referring to material damage only has no safety alert symbol. The notices shown below are graded according to the level of danger (from most to least dangerous):



Danger

Indicates that death or serious injury **will** result if proper precautions are not taken.



Warning

Indicates that death or serious injury **may** result if proper precautions are not taken.



Caution

With a safety alert symbol, indicates that minor injury **may** result if proper precautions are not taken.

Caution

Without a safety alert symbol, indicates that material damage may result if proper precautions are not taken.

Notice

Indicates that an undesirable result or state may occur if the relevant instructions are not observed.

If more than one level of danger exists, the warning for the highest level of danger is always used. A warning with a safety alert symbol indicating possible personal injury may also include a warning relating to material damage.

Qualified personnel

The associated device/system may only be installed and operated in conjunction with this documentation. The device/system may only be commissioned and operated by **qualified personnel**. For the purpose of the safety information in this manual, "qualified personnel" are those authorized to commission, ground, and tag devices, systems, and circuits in accordance with established safety engineering standards.

Use as prescribed

Please note the following:



Warning

The device may be used only for the applications described in the catalog or the technical description, and only in combination with equipment, components, and devices supplied by other -manufacturers where recommended or permitted by Siemens.

This product can only function correctly and safely if it is transported, stored, set up, and installed correctly, and operated and maintained as recommended.

Trademarks

All names shown with the trademark symbol ® are registered trademarks of Siemens AG. If third parties use any other names in this document which refer to trademarks for their own purposes, this might infringe upon the rights of the trademark owners.

Disclaimer of liability

We have checked the contents of this publication for consistency with the hardware and software described. However, since variance cannot be precluded entirely, we cannot guarantee full consistency. Having said that, the information in this publication is reviewed regularly and any necessary corrections are included in subsequent editions.

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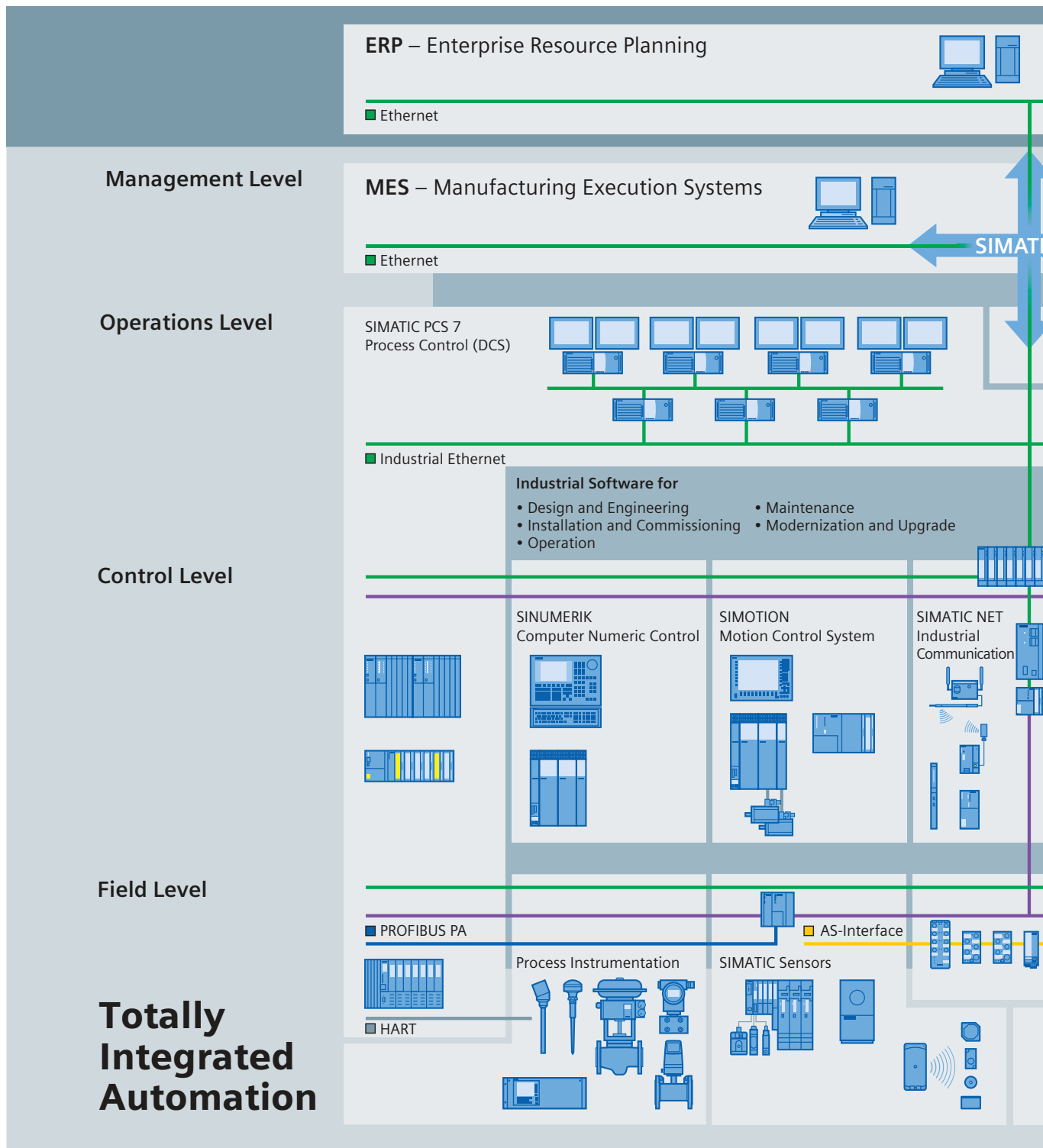
Introduction

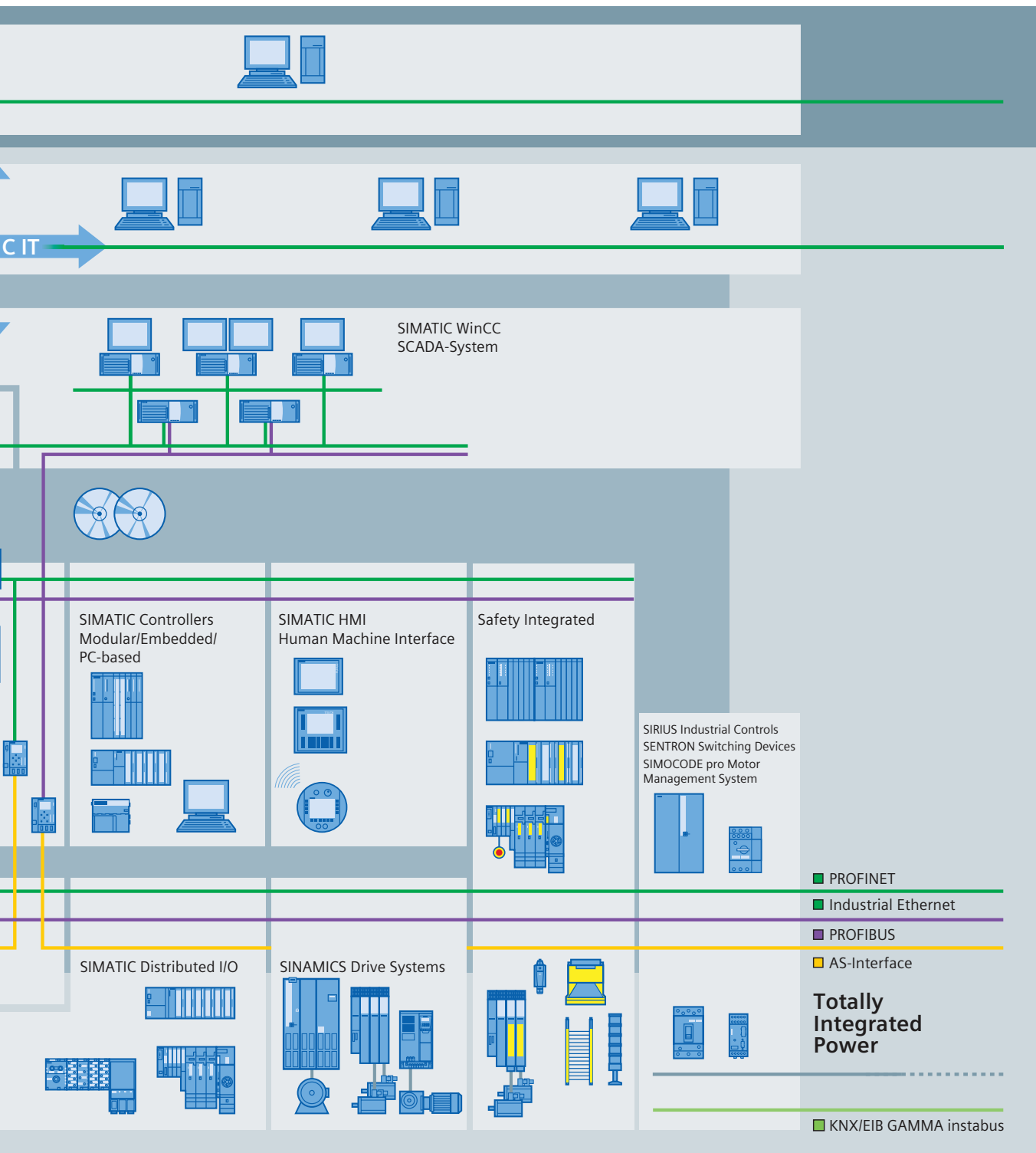
Siemens Industry Automation and Drive Technologies provides comprehensive solutions for the manufacturing and process industries. Totally Integrated Automation is a platform for integrated drive and automation solutions and is only offered by Siemens Industry Automation and Drive Technologies.

Siemens geared motors, low-voltage motors, and frequency converters enable individual solutions to be developed for a wide range of drive technology tasks.

This manual uses examples to describe the procedures to be followed for various drive tasks. The examples illustrate how the torque and motor power are calculated from the drive's mechanical data and how gearboxes, motors, and frequency converters are selected.

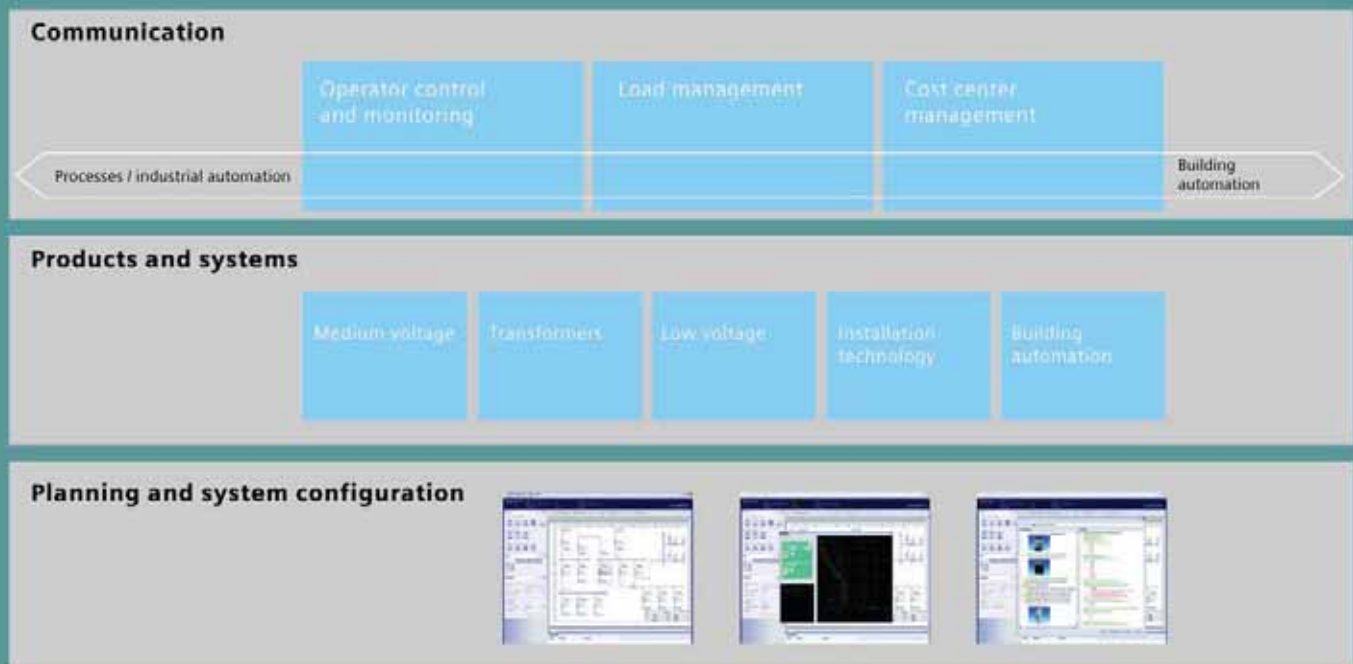
The examples only deal with the basic aspects of each case; the specific requirements and boundary conditions associated with the application in question always need to be taken into account too.





With Totally Integrated Power, Siemens offers integrated solutions for power distribution in non-residential and industrial buildings from the medium-voltage line through to the socket outlet.

Totally Integrated Power is based on integrated planning and configuration procedures, as well as on coordinated products and systems. It offers communication and software modules for connecting the power distribution systems to the industrial and building automation systems, thus enabling considerable savings to be made.



Notes on gearboxes, motors, brakes, and frequency converters

2

2.1 Gearboxes

This manual deals with the helical, bevel helical, parallel shaft, helical worm, and worm standard types of gearbox.

Helical gearbox



Bevel helical gearbox



Helical worm gearbox



Parallel shaft gearbox



Worm gearbox



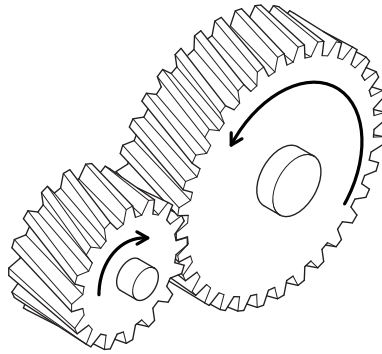
Fig. 2-1 Overview: Helical, bevel helical, helical worm, parallel shaft, and worm gearboxes

The Siemens product range also contains further versions for specific applications:

- Agitators and mixers
- Overhead monorail conveyors
- Tandem gearboxes
- Car washes
- Cooling towers
- Extruders

Description and method of operation

A gearbox consists of one or more gear wheel stages (wheel sets). A stage (wheel set) consists of a pair of gear wheels with different diameters.



Zahnräder.eps

Fig. 2-2 Principle of operation for a gearbox

Due to their different diameters, the two wheels run at different speeds.

The gear ratio i refers to the ratio of the two speeds to one another. One or more wheel sets are combined within a gearbox housing, depending on what ratio is required.

$$i_{\text{total}} = i_1 \cdot i_2 \cdot i_3$$

A gearbox is a speed and torque variator.

$$i = \frac{n_1}{n_2} \quad i = \frac{M_1}{M_2}$$

i	Gear ratio
n	Speed
M	Torque
Index 1	Small wheel
Index 2	Large wheel

Efficiency η

In the main, the gearbox losses can be traced back to the gear teeth, bearings, and seals. The efficiency does not usually depend on the speed or the load.

An initial estimate of a 2 % drop in efficiency can be assumed for each gear wheel stage of helical, parallel shaft, and bevel helical gearboxes.

Helical worm and worm gearboxes

The efficiency of gearboxes with worm gear teeth is heavily dependent on the transmission ratio. While at a transmission ratio of approximately 100:1 the efficiency will only be about 40 to 50 %, at a transmission ratio of 7:1 the worm efficiency is likely to be around 90 %.

Helical worm gearboxes are usually supplied with low worm transmission ratios and, as a result, a relatively high efficiency. This means that such gearboxes still compare favorably with bevel gearboxes, for example.

For the efficiencies of helical worm and worm gearboxes, see the specifications in the ordering and selection data. The efficiency will be lower during the run-in process.

Starting efficiency is never as great as the efficiency at operating speed. This fact should be taken into account when starting a machine at full load, depending on the starting characteristics of the motor.

The efficiency η applies to driving worms; for restoring torques, an efficiency of $\eta' = 2 - \frac{1}{\eta}$ must be assumed.

Self-locking

Static self-locking mostly occurs if the efficiency of a driving worm wheel is ≤ 0.5 . This makes startup impossible. Static self-locking can be canceled by means of vibrations. Self-locking gear teeth, therefore, cannot always replace a brake or backstop.

Experience has shown that dynamic self-locking occurs when the efficiency during operation is ≤ 0.5 .

Example:

Efficiency ($P = 0.37 \text{ kW}$, $n_{\text{OUT}} = 23 \text{ min}^{-1}$, $f_s \approx 1.5$)

Type	T2 (Nm)	η (%)
Helical	150	94
Bevel	169	96
Helical worm	132	87
Worm	102	66

Losses

Losses are caused by friction at the meshing, in the bearings and in the shaft sealing rings, as well as by splashing during oil splash lubrication. Higher losses are experienced with worm and helical worm gearboxes.

The higher the gearbox input speed (which is usually the motor speed), the higher the losses will be.

Losses of splashing: With some gearbox types of construction, the first stage can become completely immersed in the gearbox oil. Large gearboxes with high input speeds can lead to losses of splashing which cannot be ignored.

Service factor

Service factor of a geared motor f_s

This factor is not standardized. The definition used by Siemens is that f_s corresponds to the ratio of the gearbox rated power to the motor rated power. As the power is proportional to the torque • speed, f_s can be determined as follows:

$$f_s = \frac{P_{N, \text{gearbox}}}{P_{N, \text{Motor}}} \Rightarrow \frac{M_{N, \text{gearbox}} \cdot n_{\text{out, gearbox}}}{M_{N, \text{motor}} \cdot n_{\text{motor}}} = \frac{M_{N, \text{gearbox}}}{M_{N, \text{motor}} \cdot i}$$

The gearbox efficiency also has to be taken into account, depending on the gearbox type.

Required service factor of the application $f_{s, \text{req}}$

When gearboxes are being developed for series production, it is not possible to take into account the precise rated conditions to which the gearboxes will subsequently be subjected. To ensure that the drive meets these conditions, a required service factor $f_{s, \text{req}}$ is calculated, which represents the specific case in question. This factor determines the gearbox size in relation to the drive motor's power, but it does not determine that power.

Siemens will be pleased to assist you in making the necessary calculations for drives operating under special rated conditions, such as frequent reversing, short-time and intermittent duty, abnormal temperature conditions, reversal braking, extreme transverse forces on the gearbox output shaft, etc.

In general, if a braking torque which is more than 2.5 times the rated torque has been selected, please contact Siemens to check the drive calculations. This also applies if motors with a starting torque of more than 2.5 times the rated torque are selected or if required service factors > 1.8 result from large mass acceleration factors (> 10) or a great deal of backlash in the transmission elements, for example.

For helical, bevel, parallel shaft, and helical worm gearboxes:

$$f_{s, \text{req}} = f_{s1}$$

With worm gearboxes, two other service factors are also used, which take the duty cycle and ambient temperature into account:

$$f_{s, \text{req}} = f_{s1} \cdot f_{s2} \cdot f_{s3}$$

Assessment of the expected rated conditions according to the shock load (Table 2-1) or mass acceleration factor (formulas):

Table 2-1 Shock load and operating conditions

Shock load	Driving machine
I Light shocks	Mass acceleration factor ≤ 0.3 : Electric generators, belt conveyors, apron conveyors, screw conveyors, lightweight elevators, electric hoists, machine tool feed drives, turbo blowers, centrifugal compressors, agitators, and mixers for uniform densities.
II Moderate shocks	Mass acceleration factor ≤ 3 : Machine tool main drives, heavyweight elevators, slewing gears, cranes, shaft ventilators, agitators and mixers for non-uniform densities, piston pumps with multiple cylinders, metering pumps.
III Heavy shocks	Mass acceleration factor ≤ 10 : Punching presses, shears, rubber kneaders, machinery used in rolling mills and the iron and steel industry, mechanical shovels, heavyweight centrifuges, heavyweight metering pumps, rotary drilling rigs, briquetting presses, pug mills.

$$\text{Mass acceleration factor} = \frac{\text{all external moments of inertia}}{\text{moment of inertia of the drive motor}}$$

The moments of inertia of the gearbox attachments and the production machine are "all external moments of inertia", converted into the motor speed. The calculation is done using the following formula:

$$J_{\text{load}} = J_2 \cdot \left(\frac{n_2}{n_1}\right)^2$$

$$\begin{aligned} J_{\text{load}} &= \text{All external moments of inertia (based on the motor shaft)} \\ J_2 &= \text{Moment of inertia based on the output speed of the gearbox} \\ n_2 &= \text{Output speed of the gearbox} \\ n_1 &= \text{Input speed (motor speed)} \end{aligned}$$

The moment of inertia of the drive motor (motor side) is calculated as follows:

- J_M moment of inertia of the motor
- J_B moment of inertia of the brake
- J_{M+} additional moment of inertia (e.g. flywheel or heavyweight fan)

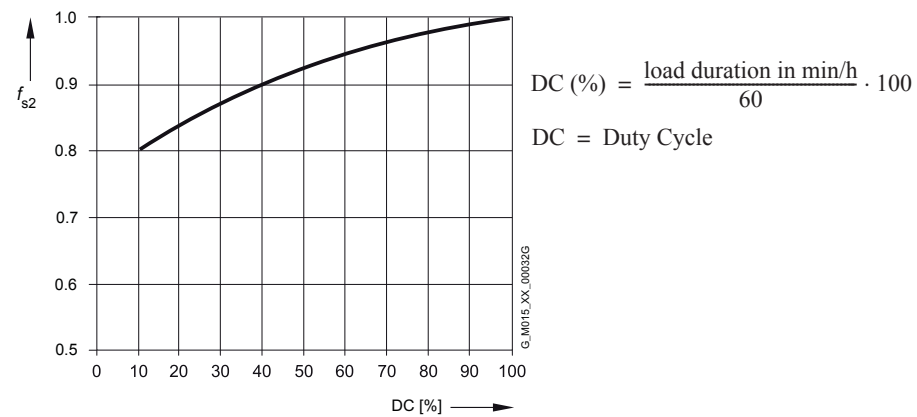
Table 2-2 Service factors f_{s1}

Daily operating duration		4 hours			8 hours			16 hours			24 hours		
Operations*/h		< 10	10 - 200	> 200	<10	10 - 200	> 200	< 10	10 - 200	> 200	< 10	10 - 200	> 200
Shock load	I	0.8	0.9	1.0	0.9	1.0	1.1	1.0	1.1	1.2	1.2	1.3	1.5
	II	1.0	1.1	1.3	1.1	1.2	1.3	1.2	1.4	1.6	1.4	1.5	1.6
	III	1.3	1.4	1.5	1.4	1.5	1.6	1.5	1.6	1.7	1.6	1.7	1.8

* "Operations" refers to the total number of switch-on, braking, and reversal operations.

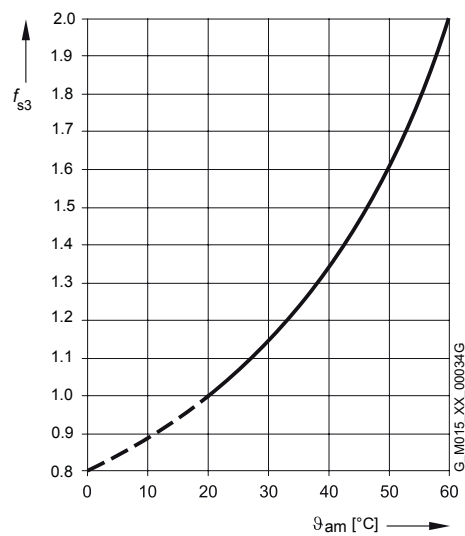
Lower f_{s1} factors can be selected if the operating duration is less than 4 hours a day or if flexible couplings or belt drives are used. In such cases, please consult our specialist support team.

Service factors f_{s2} for short-time duty



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Fig. 2-3 Service factors f_{s2} for short-time duty

Service factors f_{s3} for ambient temperature

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Fig. 2-4 Service factors f_{s3} for ambient temperature

With a service factor $f_{s3} < 1$ for temperatures below +20 °C, please contact us.

Dimensioning the gearbox

A geared motor is selected where the following applies: $f_s \geq f_{s, req}$

f_s = Available service factor of the geared motor

$f_{s, req}$ = Required service factor

Radial and axial forces

If these forces occur, they can reach impermissibly high values irrespective of the rated torques and service factors. As a result, they represent additional criteria to be considered when selecting the gearbox.

Radial forces (transverse forces)

Radial forces occur if no coupling is used and the driving forces are transmitted to the gearbox output shaft in a non-positive or positive manner by means of chain wheels, for example. The chain's recoil is then applied to the gearbox shaft as a radial force, as are the pretensioning forces of belts or friction wheels, for example.

The radial force available at the shaft extension is calculated from the geared motor's available output torque, the diameter, and the type of output element. The type of output element determines factor C , by which the available radial force is to be increased.

$$F = \frac{2 \cdot M \cdot C}{d}$$

F Available radial force in N

M Available torque in Nm

d Diameter of the output element in m

C Factor for the type of output element (without unit)

Table 2-3 Factor C for the type of the output element

Input element	Version	C
Gear wheel	> 17 teeth	1.00
	≤ 17 teeth	1.15
Chain wheel	≤ 20 teeth	1.00
	14 to 19 teeth	1.25
	≤ 13 teeth	1.40
Toothed belt ¹⁾	Pretensioning force	1.50
V belt ¹⁾	Pretensioning force	2.00
Flat belt ¹⁾	Pretensioning force	2.50
Agitator/mixer	Rotating radial force	2.00

¹⁾ Pretensioning in accordance with the belt manufacturer's instructions

The available radial force must not exceed the level that is permissible for the gearbox.

The shaft strength and bearing load rating are crucial factors when it comes to determining the permissible radial forces. You will find the relevant values in the *Geared Motors* catalog. These values apply for forces applied to the center of the gearbox shaft extension. If forces are applied somewhere other than the center, please use the formulas in the *Geared Motors* catalog or the *calculation of output shaft bearing arrangement* wizard in the electronic catalog.

Axial forces

Axial forces result from pressure or tension being exerted on the gearbox output shaft, for example, if the gearbox shaft is used as a turning point on a vertical geared motor (gearbox at top, motor at bottom). If no transverse force load is present, an axial force F_{Ax} (tension or compression) of around 50 % of the specified radial force with standard bearings can be permitted for gearbox sizes 18 to 148. You can use our *calculation of output shaft bearing arrangement* wizard in the electronic catalog to calculate the permissible forces. Combined forces with an axial and a radial component can also be calculated. Please contact us in case of doubt.

Special environmental requirements

The standard temperature range is -20 to +40 °C. Care must be taken to choose a suitable lubricant; see the *Geared Motors* catalog.

Higher ambient temperatures

A higher ambient temperature will inevitably cause the oil sump temperature to increase, necessitating a thermal check of the gearbox. Viton shaft sealing rings must be provided.

Lower ambient temperatures

In the case of frozen driving machines, higher torques and loads may have to be taken into account. The gearbox materials will have to be checked in terms of their suitability for use at the required temperature in each case.

Dust, humidity, acids

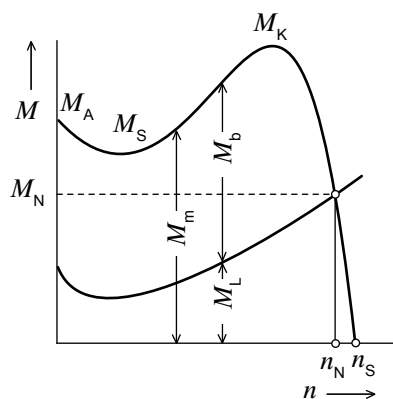
Special shaft sealing rings may be required, as may vent valves, in place of the vent plugs which are used as standard. A higher degree of protection can be used too. Our drive experts will be happy to provide assistance in determining what measures are necessary for your particular case.

2.2 Motors

The motors referred to here are three-phase asynchronous motors with squirrel-cage rotors.

Method of operation

The figure below shows the typical torque (M) - speed (n) characteristic of a three-phase asynchronous motor. The characteristic of the load torque shown is simply an example. On start-up, the motor follows its characteristic curve until this intersects with the load torque characteristic curve.



DrehmomentAsynchron.eps

Fig. 2-5 Torque characteristic of a three-phase asynchronous motor with squirrel-cage rotor

M_N	Rated torque	M_K	Breakdown torque
M_m	Motor torque	M_S	Pull-up torque
M_L	Load torque	n_N	Rated speed
M_b	Acceleration torque	n_s	Synchronous speed = no-load operation
M_A	Starting torque		

Number of poles

The synchronous speed of the three-phase motor is inversely proportional to the number of pole pairs p and proportional to the supply frequency f .

$$n_s = \frac{f}{p}$$

Example:

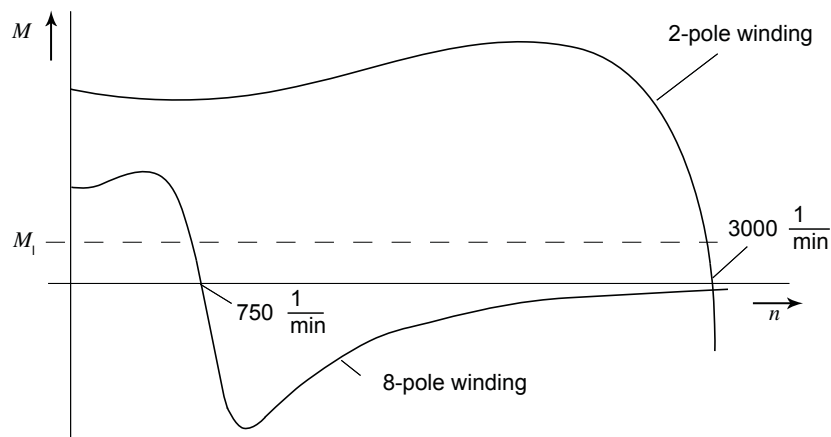
A 4-pole motor (number of poles $2p = 4$) at a supply frequency of $f = 50$ Hz has a synchronous speed of

$$n_s = \frac{f}{p} = \frac{50 \text{ Hz}}{2} = \frac{50 \frac{1}{s}}{2} = 25 \frac{1}{s} = 1500 \frac{1}{\text{min}}$$

2, 4, 6, and 8 poles are standard; 4-pole motors are normally used. 2-pole motors generate more noise than types with more poles. Motors with a higher number of poles require a larger unit volume at the same power, meaning that they are usually more expensive.

Pole-changing

If the stator is equipped with windings for several different numbers of poles, this motor can be operated in several corresponding speed steps. The lower speed can be used for positioning, for example. Fig. 2-6 shows that, when switching from the high speed to the low speed, the low-speed winding is operated regeneratively until the load characteristic curve intersects with the motor characteristic curve of the low-speed winding. This is also known as "regenerative braking".



GeneratorBremsen.eps

Fig. 2-6 Regenerative braking, schematic representation

The regenerative breakdown torque can be higher than the breakdown torque when motoring, which would subject mechanical attachments, such as the gearbox, to extreme loads too. The motor's current load is also very high, which causes the motor to heat up significantly, thus reducing the permissible switching frequency.

For these reasons it can be a good idea to reduce the "switching shock". One way of doing this is to use 2-phase regenerative braking, which lowers the torque. However, this is not normally permitted for vertical lift drives, for safety reasons. A motor with a higher moment of inertia could also be used; although the boundary conditions of the application would need to be taken into account.

Service factor

If a service factor is specified on the nameplate, this means that the motor can be continuously operated with an overload. A service factor of 1.1 means that the motor can be permanently operated at 1.1 times its rated power.

Slip

As can be seen in Fig. 2-5 showing the asynchronous motor's method of operation, in motoring operation the rated speed is lower than the synchronous speed. This lag is called the **slip speed**. The **slip** s is defined as follows and is usually given in %.

$$s = \frac{n_s - n_{\text{rated}}}{n_s} \cdot 100 \%$$

n_s Synchronous speed

n_{rated} Rated speed

Power factor $\cos \varphi$ and efficiency η

The power factor is a measure of the ratio of the motor's active power consumption to its reactive power consumption. The current lags behind the voltage by angle φ . The smaller this angle is, the higher the power factor and, as a result, the higher the active power component consumed.

$$P_{\text{ap}} = \sqrt{3} \cdot U \cdot I \quad P_r = \sqrt{3} \cdot U \cdot I \cdot \sin \varphi \quad P_a = \sqrt{3} \cdot U \cdot I \cdot \cos \varphi$$

P_{ap} Apparent power

P_r Reactive power

P_a Active power

U Line voltage in V

I Phase current in A

$$P_{\text{rated}} = P_a \cdot \eta$$

P_{rated} Rated power of the motor (= shaft output) in W

η Efficiency of the motor

The product range of geared motors exclusively comprises motors in the EU efficiency classes EFF1 (High Efficiency) and EFF2 (Improved Efficiency).

Tolerances

The following tolerances are permitted for the ratings, in accordance with EN 60034-1:

Efficiency	$-0.15 \cdot (1-\eta)$ for rated power ≤ 50 kW
Power factor	$-(1 - \cos \varphi)/6$
Minimum	0.02
Maximum	0.07
Slip	± 20 % for rated power > 1 kW ± 30 % for rated power < 1 kW
Locked-rotor current	+20 %
Starting torque	-15 % to +25 %
Breakdown torque	-10 %
Moment of inertia	± 10 %

Derating

Unless otherwise specified, the rated power of a motor applies for a maximum coolant temperature of 40 °C and a maximum installation altitude of 1,000 m above sea level.

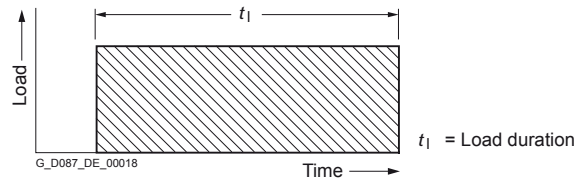
For an approximate selection at higher coolant temperatures and/or installation altitudes, the motor power should be reduced by the factor k_{AT} .

Table 2-4 Factor k_{AT} for different installation altitudes and/or coolant temperatures

Installation altitude (IA) m	Coolant temperature (CT)					
	< 30 °C	30 to 40 °C	45 °C	50 °C	55 °C	60 °C
1,000	1.07	1.00	0.96	0.92	0.87	0.82
1,500	1.04	0.97	0.93	0.89	0.84	0.79
2,000	1.00	0.94	0.90	0.86	0.82	0.77
2,500	0.96	0.90	0.86	0.83	0.79	0.74
3,000	0.92	0.86	0.82	0.79	0.75	0.70
3,500	0.88	0.82	0.79	0.75	0.71	0.67
4,000	0.82	0.77	0.74	0.71	0.67	0.63

Operating modes

The rated powers apply for the S1 operating mode (uninterrupted duty with constant load) according to EN 60034-1. This refers to operation with a constant load condition, which lasts long enough to enable the thermal steady state to be achieved. This same regulation differentiates between the following groups of operating modes:



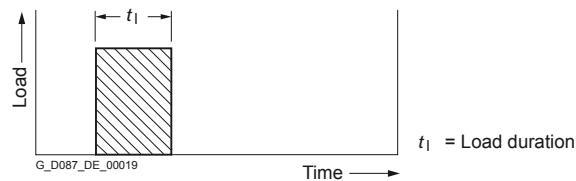
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Fig. 2-7 Load/time diagram S1

Operating modes where starting and electrical braking do not affect the temperature

Operating mode S2: Short-time duty

Operating times of 10, 30, 60, and 90 minutes are recommended. After each period of duty the motor remains at zero current until the winding has cooled down to the coolant temperature.

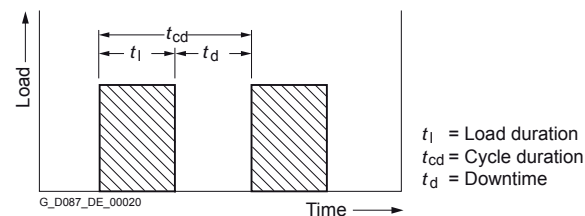


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Fig. 2-8 Load/time diagram S2

Operating mode S3: Intermittent duty

Starting does not affect the temperature. Unless any agreement is made to the contrary, the cycle duration is 10 minutes. Values of 15 %, 25 %, 40 %, and 60 % are recommended for the duty ratio.



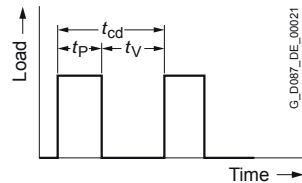
$$\text{Duty ratio based on 10 min} = \frac{t_l}{t_l + t_d} \cdot 100\%$$

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Fig. 2-9 Load/time diagram S3

Operating mode S6: Continuous operation with intermittent load

Unless any agreement is made to the contrary, the cycle duration here is also 10 minutes. Values of 15 %, 25 %, 40 %, and 60 % are recommended for the load duration factor.



t_{cd} Cycle duration
 t_p Operating time with constant load
 t_v Idle time

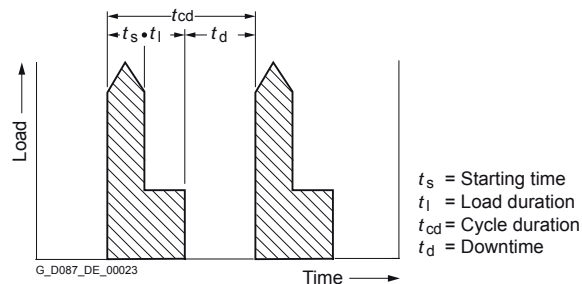
$$\text{Duty ratio based on 10 min} = \frac{t_p}{t_{cd}} \cdot 100 \%$$

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Fig. 2-10 Load/time diagram S6

Operating mode S10: Operation with discrete constant loads

In this mode a maximum of four discrete loads are available, of which each load achieves the thermal steady state. A load of the same value as the one used in S1 operating mode should be selected for this operating mode.

Operating modes where starting and electrical braking do affect the temperature**Operating mode S4:** Intermittent duty where starting affects the temperature

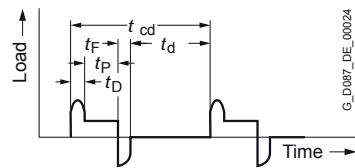
t_s = Starting time
 t_l = Load duration
 t_{cd} = Cycle duration
 t_d = Downtime

$$\text{Duty ratio based on 10 min} = \frac{t_s + t_l}{t_s + t_l + t_d} \cdot 100 \%$$

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Fig. 2-11 Load/time diagram S4

Operating mode S5: Intermittent duty where starting and braking affects the temperature



t_{cd} Cycle duration
 t_D Starting time
 t_P Operating time with constant load
 t_F Time with electrical braking
 t_d Downtime with windings at zero current

$$\text{Duty ratio based on 10 min} = \frac{t_D + t_P + t_F}{t_{cd}} \cdot 100 \%$$

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Fig. 2-12 Load/time diagram S5

For the S4 and S5 operating modes, this code should be followed by the duty ratio, the moment of inertia of the motor (J_M), and the moment of inertia of the load (J_{ext}), both based on the motor shaft.

Unless any agreement is made to the contrary, the cycle duration here is also 10 minutes. Values of 15 %, 25 %, 40 %, and 60 % are recommended for the duty ratio.

Operating mode S7: Continuous periodic duty with starting and braking

Operating mode S8: Continuous operation with periodic speed variations

For the **S7** and **S8** operating modes, the moment of inertia of the load based on the motor shaft must be known.

Operating mode S9: Operation with non-periodic load and speed variations

The cycle duration for the periodic operating modes is 10 minutes.

According to the table below, the motor list powers can be converted to the lower duty cycle using the corresponding k_{DR} factors for the **S2** and **S3** operating modes. This causes both the power and the rated torque to increase, but not the breakdown torque.

Table 2-5 Operating modes S2 and S3

Operating mode	k_{DR}	Explanation	
S2	1.1	Load duration	60 min
	1.2	Load duration	30 min
	1.4	Load duration	10 min
S3	1.1	Duty ratio	60 %
	1.15	Duty ratio	40 %
	1.3	Duty ratio	25 %
	1.4	Duty ratio	15 %

The operating mode and the required motor power can be determined on request for operating modes S4 to S10.

Temperature classes

EN 60034-1 categorizes the temperature classes, to which precisely predefined temperature values are also assigned. The temperature rise limit (also known as temperature increase) is a mean value of the motor winding temperature. The maximum permissible steady-state temperature applies to the hottest point on the winding.

Siemens motors are designed in accordance with temperature class F. At rated power and with direct on-line operation, the utilization corresponds to temperature class B.

The values in the table below apply for a maximum coolant temperature of 40 °C.

Table 2-6 Temperature classes

Temperature class	Temperature rise limit based on a coolant temperature of 40 °C	Maximum permissible steady-state temperature
B	80 K	130 °C
F	105 K	155 °C
H	125 K	180 °C

Degree of protection

A suitable degree of protection must be selected to protect the machine and personnel from the following hazards, depending on the relevant operating conditions and environmental requirements:

- Damaging effects of water, foreign bodies, and dust
- Contact with rotating internal parts or live parts

The degrees of protection for electrical machines are indicated by means of a code consisting of the two letters IP (= International Protection) and two digits, in accordance with EN 60034 Part 5.

Motors and geared motors are supplied with the IP55 degree of protection as standard.

Table 2-7 Degrees of protection

IP	1st code number, protection against foreign bodies	2nd code number, protection against water
0	Not protected	Not protected
1	Protected against solid foreign bodies, Ø 50 mm and larger	Drip-proof
2	Protected against solid foreign bodies, Ø 12 mm and larger	Drip-proof, if the housing is inclined by up to 15°
3	Protected against solid foreign bodies, Ø 2.5 mm and larger	Protected against spray water
4	Protected against solid foreign bodies, Ø 1 mm and larger	Protected against splash water
5	Protected against dust	Protected against jet-water
6	Dust-proof	Protected against intense jet-water
7	-	Protected against temporary immersion in water
8	-	Protected against permanent immersion in water

Motor protection

A distinction is made between current-dependent and motor-temperature-dependent protective devices.

Current-dependent protective devices

Fuses are only used to protect line supply conductors in the event of a short-circuit. They are not suitable for providing the motor with overload protection. The motors are usually protected by thermally delayed overload protection (motor circuit breakers or overload relays). This type of protection is particularly effective in the event of a locked rotor. For normal operation with short startup times and starting currents that are not excessive and for low switching frequencies, motor circuit breakers provide adequate protection. They are not suitable for heavy starting duty or high switching frequencies. Differences between the thermal time constants of the protective device and of the motor can result in unnecessary early tripping or in overloads not being detected.

Temperature-dependent protective devices

These devices are integrated into the motor winding and can be designed as temperature sensors (PTC thermistors) or temperature switches (winding thermostats, bimetallic switches). Temperature sensors undergo a sudden change in their resistance when a rated response temperature is reached. A trip unit can be used to evaluate this change and to open auxiliary circuits. Most converters can evaluate temperature sensors directly.

Motor temperature sensing for operation on the converter

The *KTY 84-130 temperature sensor* is a PTC thermistor that changes its resistance depending on temperature in accordance with a defined curve. Some converters are able to determine and evaluate the motor temperature on the basis of the resistance.

Dimensioning of a motor connected to the supply system

The thermal utilization is the selection criterion for the motor.

- Uninterrupted duty (= S1 = 100 % duty cycle)

The rated power of the motor must be equal to or greater than the power required by the driving machine.

- Intermittent duty (= S3 or S4)

Switch-on and the high starting current that results from this cause the motor to heat up significantly.

Determining the permissible switching frequency

The no-load operating frequency z_0 is specified in the technical data for the motors. The frequency of operation expresses how often a motor is able to accelerate the moment of inertia of its rotor to its no-load speed, with no counter-torque and at 50 % of the duty ratio, in one hour.

The permissible switching frequency z takes the counter-torque, the additional moment of inertia, and the duty cycle into account for the application in question.

The permissible switching frequency z can be determined using the following formula:

$$z = z_0 \cdot k_M \cdot k_{FI} \cdot k_P$$

When calculating the permissible switching frequency, it is taken for granted that the motor must be braked mechanically or allowed to slow down without braking. Electrical braking will increase the losses experienced by the motor.

In the case of reversal braking, which should be avoided in practice, the calculated switching frequency corresponds to around one quarter of the number of permissible starts without electrical braking.

The calculation of switching frequencies is an approximate assessment, which serves as a configuring tool.

If the calculated switching frequency is close to the required value, we recommend that you contact us.

Once the permissible switching frequency of the motor has been determined, the brake must be checked in terms of its suitability for this switching frequency.

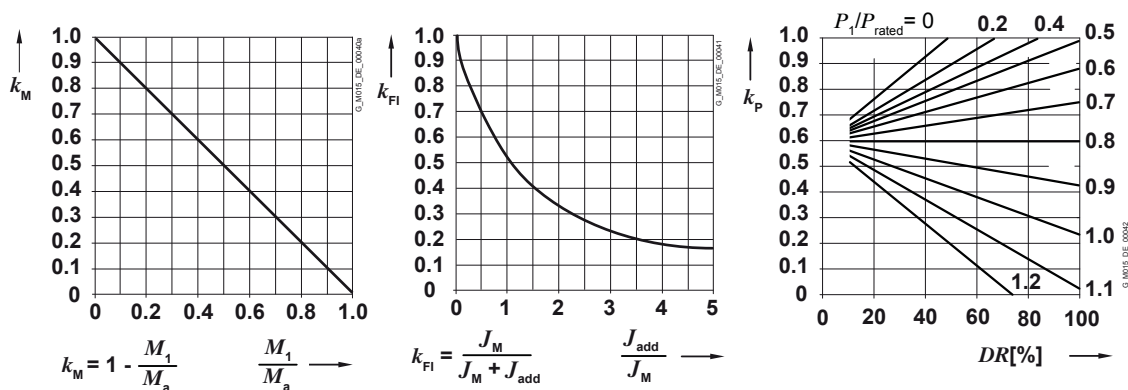


Fig. 2-13 Switching frequencies k_M , k_{FI} , and k_P

Z_0	No-load operating frequency	in h^{-1}
M_a	Acceleration torque of the motor	in Nm
M_1	Load torque during acceleration	in Nm
J_M	Moment of inertia of the motor	in kgm^2
J_{add}	All additional moments of inertia based on the motor shaft	in kgm^2
DR	Duty ratio	in %
P_{rated}	Rated motor power	in kW
P_1	Static load	in kW

Operation on a frequency converter

The torque-speed characteristic of the three-phase asynchronous motor shifts when the frequency and voltage change; see the figure below. Provided that the ratio of the voltage to the frequency remains constant, the motor can be operated with a constant flux and, as a result, a constant torque.

If the frequency is further increased once the maximum possible voltage has been reached, the rated torque drops roughly in proportion to the voltage rise and the motor breakdown torque drops quadratically. This is called the field weakening range; the motor can be operated at a constant power up to the mechanical or electrical limit speed, whichever is lower. The electrical limit speed results from the sharply decreasing breakdown torque, See Fig. 2-14. Experience has shown that the condition " $0.7 \cdot \text{breakdown torque} > \text{delivered torque}$ " should be met.

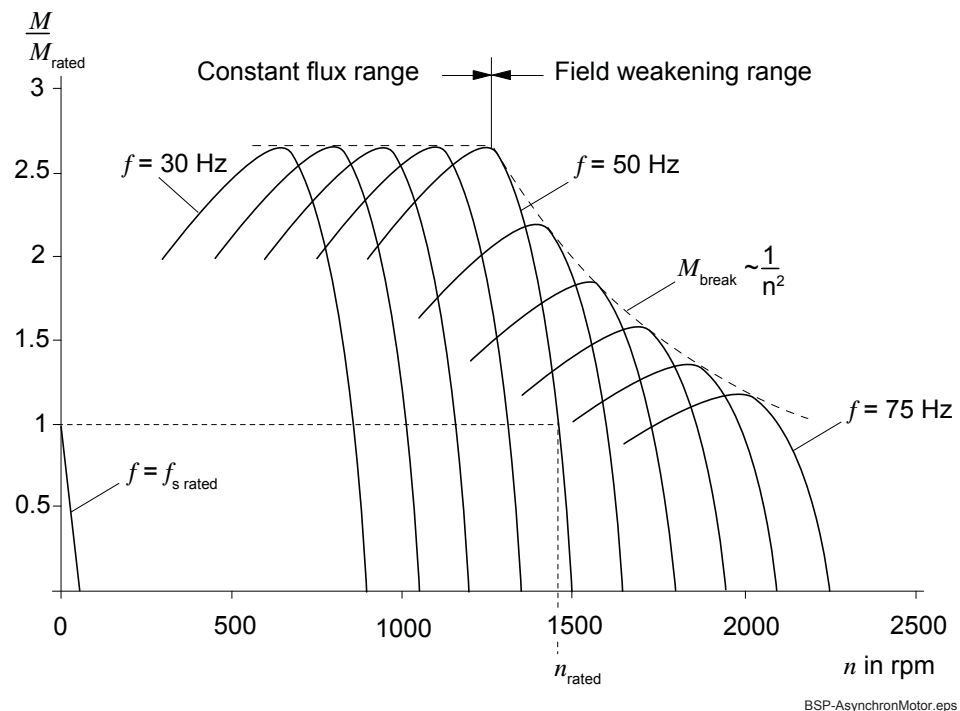
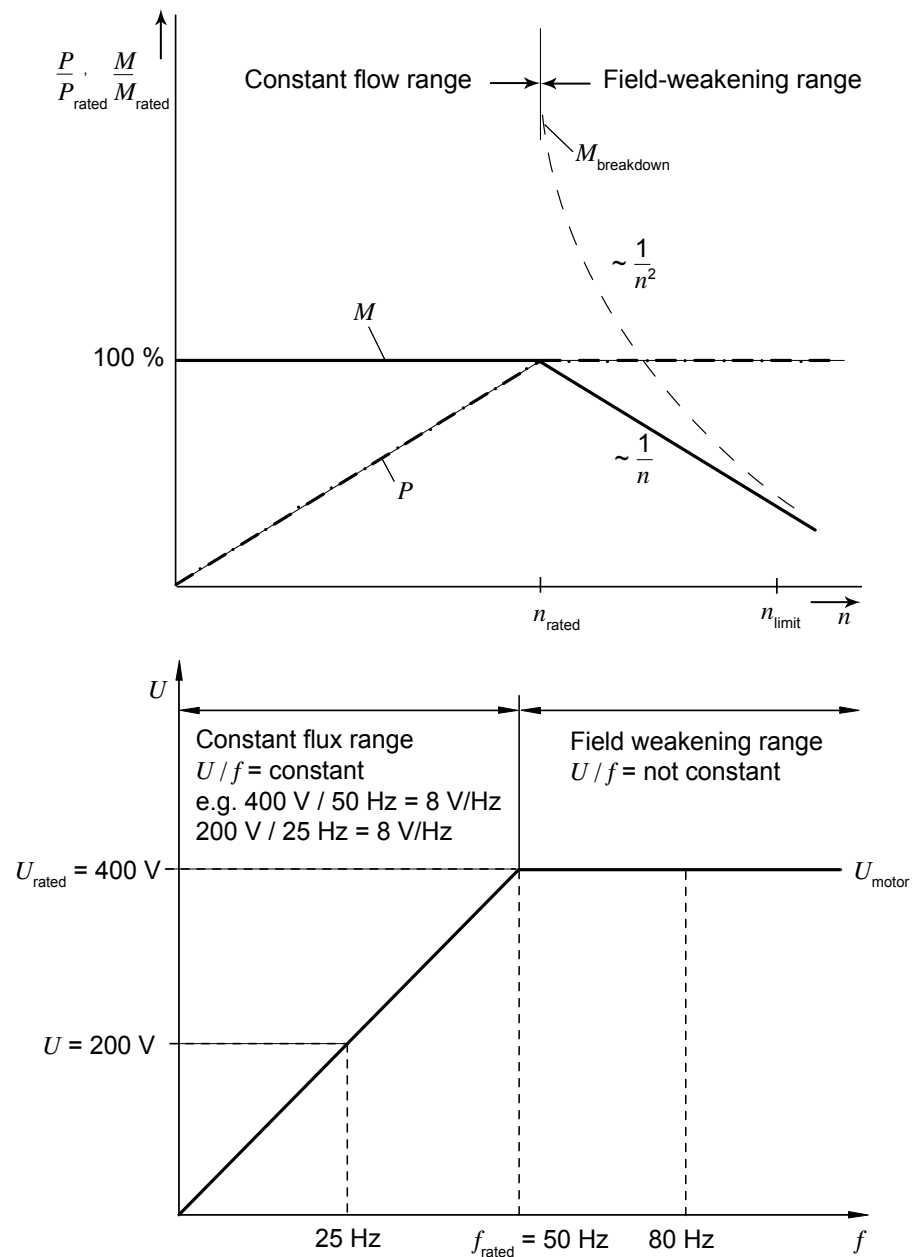


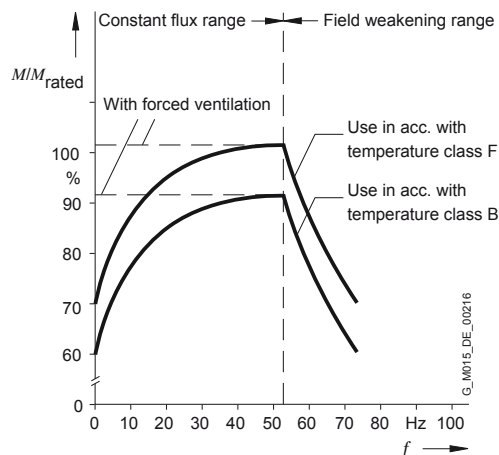
Fig. 2-14 Example for a 4-pole asynchronous motor



Feldschwä .../ Uf-Kenn ...eps

Fig. 2-15 Field weakening and constant flux range

Due to the weaker cooling effect at low speeds, the continuous permissible torque of a self-cooled motor is lower than at its rated frequency. Figure 2-16 applies to a motor with a rated frequency of 50 Hz.



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Fig. 2-16 Torque utilization

In many applications, using a self-cooled motor of the next frame size up does away with the need for an external fan.

To enable the motor to still be operated with a constant torque at speeds higher than the rated speed, the **87 Hz characteristic** is used. This characteristic exploits the fact that the frequency converter is capable of outputting frequencies higher than the line frequency. Since the ratio of the voltage to the frequency must be constant, a standard motor (winding 230/400 V - delta/star - 50 Hz) can be operated at 230 V and 50 Hz or 400 V and 87 Hz in a delta connection, for example. ($87 = \sqrt{3} \cdot 50$, $400 = \sqrt{3} \cdot 230$). This means that the rated torque can be kept constant, up to almost 87 Hz. The frequency rise results in a power of 1.73 times than offered by operation at 50 Hz ($1.73 = \sqrt{3}$).

In this case, the converter must be dimensioned for the higher motor current in the delta connection.

The higher speed means that more noise is generated.

Dimensioning of a motor connected to a frequency converter

Various criteria can affect the dimensioning of the motor, depending on the requirements of the application in question. In each case, the thermal utilization of the motor must be checked.

When operating with a high switching frequency, the motor's acceleration torque may have to be limited to its rated torque. If short acceleration and deceleration times are of paramount importance, the motor's overload capability can be utilized in conjunction with the converter provided. It is also necessary to take the entire cycle into account for the thermal check. As a general rule, the rms current must not exceed the rated motor current.

An external fan may be required for operation at frequencies below the rated frequency, depending on the operation time. If no external fan is used, the cooling effect will be reduced due to the slower speed; in the long term, the motor is only capable of delivering a torque which is lower than its rated torque. A larger motor could be selected, rather than using an external fan.

The following must be taken into consideration if the motor is operated above the rated frequency:

- Noise generation
- In the field weakening range, the limit speed based on the point at which the motor's breakdown torque and rated torque meet, See Fig. 2-15. Experience has shown that a tolerance of -30 % must be observed for the breakdown torque.
- Heating of the gearbox due to high input speeds

Special environmental requirements

The standard ambient temperature is between -20 and +40 °C.

Higher temperatures

For higher temperatures, refer to "Derating", Page 2-25. The motor and attachment materials will have to be checked in terms of their suitability for use at the required temperature in each case.

Lower temperatures

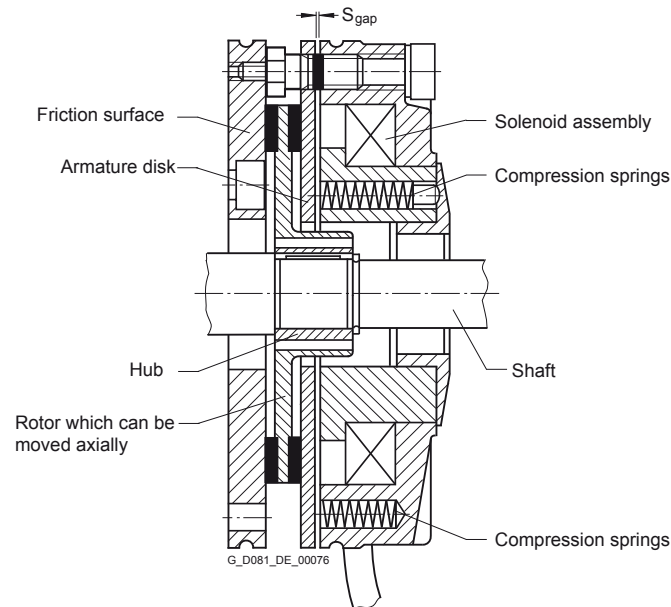
Strip heaters and special bearing grease may be required. The motor and attachment materials will have to be checked in terms of their suitability for use at the required temperature in each case.

Dust, humidity, acids

Higher degrees of protection can be used. Condensation formed due to temperature fluctuations can be combated by means of heaters and condensate drain holes. Our drive experts will be happy to provide assistance in determining what measures are necessary for your particular case.

2.3 Brakes

The brakes described here are *spring-operated single-disk brakes with two friction faces*, which work according to the quiescent current principle. This means that the brake is applied at zero current and released when energized.



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Fig. 2-17 Schematic representation of a spring-operated single-disk brake

With the exception of the shaft, the only rotating part is the rotor, which can be moved axially along the shaft.

Braking/Applying

Compression springs generate the braking torque at zero current by means of friction locking - the brake is applied (i.e. engaged). The rotor which can be moved axially is pressed against the opposite friction surface by the armature disk.

Disconnecting/Releasing

The brake is disconnected (released/disengaged) electromagnetically when the brake coil is supplied with voltage. The resulting magnetic force pulls the armature disk against the spring force on to the solenoid component. The rotor is released from the spring force and can rotate freely.

Voltage supply

The brake features a DC voltage coil, which is supplied with power by a rectifier in the motor terminal box as standard. A direct DC voltage supply or a rectifier with supplementary functions can also be used.

Braking torques

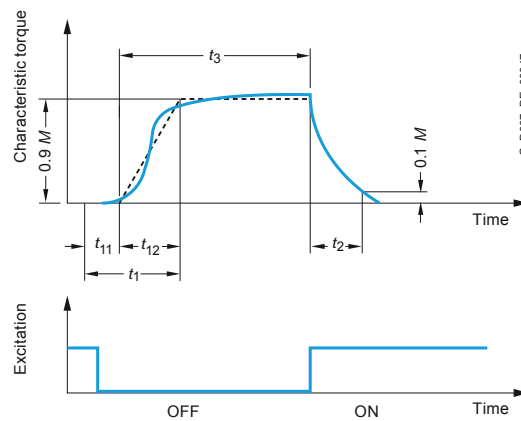
The torque can be set within certain limits for every brake size.

Holding brake/Operational brake

Brakes can be used as holding brakes if the motor needs to be mechanically locked in a certain position for safety reasons, for example.

They can also be used as operational brakes if the motor has to be braked mechanically.

Switching times



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Fig. 2-18 Definition of switching times (VDI 2241)

Switching times:

- t_{11} Response time
- t_{12} Rise time
- t_1 Application time
- t_2 Disconnection time
- t_3 Slipping time

Application time, $t_1 = t_{11} + t_{12}$ (= engaging time)

The time it takes to build up the braking torque comprises two different components: the response time t_{11} , for example due to bus runtimes, CPU times, etc., and the rise time t_{12} , during which the mechanical braking torque builds up to 90 % of the set rated torque.

The brake coil voltage is disconnected at the AC side of the rectifier as standard. The AC voltage is either picked off from the motor winding directly or fed to the rectifier separately. Disconnection on the DC side enables significantly shorter engaging times to be achieved. This always takes place with switching on the DC and AC side. Disconnection on the DC side can be executed by means of a rectifier with a supplementary function, for example, so no additional cables are required. If an external switch is used, please note that it may require a protective circuit, depending on the prevailing requirements in terms of the switching frequency and EMC.

The brake may need a separate voltage supply in the case of hoisting gears, other overhauling loads, or applications with extremely high moments of inertia. We also recommend that the brake is disconnected on the DC and AC sides.

Stopping time, $t_{11} + t_3$

The time it takes the motor to come to a standstill comprises two components: the response time t_{11} , for example due to bus runtimes, CPU times, etc., and the brake's slipping time t_3 (the actual braking time required to dissipate the drive's kinetic energy).

Disconnection time, t_2 (= release time)

The *over-excitation* (also called *high-speed excitation*) option is available for shortening the release time. A shorter release time can be of benefit for drives connected to the supply system with a high switching frequency. The high-speed release of the brake increases the brake motor's no-load operating frequency.

Slipping time, t_3 (= braking time)

The slipping time is the time during which the mechanical brake converts rotary (kinetic) energy into heat by means of friction. The friction lining "slips". The slipping time is usually determined by the kinetic energy which is stored in the moving objects and which is to be converted, as well as by the available braking torque.

Mechanical release

The brake can be equipped with a manual release lever so that it can be released manually at zero current.

Dimensioning of a brake

The application's requirements must be taken into account when dimensioning the brake, such as:

- Static safety, so a motor can be held at a standstill, for example
- Maximum braking time, so a saw does not continue to run for too long, for example
- Maximum brake deceleration, so the material being conveyed does not tip over and wheels do not slip, for example
- Maximum braking distance, so a machine does not collide with a solid obstacle, for example

The following should then be checked for the selected brake:

- Maximum permissible no-load and operating speeds
- Wear

A measure of the wear is the braking energy (also known as "friction energy" or "operating energy") which is converted into heat in the brake linings. The wear to the brake lining depends on the following factors:

- Masses to be decelerated
- Braking speed
- Switching frequency

The permissible switching frequency for the calculated braking energy and the selected brake can be found in the relevant diagram in the Geared Motors catalog. It is particularly important for the maximum permissible operating energy for one operation to be checked when operating on a frequency converter or when the brake is being used as a holding brake.

Depending on the application, the service life (also known as "endurance") offered by the brake lining until adjustment and/or replacement is required may be of interest.

Special environmental requirements

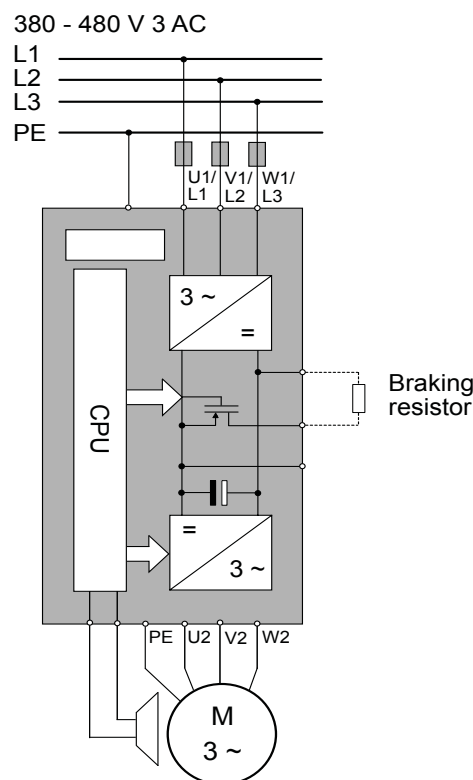
If you intend to operate the brake outside of the standard ambient temperature range of -20 to +40 °C or in an environment containing abrasive materials, acidic, and/or humid air, our drive experts will be happy to provide assistance in determining what measures are necessary for your particular case.

2.4 Frequency converters

To facilitate stepless speed adjustment for electric motors, Siemens provides frequency converters for installation in the control cabinet, as well as versions for distributed installation either local to or integrated in the motor. The SINAMICS range also offers a power recovery option.

Principle of operation

A supply system with a fixed voltage and frequency generates a variable voltage and frequency in order to supply the motor windings with power. The user can define both of these variables. The motor's speed can be adjusted by changing the frequency. The voltage usually varies in proportion to the frequency in order to keep the magnetic flux in the motor constant in its rated operating state. A torque precisely tailored to the application in question can be delivered by assigning parameters to the converter and using control methods within it.



FU-Block_1.eps

Fig. 2-19 Frequency converter block diagram

Brief description of functions

- The voltage provided by the supply system is rectified (DC-link voltage) and fed to the inverter, which then uses the rectified voltage to generate a motor voltage that is variable in terms of frequency and voltage level.
- All the converter functions are controlled (open-loop and closed-loop) via a processor module (CPU), which also serves as the interface with the user (terminals, bus).
- The (optional) braking resistor is used when the converter does not have access to power recovery and the motor creates regenerative power. Examples are: stretching unit of a belt conveyor system, hoisting gear when lowering, centrifuge when braking.
- Certain applications require a holding brake, which is usually controlled by the converter.

Dimensioning of a frequency converter

The characteristic used for selection is the required motor current, which has to be lower than or equal to the converter current. Acceleration and deceleration procedures normally have to be taken into account. The converter characteristics such as rated current and overload capability can be found in the relevant catalogs.

Line harmonic distortions - Line commutating reactor

The most effective way of limiting the frequency converter's line harmonic distortions to the permissible values is to increase the inductance of the line supply conductors by using a line reactor. The line reactor restricts the current increase, extends the current flow time, reduces the current amplitudes and, as a result, brings down the harmonic currents.

The question of whether or not a line commutating reactor is required depends to a large extent on the ratio of the rated converter power S_{con} to the line short-circuit power $S_{\text{s, line}}$ which is permissible for the particular type of converter used. A line reactor must be used for the Micromaster 4, for example, if the following applies to the line short-circuit power:

$$S_{\text{s, line}} = 100 \times S_{\text{con}}$$

$$S_{\text{con}} = \sqrt{3} \times U_{\text{line}} \times I_{\text{con_input}}$$

As the line short-circuit power $S_{\text{s, line}}$ is not known in most cases, we recommend that a line reactor is always connected upstream of the frequency converter.

SINAMICS G120 converters support the use of power units which are capable of power recovery. In cases such as this, a line reactor must not be used.

EMC (electromagnetic compatibility) - Line filter

Electromagnetic compatibility (EMC) describes the capability of an electrical device to function satisfactorily in a predetermined electromagnetic environment without itself causing interference which is unacceptable for other devices in the environment.

EMC, therefore, represents a quality feature for:

- Internal noise immunity: resistance to internal electrical disturbance variables
- External noise immunity: resistance to external electromagnetic disturbances
- Noise emission level: environmental effects caused by electromagnetic emissions

To ensure that the cabinet unit functions satisfactorily in the system, the environment subject to interference must not be neglected. For this reason, special requirements exist regarding the structure of the system in terms of EMC.

Operational reliability and noise immunity

In order to achieve the greatest possible operational reliability and immunity to noise of a complete system (converter, automation, drive machine, etc.), measures must be taken by the converter manufacturer and the user. Only when all these measures are fulfilled can the faultless functioning of the converter be guaranteed and the specified legal requirements (89/336/EEC) be met.

Noise emissions

Product standard EN 61800-3 outlines the EMC requirements for variable-speed drive systems. It specifies requirements for converters with operating voltages of less than 1,000 V. Different environments and categories are defined depending on where the drive system is installed.

Definition of first and second environments:

- First environment
Residential buildings or locations where the drive system is connected to a public low-voltage network without a transformer.
- Second environment
Industrial locations supplied by a medium-voltage network via a separate transformer.

First environment	C1	Second environment
	C2	
	C3	
	C4	

1-2_Umgebung.eps

Fig. 2-20 First and second environment

Definition of categories C1 to C4:

- Category C1
Rated voltage < 1,000 V; for unrestricted use in the first environment.
- Category C2
Rated voltage for stationary drive systems < 1,000 V; for use in the second environment. For use in the first environment only when sold and installed by skilled personnel.
- Category C3
Rated voltage < 1,000 V; for use in the second environment only.
- Category C4
Rated voltage < 1,000 V or for rated currents > 400 A in complex systems in the second environment.

EMC classes

In order to comply with the specified EMC class, a suitable filter must be used.

The following three general classes of EMC behavior have to be taken into account:

- Class 1 – General industrial use
Compliance with the EMC *Product Standard for Power Drive Systems EN 68100-3* for use in the second order environment (industrial) and with limited propagation.
For Class 1, the use of converters without filters is permitted.
- Class 2 – Industrial use with Class A filter
The behavior limit values correspond to the regulations contained in the *Generic Industrial Emissions and Immunity Standards EN 50081-2 and EN 50082-2*.
Class 2 is fulfilled when the converter is used with a Class A filter.
- Class 3 – Residential areas, commercial use, and light industry with Class B filter
The behavior limit values correspond to the regulations contained in the *Generic Industrial Emissions and Immunity Standards EN 50081-1 and EN 50082-1*.
Class 3 is fulfilled when the converter is used with a Class B filter.

Maximum cable length between the converter and the motor – Output reactor

Motor cables have a certain capacitance per unit length. The longer the cable, the greater the resulting cable capacitance. These capacitances are recharged with every commutation process. This results in a discharge current of the cable capacitances every time, which is superimposed on the actual motor current. These discharge currents must be provided by the converter in addition to the motor current. The greater the cable capacitance and, therefore, the cable length, the greater the amplitudes of these discharge currents. In order to ensure that the converter does not shut down with the fault message "Overcurrent" because of this effect with longer cables, smoothing reactors or filters must be provided at the converter output.

The catalog contains specifications relating to the maximum cable lengths between the converter and the motor, as well as the required options, such as output reactors or LC filters.

Group drives

The following applies to group drives (several motors operated in parallel on one converter):

The total of all motor currents must be lower than or equal to the rated converter current.

The maximum cable length between the converter and the motor must not be exceeded. The total of the individual motor cable lengths is the effective overall length.

Example:

5 motors, each with a 15 m motor cable to the converter, result in an overall length of 75 m.

Vertical lift drives

The rated converter current must be equal to or greater than the rated motor current. A check must also be performed to ensure that the motor's acceleration/deceleration current does not exceed the converter's limit values during the relevant periods.

Other drives

The rms value of the motor current must not exceed the rated converter current. A check must also be performed to ensure that the motor's acceleration/deceleration current does not exceed the converter's limit values during the relevant periods.

Electrical braking

The required electrical braking energy is a measure for determining/checking the braking chopper and the braking resistance.

The required braking power is calculated from the acceleration power (P_{Dyn}) and the static power (P_{Stat}).

$$P_{\text{Brake}} = (P_{\text{Dyn}} - P_{\text{Stat}}) \cdot \eta_{\text{Installation}}^2 \cdot \eta_{\text{Gearmotor}}$$

The maximum permissible braking power is calculated from the DC-link voltage U_{B} and the braking resistance value R_{B} .

$$P_{\text{B}} = \frac{U_{\text{B}}^2}{R_{\text{B}}}$$

Special environmental requirements

The standard temperature range is 0 to +40 °C.

If you intend to operate the frequency converter at higher or lower temperatures or in an environment containing abrasive materials, acidic, and/or humid air, refer to our product catalogs for advice. As not only control cabinet units, but also distributed converters can be used, we recommend that you consult our drive experts for assistance in finding the ideal solution for you, taking the overall concept into account.



Units and tables

SI units

SI is the abbreviation of the French *Système International d'Unités* (international system of units)

Table 3-1 SI units

Basic variable	SI base unit	
	Name	Letter/Abbreviation
Length	Meter	m
Mass	Kilogram	kg
Time	Second	s
Electrical current	Ampere	A
Temperature	Kelvin	K

Units derived from SI units

Table 3-2 Derived units

Variable	SI unit		Explanation
	Name	Letter/Abbreviation	
Plane angle	Radian	rad	1 rad = 1 m/m
Solid angle	Steradian	sr	1 sr = 1 m ² /m ²
Frequency of a periodic process	Hertz	Hz	1 Hz = 1 s ⁻¹
Activity of a radioactive substance	Becquerel	Bq	1 Bq = 1 s ⁻¹
Force	Newton	N	1 N = 1 kg·m/s ²
Pressure, mechanical tension	Pascal	Pa	1 Pa = 1 N/m ²
Energy, work, quantity of heat	Joule	J	1 J = 1 N · m = 1 W · s
Power, heat flow	Watt	W	1 W = 1 J/s
Absorbed dose	Gray	Gy	1 Gy = 1 J/kg
Electric charge, quantity of electricity	Coulomb	C	1 C = 1 A · s

Table 3-2 Derived units

Variable	SI unit		Explanation
	Name	Letter/ Abbreviation	
Electrical potential, electrical voltage	Volt	V	$1 \text{ V} = 1 \text{ J/C}$
Electrical capacitance	Farad	F	$1 \text{ F} = 1 \text{ C/V}$
Electrical resistance	Ohm	Ω	$1 \Omega = 1 \text{ V/A}$
Electrical conductance	Siemens	S	$1 \text{ S} = 1 \Omega^{-1}$
Magnetic flux	Weber	Wb	$1 \text{ Wb} = 1 \text{ V} \cdot \text{s}$
Magnetic flux density, magnetic induction	Tesla	T	$1 \text{ T} = 1 \text{ Wb/m}^2$
Inductance	Henry	H	$1 \text{ H} = 1 \text{ Wb/A}$
Temperature	Degrees Celsius	°C	$1 \text{ °C} = 1 \text{ K}$
Luminous flux	Lumen	lm	$1 \text{ lm} = 1 \text{ cd} \cdot \text{sr}$
Illuminance	Lux	lx	$1 \text{ lx} = 1 \text{ lm/m}^2$

Other units

Table 3-3 Other units

Variable	Unit name	Unit letter/abbrevi- ation	Definition
Plane angle	Minute	'	$1' = (1/60)^\circ$
	Second	"	$1'' = (1/60)'$ $1'' = (1/3,600)^\circ$
Volume	Liter	l	$1 \text{ l} = 1 \text{ dm}^3$
Mass	Metric ton	t	$1 \text{ t} = 10^3 \text{ kg} = 1 \text{ Mg}$
Pressure	Bar	bar	$1 \text{ bar} = 10^5 \text{ Pa}$
Area	Hectare	ha	$1 \text{ ha} = 10,000 \text{ m}^2$

Unit prefixes

Table 3-4 Unit prefixes

Name	Symbol	Factor
Atto	a	10^{-18}
Femto	f	10^{-15}
Pico	p	10^{-12}
Nano	n	10^{-9}
Micro	μ	10^{-6}
Milli	m	$10^{-3} = 0.001$
Centi	c	$10^{-2} = 0.01$
Deci	d	$10^{-1} = 0.1$
Deca	da	$10^1 = 10$
Hecto	h	$10^2 = 100$
Kilo	k	$10^3 = 1,000$
Mega	M	10^6
Giga	G	10^9
Tera	T	10^{12}
Peta	P	10^{15}
Exa	E	10^{18}

Rolling friction

Table 3-5 Rolling friction

Material combinations		f = Lever arm of the rolling friction in mm
Steel/steel		0.5
Wood/steel		1.2
Plastic/steel		2.0
Hard rubber/steel		7.0
Plastic/concrete		5.0
Hard rubber/concrete		10 to 20
Rubber/concrete		15 to 35
Polyurethane on steel ¹⁾	Ø 100 mm	0.75
	Ø 125 mm	0.9
	Ø 200 mm	1.5
	Ø 415 mm	3.1

- ¹⁾ The specified values are valid during operation. Considerably higher values may occur during startup so we recommend that you consult the wheel manufacturer.

Bearing friction

Table 3-6 Bearing friction

Bearing type	Lubrication	Friction coefficient μ_f
Steel in white metal	Well lubricated	0.02 to 0.06
	Poorly lubricated	0.08 to 0.1
Steel in bronze	Well lubricated	0.02 to 0.06
	Poorly lubricated	0.08 to 0.1
Pin in cast iron	Grease lubrication	0.1
Ball and roller bearing	Well lubricated	0.001 to 0.003
Trapezoidal spindle		0.2 to 0.3
Roller screw		0.005 to 0.01

Tracking friction

Table 3-7 Tracking friction

Type of bearing	Friction coefficient c
Wheels with roller bearings	0.003
Wheels with journal bearings	0.005
Lateral guide rollers	0.002

Stiction/Sliding friction

Table 3-8 Stiction and sliding friction

	μ_s	μ_{sl}
Steel on steel	0.12 to 0.6 (dry)	0.08 to 0.5 (dry)
	0.12 to 0.35 (greased)	0.04 to 0.25 (greased)
Wood on steel	0.45 to 0.75 (dry)	0.3 to 0.6 (dry)
Wood on wood	0.4 to 0.75 (dry)	0.3 to 0.5 (dry)
Plastic belts on steel	0.25 to 0.45 (dry)	0.18 to 0.35 (dry)
Steel on plastic	0.2 to 0.45 (dry)	0.18 to 0.35 (dry)
Ductile cast iron on steel	0.15 to 0.25 (dry)	0.08 to 0.12 (dry)
Polyurethane on steel	0.5 to 0.66 (dry)	0.3 to 0.4 (dry)
Steel on polyamide	0.18 to 0.36 (dry)	0.08 to 0.15 (dry)

Density ρ of different materialsTable 3-9 Density ρ of different materials

	Density ρ	Unit
Aluminum	2,700	kg/m ³
Cast iron	7,600	kg/m ³
Copper	8,960	kg/m ³
Brass	8,400 to 8,900	kg/m ³
Steel	7,860	kg/m ³
Zinc	7,130	kg/m ³
Tin	7,290	kg/m ³
Epoxy resin	1,200	kg/m ³
Rubber	920 to 990	kg/m ³
Phenolic resin, type 31	1,400	kg/m ³
Polyethylene	900 to 950	kg/m ³
PVC	1,300 to 1,400	kg/m ³

Temperature conversions

Table 3-10 Temperature conversions

		T_K in Kelvin (K) T_C in degrees Celsius (°C) T_F in degrees Fahrenheit (°F)
$T_C = T_K - 273.15$	$T_F = \frac{9}{5} \cdot T_K - 459.67$	$T_K = T_C + 273.15$
$T_C = \frac{5}{9} \cdot (T_F - 32)$	$T_F = \frac{9}{5} \cdot T_C + 32$	$T_K = \frac{9}{5} \cdot (T_F + 459.67)$

Length conversion table

Table 3-11 Length conversion table

	mm	cm	m	in	Foot ft	Yard yd	km	Furlong fur	mile	Nautical mile (sea mile, naut mile ¹⁾)
1 mm	1	10 ⁻¹	10 ⁻³	3.93701·10 ⁻²	3.28084·10 ⁻³	1.09361·10 ⁻³	10 ⁻⁶	4.9710·10 ⁻⁶	6.21371·10 ⁻⁷	5.39957·10 ⁻⁷
1 cm	10	1	10 ⁻²	3.93701·10 ⁻¹	3.28084·10 ⁻²	1.09361·10 ⁻²	10 ⁻⁵	4.9710·10 ⁻⁵	6.21371·10 ⁻⁶	5.39957·10 ⁻⁶
1 m	1,000	100	1	39.37	3.28084	1.09361	10 ⁻³	4.9710·10 ⁻³	6.21371·10 ⁻⁴	5.39957·10 ⁻⁴
1 in	25.4	2.54	2.54·10 ⁻²	1	8.33333·10 ⁻²	2.77778·10 ⁻²	2.54·10 ⁻⁵	1.2626·10 ⁻⁴	1.57828·10 ⁻⁵	1.37149·10 ⁻⁵
1 ft	304.8	30.48	0.3048	12	1	3.33333·10 ⁻¹	3.048·10 ⁻⁴	1.5152·10 ⁻³	1.89394·10 ⁻⁴	1.64579·10 ⁻⁴
1 yd	914.4	91.44	0.9144	36	3	1	9.144·10 ⁻⁴	4.5455·10 ⁻³	5.68182·10 ⁻⁴	4.93737·10 ⁻⁴
1 km	10 ⁶	10 ⁵	1,000	39,370.1	3,280.84	1,093.61	1	4.9710	6.21371·10 ⁻¹	5.39957·10 ⁻¹
1 fur	2.01168·10 ⁵	20,116.75	201.168	7,920	660	220	2.01168·10 ⁻¹	1	0.125	1.08622·10 ⁻¹
1 mile	1.60934·10 ⁶	160,934	1,609.34	63,360	5,280	1,760	1.60934	8	1	8.68976·10 ⁻¹
1 naut mile ¹⁾	1.852·10 ⁶	185,200	1,852	72,913.4	6,076.12	2,025.37	1.852	9.2062	1.15078	1

¹⁾ The following applies in the UK: 1 imp naut mile = 1,853 m

1 ft = 12 in, 1 yd = 3 ft = 36 in, 1 fathom = 2 yd,

1 rod = 1 pole = 1 perch = 5.5 yd = 5.0292 m,

1 link = 0.201168 m, 1 chain = 4 rods = 22 yd = 100 links = 20.1168 m,

1 fur = 10 chains = 220 yd = 1,000 links = 201.168 m,

1 mile = 8 fur = 80 chains = 1,760 yd = 1,609.344 m, 1 mil = 0.001 in = 0.0254 mm

Area conversion table

Table 3-12 Area conversion table

	cm ²	m ²	Are a	Hectare ha	km ²	Square inch in ²	Square foot ft ²	Square yard yd ²	Square mile mile ²	Rood	Acre
1 cm ²	1	10 ⁻⁴	10 ⁻⁶	10 ⁻⁸	10 ⁻¹⁰	1.55000· 10 ⁻¹	1.07639· 10 ⁻³	1.19599· 10 ⁻⁴	3.86102· 10 ⁻¹¹	9.88430· 10 ⁻⁸	2.47105· 10 ⁻⁸
1 m ²	10,000	1	10 ⁻²	10 ⁻⁴	10 ⁻⁶	1,550.00	10.7639	1.19599	3.86102· 10 ⁻⁷	9.88430· 10 ⁻⁴	2.47105· 10 ⁻⁴
1 a	10 ⁶	100	1	10 ⁻²	10 ⁻⁴	155,000	1,076.39	119.599	3.86102· 10 ⁻⁵	9.88430· 10 ⁻²	2.47105· 10 ⁻²
1 ha	10 ⁸	10,000	100	1	10 ⁻²	1.55000· 10 ⁷	107,639	11,959.9	3.86102· 10 ⁻³	9.88430	2.47105
1 km ²	10 ¹⁰	10 ⁶	10,000	100	1	1.55000· 10 ⁹	1.07639· 10 ⁷	1.19599· 10 ⁶	3.86102· 10 ⁻¹	988.43	247.105
1 in ²	6.45160	6.45160· 10 ⁻⁴	6.45160· 10 ⁻⁶	6.45160· 10 ⁻⁸	6.45160· 10 ⁻¹⁰	1	6.94444· 10 ⁻³	7.71605· 10 ⁻⁴	2.49098· 10 ⁻¹⁰	6.37692· 10 ⁻⁷	1.59423· 10 ⁻⁷
1 ft ²	929.030	9.29030· 10 ⁻²	9.29030· 10 ⁻⁴	9.29030· 10 ⁻⁶	9.29030· 10 ⁻⁸	144	1	1.11111· 10 ⁻¹	3.58701· 10 ⁻⁸	9.18272· 10 ⁻⁵	2.29568· 10 ⁻⁵
1 yd ²	8,361.27	8.36127· 10 ⁻¹	8.36127· 10 ⁻³	8.36127· 10 ⁻⁵	8.36127· 10 ⁻⁷	1,296	9	1	3.22831· 10 ⁻⁷	8.26448· 10 ⁻⁴	2.06612· 10 ⁻⁴
1 mile ²	2.58999· 10 ¹⁰	2.58999· 10 ⁶	25,899.9	258.999	2.58999	4.01449· 10 ⁹	2.78784· 10 ⁷	3.09760· 10 ⁶	1	2,560	640
1 rood	1.01172· 10 ⁷	1,011.72	10.1172	1.01172· 10 ⁻¹	1.01172· 10 ⁻³	1.56816· 10 ⁶	10,890.0	1,210	3.90625· 10 ⁻⁴	1	0.25
1 acre	4.04686· 10 ⁷	4,046.86	40.4686	4.04686· 10 ⁻¹	4.04686· 10 ⁻³	6.27264· 10 ⁶	43,560.0	4,840	1.56250· 10 ⁻³	4	1

$$1 \text{ acre} = 4 \text{ roods} = 10 \text{ sq. chains} = 4,840 \text{ yd}^2 = 4,046.856 \text{ m}^2$$

Volume conversion table

Table 3-13 Volume conversion table

	cm ³	Liter dm ³ = 1	Cubic inch in ³	Cubic foot ft ³	Cubic yard yd ³	Fluid ounce US fl oz	Fluid ounce imp fl oz	Gallon US gal	Gallon imp gal	Pint imp pint
1 cm ³	1	10 ⁻³	6.10237· 10 ⁻²	3.53147· 10 ⁻⁵	1.30795· 10 ⁻⁶	3.38140· 10 ⁻²	3.51951· 10 ⁻²	2.64172· 10 ⁻⁴	2.19969· 10 ⁻⁴	1.75975· 10 ⁻³
1 dm ³ = 1 l	1,000	1	61.0237	3.53147· 10 ⁻²	1.30795· 10 ⁻³	33.8140	35.1951	2.64172· 10 ⁻¹	2.19969· 10 ⁻¹	1.75975
1 in ³	16.3871	1.63871 10 ⁻²	1	5.78704· 10 ⁻⁴	2.14335· 10 ⁻⁵	5.54113· 10 ⁻¹	5.76744· 10 ⁻¹	4.32900· 10 ⁻³	3.60465· 10 ⁻³	2.88372· 10 ⁻²
1 ft ³	28,316.8	28.3168	1,728	1	3.70370· 10 ⁻²	957.506	996.614	7.48052	6.22884	49.8307
1 yd ³	764,555	764.555	46,656	27	1	25,852.7	26,908.6	201.974	168.179	1,345.43
1 US fl oz	29.5735	2.95735· 10 ⁻²	1.80469	1.04438· 10 ⁻³	3.86807· 10 ⁻⁵	1	1.04084	7.8125· 10 ⁻³	6.50527· 10 ⁻³	5.20421· 10 ⁻²
1 imp fl oz	28.4131	2.84131· 10 ⁻²	1.73387	1.00340· 10 ⁻³	3.71629· 10 ⁻⁵	9.60760· 10 ⁻¹	1	7.50594· 10 ⁻³	6.25·10 ⁻³	5·10 ⁻²
1 US gal	3,785.41	3.78541	231	1.33681· 10 ⁻¹	4.95113· 10 ⁻³	128	133.228	1	8.32674· 10 ⁻¹	6.66139
1 imp gal	4,546.09	4.54609	277.419	1.60544· 10 ⁻¹	5.94606· 10 ⁻³	153.722	160	1.20095	1	8
1 imp pint	568.261	5.68261 10 ⁻¹	34.6774	2.00680· 10 ⁻²	7.43258· 10 ⁻⁴	19.2152	20	1.50119· 10 ⁻¹	1.25·10 ⁻¹	1

UK units:

- 1 bushel = 8 gallons = 36.3687 dm³
- 1 gallon = 4 quarts = 4.5461 dm³
- 1 quart = 2 pints = 1.1365 dm³
- 1 pint = 4 gills = 0.5682 dm³
- 1 gill = 5 fluid ounces = 142.065 cm³
- 1 fluid ounce = 8 fluid drams = 28.4131 cm³
- 1 fluid dram = 60 minims = 3.5516 cm³

Other units:

- 1 US barrel = 42 US gallons = 158.987 dm³
- Gross register ton (GRT); unit for designating the internal volume of a ship,
1 GRT = 2.832 m³

Velocity conversion table

Table 3-14 Velocity conversion table

Unit	Meter per second	Kilometer per hour	Mile per hour	Knot	Foot per second
	m/s	km/h	mile/h	kn	ft/s
1 m/s	1	3.6	2.2369	1.9438	3.2808
1 km/h	0.2777	1	0.6214	0.5400	0.9113
1 mile/h	0.4470	1.6093	1	0.8690	1.4667
1 kn	0.5144	1.852	1.1508	1	1.6878
1 ft/s	0.3048	1.0973	0.6818	0.5925	1

Knot = 1 int. nautical mile per hour

Standard acceleration due to gravity: $g_n = 9.80665 \text{ m/s}^2 = 32.17405 \text{ ft/s}^2$ **Mass conversion table**

Table 3-15 Mass conversion table

Unit	Kilogram	Gram	Metric ton	Long ton	Short ton	Hundred-weight	Pound (av)*	Pound (tr)*		Metr. carat	Ounce (av)*	Ounce (tr)*
	kg	g	t	ton	sh tn	cwt	lb (av)*	lb (tr)*	slug	ct	oz (av)*	oz (tr)*
1 kg	1	$1 \cdot 10^3$	$1 \cdot 10^{-3}$	$9.842 \cdot 10^{-4}$	$1.1023 \cdot 10^{-3}$	$1.968 \cdot 10^{-2}$	2.2046	2.6792	$68.5218 \cdot 10^{-3}$	$5 \cdot 10^3$	35.273	32.1505
1 g	$1 \cdot 10^{-3}$	1	$1 \cdot 10^{-6}$	$9.842 \cdot 10^{-7}$	$1.1023 \cdot 10^{-6}$	$1.968 \cdot 10^{-5}$	$2.2046 \cdot 10^{-3}$	$2.6792 \cdot 10^{-3}$	$68.5218 \cdot 10^{-6}$	5	$35.2740 \cdot 10^{-3}$	$32.1507 \cdot 10^{-3}$
1 t	1,000	$1 \cdot 10^6$	1	0.9842	1.1023	19.6841	2,204.6	2,679.35	68.5218	$5 \cdot 10^6$	$3.5273 \cdot 10^4$	$3.2150 \cdot 10^4$
1 ton	1,016.057	$1.016 \cdot 10^6$	1.0160	1	1.12	20	2,240	2,722.22	69.622	$5.08 \cdot 10^6$	$3.584 \cdot 10^4$	$3.2668 \cdot 10^4$
1 sh tn	907.194	$9.072 \cdot 10^5$	0.9072	0.8929	1	17.8571	2,000	2,430.55	62.1625	$4.536 \cdot 10^6$	$3.2 \cdot 10^4$	$2.9168 \cdot 10^4$
1 cwt	50.80	$5.08 \cdot 10^4$	$50.8023 \cdot 10^{-3}$	$50 \cdot 10^{-3}$	$56 \cdot 10^{-3}$	1	112	136.111	3.4811	$2.54 \cdot 10^5$	$1.792 \cdot 10^3$	$1.6334 \cdot 10^3$
1 lb (av)	0.4536	453.592	$4.536 \cdot 10^{-4}$	$4.464 \cdot 10^{-4}$	$5 \cdot 10^{-4}$	$8.9268 \cdot 10^{-3}$	1	1.2153	$31.0809 \cdot 10^{-3}$	2,267.968	16	14.584
1 lb (tr)	0.3732	373.26	$3.7324 \cdot 10^{-4}$	$3.6734 \cdot 10^{-4}$	$4.1143 \cdot 10^{-4}$	$7.3459 \cdot 10^{-3}$	0.8229	1	$25.5752 \cdot 10^{-3}$	1,866.31	13.1664	12
1 slug	14.5939	$1.4594 \cdot 10^4$	$1.459 \cdot 10^{-2}$	$1.436 \cdot 10^{-2}$	$1.6087 \cdot 10^{-2}$	$2.8721 \cdot 10^{-1}$	32.1740	39.1	1	$7.2973 \cdot 10^4$	514.806	469.2
1 ct	$2 \cdot 10^{-4}$	0.2	$2 \cdot 10^{-7}$	$1.9684 \cdot 10^{-7}$	$2.2046 \cdot 10^{-7}$	$3.936 \cdot 10^{-6}$	$4.4092 \cdot 10^{-4}$	$5.3584 \cdot 10^{-4}$	$13.70436 \cdot 10^{-6}$	1	$7.0548 \cdot 10^{-3}$	$6.4301 \cdot 10^{-3}$
1 oz (av)	$2.835 \cdot 10^{-2}$	28.3495	$2.83495 \cdot 10^{-5}$	$2.7902 \cdot 10^{-5}$	$3.125 \cdot 10^{-5}$	$5.579 \cdot 10^{-4}$	$6.25 \cdot 10^{-2}$	$7.5955 \cdot 10^{-2}$	$1.9426 \cdot 10^{-3}$	141.748	1	0.9115
1 oz (tr)	$3.1103 \cdot 10^{-2}$	31.1035	$3.1103 \cdot 10^{-5}$	$3.0612 \cdot 10^{-5}$	$3.4286 \cdot 10^{-5}$	$6.122 \cdot 10^{-4}$	$6.8571 \cdot 10^{-2}$	$8.3333 \cdot 10^{-2}$	$2.131 \cdot 10^{-3}$	155.517	1.0971	1

av = avoirdupois = standard commercial weight

tr = troy = fine weight for precious metals and stones

Other units:

- 1 oz (av) = 437.5 grains (av) = 16 drams (av)
- 1 dram (av) = 1.7718 g
- 1 grain (av) = 1/7,000 lb (av) = 64.7989 mg
- 1 dram = 60 grains (av) = 60/7,000 lb (av) = 3.8879 g
- 1 oz (tr) = 480/7,000 lb (av)
- 1 lb (tr) = 5,760/7,000 lb (av)
- tdw: ton deadweight, unit of mass specifying how much cargo or burden ships can carry safely. 1 tdw = 1,016 kg

Pressure conversion table

Table 3-16 Pressure conversion table

Unit	Newton per square millimeter	Kilopond per square centimeter	Kilopond per square millimeter	Pound-force per square inch (psi)	Millibar	Standard atmosphere	Torr (1 Torr \approx 1 mm Hg)	Millimeter of water column (1 mm WC = 1 kp/m ²)	Inch of mercury	Foot of water	Pascal	
	N/mm ²	kp/cm ²	kp/mm ²	lb/in ²	mbar	atm	Torr	mm WC			Pa	bar
1 N/mm ²	1	10.1972	0.101972	145.038	1·10 ⁴	9.86842	7.5·10 ⁻³	10.1972·10 ⁴	295.2999	334.5453	10 ⁶	10
1 kp/cm ²	98.0665·10 ⁻³	1	0.01	14.2233	980.665	0.96784	735.559	10 ⁴	28.95949	32.8082	98,068	0.98068
1 kp/mm ²	9.80665	100	1	1,422.33	98,066.5	96.78408	73,555.9	10 ⁶	2,895.949	3,280.82	9.8068·10 ⁶	98.068
1 lb/in ²	6.8948·10 ⁻³	70.3070·10 ⁻³	0.70307·10 ⁻³	1	68.94759	68.04599·10 ⁻³	51.71495	703.07237	2.03602	2.3066	6.894759·10 ³	68.94759·10 ⁻³
1 mbar	1·10 ⁻⁴	1.0197·10 ⁻³	10.197·10 ⁻⁶	14.50377·10 ⁻³	1	0.987·10 ⁻³	0.7501	10.1972	29.52999·10 ⁻³	33.45453·10 ⁻³	100	1·10 ⁻³
1 atm	0.10126	1.03322	10.3322·10 ⁻³	14.69607	1.01325·10 ³	1	760	10.332276·10 ³	29.92153	33.89810	0.101326·10 ⁶	1.01325
1 Torr	0.133231·10 ⁻³	1.3595·10 ⁻³	1.3595·10 ⁻⁵	19.33694·10 ⁻³	1.333224	1.31579·10 ⁻³	1	13.5951	39.37043·10 ⁻³	44.60276·10 ⁻³	133.3234	1.333234·10 ⁻³
1 mm WC	9.8066·10 ⁻⁶	10 ⁻⁴	1·10 ⁻⁶	1.4223·10 ⁻³	98.0665·10 ⁻³	96.78408·10 ⁻⁶	73.5559·10 ⁻³	1	2.89589·10 ⁻³	3.28075·10 ⁻³	9.80661	98.0661·10 ⁻⁶
1 in of mercury	3.3863878·10 ⁻³	34.53167·10 ⁻³	345.3167·10 ⁻⁶	0.49115	33.863878	33.42105·10 ⁻³	25.4	345.316744	1	1.1329	3.3863878·10 ³	33.863878·10 ⁻³
1 foot of water	2.989165·10 ⁻³	30.4811·10 ⁻³	304.811·10 ⁻⁶	0.433538	29.89165	29.50076·10 ⁻³	22.42058	304.811	0.8827	1	2.989165·10 ³	29.89165·10 ⁻³
1 Pa	1·10 ⁻⁶	1.02·10 ⁻⁵	0.10197·10 ⁻⁶	0.145·10 ⁻³	0.01	9.87·10 ⁻⁶	7.5·10 ⁻³	0.101972	0.2952999·10 ⁻³	0.3345453·10 ⁻³	1	1·10 ⁻⁵
1 bar	0.1	1.02	0.10197·10 ⁻¹	14.50377	1·10 ³	0.987	750	10.1972·10 ³	29.52999	33.45453	10 ⁵	1

Other unit:

- Technical atmosphere: 1 at = 1 kp/cm²

Force conversion table

Table 3-17 Force conversion table

Unit	Newton N	Kilopond kp	Pound-force lb (or lbf)	Poundal pdl
1 N	1	0.101972	0.22481	7.2230
1 kp	9.80665	1	2.20462	70.9316
1 lb (lbf)	4.44822	0.45362	1	32.1740
1 pdl	0.13826	$14.0981 \cdot 10^{-3}$	$31.0810 \cdot 10^{-3}$	1

Other unit:

- Long ton force: 1 tonf = 2,240 lb = 9.964 kN \approx 10 kN

Power conversion table

Table 3-18 Power

Unit	Watt W	Kilowatt kW	Kilogram calorie per hour kcal/h	British ther- mal unit per hour Btu/h	Foot pound force per second ft·lb/s	Megapond meter per second Mp m/s	Metric horsepower PS	Horsepower hp
1 W	1	0.001	0.8593	3.4121	0.7376	$0.10197 \cdot 10^{-3}$	$1.3596 \cdot 10^{-3}$	$1.3410 \cdot 10^{-3}$
1 kW	1,000	1	859.3	3,412.1	737.6	0.10197	1.3596	1.3410
1 kcal/h	1.163	$1.163 \cdot 10^{-3}$	1	3.9683	0.8578	$0.11859 \cdot 10^{-3}$	$1.58 \cdot 10^{-3}$	$1.5596 \cdot 10^{-3}$
1 Btu/h	0.2931	$0.2931 \cdot 10^{-3}$	0.2520	1	0.2162	$29.8874 \cdot 10^{-6}$	$0.398 \cdot 10^{-3}$	$0.393 \cdot 10^{-3}$
1 ft·lb/s	1.3558	$1.3558 \cdot 10^{-3}$	1.1658	4.6263	1	$0.13825 \cdot 10^{-3}$	$1.8433 \cdot 10^{-3}$	$1.8181 \cdot 10^{-3}$
1 Mp m/s	9,806.65	9.80665	8,432	33,461	7,233.385	1	13.3333	13.1509
1 PS	735.5	0.73550	632.015	2,509.6	542.5	0.075	1	0.9863
1 hp	745.69	0.74570	641.19	2,544.4	550	0.07604	1.0139	1

Energy conversion table

Table 3-19 Energy conversion table

Unit	Joule J	Kilowatt hour kWh	Kilopond meter kp m	Kilogram calorie kcal	British thermal unit BTU	Metric ton of coal equivalent TCE	Foot pound force ft·lb
1 J	1	$2.77778 \cdot 10^{-7}$	0.101972	$2.38846 \cdot 10^{-4}$	$9.47817 \cdot 10^{-4}$	$34.12 \cdot 10^{-9}$	0.7376
1 kWh	$3.6 \cdot 10^6$	1	367,098	859.845	3,412.14	0.1228	$2.655 \cdot 10^6$
1 kp m	9.80665	$2.72407 \cdot 10^{-6}$	1	$2.34228 \cdot 10^{-3}$	$9.29491 \cdot 10^{-3}$	$0.3346 \cdot 10^{-6}$	7.2330
1 kcal	4,186.8	$1.163 \cdot 10^{-3}$	426.935	1	3.9683	$1.429 \cdot 10^{-4}$	3,088.18
1 BTU	1,055.06	$2.93071 \cdot 10^{-4}$	107.586	0.252	1	$36.00 \cdot 10^{-6}$	778.21
1 TCE	$29.308 \cdot 10^6$	8.141	$2.989 \cdot 10^6$	7,000	$27.78 \cdot 10^3$	1	$21.617 \cdot 10^6$
1 ft·lb	1.35582	$3.766 \cdot 10^{-7}$	0.13826	$3.23815 \cdot 10^{-4}$	$1.285 \cdot 10^{-3}$	$46.25 \cdot 10^{-9}$	1

Moment of inertia conversion table

Table 3-20 Moment of inertia conversion table

	kg cm ²	kp cm s ²	kg m ²	kp m s ²	oz in ²	ozf in s ²	lb in ²	lbf in s ²	lb ft ²	lbf ft s ²
1 kg cm ²	1	$1.01972 \cdot 10^{-3}$	10^{-4}	$1.01972 \cdot 10^{-5}$	5.46748	$1.41612 \cdot 10^{-2}$	$3.41717 \cdot 10^{-1}$	$8.85075 \cdot 10^{-4}$	$2.37304 \cdot 10^{-3}$	$7.37562 \cdot 10^{-5}$
1 kp cm s ²	980.665	1	$9.80665 \cdot 10^{-2}$	10^{-2}	5,361.76	13.8874	335.110	$8.67962 \cdot 10^{-1}$	2.32715	$7.23301 \cdot 10^{-2}$
1 kg m ²	10^4	10,1972	1	$1.01972 \cdot 10^{-1}$	54,674.8	141.612	3,417.17	8.85075	23.7304	$7.37562 \cdot 10^{-1}$
1 kp m s ²	98,066.5	100	9.80665	1	536,176	1,388.74	33,511.0	86.7962	232.715	7.23301
1 oz in ²	$1.82900 \cdot 10^{-1}$	$1.86506 \cdot 10^{-4}$	$1.82900 \cdot 10^{-5}$	$1.86506 \cdot 10^{-6}$	1	$2.59008 \cdot 10^{-3}$	$6.25 \cdot 10^{-2}$	$1.61880 \cdot 10^{-4}$	$4.34028 \cdot 10^{-4}$	$1.34900 \cdot 10^{-5}$
1 ozf in s ²	70.6155	$7.20078 \cdot 10^{-2}$	$7.06155 \cdot 10^{-3}$	$7.20078 \cdot 10^{-4}$	386.089	1	24.1305	$6.25 \cdot 10^{-2}$	$1.67573 \cdot 10^{-1}$	$5.20833 \cdot 10^{-3}$
1 lb in ²	2.92640	$2.98409 \cdot 10^{-3}$	$2.92640 \cdot 10^{-4}$	$2.98409 \cdot 10^{-5}$	16	$4.14413 \cdot 10^{-2}$	1	$2.59008 \cdot 10^{-3}$	$6.94444 \cdot 10^{-3}$	$2.15840 \cdot 10^{-4}$
1 lbf in s ²	1,129.85	1.15212	$1.12985 \cdot 10^{-1}$	$1.15212 \cdot 10^{-2}$	6,177.42	16	386.089	1	2.68117	$8.33333 \cdot 10^{-2}$
1 lb ft ²	421.401	$4.29710 \cdot 10^{-1}$	$4.21401 \cdot 10^{-2}$	$4.29710 \cdot 10^{-3}$	2,304.00	5.96754	144	$3.72971 \cdot 10^{-1}$	1	$3.10810 \cdot 10^{-2}$
1 lbf ft s ²	13,558.2	13.8255	1.35582	$1.38255 \cdot 10^{-1}$	74,129.0	192	4,633.06	12	32.1740	1

The numerical value of the flywheel effect GD^2 in kpm^2 is 4 times as large as the value of moment of inertia J in kgm^2 .

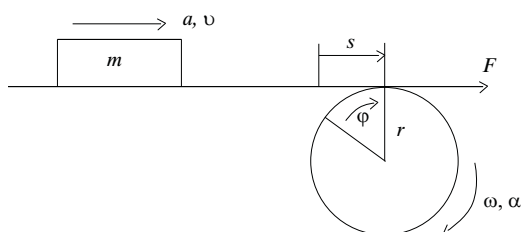
Torque conversion table

	N cm	N m	kp cm	kp m	p cm	ozf in	lbf in	lbf ft
1 N cm	1	10^{-2}	$1.01972 \cdot 10^{-1}$	$1.01972 \cdot 10^{-3}$	101.972	1.41612	$8.85075 \cdot 10^{-2}$	$7.37562 \cdot 10^{-3}$
1 N m	100	1	10.1972	$1.01972 \cdot 10^{-1}$	10,197.2	141.612	8.85075	$7.37562 \cdot 10^{-1}$
1 kp cm	9.80665	$9.80655 \cdot 10^{-2}$	1	10^{-2}	1,000	13.8874	$86.7962 \cdot 10^{-2}$	$7.23301 \cdot 10^{-2}$
1 kp m	980.665	9.80665	100	1	10^5	1,388.74	86.7962	$7.23301 \cdot 10^{-1}$
1 p cm	$9.80665 \cdot 10^{-3}$	$9.80665 \cdot 10^{-5}$	10^{-3}	10^{-5}	1	$1.38874 \cdot 10^{-2}$	$8.67962 \cdot 10^{-4}$	$7.23301 \cdot 10^{-5}$
1 ozf in	$7.06155 \cdot 10^{-1}$	$7.06155 \cdot 10^{-3}$	$7.20078 \cdot 10^{-2}$	$7.20078 \cdot 10^{-4}$	72.0078	1	$6.25 \cdot 10^{-2}$	$5.20833 \cdot 10^{-3}$
1 lbf in	11.2985	$1.12985 \cdot 10^{-1}$	1.15212	$1.15212 \cdot 10^{-2}$	1,152.12	16	1	$8.33333 \cdot 10^{-2}$
1 lbf ft	135.582	1.35582	13.8255	$1.38255 \cdot 10^{-1}$	13,825.5	192	12	1



Formulas

4.1 Comparison of the calculation variables involved in translation and rotation

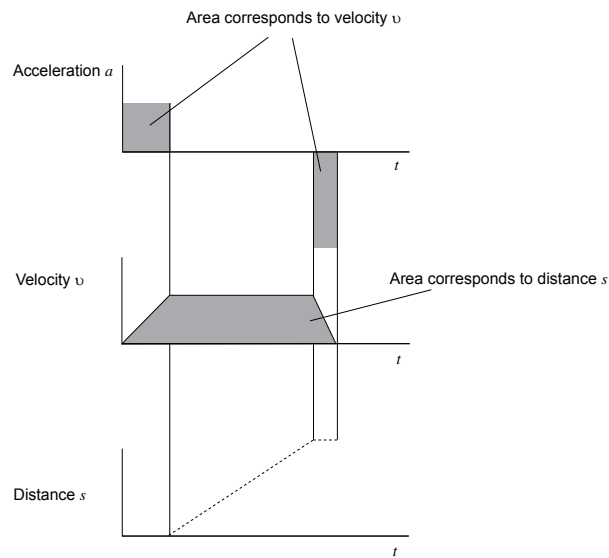


aMasse_Transportband.eps

Fig. 4-1 Example: Acceleration of a mass on a conveyor belt at a constant acceleration

$s = \varphi \cdot r$	Distance	[m]	φ : Radian angle
$v = \omega \cdot r$	Velocity	[m/s]	ω : Angular velocity
			$\omega = 2 \cdot \pi \cdot n$:
			n : Rotational frequency (or speed) in s^{-1}
$a = \alpha \cdot r$	Acceleration	$[m/s^2]$	α : Angular acceleration s^{-2}

Graphical description of acceleration, velocity, and distance



a-v-s-Grafik.eps

Fig. 4-2 Acceleration/velocity/distance diagram

Table 4-1 Comparison of translation and rotation

Translation			Rotation		
$a = \frac{dv}{dt}$	[m/s ²]	Acceleration	$\alpha = \frac{d\omega}{dt}$	[s ⁻²]	Angular acceleration
$v = \frac{ds}{dt}$	[m/s]	Velocity	$\omega = \frac{d\phi}{dt}$	[s ⁻¹]	Angular velocity
s	[m]	Distance	ϕ	[rad]	Radian angle
m	[kg]	Mass	J	[kgm ²]	Moment of inertia
F	[N]	Force	M	[Nm]	Torque
$F = m \cdot a$	[N]	Accel. force	$M = J \cdot \alpha$	[Nm]	Accel. torque
$P = F \cdot v$	[W]	Power	$P = M \cdot \omega$	[W]	Power
$W = F \cdot s$	[Ws]	Work (= energy)	$W = M \cdot \phi$	[Ws]	Work (= energy)
$W = \frac{1}{2} \cdot m \cdot v^2$	[Ws]	Kinetic energy	$W = \frac{1}{2} \cdot J \cdot \omega^2$	[Ws]	Kinetic energy

Comparison of the calculation variables involved in translation and rotation

Translation			Rotation	
	$v = \text{konstant}$	$a = \text{konstant}$	$\omega = \text{konstant}$	$\alpha = \text{konstant}$
Distance	$s = v \cdot t$	$s = \frac{v \cdot t}{2} = \frac{v^2}{2 \cdot a}$	$\varphi = \omega \cdot t$	$\varphi = \frac{\omega^2}{2 \cdot \alpha}$
Velocity	$v = \frac{s}{t}$	$v = a \cdot t$	$\omega = \frac{\varphi}{t}$	$\omega = \alpha \cdot t$
Acceleration	$a = 0$	$a = \frac{v^2}{2 \cdot s}$	$\alpha = 0$	$\alpha = \frac{\omega^2}{2 \cdot \varphi}$
Time	$t = \frac{s}{v}$	$t = \frac{2 \cdot s}{v}$	$t = \frac{\varphi}{\omega}$	$t = \frac{2 \cdot \varphi}{\omega}$

4.2 Moments of inertia

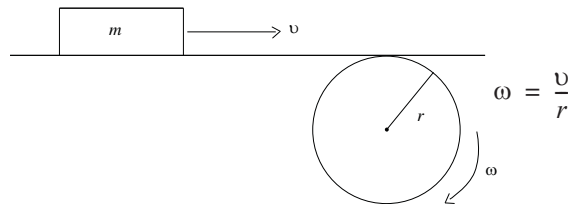
The moment of inertia J specifies the inertia in terms of a rotary motion.

Since a motor performs rotary movement, it is usual to make calculations using the rotary system. It may be necessary, therefore, to convert a mass which is subject to translatory movement into a rotary equivalent, called the moment of inertia.

Converting masses subject to translatory movement into moments of inertia

The conversion is performed by equating the kinetic energy with the kinetic energy of rotation.

$$\frac{1}{2} \cdot m \cdot v^2 = \frac{1}{2} \cdot J \cdot \omega^2 \quad \rightarrow \quad J = m \cdot \left(\frac{v}{\omega} \right)^2$$



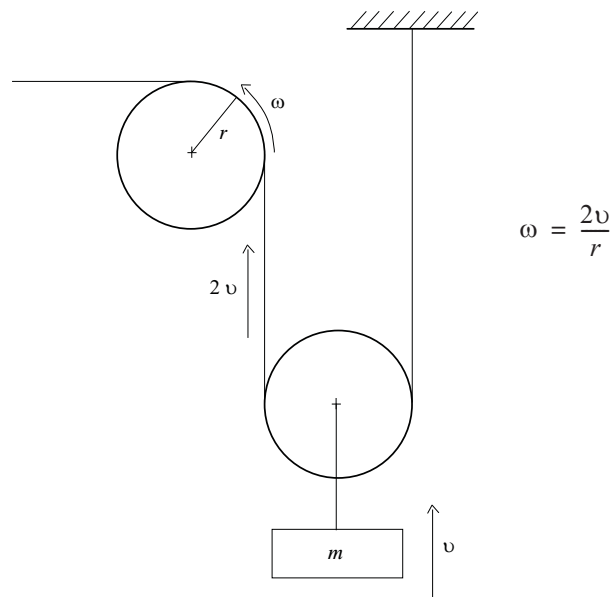
Translat-Umrechnung.eps

Fig. 4-3 Schematic: Translation \leftrightarrow rotation conversion

Moment of inertia of mass m in relation to the roller:

$$J = m \cdot r^2$$

m	in kg
r	in m
J	in kgm^2



Hub.eps

Fig. 4-4 Vertical lift drive with a reeve of 1:2

Moment of inertia of mass m in relation to the fixed roller:

$$J = \frac{1}{4} \cdot m \cdot r^2$$

m in kg
 r in m
 J in kgm^2

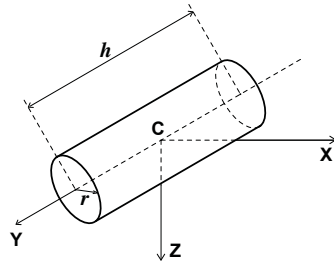
Converting moments of inertia in relation to the motor shaft

$$J_{\text{load}}^* = \frac{J_{\text{load}}}{i^2}$$

J Load moment of inertia in kgm^2
 J^* Load moment of inertia in relation to the motor shaft in kgm^2
 i Gear ratio

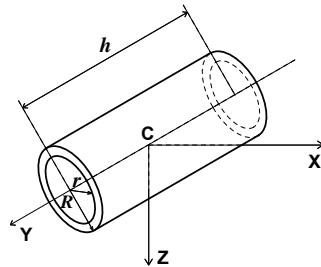
Moments of inertia of different objects

J	in kgm^2
l, r, R, r_m	in m
m	in kg
C	Center of gravity of the object



Full cylinder

$$J_Y = \frac{1}{2} \cdot m \cdot r^2$$

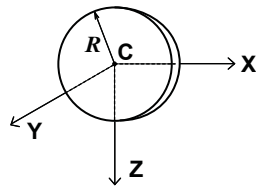


Hollow cylinder

$$J_Y = \frac{1}{2} \cdot m \cdot (R^2 + r^2)$$

Thin-walled hollow cylinder $r \approx R \approx r_m$

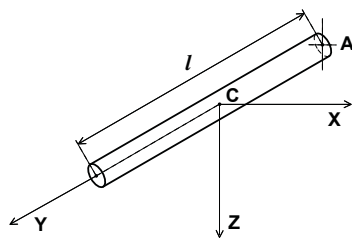
$$J_Y = m \cdot r_m^2$$



Circular disk

$$J_X = J_Z = \frac{1}{4} \cdot m \cdot R^2$$

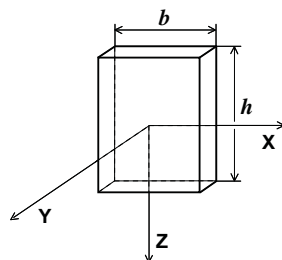
$$J_Y = \frac{1}{2} \cdot m \cdot R^2$$



Thin rod

$$J_X = J_Z = \frac{1}{12} \cdot m \cdot l^2, J_Y = 0$$

$$J_A = \frac{1}{3} \cdot m \cdot l^2$$



Thin rectangular plate

$$J_X = \frac{1}{12} \cdot m \cdot h^2$$

$$J_Y = \frac{1}{12} \cdot m \cdot (h^2 + b^2)$$

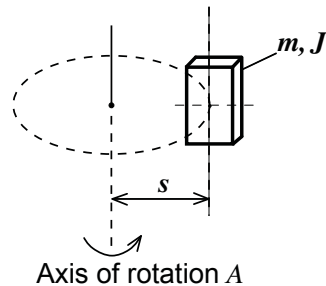
$$J_Z = \frac{1}{12} \cdot m \cdot b^2$$

Parallel axis theorem

The parallel axis theorem is applied if an object's axis of rotation does not coincide with the axis which runs through the object's centre of gravity. The object's moment of inertia about the axis of rotation A can be expressed as follows:

$$J_A = J + m \cdot s^2$$

J is the object's moment of inertia in relation to the axis which runs through its center of gravity, parallel to the axis of rotation.



Steiner.eps

Fig. 4-5 Graphic representation of the parallel axis theorem

4.3 Resisting forces

This refers to forces which act against the direction of movement. They are divided into static and dynamic resisting forces.

4.3.1 Static resisting forces

Tractive resistance for traction drives

$$F_W = m \cdot g \cdot w_F \quad \text{Resisting force in N}$$

$$m \quad \text{Load in kg}$$

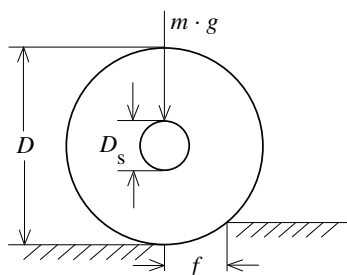
$$g \quad \text{Acceleration due to gravity} = 9.81 \frac{\text{m}}{\text{s}^2}$$

$$w_F \quad \text{Specific tractive resistance}$$

If the w_F factor is not known, it can be calculated as follows:

$$w_F = \frac{2}{D} \cdot \left(\frac{D_s}{2} \cdot \mu_r + f \right) + c$$

D	Wheel diameter
D_s	Shaft diameter for bearing friction
μ_r	Coefficient for bearing friction
f	Lever arm of the rolling friction
c	Coefficient for wheel flange friction



Rad.eps

Fig. 4-6 Wheel

The resisting force F_W comprises:

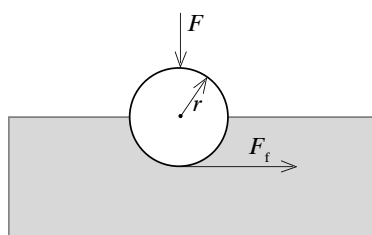
$$F_{W1} = m \cdot g \cdot \mu_r \cdot \frac{D_s}{D} \quad \text{Bearing friction}$$

Roller bearings: $\mu_r = 0.005$
 Journal bearings: $\mu_r = 0.008$

$$F_{W2} = m \cdot g \cdot \frac{2 \cdot f}{D} \quad \text{Rolling friction}$$

$$F_{W3} = m \cdot g \cdot c \quad \text{Tracking friction}$$

General bearing friction

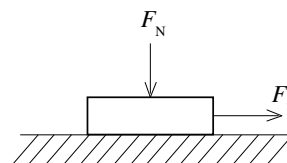
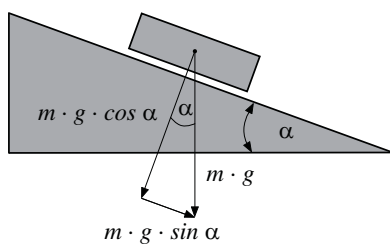


$$F_f = \mu \cdot F$$

Lagerreibung.eps

Fig. 4-7 Bearing friction

Sliding friction, stiction	F_f	Friction force [N]
$F_f = \mu \cdot F_N$	μ	Friction value
Weight force/component	F_N	Clamping force (normal force) perpendicular to the surface [N]
$F_N = m \cdot g \cdot \cos \alpha$	m	Mass
If $\alpha = 0$, then $F_N = m \cdot g$	g	Acceleration due to gravity [= 9.81 m/s ²]
Grade resistance	α	Angle of lead [°]
$F = m \cdot g \cdot \sin \alpha$		



Hangabtrieb.eps

Fig. 4-8 Grade resistance

4.3.2 Dynamic resisting forces

Acceleration force

$$F = m \cdot a$$

4.4 Torque, power, energy (= work)

Torque in Nm

$$M = F \cdot r = \frac{F \cdot D}{2}$$

Acceleration torque in Nm

$$M = J \cdot \alpha$$

Power in W

$$P = F \cdot v \text{ Translatory}$$

$$P = M \cdot \omega \text{ Rotary}$$

Kinetic energy in Js

Energy which an object has accumulated with mass m and velocity v .

$$W = \frac{1}{2} \cdot m \cdot v^2 \text{ Translatory}$$

$$W = \frac{1}{2} \cdot J \cdot \omega^2 \text{ Rotary}$$

Potential energy in Js

Energy which an object has accumulated with mass m at height h .

$$W = m \cdot g \cdot h$$

F	Resisting force in N (static and/or dynamic)
r or D	Radius or diameter of the drive element in m
v	Velocity in m/s
J	Moment of inertia of the mass to be accelerated in kgm^2
α	Angular acceleration in $1/\text{s}^2$
ω	Angular velocity in $1/\text{s}$
h	Height in m, vertical direction (direction of the force of gravity)

4.5 Efficiencies

If several efficiencies (of different parts of the system, for example) have to be taken into account, they have to be multiplied in order to obtain the total efficiency.

$$\eta_{\text{total}} = \eta_1 \cdot \eta_2 \cdot \eta_{\text{gearbox}}$$

4.6 Formulas for a motion sequence at a constant load torque

J_M	Moment of inertia of the motor plus brake in kgm^2
J_{load}	Moment of inertia of the load in kgm^2
η	Efficiency of the system
n_M	Motor speed in min^{-1}
n_S	Synchronous motor speed in min^{-1}
M_H	Average acceleration torque of the motor connected to the supply system in Nm
M_B	Braking torque in Nm
M_L	Static load torque in Nm without taking the efficiency into account
v	Velocity in m/s
t_1	Engaging time of the brake in s
z_N	Number of operations in h^{-1}
L	Braking energy until adjustment or replacement in J

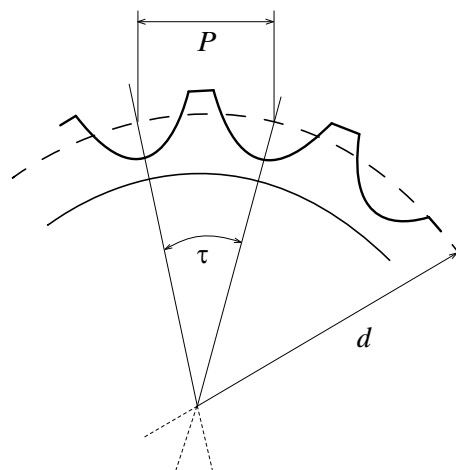
Variable	Horizontal motion and upward vertical motion (motoring operation)	Downward vertical motion (regenerative operation, simplified calculation using synchronous speed)
Acceleration time [s]	$t_A = \frac{\left(J_M + \frac{J_{\text{load}}}{\eta}\right) \cdot 2\pi \cdot n_M}{60 \cdot \left(M_H - \frac{M_L}{\eta}\right)}$	$t_A = \frac{\left(J_M + \frac{J_{\text{load}}}{\eta}\right) \cdot 2\pi \cdot n_S}{60 \cdot (M_H - M_L \cdot \eta)}$
Starting acceleration [m/s ²]	$a_A = \frac{v}{t_A}$	$a_A = \frac{v}{t_A} \cdot \frac{n_S}{n_M}$
Starting distance [mm]	$s_A = \frac{1}{2} \cdot t_A \cdot v \cdot 1000$	$s_A = \frac{1}{2} \cdot t_A \cdot \frac{n_S}{n_M} \cdot v \cdot 1000$
Braking time [s]	$t_B = \frac{(J_M + J_{\text{load}} \cdot \eta) \cdot 2\pi \cdot n_M}{60 \cdot (M_B + M_L \cdot \eta)}$	$t_B = \frac{(J_M + J_{\text{load}} \cdot \eta) \cdot 2\pi \cdot n_S}{60 \cdot (M_B - M_L \cdot \eta)}$
Brake deceleration [m/s ²]	$a_B = \frac{v}{t_B}$	$a_B = \frac{v}{t_B}$
Stopping distance [mm]	$s_B = v \cdot 1000 \cdot \left(t_1 + \frac{1}{2} \cdot t_B\right)$	$s_B = v \cdot 1000 \cdot \left(t_1 + \frac{1}{2} \cdot t_B\right)$
Stopping accuracy [mm]	$\Delta s = \pm 0.15 \cdot s_B$	$\Delta s = \pm 0.15 \cdot s_B$
Switching frequency [1/h]	$z = z_0 \cdot k_M \cdot k_{FI} \cdot k_P$ $z = z_0 \cdot \frac{1 - \frac{M_L}{M_H \cdot \eta}}{\frac{J_M + \frac{J_X}{\eta}}{J_M}} \cdot k_P$	$z = z_0 \cdot \frac{1 - \frac{M_L \cdot \eta}{M_H}}{\frac{J_M + J_X \cdot \eta}{J_M}} \cdot k_P$
Braking energy [J]	$W = \frac{M_B}{M_B + M_L \cdot \eta} \cdot \frac{(J_M + J_X \cdot \eta) \cdot 2 \cdot (n_M \cdot \pi)^2}{3600}$	$W = \frac{M_B}{M_B - M_L \cdot \eta} \cdot \frac{(J_M + J_X \cdot \eta) \cdot 2 \cdot (\pi \cdot n_M)^2}{3600}$
Brake life time [h]	$L_B = \frac{L}{W \cdot Z_N}$	$L_B = \frac{L}{W \cdot Z_N}$

4.7 Calculating the velocity, if the acceleration, time, and distance are predefined

$$v_{\max} = \frac{a_{\max} \cdot t_{\text{total}}}{2} - \sqrt{\left(\frac{a_{\max} \cdot t_{\text{total}}}{2}\right)^2 - a_{\max} \cdot s_{\text{total}}}$$

4.8 Chain wheels

d	Pitch diameter
z	Number of teeth
m	Module
P	Pitch
τ	Angular pitch



$$d = z \cdot m$$

$$P = m \cdot \pi$$

$$d = z \cdot \frac{P}{\pi}$$

$$\tau = \frac{360^\circ}{z}$$

$$d = \frac{P}{\sin \frac{180^\circ}{z}}$$

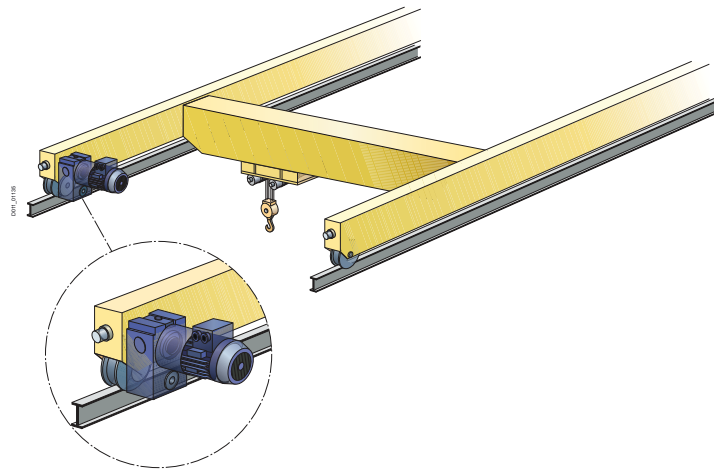
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Fig. 4-9 Chain wheel



Example of a traction drive with a frequency converter

5



070808_Kranfahrwerk.pdf

Fig. 5-1 Gantry

Self-weight of the crane	m_{crane}	25,000 kg (including trolley)
Weight of the trolley	m_{trolley}	5,000 kg
Load weight	m_L	25,000 kg
Travelling speed	v	60 m/min
Accelerating time	t	2 s
Force of the wind	F_{wind}	Not applicable, as this is a workshop crane
Wheel diameter	D	500 mm
Shaft diameter	D_s	$0.2 \cdot D = 100 \text{ mm}$
Number of wheels	z	4
Wheels with roller bearings	μ_r	0.005
Material combination of wheel/rail, steel on steel	f	Lever arm of the rolling friction, $f = 0.5 \text{ mm}$
Wheels guided on flanges	c	0.003
Efficiency	η	0.9
Daily operating duration		< 16 h
Making operations per hour		60 h^{-1}

Each wheel is to be driven by a 4-pole parallel shaft geared motor. All motors are operated on one frequency converter, connected in parallel.

The force of the wind can be calculated from $F_{\text{wind}} = A \cdot q$

A : Effective area exposed to the wind in m^2 ; information to be supplied by the customer.

q : Dynamic pressure in N/m^2 ; if no other requirements exist, wind strength 7 (150 N/m^2) is the limit for operating at setpoints.

5.1 Selecting a geared motor

Specific tractive resistance w_F

$$w_F = \frac{2}{D} \cdot \left(\frac{D_w}{2} \cdot \mu_r + f \right) + c = \frac{2}{0.5 \text{ m}} \cdot \left(\frac{0.1 \text{ m}}{2} \cdot 0.005 + 0.0005 \text{ m} \right) + 0.003 = 0.006$$

Tractive resistance force

$$F_w = (m_{\text{crane}} + m_L) \cdot g \cdot w_F = (25000 \text{ kg} + 25000 \text{ kg}) \cdot \frac{9.81 \text{ m}}{\text{s}^2} \cdot 0.006 = 2943 \text{ N}$$

Static power required per motor

$$P = \frac{(F_w + F_{\text{wind}}) \cdot v}{z \cdot \eta} = \frac{(2943 + 0) \text{ N} \cdot 60 \frac{\text{m}}{60 \text{ s}}}{4 \cdot 0.9} = 817.5 \text{ W}$$

Required acceleration

$$a = \frac{v}{t} = \frac{60 \frac{\text{m}}{60 \text{ s}}}{2 \text{ s}} = 0.5 \frac{\text{m}}{\text{s}^2}$$

Required output speed at the gearbox

$$n_{\text{out}} = \frac{v}{\pi \cdot D} = \frac{60 \cdot \frac{\text{m}}{\text{min}}}{\pi \cdot 0.5 \text{ m}} = 38.20 \text{ min}^{-1}$$

Required gear ratio

Initially, the speed of the 4-pole motor is assumed to be $1,400 \text{ min}^{-1}$.

$$i_{\text{req}} = \frac{n_N}{n_{\text{out}}} = \frac{1400 \text{ min}^{-1}}{38.2 \text{ min}^{-1}} = 36.65$$

Static torque required per motor

$$M_{\text{stat}} = \frac{(F_w + F_{\text{wind}}) \cdot D}{z \cdot \eta \cdot 2 \cdot i_{\text{req}}} = \frac{(2943 + 0) \cdot \text{N} \cdot 0.5 \text{ m}}{4 \cdot 0.9 \cdot 2 \cdot 36.65} = 5.6 \text{ Nm}$$

Dynamic torque per motor

$$M_{\text{dyn}} = \frac{(m_{\text{crane}} + m_{\text{load}}) \cdot a \cdot D}{z \cdot \eta \cdot 2 \cdot i_{\text{req}}} = \frac{(25000 \text{ kg} + 25000 \text{ kg}) \cdot 0.5 \frac{\text{m}}{\text{s}^2} \cdot 0.5 \text{ m}}{4 \cdot 0.9 \cdot 2 \cdot 36.65} = 47.4 \text{ Nm}$$

Acceleration torque required per motor

$$M_{\text{acc}} = M_{\text{stat}} + M_{\text{dyn}} = 5.6 \text{ Nm} + 47.4 \text{ Nm} = 53.0 \text{ Nm}$$

Selected motor

When operating on a frequency converter, 1.5 to 2 times the rated motor torque is usually used as the acceleration value. In this example, 1.6 times the rated torque has been assumed.

$$M_{\text{rated, motor, req}} = \frac{M_{\text{acc}}}{1.6} = \frac{53.0}{1.6} = 33.1 \text{ Nm}$$

The catalog states that the power for a 4-pole motor is 5.5 kW.

Motor data:

$$P_{\text{rated}} = 5.5 \text{ kW}$$

$$n_{\text{rated}} = 1,450 \text{ min}^{-1}$$

$$M_{\text{rated}} = 36 \text{ Nm}$$

$$J_M = 0.024 \text{ kgm}^2 \text{ including brake}$$

$$I_{\text{rated}} = 11 \text{ A}$$

$$\eta = 0.86$$

External moment of inertia

$$J_{\text{ext}} = (m_L + m_{\text{crane}}) \cdot \left(\frac{D}{2}\right)^2 \cdot \frac{1}{i_{\text{req}}^2} = (25000 \text{ kg} + 25000 \text{ kg}) \cdot \left(\frac{0.5 \text{ m}}{2}\right)^2 \cdot \frac{1}{36.65^2} = 2.3 \text{ kgm}^2$$

This equates to shock load II.

Required service factor

According to Chapter 2.1, shock load II, 60 making operations + 60 braking operations = 120 operations per hour, and a daily operation time of under 16 h results in a service factor f_{s1} of 1.4.

$$f_{s, \text{req}} = f_{s1} = 1.4$$

Required gearbox rated torque

$$M_{\text{rated, gearbox, req}} = M_{\text{rated, motor}} \cdot i_{\text{req}} \cdot f_{s, \text{req}} = 36 \text{ Nm} \cdot 36.65 \cdot 1.4 = 1847 \text{ Nm}$$

Selected gearbox

Gearbox data:

FZ 88B

$i = 35.3$

$M_{\text{rated}} = 1,900 \text{ Nm}$

$\eta_{\text{gearbox}} = 0.96$

$$f_s = \frac{1900 \text{ Nm}}{35.3 \cdot 36 \text{ Nm}} = 1.5$$

Recalculating the selected motor

The selected motor is recalculated using the actual transmission ratio of the selected gearbox. The required acceleration torque is determined for this purpose.

$$M_{\text{stat}} = \frac{(F_w + F_{\text{wind}}) \cdot D}{z \cdot \eta \cdot \eta_{\text{gearbox}} \cdot 2 \cdot i} = \frac{(2943 + 0) \text{ N} \cdot 0.5}{4 \cdot 0.9 \cdot 0.96 \cdot 2 \cdot 35.3} = 6 \text{ Nm}$$

$$J_{\text{ext}} = (m_L + m_{\text{crane}}) \cdot \left(\frac{D}{2}\right)^2 \cdot \frac{1}{i^2} = (25000 \text{ kg} + 25000 \text{ kg}) \cdot \left(\frac{0.5 \text{ m}}{2}\right)^2 \cdot \frac{1}{35.3^2} = 2.5 \text{ kgm}^2$$

$$M_{\text{dyn}} = \frac{\left(J_M + \frac{J_{\text{ext}}}{\eta \cdot \eta_{\text{gearbox}}}\right) \cdot 2\pi \cdot n_{\text{mot}}}{z \cdot t} = \frac{\left(J_M + \frac{J_{\text{ext}}}{\eta \cdot \eta_{\text{gearbox}}}\right) \cdot a \cdot i \cdot 2}{z \cdot D} =$$

$$= \frac{\left(0.024 + \frac{2.5}{0.9 \cdot 0.96}\right) \cdot \text{kgm}^2 \cdot 0.5 \frac{\text{m}}{\text{s}^2} \cdot 35.3 \cdot 2}{4 \cdot 0.5 \text{ m}} = 51.5 \text{ Nm}$$

Acceleration torque required per motor

$$M_{\text{acc}} = M_{\text{stat}} + M_{\text{dyn}} = 6 \text{ Nm} + 51.5 \text{ Nm} = 57.5 \text{ Nm}$$

$$\frac{M_{\text{acc}}}{M_{\text{rated, motor}}} = \frac{57.5 \text{ Nm}}{36 \text{ Nm}} = 1.6$$

This value lies within the range 1.5 to 2 mentioned above.

The motor is suitable.

At the set velocity, the motor speed is

$$n_{\text{mot, actual}} = n_{\text{out}} \cdot i = 38.2 \text{ min}^{-1} \cdot 35.3 = 1349 \text{ min}^{-1} = 22.5 \text{ s}^{-1}$$

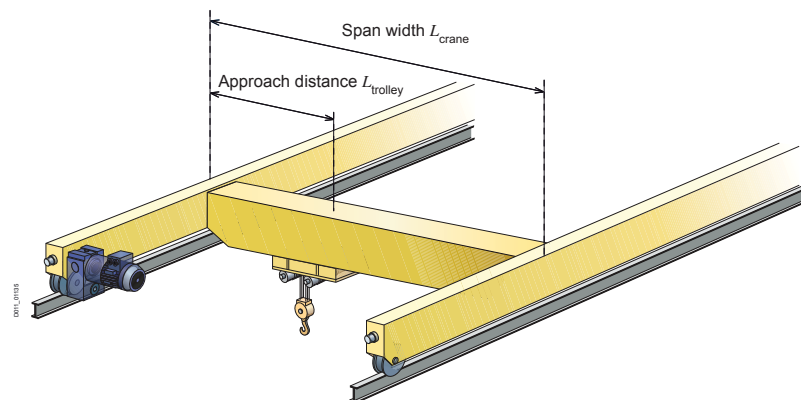
Checking the anti-slip resistance

Here, the maximum permissible acceleration at which the wheels do not slip is calculated.

The following generally applies:

$$a_{\text{perm}} = \frac{\mu \cdot F_{\text{min}} \cdot \cos \alpha \cdot z_{\text{dr}} - F_{\text{add}}}{m} - g \cdot \sin \alpha - \text{signum} \cdot g \cdot w_F \cdot \cos \alpha$$

μ	Friction coefficient between the driven wheel and the rail
F_{min}	Minimum force on a driven wheel in the direction of gravity
F_{add}	Additional force, such as the force of the wind
α	Angle of rise
m	Total of all masses to be moved
signum (sign)	+ for acceleration, - for deceleration
z_{dr}	Number of driven wheels



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Fig. 5-2 Anti-slip resistance

The following must be taken into account for the permissible acceleration in this case:

Span width of the crane $L_{\text{crane}} = 20 \text{ m}$

Approach distance of the trolley $L_{\text{trolley}} = 1.5 \text{ m}$

$z_{\text{total}} = \text{total number of wheels} = 4$

$\mu = 0.15$

$$F_{\min} = \frac{g}{z_{\text{total}}} \cdot \left[(m_{\text{crane}} - m_{\text{trolley}}) + (m_{\text{trolley}} + m_{\text{load}}) \cdot 2 \cdot \frac{L_{\text{trolley}}}{L_{\text{crane}}} \right]$$

$$F_{\min} = \frac{9.81 \cdot \frac{\text{m}}{\text{s}^2}}{4} \cdot \left[(25000 \text{ kg} - 5000 \text{ kg}) + (5000 \text{ kg} + 25000 \text{ kg}) \cdot 2 \cdot \frac{1.5 \text{ m}}{20 \text{ m}} \right] = 60086.25 \text{ N}$$

$$F_{\text{add}} = 0$$

$$\alpha = 0$$

$$m = m_{\text{load}} + m_{\text{crane}} = 25,000 \text{ kg} + 25,000 \text{ kg} = 50,000 \text{ kg}$$

$$z_{\text{dr}} = 4$$

This enables the permissible acceleration to be calculated:

$$a_{\text{perm}} = \frac{0.15 \cdot F_{\min} \cdot \cos \alpha \cdot z_{\text{dr}} - F_{\text{add}}}{m} - g \cdot \sin \alpha - g \cdot w_{\text{F}} \cdot \cos \alpha$$

$$a_{\text{perm}} = \frac{0.15 \cdot 60086.25 \text{ N} \cdot \cos 0 \cdot 4 - 0}{50000 \text{ kg}} - 9.81 \frac{\text{m}}{\text{s}^2} \cdot \sin 0 - 9.81 \frac{\text{m}}{\text{s}^2} \cdot 0.006 \cdot \cos 0 = 0.66 \frac{\text{m}}{\text{s}^2}$$

Since the required acceleration of 0.5 m/s^2 is lower than the permissible acceleration, acceleration will be performed without slipping, provided that the rails are dry.

5.2 Selecting the frequency converter

Notes on dimensioning

- As far as the acceleration procedure is concerned, the converter overload behavior must support a duty cycle which is typical of the application.
- For a traversing gear, the maximum converter output current is a temporary acceleration current. For a gantry crane (no force of the wind, level route), however, the steady current during constant travel is very low.
- It is essential that the force of the wind be taken into account if the crane is installed outside.
- The following applies to the G120:
Utilization of 1.5 times the overload, with a base-load current of I_H ; 2 times the overload is not normally used.
Duty cycle: $1.5 \cdot I_H$ over 57 s at a cycle time of 300 s
- If there is any doubt, the duty cycle should be calculated first and the converter dimensioned afterwards.

Procedure

- Calculating the maximum motor current and dimensioning the converter based on the acceleration current
- You can also check whether or not the converter will be able to carry the steady motor current continuously.

If the force of the wind has to be taken into account, this check must always be performed.

Calculating the maximum motor current

$$I_{\text{MotMax}} = \frac{M_{\text{acceleration}}}{M_N} \cdot I_N = \frac{57.5 \text{ Nm}}{36 \text{ Nm}} \cdot 11 \text{ A} = 17.6 \text{ A}$$

The traversing gear consists of 4 motors, so the maximum converter output current is

$$I_{\text{ConverterMax}} = 4 \cdot I_{\text{MotMax}} = 71 \text{ A}$$

$$I_{\text{ConverterBase-LoadCurrentIH}} = I_{\text{ConverterMax}} / 1.5 = 71 \text{ A} / 1.5 = 47 \text{ A}$$

1.5 = Overload factor

So, in this example a 30 kW PM240 power unit is selected, order number: 6SL3224-0BE33-0AA0

Additional check

For added safety, an additional check can be performed to see whether or not the converter can cope with the steady motor current.

In this case, the steady motor torque is small compared to the rated torque, so it does not have to be included in the converter dimensioning calculations.

5.3 Dimensioning the braking resistor

Notes:

When braking by means of a chopper and braking resistor or when performing power recovery, the overload behavior must always be clarified. It cannot be assumed that the overload behavior will be the same during motoring operation (acceleration) and regenerative operation (deceleration).

The following system data is also required:

1. Mass of the crane and the load (the rotating masses of the motors and gearboxes, for example, are not taken into consideration)
2. Travelling speed of the crane
3. Efficiencies of the crane system, gearbox, motor, and converter

The following values result for this example:

1. 50,000 kg
2. 60 m/min
3. $\eta = \eta_{\text{gearbox}} \cdot \eta_{\text{motor}} \cdot \eta_{\text{converter}} = 0.9 \cdot 0.96 \cdot 0.86 \cdot 0.98 = 0.73$

According to the catalog, the selected braking resistor has a continuous braking power P_{CB} of 2.2 kW and a peak power P_{max} of 44 kW. P_{max} is permitted for 12 s during a cycle time of $t_{\text{ZR}} = 240$ s.

Note:

The resistor values given here (particularly P_{max} and t_{ZR}) are manufacturer-specific and must be identified so that they can be used for the subsequent verification.

Procedure

- Calculate the peak braking power
- Calculate the average braking power $P_{\text{average, R}}$

Calculating the peak braking power

$$M_{\text{stat}} = \frac{(F_w + F_{\text{wind}}) \cdot D \cdot \eta \cdot \eta_{\text{gearbox}}}{z \cdot 2 \cdot i} = \frac{(2943 + 0) \text{ N} \cdot 0.5 \text{ m} \cdot 0.9 \cdot 0.96}{4 \cdot 2 \cdot 35.3} = 4.5 \text{ Nm}$$

$$M_{\text{dyn}} = \frac{(J_M + J_{\text{ext}} \cdot \eta \cdot \eta_{\text{gearbox}}) \cdot 2\pi \cdot n_{\text{mot}}}{z \cdot t} = \frac{(J_M + J_{\text{ext}} \cdot \eta \cdot \eta_{\text{gearbox}}) \cdot a \cdot i \cdot 2}{z \cdot D} =$$

$$= \frac{(0.024 + 2.5 \cdot 0.9 \cdot 0.96) \text{ kgm}^2 \cdot 0.5 \frac{\text{m}}{\text{s}^2} \cdot 35.3 \cdot 2}{4 \cdot 0.5 \text{ m}} = 38.5 \text{ Nm}$$

$$P_{\text{brake, max}} = z \cdot (M_{\text{dyn}} - M_{\text{stat}}) \cdot 2\pi \cdot n_{\text{mot, actual}} = 4 \cdot (38.5 - 4.5) \text{ Nm} \cdot 2\pi \cdot 22.4 \text{ s}^{-1} = 19 \text{ kW}$$

Calculating the average braking power $P_{\text{average, R}}$

The mean value for the electrical braking power during a braking operation P_{average} is $19 \text{ kW} / 2 = 9.5 \text{ kW}$. It occurs during the braking time of 2 s.

The crane is moved a maximum of 60 times per hour. This means that there are 60 s available for each crane cycle (acceleration, constant travel, deceleration). During t_{ZR} , therefore, the crane brakes a maximum of 4 times (braking time t), so the total braking time t_{S} during t_{ZR} is as follows:

$$t_{\text{S}} = 4 \cdot t = 4 \cdot 2 \text{ s} = 8 \text{ s}$$

So, in 8 s a braking energy of W_{res} occurs at the resistor:

$$W_{\text{res}} = P_{\text{average}} \cdot t_{\text{S}} = 9.5 \text{ kW} \cdot 8 \text{ s} = 76 \text{ kWs}$$

This means that the average power dissipated at the braking resistor is:

$$P_{\text{average, R}} = \frac{W_{\text{res}}}{t_{\text{ZR}}} = \frac{76 \text{ kWs}}{240 \text{ s}} = 0.3 \text{ kW}$$

Therefore, the braking resistor is suitable, as:

- At 19 kW, the peak braking power of the gantry is lower than the braking resistor's permissible peak power.
- $P_{\text{average, R}}$ is lower than P_{CB} .

Note on dimensioning the PM250 power unit with power recovery

- Rated data must be taken into account (see catalog).
- Motoring operation:
 - The PM250 is selected using the same criteria as for the PM240.
- Regenerative operation:
 - The braking resistor does not have to be dimensioned in this case.
 - The PM250 can continuously recover the rated device power (based on I_{H}) and feed it back into the system.
 - NOTICE, POWER RECOVERY!

The rated device power also represents the peak power for power recovery, so no overload is possible in terms of the regenerative power!

The inverter's torque overload capability is not affected by this, provided that the total power recovered and fed back into the system does not exceed the permissible value.

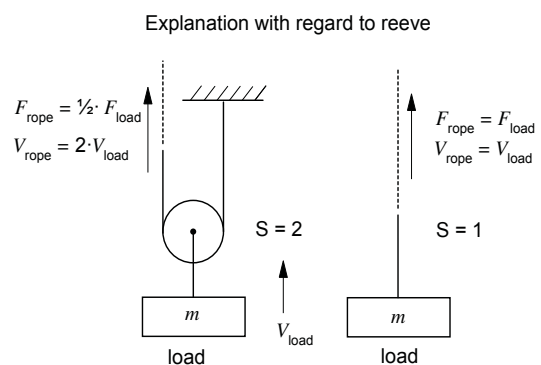
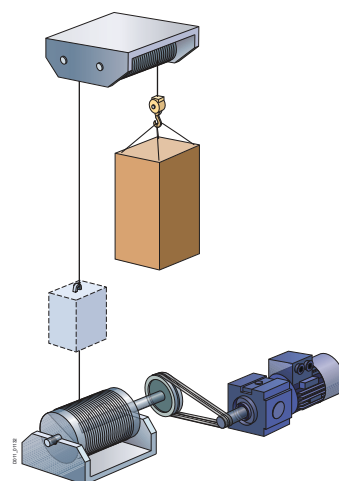


Example of hoisting gear with a frequency converter

6

Load mass including load suspension device	m_L	10,000 kg
Counterweight	m_G	5,000 kg
Hoisting velocity	v_h	0.3 m/s
Hoisting height	h	8 m
Rope drum diameter	d	600 mm
Moment of inertia of the drum	J_T	16 kgm ²
Reeve	s	2
Starting time	t_s	2 s
Efficiency of the system	η	0.93
Daily operating duration		16 h
Hoisting movements per hour		22 h ⁻¹

The vertical lift drive should consist of a frequency-controlled helical geared motor.



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Fig. 6-1 Vertical lift drive

6.1 Selecting a geared motor

Required steady output power P_{req}

$$P_{\text{req}} = \frac{(m_L + m_G)}{\eta} \cdot g \cdot v_H$$

$$P_{\text{req}} = \frac{(10000 \text{ kg} + 5000 \text{ kg})}{0.93} \cdot 9.81 \frac{\text{m}}{\text{s}^2} \cdot 0.3 \frac{\text{m}}{\text{s}} = 15823 \text{ W}$$

Selected motor: 18.5 kW, 4-pole

Motor data:

$$P_{\text{rated}} = 18.5 \text{ kW}$$

$$U_{\text{rated}} = 400 \text{ V}, f_{\text{rated}} = 50 \text{ Hz}$$

$$I_{\text{rated}} = 35.5 \text{ A}$$

$$\eta_{\text{motor}} = 0.91$$

$$M_{\text{rated}} = 121 \text{ Nm}$$

$$M_{\text{tilting}}/M_{\text{rated}} = 3$$

$$n_{\text{rated}} = 1,460 \text{ min}^{-1}$$

$$J_{\text{mot}} = 0.13 \text{ kgm}^2$$

Required output speed n_{out}

$$n_{\text{out}} = n_{\text{drum}} = \frac{s \cdot v_H}{\pi \cdot d}$$

$$n_{\text{out}} = \frac{2 \cdot 18 \frac{\text{m}}{\text{min}}}{\pi \cdot 0.6 \text{ m}} = 19.1 \text{ min}^{-1}$$

Required gear ratio i_{set}

$$i_{\text{set}} = \frac{n_{\text{motor}}}{n_{\text{out}}} = \frac{1460 \text{ min}^{-1}}{19.1 \text{ min}^{-1}} = 76.44$$

Calculating the moments of inertia

$$J_2 = (m_L + m_G) \cdot \left(\frac{d}{2}\right)^2$$

$$J_2 = (10000 \text{ kg} + 5000 \text{ kg}) \cdot \left(\frac{0.6 \text{ m}}{2}\right)^2 = 1350 \text{ kgm}^2$$

Converted for the motor side:

$$J_L = \frac{J_2}{i_{\text{set}}^2 \cdot s^2} = \frac{1350 \text{ kgm}^2}{76.44^2 \cdot 2^2} = 0.05776 \text{ kgm}^2$$

$$J_{L, \text{total}} = J_L + J_T \cdot \frac{1}{i_{\text{set}}^2} = 0.05776 \text{ kgm}^2 + 16 \text{ kgm}^2 \cdot \frac{1}{76.44^2} = 0.06 \text{ kgm}^2$$

$$\frac{J_{L, \text{total}}}{J_{\text{motor}}} = \frac{0.06 \text{ kgm}^2}{0.13 \text{ kgm}^2} = 0.46$$

The moment of inertia of a brake is not taken into account here. This is permitted because the acceleration procedure has practically no effect on motor dimensioning, as we will see a little later when it comes to checking the acceleration procedure.

Required service factor

As shown in Chapter 2.1, if $0.3 < J_{L, \text{total}}/J_{\text{motor}} < 3$ and with shock load II, operating duration 16 hours, and 44 operations/h, this will result in:

$$f_{s, \text{req}} = 1.4$$

Calculating the required rated output torque at the drum

$$M_{\text{rated, out, req}} = M_{\text{rated, mot}} \cdot i_{\text{set}} \cdot f_{B \text{ req}} = 121 \text{ Nm} \cdot 76.44 \cdot 1.4 = 12949 \text{ Nm}$$

Selected gearbox:

- D168 helical gearbox, 3-stage
- $i = 72.4$
- $\eta_{\text{gearbox}} = 0.94$
- $M_{\text{rated, out}} = 14,000 \text{ Nm}$
- $f_s = \frac{14000 \text{ Nm}}{121 \text{ Nm} \cdot 72.4} = 1.6$

Shaft speed at the motor with rated hoisting velocity

$$n_{\text{hoist}} = \frac{v_H}{\pi \cdot d} \cdot s \cdot i = \frac{0.3 \frac{\text{m}}{\text{s}}}{\pi \cdot 0.6 \text{ m}} \cdot 2 \cdot 72.4 = 23 \frac{1}{\text{s}} = 1383 \text{ min}^{-1}$$

Required acceleration torque at the motor side

$$M_{\text{accel}} = M_{\text{stat}} + M_{\text{dyn}}$$

$$M_{\text{stat}} = (m_L - m_G) \cdot \frac{g}{\eta \cdot \eta_{\text{gearbox}}} \cdot \frac{d}{2} \cdot \frac{1}{s} \cdot \frac{1}{i} = 5000 \text{ kg} \cdot \frac{9.81 \frac{\text{m}}{\text{s}^2}}{0.93 \cdot 0.94} \cdot \frac{0.6 \text{ m}}{2} \cdot \frac{1}{2} \cdot \frac{1}{72.4} = 116 \text{ Nm}$$

$$M_{\text{dyn}} = \frac{(m_L + m_G) \cdot a \cdot d}{s \cdot i \cdot 2 \cdot \eta \cdot \eta_{\text{gearbox}}} = \frac{15000 \text{ kg} \cdot 0.15 \frac{\text{m}}{\text{s}^2} \cdot 0.6 \text{ m}}{2 \cdot 72.4 \cdot 2 \cdot 0.93 \cdot 0.94} = 5.3 \text{ Nm}$$

$$\text{The required acceleration is } a = \frac{v_H}{t_A} = \frac{0.3 \frac{\text{m}}{\text{s}}}{2 \text{ s}} = 0.15 \frac{\text{m}}{\text{s}^2}$$

$$M_{\text{accel}} = M_{\text{stat}} + M_{\text{dyn}} = 116 \text{ Nm} + 5.3 \text{ Nm} = 121.3 \text{ Nm}$$

Checking the selected motor

1. At 121 Nm, the rated motor torque is higher than the steady hoisting torque M_{stat} .

2. The motor can deliver a maximum torque of:

$$M_{\text{MaxMotor}} = 0.77 \cdot (M_{\text{tilting}}/M_{\text{rated}}) \cdot M_{\text{rated}} = 0.77 \cdot 3 \cdot 121 \text{ Nm} = 280 \text{ Nm}$$

This value is higher than M_{accel} .

Note on factor 0.77:

0.77 is a safety factor, which ensures that the motor on the frequency converter is operated in a stable manner in the torque overload range.

Result:

As conditions 1 and 2 are met, the motor can be used in this hoisting gear.

6.2 Selecting the frequency converter

Procedure:

1. Initially, the converter is dimensioned according to the motor current at the rated hoisting velocity (I_{stat}). For the G120, for example, the current for "high overload" I_{H} must be used here!

Using the G120 (PM240/PM250 power units) as an example, the following selection would be made:

$$I_{\text{stat}} = \frac{M_{\text{stat}}}{M_{\text{N}}} \cdot I_{\text{N}} = \frac{116 \text{ Nm}}{121 \text{ Nm}} \cdot 35.5 \text{ A} = 34 \text{ A}$$

As a result, a power unit with $I_{\text{H}} = 38 \text{ A}$ is selected.

2. As far as the acceleration procedure is concerned, the converter overload behavior must support a duty cycle which is typical of the application. In this example and in the case of the G120, this is the "high overload" (HO), where 1.5 times the overload is normally used ($1.5 \cdot I_{\text{H}}$ over 57 s at a cycle time of 300 s).

Note:

If there is any doubt, the hoisting gear duty cycle should be calculated first and the converter dimensioned afterwards.

Calculation of the max. motor current:

$$I_{\text{MotMax}} = \frac{M_{\text{accel}}}{M_{\text{N}}} \cdot I_{\text{N}} = \frac{121 \text{ Nm}}{121 \text{ Nm}} \cdot 35.5 \text{ A} = 35.5 \text{ A}$$

Since the required current I_{MotMax} is less than $1.5 \cdot 38 \text{ A}$, the power unit investigated in the first step can also provide the required overload current for the acceleration phase.

6.3 Dimensioning the braking resistor

Notes

When braking by means of a chopper and braking resistor or when performing power recovery, the overload behavior must always be clarified. It cannot be assumed that the overload behavior will be the same during motoring operation (hoisting) and regenerative operation (lowering).

The following data is also required:

- System:
 - Duration of the max. possible lowering operation or that which is actually performed
 - Duty cycle of the hoisting gear or number of movements
 - Max. lowering velocity
- Converter:
 - Maximum chopper current (or minimum resistance value of the braking resistor)
 - DC-link voltage for chopper operation

The following values result for this example:

- System:
 - Duration of the max. possible lowering operation or that which is actually performed, approx. 27 s; see below for calculation
 - Duty cycle of the hoisting gear, 33 %; see below for calculation
 - Max. lowering velocity = rated hoisting velocity = 0.3 m/s
- Converter:
 - Resistance value of the braking resistor, e.g. PM240 power unit, frame size D, order number: 6SL3224-0BE31-8AA0; $I_H = 38 \text{ A}$, $R_{\text{BrakeR}} = 27 \Omega$
 - DC-link voltage $U_{\text{DC link}}$ for chopper operation, power unit as stated:
 $U_{\text{DC link}} = 800 \text{ V}$

Approximate calculation of the maximum possible lowering time

$$\frac{h}{v_H} = \frac{8 \text{ m}}{0.3 \frac{\text{m}}{\text{s}}} = 26.7 \text{ s} = t_{\text{lower}}$$

Calculation of the duty cycle

Here, Number of lifting movements = Number of lowering movements = 22 h^{-1}

$$\text{Motion time per hour} = 22 \frac{1}{\text{h}} \cdot 2 \cdot 27 \text{ s} = 1188 \frac{\text{s}}{\text{h}}$$

$$\text{Duty Cycle} = \frac{1188 \text{ s}}{3600 \text{ s}} = 0.33 \Rightarrow \text{DC} = 33 \%$$

Procedure

- Calculating the steady braking power P_{brake}
- Calculating the peak braking power $P_{\text{brake, max}}$
- Checking whether the braking resistor (in this case, 27Ω) can cope with the peak braking power
- Calculating the effective braking power
- Finding a suitable braking resistor

Calculating the steady braking power P_{brake}

$$P_{\text{brake}} = (m_{\text{L}} - m_{\text{G}}) \cdot \eta \cdot \eta_{\text{mot}} \cdot \eta_{\text{gearbox}} \cdot \eta_{\text{converter}} \cdot g \cdot v_{\text{H}}$$

$$P_{\text{brake}} = (10000 \text{ kg} - 5000 \text{ kg}) \cdot 0.93 \cdot 0.91 \cdot 0.94 \cdot 0.98 \cdot 9.81 \frac{\text{m}}{\text{s}^2} \cdot 0.3 \frac{\text{m}}{\text{s}} = 11472 \text{ W}$$

Calculating the peak braking power $P_{\text{brake, max}}$

$$P_{\text{brake, max}} = P_{\text{brake}} + M_{\text{dyn}} \cdot 2 \cdot \pi \cdot n_{\text{hoisting}}$$

$$M_{\text{dyn}} = \frac{(m_{\text{L}} + m_{\text{G}}) \cdot a \cdot d \cdot \eta \cdot \eta_{\text{gearbox}}}{s \cdot i \cdot 2} = \frac{15000 \text{ kg} \cdot 0.15 \frac{\text{m}}{\text{s}^2} \cdot 0.6 \text{ m} \cdot 0.93 \cdot 0.94}{2 \cdot 72.4 \cdot 2} = 4.1 \text{ Nm}$$

$$P_{\text{brake, max}} = 11472 \text{ W} + 4.1 \text{ Nm} \cdot 2 \cdot \pi \cdot 1383 \frac{1}{60 \text{ s}} = 12065 \text{ W}$$

Checking the braking resistor

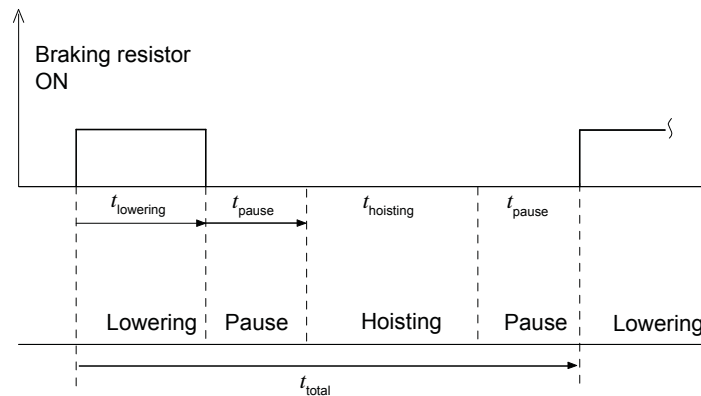
The braking resistor can dissipate a peak braking power $P_{\text{BrakeElMax}}$ of
 $P_{\text{BrakeElMax}} = (U_{\text{DC link}})^2 / R_{\text{BrakeR}} = (800 \text{ V})^2 / 27 \Omega = 24 \text{ kW}$.

A comparison with $P_{\text{brake, max}}$ shows that the required peak braking power can be delivered if the braking resistor referred to above (which has been adapted to the device) is used.

Calculation of the effective braking power

Assumption:

The acceleration and deceleration ramps are not taken into account; instead, it is assumed that the steady braking power is effective throughout the entire lowering time. It is also assumed that $t_{\text{lowering}} = t_{\text{hoisting}}$.



Zyklus_1.eps

Fig. 6-2 Cycle for the braking resistor

At 22 hoisting movements per hour, t_{total} results in

$$t_{\text{total}} = \frac{3600 \text{ s}}{22} = 164 \text{ s}$$

$$P_{\text{eff}} = \sqrt{\frac{P_{\text{brake}}^2 \cdot t_{\text{lowering}}}{t_{\text{total}}}} = \sqrt{\frac{(11472 \text{ W})^2 \cdot 27 \text{ s}}{164 \text{ s}}} = 4.7 \text{ kW}$$

Summary of the values required to find a suitable braking resistor:

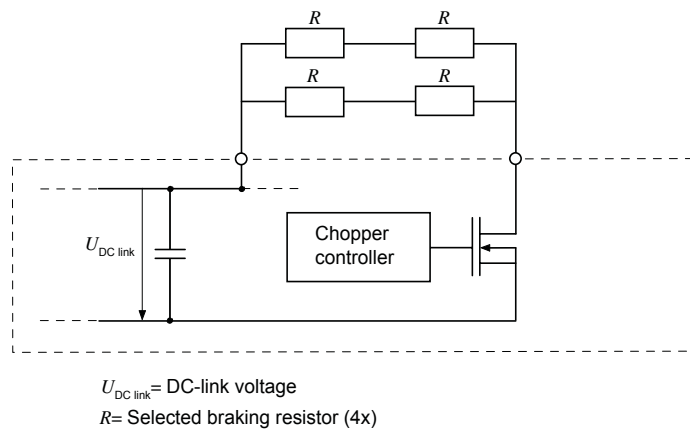
- Resistance value: 27Ω
- Braking power when lowering: 11.5 kW for 27 s
- Effective braking power: 4.7 kW

Finding a suitable braking resistor

Dimensioning example based on the resistor assigned to the PM240 power unit referred to above (see catalog D11.1):

- Type: 6SE6400-4BD21-2DA0
- Resistance value: 27Ω
- Peak braking power 24 kW for 12 s , total cycle time 240 s
- Rated braking power (effective braking power) 1.2 kW

Connecting the resistors in parallel and in series (RPS), See Fig. 6-3, enables a continuous braking power of $4 \cdot 1.2 \text{ kW} = 4.8 \text{ kW}$ to be achieved.



Chopper.eps

Fig. 6-3 Chopper

The RPS only applies 1/4 of the resulting braking power to each individual resistor. This means that the peak braking power of the RPS is restricted to 48 s max. (and lasts for just 27 s in this example).

Note on dimensioning the PM250 power unit with power recovery

- Rated data must be taken into account (see catalog).
- Hoisting operation (motoring operation):
 - The PM250 is selected using the same criteria as for the PM240.
- Lowering operation (regenerative operation):
 - The braking resistor does not have to be dimensioned in this case.
 - The PM250 can continuously recover the rated device power (based on I_H) and feed it back into the system.
 - NOTICE, POWER RECOVERY!

The rated device power also represents the peak power for power recovery, so no overload is possible in terms of the regenerative power!

The inverter's torque overload capability is not affected by this, provided that the total power recovered and fed back into the system does not exceed the permissible value.



Example of a roller table drive with a frequency converter

7

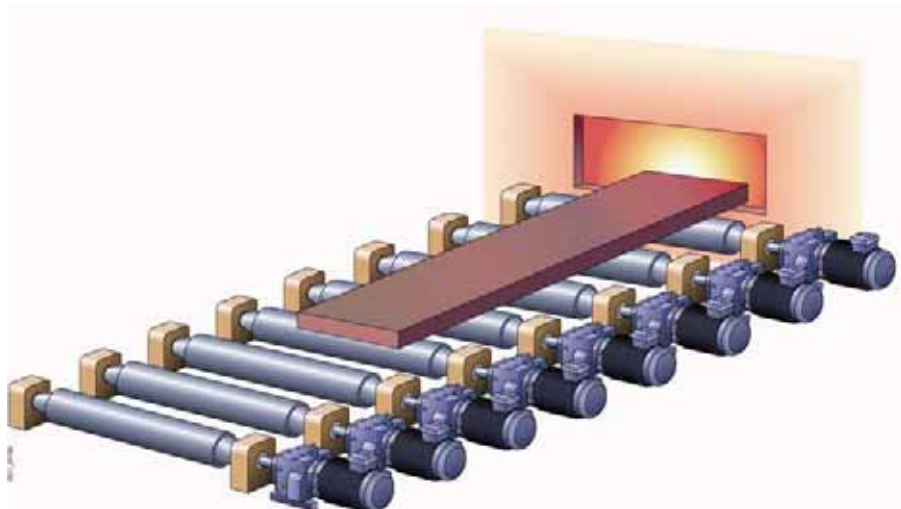
The drives used for roller tables in rolling mills need to be selected according to special criteria, which are very different to those used when configuring other drives. Although most drives must be rated according to the required continuous power, in the case of the roller table motor, the opposing-field braking operations, starts, and no-load periods alternate in quick succession. The actual conveying power itself is relatively low. On top of these special requirements, it is primarily because of the mechanical stresses (which are often extreme) and the temperature effects (which can sometimes be quite considerable) involved that roller table drives have to be configured very carefully indeed.

Application examples

Roller tables are used to convey cold or hot materials in the form of blocks, rods, pipes, or plates in rolling mills or metal processing plants.

- Working roller tables on both sides of the roll stand are meant to decelerate the material, which is usually red hot, in the fastest pass sequence possible, then accelerate it toward the stand.
- Entry tables convey the blocks or slabs from the oven to the working roller table.
- Delivery roller tables receive the product which has been rolled out into billets or rods and feeds it on for further machining (shears, straightening press, etc.).
- Cooling bed roller tables enable the material to be cooled down evenly by using a slow, oscillating motion.

Depending on the application in question, different criteria will come into play for selecting and dimensioning roller table drives.



Rollgang.JPG

Fig. 7-1 Roller table

Roller table

Outer roller radius	r_o	0.225 m
Inner roller radius	r_i	0.175 m
Surface length of roll (roller length)	l	2 m
Roller distance		1.25 m
Roller speed	n_{out}	52 min ⁻¹
Roller density	ρ	7,850 kg/m ³
Transport speed	v	1.25 m/s
Acceleration	a	1.25 m/s ²
Starting time	t_s	1 s
Efficiency	η	0.85

Rolling stock

Length		10 m
Maximum weight		38,000 kg
Minimum weight		3,000 kg
Number of supporting rollers (assumption: only every second roller supports any weight)	n	$= \frac{10 \text{ m}}{1.25 \text{ m} \cdot 2} = 4$
Max. load per roller	m_{max}	$= \frac{38000 \text{ kg}}{4} = 9500 \text{ kg}$
Min. load per roller	m_{min}	$= \frac{3000 \text{ kg}}{4} = 750 \text{ kg}$

Operating cycles

Switching frequency	120 making operations/h
Ambient temperature	40 °C
Temperature class/utilization of the temperature class	F/F

Derating factors for the motor:

At higher temperatures, the usual derating factors must be taken into account for the power and torque (see catalog D81.1, Low-Voltage Motors or catalog D87.1 MOTOX Geared Motors or SIZER)

40 °C to 45 °C = 0.96

45 °C to 50 °C = 0.92

The following derating factors apply at installation altitudes of over 1,000 m above sea level

1,000 m to 1,500 m = 0.97

1,500 m to 2,000 m = 0.94

For the utilization of the motor's temperature class

Utilization F/F = 1.0

Utilization F/B = 0.85

Roller mass

$$m = (r_o^2 - r_i^2) \cdot \pi \cdot l \cdot \rho = (0.225^2 - 0.175^2) \text{ m}^2 \cdot \pi \cdot 2 \text{ m} \cdot 7850 \frac{\text{kg}}{\text{m}^3} = 986 \text{ kg}$$

Moments of inertia

$$J_{\text{roller}} = \frac{1}{2} \cdot m \cdot (r_o^2 + r_i^2) = \frac{1}{2} \cdot 986 \text{ kg} \cdot (0.225^2 + 0.175^2) \text{ m}^2 = 40 \text{ kgm}^2$$

$$J_{\text{load max}} = m_{\text{max}} \cdot r_o^2 = 9500 \text{ kg} \cdot 0.225^2 \text{ m}^2 = 481 \text{ kgm}^2$$

$$J_{\text{load min}} = m_{\text{min}} \cdot r_o^2 = 750 \text{ kg} \cdot 0.225^2 \text{ m}^2 = 38 \text{ kgm}^2$$

$$J_{\text{roller}} + J_{\text{load, max}} = (40 + 481) \text{ kgm}^2 = 521 \text{ kgm}^2$$

Output speed of the geared motor

$$n_{\text{out}} = \frac{v}{2 \cdot r_o \cdot \pi} = \frac{1.25 \frac{\text{m}}{\text{s}}}{2 \cdot 0.225 \text{ m} \cdot \pi} = 0.88 \frac{1}{\text{s}} = 53 \text{ min}^{-1}$$

Torque for mass acceleration on a roller

$$M_{\text{a, roller}} = \frac{(J_{\text{roller}} + J_{\text{load, max}}) \cdot 2\pi \cdot \frac{n_{\text{out}}}{60}}{t_s} = \frac{521 \text{ kgm}^2 \cdot 2\pi \cdot \frac{53}{60 \text{ s}}}{1 \text{ s}} = 2881 \text{ Nm}$$

Required gear ratio

$$\begin{aligned} n_{\text{mot}} &= 1500 \text{ min}^{-1} \\ n_{\text{out}} &= 53 \text{ min}^{-1} \\ i_{\text{req}} &= \frac{n_{\text{mot}}}{n_{\text{out}}} = \frac{1500 \text{ min}^{-1}}{53 \text{ min}^{-1}} = 28.3 \end{aligned}$$

Selecting a motor

$$M_{\text{a, mot}} = \frac{M_{\text{a, roller}}}{i \cdot \eta} = \frac{2881 \text{ Nm}}{28.3 \cdot 0.85} = 120 \text{ Nm}$$

The impulse torque of the motor must be higher than this value.

Selected motor 1LP3 166-4RL...

Motor data:

$$P = 7.5 \text{ kW}, n_{\text{rated}} = 1,480 \text{ min}^{-1}, U = 400 \text{ V}, I = 17 \text{ A}$$

$$I_{\text{impulse}} = 51 \text{ A}, M_{\text{rated, motor}} = 48 \text{ Nm},$$

$$M_{\text{impulse}} = 144 \text{ Nm}, \cos \varphi = 0.74, \eta_{\text{mot}} = 0.88, J_{\text{motor}} = 0.052 \text{ kgm}^2$$

Selecting a gearbox

In the majority of cases, helical gearboxes are used for roller tables, but in some cases parallel shaft gearboxes may be used instead. The gearbox housing is made of ductile cast iron (GGG40), as is the roller table motor. Since the standard housing is made of cast iron, the appropriate option must be specified when ordering.

The motor is connected to the gearbox via a K4 distance piece.

No details are given here of any options or customer requirements such as lacquering, shaft diameter, etc.

A helical gearbox of the B3 type of construction (foot-mounted type) is needed here. The rated torque must be higher than the acceleration torque $M_{a, \text{roller}}$ calculated above. In this case, a gearbox with a rated torque around 1.5 times higher than $M_{a, \text{roller}}$ is selected.

$$M_{\text{rated, gearbox}} > 1.5 \cdot 2,881 \text{ Nm} = 4,321.5 \text{ Nm}$$

Gearbox Z128-K4-160 has been selected with the help of catalog D87.1 (MOTOX Geared Motors) and/or the configurator:

Data:

Z128-K4-160

$$i = 30.28$$

$$M_{\text{rated, gearbox}} = 5,100 \text{ Nm}$$

The rated input torque of the K4 distance piece is 98 Nm.

$$M_{\text{in}} = 98 \text{ Nm}$$

Recalculating the motor

As neither the moment of inertia of the motor nor the precise gear ratio has been taken into account yet, the motor needs to be recalculated.

Converting the external load moment of inertia into the motor speed

$$J^* = \frac{J_{\text{roller}} + J_{\text{load, max}}}{i^2} = \frac{521 \text{ kgm}^2}{30.28^2} = 0.568 \text{ kgm}^2$$

Total moment of inertia with motor

$$J_{\text{tot}} = J^* + J_{\text{motor}} = (0.568 + 0.052) \text{ kgm}^2 = 0.62 \text{ kgm}^2$$

$$M_{a, \text{mot}} = \frac{J_{\text{tot}} \cdot 2\pi \cdot \frac{n_{\text{mot}}}{60}}{t_o \cdot \eta \cdot \eta_{\text{mot}}} = \frac{0.62 \text{ kgm}^2 \cdot 2\pi \cdot \frac{1480}{60 \text{ s}}}{1 \text{ s} \cdot 0.85 \cdot 0.88} = 128 \text{ Nm}$$

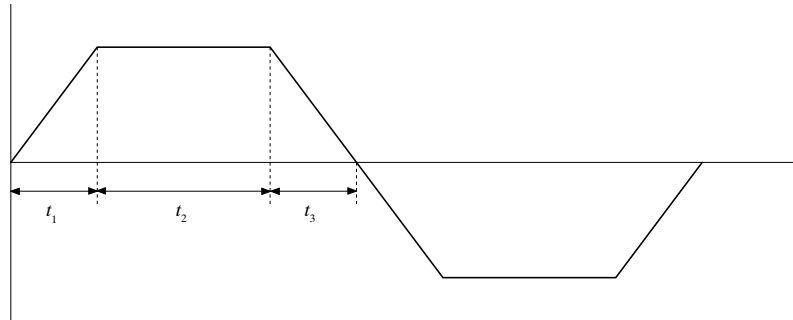
So, the following requirement is met

$$\underline{M_{\text{impulse}} \geq M_{a, \text{mot}}} \quad (144 \text{ Nm} > 128 \text{ Nm})$$

Thermal check of the motor

The effective torque M_{eff} and rms current I_{eff} are calculated for this purpose.

Duty cycle: 120 making operations/h



Zyklus.eps

Fig. 7-2 Cycle

So, one cycle lasts for $3,600 \text{ sec} / 240 = 15 \text{ sec}$.

$$t_1 + t_2 + t_3 = 15 \text{ s}$$

$t_1 = 1 \text{ sec}$ acceleration

$t_3 = 1 \text{ sec}$ deceleration (assumed)

$t_2 = 13 \text{ sec}$ constant travel

Effective torque

$$M_{\text{eff}} = \sqrt{\frac{M_1^2 \cdot t_1 + M_2^2 \cdot t_2 + M_3^2 \cdot t_3}{t_1 + t_2 + t_3}}$$

1. Acceleration

$$M_1 = M_{a, \text{mot}} = 128 \text{ Nm}$$

$$t_1 = 1 \text{ s}$$

2. Constant load

$$M_2 = 12 \text{ Nm}$$

(Assumption: 25 % of the rated torque $\rightarrow 0.25 \cdot 48 \text{ Nm} = 12 \text{ Nm}$)

$$t_2 = 13 \text{ s}$$

3. Deceleration

$$M_3 = J_{\text{tot}} \cdot 2\pi \cdot \frac{\eta_{\text{mot}}}{60} \cdot \eta \cdot \eta_{\text{mot}} = 0.62 \text{ kgm}^2 \cdot 2\pi \cdot \frac{1480}{60 \text{ s}} \cdot 0.85 \cdot 0.88 = 72 \text{ Nm}$$

$$t_3 = 1 \text{ s}$$

$$M_{\text{eff}} = \sqrt{\frac{(128 \text{ Nm})^2 \cdot 1\text{s} + (12 \text{ Nm})^2 \cdot 13\text{s} + (72 \text{ Nm})^2 \cdot 1\text{s}}{1\text{s} + 13\text{s} + 1\text{s}}} = 40 \text{ Nm}$$

As a result, the effective torque is lower than the rated motor torque and the rated input torque of the gearbox's K4 distance piece.

$$M_{\text{eff}} < M_{\text{rated_motor}} < M_{\text{in}} = 40 \text{ Nm} < 48 \text{ Nm} < 98 \text{ Nm}$$

rms current

$$I_{\text{eff}} = \sqrt{\frac{I_1^2 \cdot t_1 + I_2^2 \cdot t_2 + I_3^2 \cdot t_3}{t_1 + t_2 + t_3}}$$

Rated motor current $I_{\text{mot}} = 17 \text{ A}$, power factor $\cos \varphi = 0.74$

$$I_{\text{active, rated}} = I_{\text{mot}} \cdot \cos \varphi = 17 \text{ A} \cdot 0.74 = 12.58 \text{ A}$$

$$I_{\text{reactive}} = \sqrt{I_{\text{mot}}^2 - I_{\text{active, rated}}^2} = \sqrt{(17 \text{ A})^2 - (12.58 \text{ A})^2} = 11.43 \text{ A}$$

$$I_{\text{active1}} = \frac{M_1}{M_{\text{rated}}} \cdot I_{\text{active, rated}} = \frac{128}{48} \cdot 12.58 \text{ A} = 33.55 \text{ A}$$

$$I_1 = \sqrt{I_{\text{active1}}^2 + I_{\text{reactive}}^2} = \sqrt{(33.55 \text{ A})^2 + (11.43 \text{ A})^2} = 35.44 \text{ A}$$

$$I_{\text{active2}} = \frac{M_2}{M_{\text{rated}}} \cdot I_{\text{active, rated}} = \frac{12}{48} \cdot 12.58 \text{ A} = 3.15 \text{ A}$$

$$I_2 = \sqrt{I_{\text{active2}}^2 + I_{\text{reactive}}^2} = \sqrt{(3.15 \text{ A})^2 + (11.43 \text{ A})^2} = 11.86 \text{ A}$$

$$I_{\text{active3}} = \frac{M_3}{M_{\text{rated}}} \cdot I_{\text{active, rated}} = \frac{72}{48} \cdot 12.58 \text{ A} = 18.87 \text{ A}$$

$$I_3 = \sqrt{I_{\text{active3}}^2 + I_{\text{reactive}}^2} = \sqrt{(18.87 \text{ A})^2 + (11.43 \text{ A})^2} = 22.06 \text{ A}$$

$$I_{\text{eff}} = \sqrt{\frac{(35.44 \text{ A})^2 \cdot 1 \text{ s} + (11.86 \text{ A})^2 \cdot 13 \text{ s} + (22.06 \text{ A})^2 \cdot 1 \text{ s}}{1 \text{ s} + 13 \text{ s} + 1 \text{ s}}} = 15.4 \text{ A}$$

This means that the rms current is lower than the rated motor current.

$$I_{\text{eff}} < I_{\text{mot}} \Rightarrow 15.4 \text{ A} < 17 \text{ A}$$

The motor is OK.

Selecting the converter

Since reversing operation is assumed in this example, a converter that is capable of power recovery is recommended.

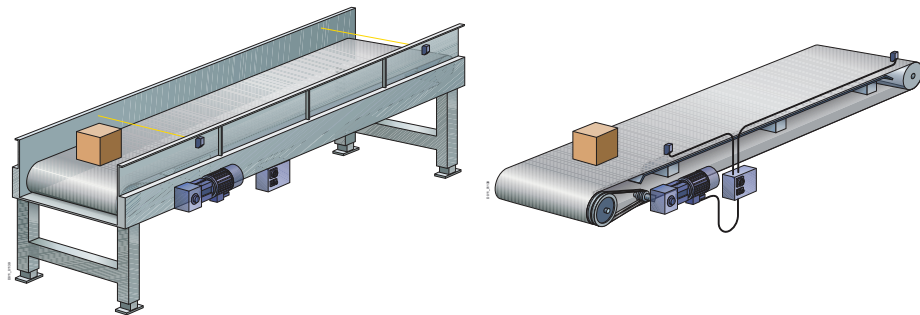
The converter is selected in the usual way. The maximum current of the motor or, in the case of group drives, the maximum total current of the motors must be lower than or equal to the maximum current of the converter. Furthermore, I_{eff} or the total of the rms currents must be lower than the rated current of the converter.

The choice of converter system (one single converter per motor, one single converter with several motors, or a multi-axis system with an infeed/regenerative feedback unit and several inverter modules) will ultimately depend on the customer's requirements.



Example of a belt conveyor

Belt conveyors are used for piece goods, for example, which are transported by means of a conveying belt. The belt glides over a surface (table, plate, etc.) and can be fed back directly as shown in the image, or via additional guide pulleys. This type of belt is used in airport baggage handling systems, for example, as it is suitable for bags of various sizes and strengths.



...Gurtbahn.../ ...Gurt.PDF

Fig. 8-1 Belt path and belt

Loading	m_l	25 kg/m
Belt weight	m_b	4.2 kg/m
Length of the conveyor	L	6 m
Velocity	v	72 m/min = 1.2 m/s
Tractive resistance	w	3.5 N/kg
Driving roller diameter	d	240 mm
Incline	α	15°
Efficiency of the system	η	0.85
Required service factor	$f_{s, req}$	1.5
Required switching frequency	z_{req}	350 h ⁻¹
Duty ratio	DR	80 %

The moments of inertia of the driving and the tensioning rollers can be left out of consideration.

The drive is to take the form of a single-speed bevel helical geared motor.

Required steady power P_{req}

$$P_{\text{req}} = \frac{L \cdot (m_l + m_b)}{\eta} \cdot (w \cdot \cos \alpha + g \cdot \sin \alpha) \cdot v$$

$$P_{\text{req}} = \frac{6 \text{ m} \cdot \left(25 \frac{\text{kg}}{\text{m}} + 4.2 \frac{\text{kg}}{\text{m}}\right)}{0.85} \cdot \left(3.5 \frac{\text{N}}{\text{kg}} \cdot \cos 15^\circ + 9.81 \frac{\text{m}}{\text{s}^2} \cdot \sin 15^\circ\right) \cdot 1.2 \frac{\text{m}}{\text{s}} = 1464 \text{ W}$$

Selected motor: 2.2 kW, 4-pole

$$P_{\text{rated}} = 2.2$$

$$M_{\text{rated}} = 14.8 \text{ Nm}$$

$$n_{\text{rated}} = 1,420 \text{ min}^{-1}$$

$$J_m = 4.66 \cdot 10^{-3} \text{ kgm}^2 \text{ including brake}$$

Required output speed n_{out}

$$n_{\text{out}} = \frac{v_H}{\pi \cdot d} = \frac{72 \frac{\text{m}}{\text{min}}}{\pi \cdot 0.24 \text{ m}} = 95.49 \text{ min}^{-1}$$

Required gear ratio i_{set}

$$i_{\text{set}} = \frac{n_{\text{motor}}}{n_{\text{out}}} = \frac{1420 \text{ min}^{-1}}{95.49 \text{ min}^{-1}} = 14.87$$

Selected gearbox: K 48

This gearbox was selected due to the required gear ratio and the required service factor of 1.5.

Data:

$$i = 15.42$$

$$M_{\text{rated}} = 450 \text{ Nm}$$

$$f_s = \frac{450 \text{ Nm}}{14.8 \text{ Nm} \cdot 15.42} = 2$$

Permissible switching frequency

$z_0 = 6500 \text{ h}^{-1}$ (applies to a motor with a brake and brake rectifier with over-excitation)

$$M_H = 37 \text{ Nm}$$

The constant load torque M_1 which occurs and the external moment of inertia J_{ext} also need to be calculated.

$$M_1 = \frac{P_{\text{req}}}{2\pi \cdot n_{\text{rated}}} = \frac{60 \cdot 1464 \text{ W}}{2\pi \cdot 1420 \frac{1}{\text{min}}} = 9.8 \text{ Nm}$$

The use of the rated motor speed n_{rated} is an approximation; as the motor is not operated at its rated power, its speed will change accordingly.

$$k_M = 1 - \frac{M_1}{M_H} = 1 - \frac{9.8 \text{ Nm}}{37 \text{ Nm}} = 0.735$$

$$J_{\text{ext}} = L \cdot (m_1 + 2 \cdot m_b) \cdot \left(\frac{d}{2}\right)^2 \cdot \frac{1}{i^2} = 6 \text{ m} \cdot \left(25 \frac{\text{kg}}{\text{m}} + 2 \cdot 4.2 \frac{\text{kg}}{\text{m}}\right) \cdot \left(\frac{0.24 \text{ m}}{2}\right)^2 \cdot \frac{1}{15.42^2} = 0.0121 \text{ kgm}^2$$

$$k_{\text{Fi}} = \frac{J_M}{J_M + J_{\text{ext}}} = \frac{4.66 \cdot 10^{-3}}{4.66 \cdot 10^{-3} + 0.0121} = 0.278$$

The k_p switching frequency diagram (see Fig. 2-13 in Chapter 2.2) is used to determine factor k_p .

$$\text{With } \frac{P_1}{P_N} = \frac{1.46 \text{ kW}}{2.2 \text{ kW}} = 0.66 \text{ and } DR = 80 \%, \text{ this results in: } k_p = 0.75$$

$$z = z_0 \cdot k_M \cdot k_{\text{Fi}} \cdot k_p = 6500 \frac{1}{\text{h}} \cdot 0.735 \cdot 0.278 \cdot 0.75 = 996 \text{ h}^{-1}$$

Calculation of the acceleration which arises

$$\text{The following generally applies: } a = \frac{v}{t_{\text{start}}}$$

Starting time t_{start} also needs to be calculated, therefore, along with the actual rated velocity v , which results from the rated motor speed.

$$t_{\text{start}} = \frac{\left(J_M + \frac{J_{\text{ext}}}{\eta}\right) \cdot n_{\text{mot}} \cdot 2\pi}{M_H - M_1} = \frac{\left(4.66 \cdot 10^{-3} + \frac{0.0121}{0.85}\right) \text{ kgm}^2 \cdot \frac{1420}{60 \text{ s}} \cdot 2\pi}{(37 - 9.8) \text{ Nm}} = 0.1 \text{ s}$$

$$v = \frac{\pi \cdot n_{\text{mot}} \cdot d}{i} = \frac{\pi \cdot 1420 \text{ min}^{-1} \cdot 0.24 \text{ m}}{15.42} = 69 \frac{\text{m}}{\text{min}} = 1.16 \frac{\text{m}}{\text{s}}$$

$$a = \frac{v}{t_{\text{start}}} = \frac{1.16 \frac{\text{m}}{\text{s}}}{0.1 \text{ s}} = 11.6 \frac{\text{m}}{\text{s}^2}$$

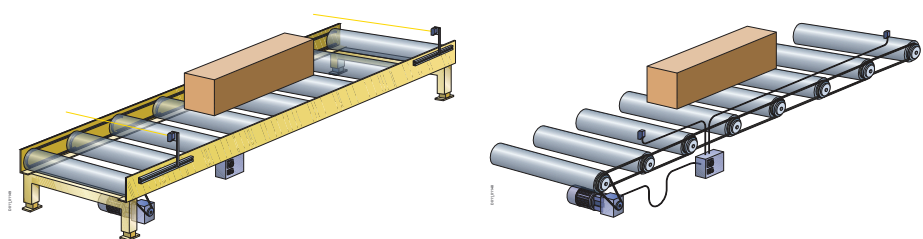
Result

The motor meets the requirements in question. The required switching frequency would not be achieved if the next motor size down (with a rating of 1.5 kW) were to be selected. The acceleration which arises must be discussed with the customer. If it is too high, a frequency converter can be used.



Example of a roller conveyor

Wooden pallets containing stones are to be transported on a roller conveyor.



... Rollenbahn.../...mit Aufbau ...pdf

Fig. 9-1 Example: Roller conveyors

The efficiency for each complete chain wrap is assumed to be 0.99. Therefore, the total efficiency η_{tot} of the chain transfers is $\eta_{\text{tot}} = 0.99^n$, where n is the number of driven rollers.

Transport speed	v	25 m/min
Maximum permissible acceleration	a	0.5 m/s ²
Mass of the load	m_l	4,000 kg
Mass of one roller	m_r	60 kg
Roller diameter (outer)	D	100 mm
Pin diameter	D_s	40 mm
Number of driven rollers	z_r	10
Auxiliary transmission	i_{tra}	1.67
Operating hours per day		16 h
Operations per hour		10 h ⁻¹

Specific tractive resistance

$$w_F = \frac{2}{D} \cdot \left(\frac{D_s}{2} \cdot \mu_r + f \right) + c$$

The rollers are fitted with ball bearings, so $\mu_r = 0.005$.

Wooden pallets on steel rollers equates to $f = 1.2$ mm.

As no lateral traction is applied, $c = 0$.

$$w_F = \frac{2}{100} \cdot \left(\frac{40}{2} \cdot 0.005 + 1.2 \right) = 0.026$$

Tractive resistance force

$$F_w = m_l \cdot g \cdot w_F = 4000 \text{ kg} \cdot 9.81 \frac{\text{m}}{\text{s}^2} \cdot 0.026 = 1020 \text{ N}$$

Efficiency

$$\eta_{\text{tot}} = 0.99^n = 0.99^{10} = 0.9$$

Required steady power

$$P = \frac{F_w \cdot v}{\eta_{\text{tot}}} = \frac{1020 \text{ N} \cdot 25 \frac{\text{m}}{60 \text{ s}}}{0.9} = 472 \text{ W}$$

Output speed at the gearbox

$$n_{\text{out}} = \frac{v \cdot i_{\text{tra}}}{\pi \cdot D} = \frac{25 \frac{\text{m}}{\text{min}} \cdot 1.67}{\pi \cdot 0.1 \text{ m}} = 133 \text{ min}^{-1}$$

Required gear ratio

For a 4-pole motor, the rated speed is around $1,450 \text{ min}^{-1}$.

$$i_{\text{req}} = \frac{n_{\text{mot}}}{n_{\text{out}}} = \frac{1450 \text{ min}^{-1}}{133 \text{ min}^{-1}} = 10.9$$

Required steady torque at the motor

$$M_{\text{stat}} = \frac{F_w \cdot D}{\eta_{\text{tot}} \cdot 2 \cdot i_{\text{tra}} \cdot i_{\text{req}}} = \frac{1020 \text{ N} \cdot 0.1 \text{ m}}{0.9 \cdot 2 \cdot 1.67 \cdot 10.9} = 3.11 \text{ Nm}$$

External moments of inertia

The external moment of inertia is comprised of that of the load and that of the rollers.

$$J_{\text{ext}} = J_l + J_r$$

$$J_l = m_l \cdot \left(\frac{D}{2}\right)^2 = 4000 \text{ kg} \cdot \left(\frac{0.1 \text{ m}}{2}\right)^2 = 10 \text{ kgm}^2$$

$$J_r = z_r \cdot \frac{1}{2} \cdot m_r \cdot \left(\frac{D}{2}\right)^2 = 10 \cdot \frac{1}{2} \cdot 60 \text{ kg} \cdot \left(\frac{0.1 \text{ m}}{2}\right)^2 = 0.75 \text{ kgm}^2$$

In relation to the motor side, the following results

$$J_{\text{ext}} = \frac{J_l + J_r}{(i_{\text{tra}} \cdot i_{\text{req}})^2} = \frac{(10 + 0.75) \text{ kgm}^2}{(1.67 \cdot 10.9)^2} = 32.6 \cdot 10^{-3} \text{ kgm}^2$$

Required dynamic torque at the motor

(without taking the moment of inertia of the motor into account, for making a rough calculation of the motor)

$$M_{\text{dyn}} = \frac{(m_l + m_r^*) \cdot a \cdot D}{\eta_{\text{tot}} \cdot 2 \cdot i_{\text{tra}} \cdot i_{\text{req}}}$$

Here, m_r^* is the mass of all rollers, converted into a linear motion.

$$m_r^* = \frac{J_r}{\left(\frac{D}{2}\right)^2} = \frac{0.75 \text{ kgm}^2}{\left(\frac{0.1 \text{ m}}{2}\right)^2} = 300 \text{ kg}$$

The acceleration is assumed to be 0.2 m/s^2 .

$$M_{\text{dyn}} = \frac{(m_l + m_r^*) \cdot a \cdot D}{\eta_{\text{tot}} \cdot 2 \cdot i_{\text{tra}} \cdot i_{\text{req}}} = \frac{(4000 + 300) \text{ kg} \cdot 0.2 \frac{\text{m}}{\text{s}^2} \cdot 0.1 \text{ m}}{0.9 \cdot 2 \cdot 1.67 \cdot 10.9} = 2.62 \text{ Nm}$$

Acceleration torque required at the motor

(without taking the moment of inertia of the motor into account, for making a rough calculation of the motor)

$$M_{\text{acc}} = M_{\text{stat}} + M_{\text{dyn}} = 3.11 \text{ Nm} + 2.62 \text{ Nm} = 5.73 \text{ Nm}$$

Selected motor

$$P_{\text{rated}} = 0.55 \text{ kW}$$

$$M_{\text{rated}} = 3.8 \text{ Nm}$$

$$n_{\text{mot}} = 1,375 \text{ min}^{-1}$$

$$M_{\text{H}} = 8 \text{ Nm (acceleration torque)}$$

$$J_{\text{mot}} = 0.0011 \text{ kgm}^2$$

J_{brake} is not taken into account here.

Required service factor for the gearbox

Here, the shock load or the ratio of J_{ext} to J_{mot} must be determined first.

$$\frac{J_{\text{ext}}}{J_{\text{mot}}} = \frac{32.6 \cdot 10^{-3}}{1.1 \cdot 10^{-3}} = 29$$

The moment of inertia of the motor can be increased by using a *flywheel fan*, which would result in

$$\frac{J_{\text{ext}}}{(J_{\text{mot}} + J_{\text{add}})} = \frac{32.6 \cdot 10^{-3}}{(1.1 + 1.71) \cdot 10^{-3}} = 11$$

This corresponds to shock load III.

With an operation time of 16 h per day, 10 making operations per hour (i.e. 20 on and off switching operations), and shock load III, $f_{\text{s, req}} = 1.6$.

Selected gearbox

Z18

$$i_{\text{g}} = 10.88$$

$$M_{\text{rated, gearbox}} = 87 \text{ Nm}$$

$$f_{\text{s}} = \frac{M_{\text{rated, gearbox}}}{i_{\text{g}} \cdot M_{\text{rated, motor}}} \cdot \frac{87 \text{ Nm}}{10.88 \cdot 3.8 \text{ Nm}} = 2.1$$

Recalculating the acceleration

$$a = \frac{v}{t_{\text{start}}}$$

The starting time t_{start} and the actual rated velocity v based on the rated motor speed, also need to be calculated.

$$t_{\text{start}} = \frac{\left((J_M + J_{\text{add}}) + \frac{J_{\text{ext}}}{\eta_{\text{ges}}} \right) \cdot n_{\text{mot}} \cdot 2\pi}{M_H - M_{\text{stat}}}$$

The value calculated for the actual gear ratio i_g must be used for J_{ext} .

$$t_{\text{start}} = \frac{\left((1.1 + 1.71) \cdot 10^{-3} + \frac{32.6 \cdot 10^{-3}}{0.9} \right) \text{ kgm}^2 \cdot \frac{1375}{60 \text{ s}} \cdot 2\pi}{(8 - 3.11) \text{ Nm}} = 1.15 \text{ s}$$

$$v = \frac{\pi \cdot n_{\text{mot}} \cdot D}{i_{\text{tra}} \cdot i_g} = \frac{\pi \cdot 1375 \text{ min}^{-1} \cdot 0.1 \text{ m}}{1.67 \cdot 10.88} = 23.8 \frac{\text{m}}{\text{min}} = 0.4 \frac{\text{m}}{\text{s}}$$

$$a = \frac{v}{t_{\text{start}}} = \frac{0.4 \frac{\text{m}}{\text{s}}}{1.15 \text{ s}} = 0.35 \frac{\text{m}}{\text{s}^2}$$

This value is lower than the maximum permissible acceleration of 0.5 m/s^2 , so it is OK.

Permissible switching frequency

The low number of making operations means that the permissible switching frequency of the motor does not have to be recalculated.

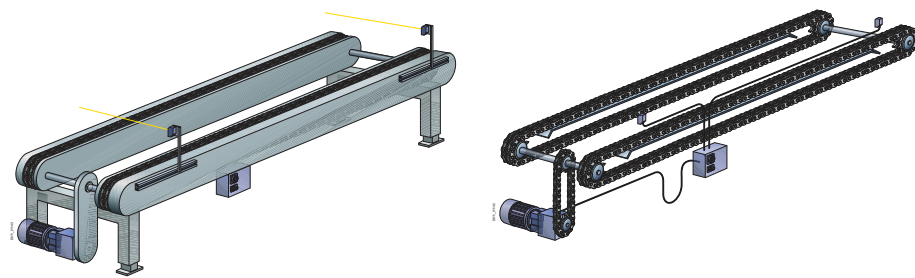
Selected geared motor

Z18 ($i = 10.88$) LA71ZMP4I (0.55 kW)



Example of a chain conveyor with a frequency converter

10



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Fig. 10-1 Chain conveyor with and without external structure

The tractive resistance of chain conveyors can lie between around 1,400 and 4,000 N/t. In most cases, you will need to consult the manufacturer of the slide rails and take the relevant environmental requirements into account very carefully.

Within the context of automobile production, chassis frames are to be transported to final assembly. The drive is to take the form of a 4-pole bevel helical geared motor. The plant operates on a three-shift rotation.

Velocity	v	24 m/min
Acceleration	a	0.1 m/s ²
Sliding friction	μ	0.3 (between chain and slide rail)
Load mass	m_l	2,100 kg
Chain weight	m_c	10 kg/m
Number of chains	z_c	2
Wheel diameter	d_c	157 mm (conveyor chains)
Wheel diameter	d_d	88 mm (driving chain)
Additional force	F_{add}	1,500 N
Additional mass	m_{add}	3,500 kg
Efficiency	η	0.85
Length of the conveyor	L	6 m

The two wheel diameters for the driving chain are identical, so the auxiliary transmission gear ratio is 1.

The additional force occurs because parts hanging over the frame are pulled along too. The weight of these parts is $m_{\text{add}} = 3,500 \text{ kg}$.

Total weight of the chains

$$m_{\text{c, total}} = m_{\text{c}} \cdot z_{\text{c}} \cdot \text{LengthChain} = m_{\text{c}} \cdot z_{\text{c}} \cdot 2 \cdot L = 10 \frac{\text{kg}}{\text{m}} \cdot 2 \cdot 2 \cdot 6 \text{ m} = 240 \text{ kg}$$

Tractive resistance force

The chains run on rails above and below, so the total weight of the chains must be taken into account.

$$F_{\text{res}} = (m_{\text{l}} + m_{\text{c, total}}) \cdot g \cdot \mu = (2100 + 240) \text{ kg} \cdot 9.81 \frac{\text{m}}{\text{s}^2} \cdot 0.3 = 6887 \text{ N}$$

Required steady power

$$P = \frac{(F_{\text{res}} + F_{\text{add}}) \cdot v}{\eta_{\text{tot}}} = \frac{(6887 \text{ N} + 1500 \text{ N}) \cdot 24 \frac{\text{m}}{60 \text{ s}}}{0.85} = 3947 \text{ W}$$

Selected motor (standard asynchronous motor)

$$P_{\text{rated}} = 5.5 \text{ kW}$$

$$M_{\text{rated}} = 36 \text{ Nm}$$

$$n_{\text{mot}} = 1,455 \text{ min}^{-1}$$

$$J_{\text{mot}} = 0.0209 \text{ kgm}^2$$

J_{brake} is not taken into account here.

Output speed at the gearbox

$$n_{\text{out}} = \frac{v \cdot i_{\text{aux}}}{\pi \cdot d_{\text{c}}} = \frac{24 \frac{\text{m}}{\text{min}} \cdot 1}{\pi \cdot 0.157 \text{ m}} = 49.66 \text{ min}^{-1}$$

Required gear ratio

For a 4-pole motor, the rated speed is around $1,450 \text{ min}^{-1}$.

$$i_{\text{req}} = \frac{n_{\text{mot}}}{n_{\text{out}}} = \frac{1450 \text{ min}^{-1}}{49.66 \text{ min}^{-1}} = 29.8$$

External moment of inertia

The external moment of inertia is comprised of the load acting on the chains (m_l), the total chain weight ($m_{c, \text{total}}$), and the parts which are pulled along too (m_{add}).

$$J_{\text{ext}} = (m_l + m_{c, \text{total}} + m_{\text{add}}) \cdot \left(\frac{d_c}{2}\right)^2 \cdot \frac{1}{i_{\text{req}}^2} =$$

$$= (2100 + 240 + 3500) \text{ kg} \cdot \left(\frac{0.157 \text{ m}}{2}\right)^2 \cdot \frac{1}{29.8^2} = 0.0405 \text{ kgm}^2$$

Required service factor for the gearbox

Here, the shock load or the ratio of J_{ext} to J_{mot} must be determined first.

$$\frac{J_{\text{ext}}}{J_{\text{mot}}} = \frac{0.0405}{0.0209} = 1.94$$

This corresponds to shock load II.

Shock load II and an operation time of 24 h per day (three-shift rotation) results in a required service factor $f_{s, \text{req}}$ of **1.4 to 1.6**, depending on the switching frequency.

Selecting the gearbox

The maximum required rated torque is

$$M_{\text{req}} = f_{s, \text{req}} \cdot i_{\text{req}} \cdot M_N = 1.6 \cdot 29.8 \cdot 36 \text{ Nm} = 1716 \text{ Nm}$$

K88

$$M_{\text{rated, g}} = 1,650 \text{ Nm}$$

$$i_g = 28.5$$

$$f_s = \frac{M_{\text{rated, g}}}{i_g \cdot M_{\text{rated, motor}}} = \frac{1650 \text{ Nm}}{28.5 \cdot 36 \text{ Nm}} = 1.61$$

Recalculating the acceleration procedure

1. Can the required acceleration be provided?

The required acceleration torque is calculated for this purpose.

$$M_{\text{acc}} = M_{\text{stat}} + M_{\text{dyn}}$$

$$M_{\text{stat}} = \frac{(F_{\text{res}} + F_{\text{add}}) \cdot d_c}{2 \cdot \eta \cdot i_g} = \frac{(6887 + 1500) \text{ N} \cdot 0.157 \text{ m}}{2 \cdot 0.85 \cdot 28.5} = 27.18 \text{ Nm}$$

$$M_{\text{dyn}} = \frac{(m_l + m_{c, \text{total}} + m_{\text{add}}) \cdot a \cdot d_c}{2 \cdot \eta \cdot i_g} = \frac{(2100 + 240 + 3500) \text{ kg} \cdot 0.1 \cdot \frac{\text{m}}{\text{s}^2} \cdot 0.157 \text{ m}}{2 \cdot 0.85 \cdot 28.5} = 1.89 \text{ Nm}$$

$$M_{\text{acc}} = M_{\text{stat}} + M_{\text{dyn}} = 27.18 \text{ Nm} + 1.89 \text{ Nm} = 28.07 \text{ Nm}$$

At 28 Nm, the acceleration torque is lower than the rated torque of the motor (36 Nm), so the required acceleration can be achieved.

With a rating of 4 kW, the next motor size down would theoretically be at the limits of the required values, so could be used for very specific rated conditions. However, considering the uncertainties stated at the beginning of this section regarding the tractive resistance and the friction value, and since a certain degree of certainty is required, the 5.5 kW motor should be used.

2. Permissible acceleration

Parts which are not actually on the chain are also pulled along. Therefore, a check needs to be performed to see whether or not the resulting additional force causes the parts and the chassis frame to slip on the chain.

The permissible force (F_{perm}) must be higher than the required driving force during acceleration (F_{acc}).

$$F_{\text{perm}} \geq F_{\text{acc}}$$

$$F_{\text{acc}} = \frac{M_{\text{acc}} \cdot 2 \cdot i_g}{d_c} = \frac{28.07 \text{ Nm} \cdot 2 \cdot 28.5}{0.157 \text{ m}} = 10191 \text{ N}$$

$$F_{\text{perm}} = (m_l + m_c \cdot z_c \cdot L) \cdot g \cdot \mu_x$$

μ_x is the friction value between the chain and the chassis frame. As a comparison with the calculation of the tractive resistance shows, μ_x must always be higher than μ .

The equations described above are used to determine the friction value μ_x , which is the minimum required value. The material on the lower side of the chassis frame should be chosen accordingly. Alternatively, a positive connection can be provided.

Radial force check for the gearbox

The radial force which arises is:
$$F = \frac{M \cdot 2 \cdot C}{d}$$

Here, M is the torque which arises at the gearbox output shaft. M needs to be determined, depending on the application in question (division of the static and dynamic component, switching frequency, for example). d is the diameter of the gear wheel, chain wheel, etc., attached to the gearbox shaft.

If chain pretensioning forces occur, these must be taken into account too.

We are using the static torque here, i.e. 27.18 Nm. Factor C , which refers to the type of output element, is 1, as we are dealing with one gear wheel with more than 17 teeth, See Chapter 2.1, Table 2-3.

$$F = \frac{M_{\text{stat}} \cdot i_g \cdot 2}{d} = \frac{27.18 \text{ Nm} \cdot 28.5 \cdot 2}{0.088 \text{ m}} = 17605 \text{ N}$$

According to the catalog, this value is higher than the permissible radial force with a standard bearing arrangement. If a **reinforced** output shaft bearing arrangement is used, the selected gearbox will be OK.

Selecting the converter

A converter which matches the power of the motor needs to be found, in accordance with the requirements relating to the degree of protection and installation approach (control cabinet/distributed). In this case, a distributed system has been set up with PROFINET, so a G120D converter is used.

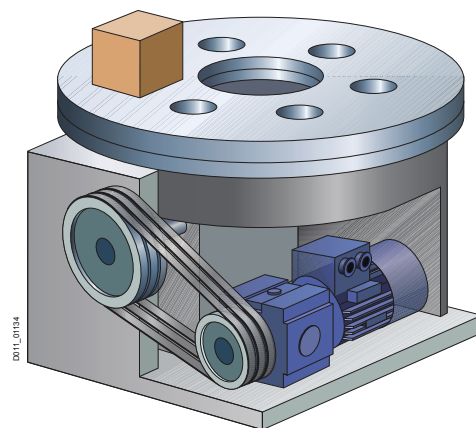
Selected products

Gearbox	Bevel helical gearbox K88 ($i = 28.5$) with reinforced output shaft bearing arrangement
Motor	Standard asynchronous motor 5.5 kW, 4-pole
Converter	Distributed, fail-safe G120D with communications interface, power 5.5 kW



Example of a turntable with a frequency converter

11



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Fig. 11-1 Turntable

Table diameter	D_{table}	3 m	
Mass of the table	m_{table}	2,000 kg	
Mass of the workpiece	m_{load}	350 kg	
Workpiece distance	L	2 m	(from the center of the table)
Auxiliary transmission gear ratio	i_{tra}	5.3	(includes all transmission ratios between the gearbox output speed and the table speed)
Rotation angle	α	90°	
Time for rotation angle		6 s	
Acceleration	a	0.6 m/s ² (maximum)	
Rangeability		1 : 10	
Switching frequency		One motion per minute	
Daily operation time		18 h	
Efficiency	η	0.9	

The table bearings take the form of a ball bearing slewing rim with $\mu = 0.008$.
Bearing running tread diameter $D_b = 1.27$ m.

The reference diameter for velocities, acceleration, and stopping accuracy is the table diameter.

Moment of inertia of the table based on the center of the table

$$J_{\text{table}} = \frac{1}{2} \cdot m_{\text{table}} \cdot \left(\frac{D_{\text{table}}}{2} \right)^2 = \frac{1}{2} \cdot 2000 \text{ kg} \cdot \left(\frac{3 \text{ m}}{2} \right)^2 = 2250 \text{ kgm}^2$$

Moment of inertia of the workpiece based on the center of the table

$$J_{\text{load}} = J_{\text{workpiece}} + m_{\text{load}} \cdot L^2$$

Here, $J_{\text{workpiece}}$ is the moment of inertia of the workpiece and $(m_{\text{load}} \cdot L^2)$ is the *parallel axis component*, as the center of rotation is the center of the table; see also Chapter 4.2, Parallel axis theorem.

In this example, we have assumed that the moment of inertia of the workpiece should be ignored in favor of that of the table, so that

$$J_{\text{load}} = m_{\text{load}} \cdot L^2 = 350 \text{ kg} \cdot (2 \text{ m})^2 = 1400 \text{ kgm}^2$$

Otherwise, the moment of inertia of the workpiece will have to be calculated using the formulas given in Chapter 4.2.

Distance around the table circumference

The rotation angle of 90° equates to one quarter of the table circumference.

$$s = \frac{1}{4} \cdot \text{TableCircumference} = \frac{1}{4} \cdot \pi \cdot D_{\text{table}} = 2.356 \text{ m}$$

Table circumferential velocity

$$v = \frac{s}{\text{Time for Rotation Angle}} = \frac{2.356 \text{ m}}{6 \text{ s}} = 0.39 \frac{\text{m}}{\text{s}}$$

Table speed

This does not take the acceleration and deceleration procedures into account.

One quarter of a revolution must be performed in 6 s:

$$n_{\text{table}} = \frac{1}{4} \cdot \frac{1}{6 \text{ s}} = 0.0417 \text{ s}^{-1} = 2.5 \text{ min}^{-1}$$

Required gear ratio

We have assumed that a 4-pole motor is being used:

$$i_{\text{req}} = \frac{n_{\text{motor}}}{n_{\text{table}} \cdot i_{\text{tra}}} = \frac{1450 \text{ min}^{-1}}{2.5 \text{ min}^{-1} \cdot 5.3} = 109.43$$

Static friction torque

$$M = \frac{\mu}{2\eta} \cdot (4.4 \cdot M_{\text{tilting}} + F_{\text{axial}} \cdot D_L + 2.2 \cdot F_{\text{radial}} \cdot D_1 \cdot 1.73) \cdot \frac{1}{i_{\text{req}} \cdot i_{\text{tra}}}$$

In this example, no radial force occurs.

The resulting axial force is:

$$F_{\text{axial}} = (m_{\text{table}} + m_{\text{load}}) \cdot g = (2000 \text{ kg} + 350 \text{ kg}) \cdot 9.81 \frac{\text{m}}{\text{s}^2} = 23054 \text{ N}$$

A tilting torque results if the load on the table is asymmetrical, as in the example.

$$M_{\text{tilting}} = m_{\text{load}} \cdot g \cdot L = 350 \text{ kg} \cdot 9.81 \frac{\text{m}}{\text{s}^2} \cdot 2 \text{ m} = 6867 \text{ Nm}$$

$$M_{\text{stat}} = \frac{0.008}{2 \cdot 0.9} \cdot (4.4 \cdot M_{\text{tilting}} + F_{\text{axial}} \cdot D_L) \cdot \frac{1}{i_{\text{req}} \cdot i_{\text{tra}}}$$

$$M_{\text{stat}} = \frac{0.008}{2 \cdot 0.9} \cdot (4.4 \cdot 6867 \text{ Nm} + 23053 \text{ N} \cdot 1.27) \cdot \frac{1}{109.43 \cdot 5.3} = 0.46 \text{ Nm}$$

Calculation of the acceleration time

$$t = \frac{v}{a} = \frac{0.39 \frac{\text{m}}{\text{s}}}{0.6 \frac{\text{m}}{\text{s}^2}} = 0.65 \text{ s}$$

Dynamic torque

$$M_{\text{dyn}} = \frac{\frac{(J_{\text{table}} + J_{\text{load}})}{\eta} \cdot \left(\frac{1}{i_{\text{req}} \cdot i_{\text{tra}}}\right)^2 \cdot 2\pi \cdot n_{\text{mot}}}{t}$$

$$M_{\text{dyn}} = \frac{\frac{(2250 \text{ kgm}^2 + 1400 \text{ kgm}^2)}{0.9} \cdot \left(\frac{1}{109.43 \cdot 5.3}\right)^2 \cdot 2\pi \cdot \frac{1450}{60 \text{ s}}}{0.65 \text{ s}} = 2.82 \text{ Nm}$$

Required acceleration torque

$$M_{\text{acc}} = M_{\text{stat}} + M_{\text{dyn}} = 0.46 \text{ Nm} + 2.82 \text{ Nm} = 3.28 \text{ Nm}$$

The dynamic torque for accelerating the motor is not yet included in this value.

Selected motor

On the frequency converter, the motor accelerates with at least 1.6 times its acceleration torque.

$$M_{\text{rated, motor, required}} = \frac{M_{\text{acc}}}{1.6} = \frac{3.28 \text{ Nm}}{1.6} = 2.1 \text{ Nm}$$

This results in a motor with the following data:

$$P_{\text{rated}} = 0.37 \text{ kW}$$

$$n_{\text{rated}} = 1,370 \text{ min}^{-1}$$

$$M_{\text{rated}} = 2.6 \text{ Nm}$$

$$J_{\text{M}} = 0.00097 \text{ kgm}^2 \text{ including brake}$$

$$I_{\text{rated}} = 1.06 \text{ A}$$

Required service factor

To calculate the required service factor, first of all the shock load must be determined, i.e. the ratio of the external moment of inertia to the motor moment of inertia.

$$\frac{J_{\text{ext}}}{J_{\text{mot}}} = \frac{(J_{\text{table}} + J_{\text{load}})}{(i_{\text{req}} \cdot i_{\text{tra}})^2} \cdot \frac{1}{J_{\text{mot}}} = \frac{(2250 \text{ kgm}^2 + 1400 \text{ kgm}^2)}{(109.43 \cdot 5.3)^2} \cdot \frac{1}{0.00097 \text{ kgm}^2} = 11$$

According to Table 2-1, this gives a shock load of III, See Chapter 2.1, Page 2-16 onwards.

The switching frequency per hour can be determined from the specification that there is one motion per minute as follows: 60 making operations + 60 braking operations = 120 operations per hour.

The daily operating duration is calculated from the daily operation time of 18 h per day, the time for one motion sequence of 6 s, and the switching frequency of one motion per minute. The gearbox moves for 6 s per minute, i.e. one tenth of the time, which equates to 1.8 h per day.

This results in a service factor $f_{\text{s, req}}$ of 1.4; see Table 2-2, Chapter 2.1.

Required gearbox rated torque

$$M_{\text{rated, gearbox, required}} = M_{\text{rated, motor}} \cdot i_{\text{req}} \cdot f_{\text{s, req}} = 2.6 \text{ Nm} \cdot 109 \cdot 1.4 = 398 \text{ Nm}$$

Selected gearbox

D48 helical gearbox

$$i_g = 103$$

$$M_{\text{rated, g}} = 450 \text{ Nm}$$

$$f_s = \frac{M_{\text{rated, g}}}{i_g \cdot M_{\text{rated, motor}}} = \frac{450 \text{ Nm}}{103 \cdot 2.6 \text{ Nm}} = 1.7$$

Recalculating the selected motor

The selected motor is recalculated using the actual transmission ratio of the selected gearbox. The required acceleration torque is determined for this purpose.

$$M_{\text{stat}} = \frac{0.008}{2\eta} \cdot (4.4 \cdot M_{\text{tilting}} + F_{\text{axial}} \cdot D_1) \cdot \frac{1}{i_g \cdot i_{\text{tra}}}$$

$$M_{\text{stat}} = \frac{0.008}{2 \cdot 0.9} \cdot (4.4 \cdot 6867 \text{ Nm} + 23053 \cdot 1.27 \text{ m}) \cdot \frac{1}{103 \cdot 5.3} = 0.5 \text{ Nm}$$

$$J_{\text{ext}} = (J_{\text{load}} + J_{\text{table}}) \cdot \frac{1}{(i_g \cdot i_{\text{tra}})^2} = (1400 \text{ kgm}^2 + 2250 \text{ kgm}^2) \cdot \frac{1}{103 \cdot 5.3^2} = 0.01225 \text{ kgm}^2$$

$$n_{\text{mot}} = n_{\text{table}} \cdot i_g \cdot i_{\text{tra}} = 2.5 \text{ min}^{-1} \cdot 103 \cdot 5.3 = 1365 \text{ min}^{-1}$$

$$M_{\text{dyn}} = \frac{\left(J_M + \frac{J_{\text{ext}}}{\eta}\right) \cdot 2\pi \cdot n_{\text{mot}}}{t} = \frac{\left(0.00097 \text{ kgm}^2 + \frac{0.01225 \text{ kgm}^2}{0.9}\right) \cdot 2\pi \cdot \frac{1365}{60 \text{ s}}}{0.65 \text{ s}} = 3.2 \text{ Nm}$$

Acceleration torque required per motor

$$M_{\text{acc}} = M_{\text{stat}} + M_{\text{dyn}} = 0.5 \text{ Nm} + 3.2 \text{ Nm} = 3.7 \text{ Nm}$$

$$\frac{M_{\text{acc}}}{M_N} = \frac{3.7}{2.6} = 1.4$$

The motor is suitable.

Stopping accuracy for mechanical braking from the positioning velocity

Diagram illustrating stopping and the stopping accuracy:

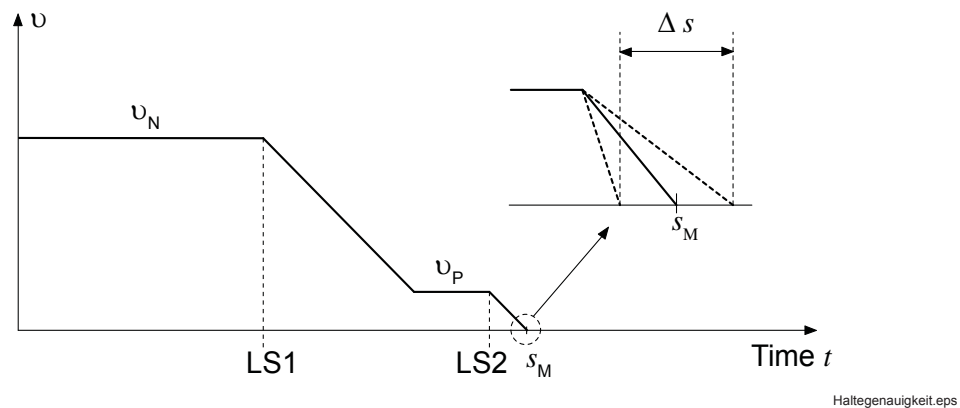


Fig. 11-2 Stopping accuracy

- v : Table circumferential velocity
- s : Distance around the table circumference
- s_{av} : Average stopping position
- Δs : Stopping accuracy
- LS1: When this limit switch is reached, the velocity is switched from the operating velocity v_N (table circumferential velocity) to the positioning v_P velocity.
- LS2: From this limit switch onward – assuming v_P – the converter is decelerated to zero speed and the brake is then applied or the brake engages as soon as LS2 has been passed.

This assumes that the brake is switched when LS2 is passed.

First of all, the braking time and stopping distance are calculated.

Braking torque $M_b = 3 \text{ Nm}$

Brake engaging time with switching on the DC and AC sides $t_1 = 30 \text{ ms}$

Since the rangeability is 1:10, $n_{mot} = 137 \text{ min}^{-1}$ approximately

$$t_B = \frac{(J_M + J_{ext} \cdot \eta) \cdot n_{mot}}{2\pi \cdot (M_b + M_l \cdot \eta^2)} = \frac{(0.00097 + 0.01225 \cdot 0.9) \text{ kgm}^2 \cdot \frac{136}{60} \text{ s}}{2\pi \cdot (3 + 0.5 \cdot 0.9^2) \text{ Nm}} = 1.3 \text{ ms}$$

The stopping distance is comprised of the distances during the brake engaging time t_1 and braking time t_B .

$$s = v \cdot \left(t_1 + \frac{1}{2} \cdot t_B \right) = 0.39 \frac{\text{m}}{\text{s}} \cdot \left(0.03 \text{ s} + \frac{1}{2} \cdot 0.0013 \text{ s} \right) = 12 \text{ mm}$$

Stopping accuracy: $\Delta s \approx 0.15 \cdot 12 \text{ mm} = 1.8 \text{ mm}$

Dimensioning the converter

The maximum motor current which arises is:

$$I_{\max} = I_{\text{rated}} \cdot \frac{(M_{\text{dyn}} + M_{\text{stat}})}{M_{\text{N}}} = 1.04 \text{ A} \cdot \frac{(3.2 + 0.5) \text{ Nm}}{2.6 \text{ Nm}} = 1.48 \text{ A}$$

A Sinamics G120 is selected in conjunction with a PM240 power unit. The rated power is 0.37 kW and $I_{\text{H}} = 1.3 \text{ A}$.

A braking resistor also has to be dimensioned for the power unit.

If a converter with IP54 degree of protection or higher is required for a distributed configuration, a Sinamics G120D is used.

Note on dimensioning the converter and the braking resistor:

You will find more detailed calculations in the examples relating to the hoisting gear and the gantry.



Example of a spindle drive with a frequency converter

12

General information

A spindle is a shaft featuring a thread, whose purpose is to convert the rotary motion of the motor into a linear motion. This can also be achieved by using linear motors, but these are too specialized and/or too cost-intensive for many applications.

A distinction is made between trapezoidal spindles and ball screws. Ball screws are considerably more efficient and as a result tend to be used in machine tools, while trapezoidal spindles are used for simpler applications where there is no requirement for an exact velocity or precise positioning.

Efficiency of the spindle

If the efficiency of the spindle (η_S) is not known, but its friction value is, η_S can be determined as shown below.

$$\eta_S = \frac{\tan \alpha}{\tan(\alpha + \rho)}$$

α : Radian angle of lead of the spindle

ρ : Radian friction angle of the spindle

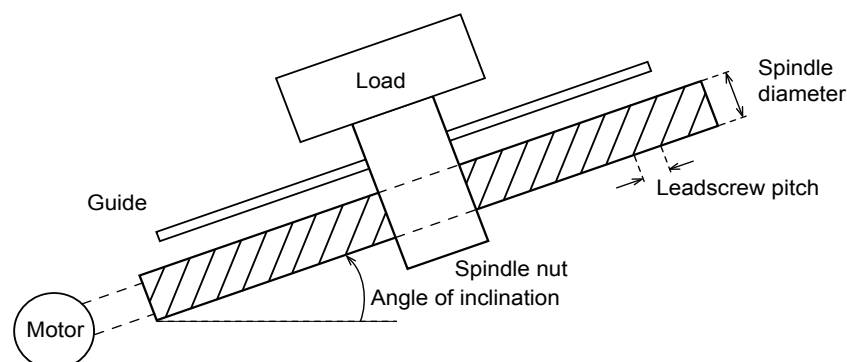
The following also applies:

$$\tan \rho = \mu_S$$

μ_S : Friction value of the spindle

$$\tan \alpha = \frac{\text{LeadscrewPitch}}{\text{SpindleCircumference}}$$

Example of a ball screw



Kugelrollspindel.eps

Fig. 12-1 Example of a ball screw

Total mass to be moved	m_1	= 100 kg
Spindle diameter	D	= 40 mm
Leadscrew pitch	h	= 10 mm
Spindle moment of inertia	J_{spindle}	= 0.002368 kgm ²
Efficiency of the spindle	η_s	= 0.88
Traversing distance		1.2 m
Max. velocity	v	= 0.22 m/s
Accelerating time	t_a	= 0.22 s
Deceleration time	t_d	= 0.22 s
Friction force	F_f	= 500 N
Angle of inclination	β	= 40°

Angle calculation

$$\alpha = \operatorname{atan} \frac{\text{LeadscrewPitch}}{\text{SpindleCircumference}} = \operatorname{atan} \frac{h}{\pi \cdot D} = \operatorname{atan} \frac{0.01 \text{ m}}{\pi \cdot 0.04 \text{ m}} = 0.079 \text{ rad}$$

$$\rho = \operatorname{atan} \left(\frac{\tan \alpha}{\eta_s} \right) - \alpha = \operatorname{atan} \left(\frac{\tan(0.079 \text{ rad})}{0.88} \right) - 0.079 \text{ rad} = 0.01 \text{ rad}$$

Spindle speed

$$n = \frac{v}{h} = \frac{0.22 \frac{\text{m}}{\text{s}}}{0.01 \text{ m}} = 22 \text{ s}^{-1} = 1320 \text{ min}^{-1}$$

Acceleration force of the load

$$F_{\text{acc}} = m_1 \cdot a = m_1 \cdot \frac{v}{t_a} = 100 \text{ kg} \cdot \frac{0.22 \frac{\text{m}}{\text{s}}}{0.22 \text{ s}} = 100 \text{ N}$$

Grade resistance

$$F_{\text{grade}} = m_1 \cdot g \cdot \sin \alpha = 100 \text{ kg} \cdot 9.81 \frac{\text{m}}{\text{s}^2} \cdot \sin 40^\circ = 631 \text{ N}$$

Torque during constant travel

$$\begin{aligned} M &= (F_{\text{grade}} + F_f) \cdot \frac{D}{2} \cdot \tan(\alpha + \rho) = (631 \text{ N} + 500 \text{ N}) \cdot \frac{0.04 \text{ m}}{2} \cdot \tan(0.079 + 0.01) = \\ &= 2.0278 \text{ Nm} \end{aligned}$$

Torque during the acceleration procedure

(without taking the motor into account)

$$M = J_{\text{spindle}} \cdot \frac{2\pi \cdot n}{t_a} + (F_{\text{acc}} + F_{\text{grade}} + F_f) \cdot \frac{D}{2} \cdot \tan(\alpha + \rho)$$

$$M = 0.002368 \text{ kgm}^2 \cdot \frac{2\pi \cdot 22 \text{ s}^{-1}}{0.22 \text{ s}} + (100 \text{ N} + 631 \text{ N} + 500 \text{ N}) \cdot \frac{0.04 \text{ m}}{2} \cdot \tan(0.079 + 0.01) = 3.68 \text{ Nm}$$

Selected motor

Standard asynchronous motor, 4-pole

Rated power 0.55 kW

Rated speed 1,395 min⁻¹

Rated torque $M_{\text{rated}} = 3.8 \text{ Nm}$

Rated current 1.44 A at 400 V, star connection

$J_{\text{motor}} = 0.0014 \text{ kgm}^2$

Operating speed when run on converter 1,320 min⁻¹

Recalculating the selected motor

Torque during the acceleration procedure, taking the motor into account:

$$M = (J_{\text{spindle}} + J_{\text{motor}}) \cdot \frac{2\pi \cdot n}{t_a} + (F_{\text{acc}} + F_{\text{grade}} + F_f) \cdot \frac{D}{2} \cdot \tan(\alpha + \rho)$$

$$M = (0.002368 + 0.0014) \text{ kgm}^2 \cdot \frac{2\pi \cdot 22 \text{ s}^{-1}}{0.22 \text{ s}} + (100 \text{ N} + 631 \text{ N} + 500 \text{ N}) \cdot \frac{0.04 \text{ m}}{2} \cdot \tan(0.079 + 0.01) = 4.56 \text{ Nm}$$

As a result, $\frac{M}{M_{\text{rated}}} = 1.2$

The limit value for M/M_{rated} is determined by looking at the breakdown torque M_{break} . A tolerance of approximately 30 % is applied to the breakdown torque value. In our example, the ratio of $M_{\text{break}}/M_{\text{rated}} = 2.2$. Therefore, the breakdown torque is at least $0.7 \cdot 2.2 \cdot M_{\text{rated}} = 1.54 \cdot M_{\text{rated}}$.

This value is higher than the calculated required ratio of 1.2.

The motor is suitable.

Selected converter

A SINAMICS G120 converter is selected in conjunction with a PM240 power unit. The rated power is 0.55 kW, rated output current (= base-load current)
 $I_{\text{rated}} = 1.7 \text{ A}$.

Checking the acceleration behavior:

The acceleration torque is so close to the rated torque that a linear torque and current characteristic can be assumed in relation to the speed. This means that the required current during the acceleration procedure is

$$I = I_N \cdot \frac{M}{M_{\text{rated}}} = 1.44 \text{ A} \cdot \frac{4.56 \text{ Nm}}{3.8 \text{ Nm}} = 1.7 \text{ A}$$

The acceleration current does not exceed the base-load current, which means that the converter is suitable.

Braking resistor

Calculating the static braking power for a downward motion:

$$M = (F_{\text{grade}} - F_f) \cdot \frac{D}{2} \cdot \tan(\alpha - \rho) = (631 \text{ N} - 500 \text{ N}) \cdot \frac{0.04 \text{ m}}{2} \cdot \tan(0.079 - 0.01) = 0.18 \text{ Nm}$$

$$P = M \cdot 2\pi \cdot n = 0.18 \text{ Nm} \cdot 2\pi \cdot \frac{1320}{60 \text{ s}} = 25 \text{ W}$$

Calculating the maximum braking power for a downward motion:

$$M = (J_{\text{spindle}} + J_{\text{motor}}) \cdot \frac{2\pi \cdot n}{t_B} + (F_{\text{acc}} + F_{\text{grade}} - F_f) \cdot \frac{D}{2} \cdot \tan(\alpha - \rho)$$

$$M = (0.002368 + 0.0014) \text{ kgm}^2 \cdot \frac{2\pi \cdot 22 \text{ s}^{-1}}{0.22 \text{ s}} + (100 \text{ N} + 631 \text{ N} - 500 \text{ N}) \cdot \frac{0.04}{2} \cdot \tan(0.079 - 0.01) = 2.7 \text{ Nm}$$

$$P = M \cdot 2\pi \cdot n = 2.7 \text{ Nm} \cdot 2\pi \cdot \frac{1320}{60 \text{ s}} = 372 \text{ W}$$

The continuous braking power of the associated resistor is 100 W and the peak power is 2 kW, so the braking resistor is suitable.



Example of a mechanical brake

13.1 Example of how to determine the braking torque

The required braking torque is calculated for three different application scenarios, depending on which of the following factors has been specified:

- Maximum braking time defined
- Maximum deceleration defined
- Maximum over-travel defined

All three examples are based on the following application data:

Moment of inertia of the load based on the motor shaft	J_{load}	=	0.1286 kgm ²
Steady load torque	M_{load}	=	4 Nm
System efficiency	η	=	0.9
Stiction (steel on plastic)	μ_{H}	=	0.2 to 0.45
Travelling speed of the load	v	=	1.2 m/s

Motor 1.5 kW with L8 brake

Moment of inertia of the motor incl. brake [= 0.0033 kgm ² (motor) + 0.000034 kgm ² (brake)]	J_{motor}	=	0.003334 kgm ²
Motor speed during rated operation	n	=	1,420 min ⁻¹
Engaging time (= application time) of the brake	t_1	=	31 ms

Maximum braking time defined

Maximum braking time $t_3 = 2.4 \text{ s}$

Once the brake supply voltage has been disconnected, time t_1 elapses before the brake linings start to slip (i.e. to brake). Therefore, speed $n_{\text{disconnect}}$, which occurs after time t_1 , is calculated first. It is assumed that the disconnection takes place at the rated motor speed n .

$$n_{\text{disconnect}} = n - \frac{60 \cdot t_1 \cdot M_{\text{load}} \cdot \eta}{2\pi \cdot (J_{\text{motor}} + J_{\text{load}} \cdot \eta)} = 1420 \frac{1}{\text{min}} - 9 \frac{1}{\text{min}} = 1411 \frac{1}{\text{min}}$$

$$M_{\text{brake}} = \frac{(J_{\text{motor}} + J_{\text{load}} \cdot \eta) \cdot n_{\text{disconnect}} \cdot 2\pi}{60 \cdot t_3} - M_{\text{load}} \cdot \eta = 7.3 \text{ Nm} - 3.6 \text{ Nm} = 3.7 \text{ Nm}$$

M_{brake} selected: 4 Nm, i.e. the L8 brake is reduced to 4 Nm \Rightarrow L8/4N.

Maximum deceleration defined

Maximum deceleration $a_{\text{brake}} = 0.5 \text{ m/s}^2$

The braking time t_3 is determined first.

$$t_3 = \frac{v}{a_{\text{brake}}} = \frac{1.2 \frac{\text{m}}{\text{s}}}{0.5 \frac{\text{m}}{\text{s}^2}} = 2.4 \text{ s}$$

$n_{\text{disconnect}}$ and M_{brake} are calculated as described above.

Maximum over-travel defined

Maximum over-travel $s_{\text{brake}} = 2.88 \text{ m}$

The braking time t_3 is determined first.

$$t_3 = \frac{s_{\text{brake}}}{v} = \frac{2.88 \text{ m}}{1.2 \frac{\text{m}}{\text{s}}} = 2.4 \text{ s}$$

$n_{\text{disconnect}}$ and M_{brake} are calculated as described above.

13.2 Example of how to recalculate the mechanical brake

The following points must be observed:

- Braking energy
- Operating speed or maximum speed
- Permissible switching frequency
- Brake life time until adjustment and wear

The magnitude of the braking torque also needs to be observed in the case of a vertical lift drive:

$$M_{\text{brake, mech}} > 2 \cdot M_{\text{load}}$$

Various different boundary conditions must be taken into account, depending on the application in question, See Chapter 2.3.

13.2.1 Vertical lift drive with motor operated on a frequency converter

During standard operation, the brake engages at standstill, so it is only needed as a holding brake. However, the exception shall be taken into account here, i.e. when the brake engages at the hoisting gear's operating velocity.

Moment of inertia of the load based on the motor shaft	J_{load}	= 0.06 kgm ²
Steady load torque	M_{load}	= 99 Nm
System efficiency	η	= 0.93

Motor 18.5 kW with L260 brake

Braking torque	M_{brake}	= 200 Nm
Moment of inertia of the motor incl. brake [= 0.1122 kgm ² (motor) + 0.0073 kgm ² (brake)]	J_{motor}	= 0.1195 kgm ²
Motor speed during rated operation	n	= 1,460 min ⁻¹
Engaging time (= application time) of the brake	t_1	= 200 ms

Magnitude of the braking torque

$$M_{\text{brake, mech}} > 2 \cdot M_{\text{load}}$$

This condition is met, as 200 Nm > 2 · 99 Nm

Braking energy

The highest braking energy usually arises when lowering with a full load. However, it can also occur under other load conditions if a counterweight is used. This means that the speed will not necessarily always increase during the brake engaging time; rather, it will decrease instead.

$$n_{\text{disconnect}} = n + \frac{60 \cdot t_1 \cdot M_{\text{load}} \cdot \eta}{2\pi \cdot (J_{\text{motor}} + J_{\text{load}} \cdot \eta)} = 1460 \text{ min}^{-1} + \frac{60 \cdot 0.2 \text{ s} \cdot 99 \text{ Nm} \cdot 0.93}{2\pi \cdot (0.1195 + 0.06 \cdot 0.93) \text{ kgm}^2} =$$

$$= 2463 \text{ min}^{-1}$$

$$W_{\text{brake}} = (J_{\text{motor}} + \eta \cdot J_{\text{load}}) \cdot 2 \cdot (\pi \cdot n_{\text{disconnect}})^2 \cdot \frac{M_{\text{brake}}}{M_{\text{brake}} - \eta \cdot M_{\text{load}}}$$

$$W_{\text{brake}} = (0.1195 + 0.93 \cdot 0.06) \text{ kgm}^2 \cdot 2 \cdot \left(\pi \cdot \frac{2463}{60 \text{ s}} \right)^2 \cdot \frac{200 \text{ Nm}}{(200 - 0.93 \cdot 99) \text{ Nm}} = 10800 \text{ J}$$

No additional time delays are taken into account here, such as those which could occur due to bus runtimes, CPU times, or mechanical arresting devices, for example.

The braking energy which arises is lower than the permitted braking energy for one operation (80 kJ) for the selected brake.

Operating speed or maximum speed

The maximum permissible speed for an emergency stop for this brake is 3,700 min⁻¹, which is not exceeded under the conditions that have been assumed here.

Permissible switching frequency and brake life times

These are not taken into account here.

13.2.2 Traction motor connected direct-on-line

Moment of inertia of the load based on the motor shaft	J_{load}	= 0.00121 kgm ²
Steady load torque	M_{load}	= 9.8 Nm
System efficiency	η	= 0.85
Required switching frequency	z_{req}	= 350 h ⁻¹
Motor 2.2 kW with L32 brake		
Braking torque	M_{brake}	= 18 Nm
Moment of inertia of the motor incl. brake [= 0.00421 kgm ² (motor) + 0.00045 kgm ² (brake)]	J_{motor}	= 0.00466 kgm ²
Motor speed during rated operation	n	= 1,420 min ⁻¹

Braking energy

The disconnection speed is not calculated here. It is assumed that the motor is decelerated from its rated speed.

$$W_{\text{brake}} = (J_{\text{motor}} + \eta \cdot J_{\text{load}}) \cdot 2 \cdot (\pi \cdot n_{\text{disconnect}})^2 \cdot \frac{M_{\text{brake}}}{M_{\text{brake}} - \eta \cdot M_{\text{load}}}$$

$$W_{\text{brake}} = (0.00466 + 0.85 \cdot 0.0121) \text{ kgm}^2 \cdot 2 \cdot \left(\pi \cdot \frac{1420}{60 \text{ s}}\right)^2 \cdot \frac{18 \text{ Nm}}{(18 - 0.85 \cdot 9.8) \text{ Nm}} = 308 \text{ J}$$

The braking energy which arises is lower than the permitted braking energy for one operation (24 kJ) for the selected brake.

Operating speed or maximum speed

The maximum permissible operating speed for this brake is 3,000 min⁻¹, which is much higher than the maximum speed in this application.

Permissible switching frequency

According to the diagram in the catalog, a braking energy of 300 J results in a permissible switching frequency of around 2,300 h⁻¹.

Brake life time until the air gap needs to be adjusted

$$T = \frac{L_{\text{adj}}}{W_{\text{brake}} \cdot z} = \frac{850 \cdot 10^6 \text{ J}}{308 \text{ J} \cdot 350 \cdot \text{h}^{-1}} = 7885 \text{ h}$$

L_{adj} = energy until the air gap needs to be adjusted.

Brake life time until the brake lining needs to be replaced

$$T = \frac{L_{\text{sl}}}{W_{\text{brake}} \cdot z} = \frac{1900 \cdot 10^6 \text{ J}}{308 \text{ J} \cdot 350 \cdot \text{h}^{-1}} = 17625 \text{ h}$$

L_{sl} = energy which corresponds to the maximum service life of the brake lining.



Example of fan startup

14.1 Basic physical principles

Due to the geometry of the rotor wheel, fans have moments of inertia which are multiples of the moment of inertia of the driving motor. These high moments of inertia mean that it takes much longer to accelerate up to the rated speed than in the case of other applications.

Moment of inertia



Radialventilatoren.tif

Fig. 14-1 Running wheels for radial fans

The crucial factors when calculating the moment of inertia of the rotor wheel are the mass and the radius. The calculation is performed according to the formulas for the full cylinder.

$$J = \frac{1}{2} \cdot m \cdot r^2$$

J = Moment of inertia in kgm^2

m = Mass of the rotor wheel in kg

r = Radius of the rotor wheel in m

For an axial fan with a discharge volume of $20,000 \text{ m}^3/\text{h}$ and a discharge pressure of 50 pa, if the wheel had a mass of 177 kg and a radius of 548 mm, for example, the moment of inertia would be 26.57 kgm^2 . The moment of inertia of the 1LG4223-4AA... driving motor is only 0.45 kgm^2 .

If speed is transformed by the gearbox or by a V belt between the motor and the fan, the moment of inertia which is relevant for the motor must be converted to the motor shaft.

$$J_1 = J_2 \cdot \frac{n_2^2}{n_1^2}$$

J_1 = Moment of inertia in kgm^2 converted for the motor side

J_2 = Moment of inertia of the rotor wheel in kgm^2

n_1 = Speed of the motor shaft, in min^{-1} , for example

n_2 = Speed of the fan, in min^{-1} , for example

So, the moment of inertia changes in relation to the speeds squared. Selecting an appropriate transmission ratio enables the moment of inertia to be reduced.

Starting time

The starting time of a drive depends on the moment of inertia to be accelerated and on the available acceleration torque of the motor.

$$t_s = \frac{J_{\text{tot}} \cdot n_{\text{rated}}}{9.55 \cdot M_a}$$

t_s = Starting time, acceleration time to the rated speed in seconds

J_{tot} = Total moment of inertia in kgm^2

n_{rated} = Rated speed of the fan in min^{-1}

M_a = Available acceleration torque in Nm

Different acceleration torques and, consequently, starting times will result, depending on the type of connection used to start the fan.

14.2 Startup on the supply system

14.2.1 Direct-on-line starting

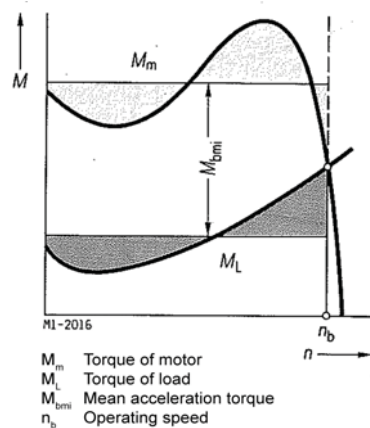
Starting the fan motor directly is easy, straightforward, and cost-effective; for these reasons, it is the preferred type of connection for smaller drives in particular. However, it is not possible to start three-phase motors with squirrel-cage rotors directly with all supply systems, as the locked-rotor current can reach up to eight times the rated current and the starting torque up to three times the rated torque. The high starting currents give rise to voltage dips in the system supplying the power, which can affect the performance of other loads. In addition, the high torques generate large mechanical loads for the drive system.

Motor protection

You must ensure that the motor circuit breakers or overload relays used have been dimensioned for the startup time concerned. These protection devices are categorized according to the following trip classes:

- CLASS 10A $2\text{ s} < t < 10\text{ s}$
- CLASS 10 $4\text{ s} < t < 10\text{ s}$
- CLASS 20 $6\text{ s} < t < 20\text{ s}$
- CLASS 30 $9\text{ s} < t < 30\text{ s}$

Starting time



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Fig. 14-2 Graphical calculation of the average acceleration torque

A simple way of establishing the acceleration torque is shown in Figure 14-2. The average motor torque and average load torque are determined graphically, with the difference being the available acceleration torque.

14.2.2 Star/delta starting

Star/delta starting is primarily used when the supply system cannot cope with the high locked-rotor current which results when the three-phase motor is started directly. The stator winding of the motor, which must be dimensioned for delta rated operation, is switched to star during acceleration. The locked-rotor current and the starting torque then drop to around one third of the values which would result if the delta connection were started directly. The starting time increases by a factor of 3, approximately.

It is important for the switch-over to only be performed once the breakdown torque has been reached. If it happens too soon, the switch-over current will be only marginally lower than with direct starting.

When switching over from star to delta, transient phenomena may also occur in the motor (aggravated by an unfavorable power frequency/rotor field constellation), which would result in higher current peaks than would be the case if the stationary motor were started directly in a delta connection. An unfavorable scenario would lead to the following problems:

- Short-circuit protective devices trip
- The delta contactor becomes welded or is subjected to a high degree of contact erosion
- The motor is subjected to a high dynamic load

A functional example produced by IA CD describes the preferred method of connection for star-delta (wye-delta) starters. Using a favorable method of connection for the main circuit will reduce the transient phenomena and peak currents which occur when switching over from a star to a delta connection. The functional example can be found in PRODIS at:

<http://support.automation.siemens.com/WW/llisapi.dll?query=CD-FE-III-001-V10-DE&func=cslib.cssearch&content=skm%2Fmain.asp&lang=de&siteid=csius&objaction=cssearch&search-inprim=0&nodeid0=20025979&x=29&y=7>

14.2.3 SPICE SD-Configurator dimensioning program

The startup data for a fan drive can be determined with the help of the *SPICE SD-Configurator* selection guide. This program is included in the interactive catalog CA01 – the Siemens Industry Automation and Drive Technologies offline mall. Select the Products group and you will then find the *SPICE SD-Configurator* under the *Selection Guides* item.

The motor type, version, frequency, rated power, synchronous speed, supply voltage, and frame size need to be specified. Once this has been done, a button labeled *Documentation* appears, which takes you to the startup calculation program.

SIEMENS

iDrives configurator

Product Selection
Your Products
Load
Save
Reset
SIZER interface
load product
save product
TLA-1LE1 Conversion

Configuration For Motors And Inverters - Please commit your data with enter. The data (optional) ✓

Project: Order Number:
 Quotation: Customer Order No.:
 Consignment No.: Item Number:
 Comment:

Choose languages for the data sheet /

List Of Products

Br.	Conf.	Order number (MLFB) (related documents)	Price* EUR	Quantity n
1		1LG4223-4AA60 IEC LV-Motor, Cast Iron (TL0) List of documents: <ul style="list-style-type: none"> 3D model manual safety and commissioning terminal connection plan EO - declaration of conformity inspection certificate 2.2 starting program characteristic curve data sheet (PDF) data sheet (RTF) dimensioned drawing 	NaN	1

* price per unit, recommended retail price

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Fig. 14-3 SPICE SD-Configurator start screen for startup calculation

Direct-on-line starting

On the startup calculation screen you need to fill in the load data, such as the torque characteristic, rated power, and external moment of inertia; the values for starting the selected drive directly on the supply system will then appear.

MOTOR DATA

System Voltage:
 Variant System Voltage:
 (Range 70 % - 110 %)
 Operating Mode:
 Y-D Starting: ☐ Yes ☒ No
 Frequency:
 MLFB:

LOAD DATA

Torque Characteristic:
 Rated Speed:
 Rated Torque:
 Rated Power:
 Moment Of Inertia:

STARTING TIME

Rated System Voltage:
 Variant System Voltage:
 Y-D Starting:
 Toggle Moment:
STARTINGS PER HOUR
 at warm condition:
 at cold condition:

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Fig. 14-4 Starting a fan directly on the supply system

The **STARTS PER HOUR** field defines how often the motor may be started from a cold or warm condition without subjecting it to a thermal overload. In this case, the motor can be started 4 times an hour from a cold condition, and just twice per hour from a warm condition.

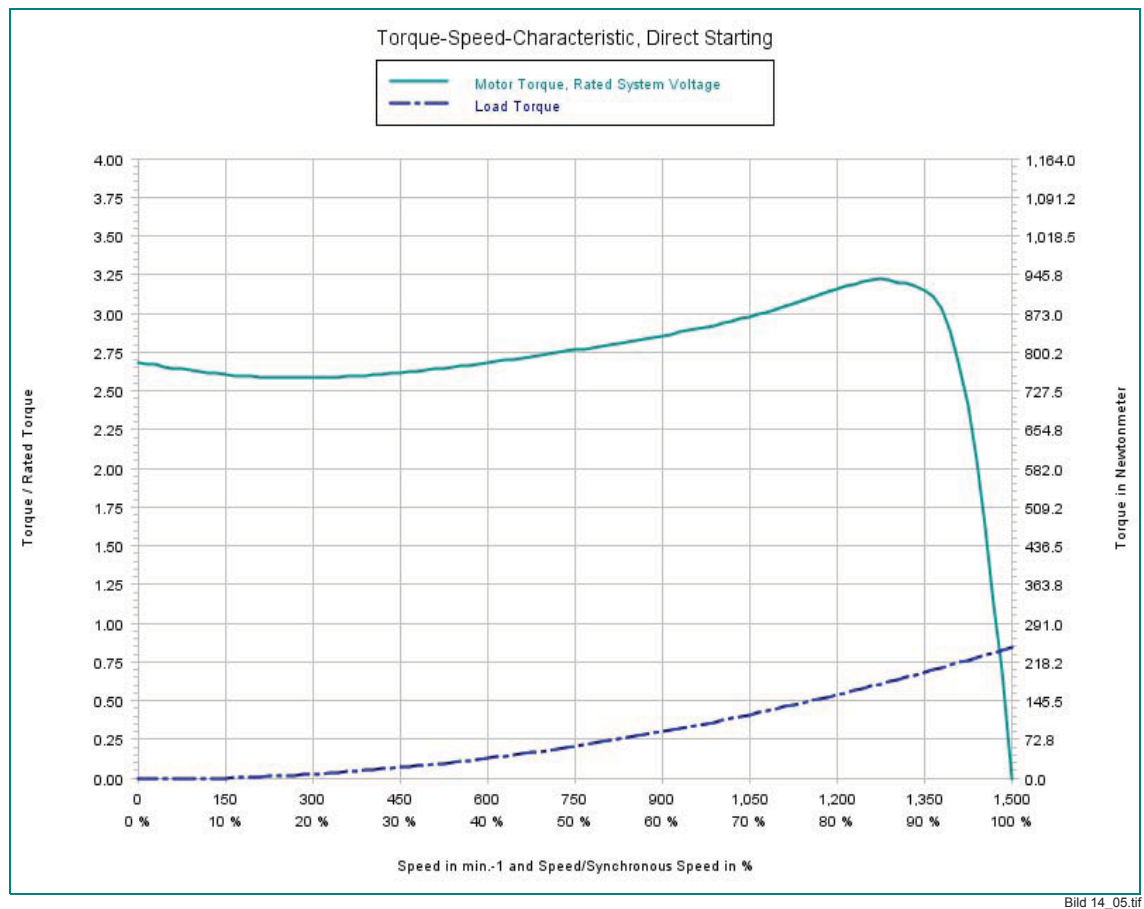


Fig. 14-5 Torque characteristic when connecting directly to the supply system

Figure 14-5 shows the torque characteristic of the motor and the load in relation to the speed. When accelerating, acceleration torques of between 250 % and 325 % of the rated torque occur.

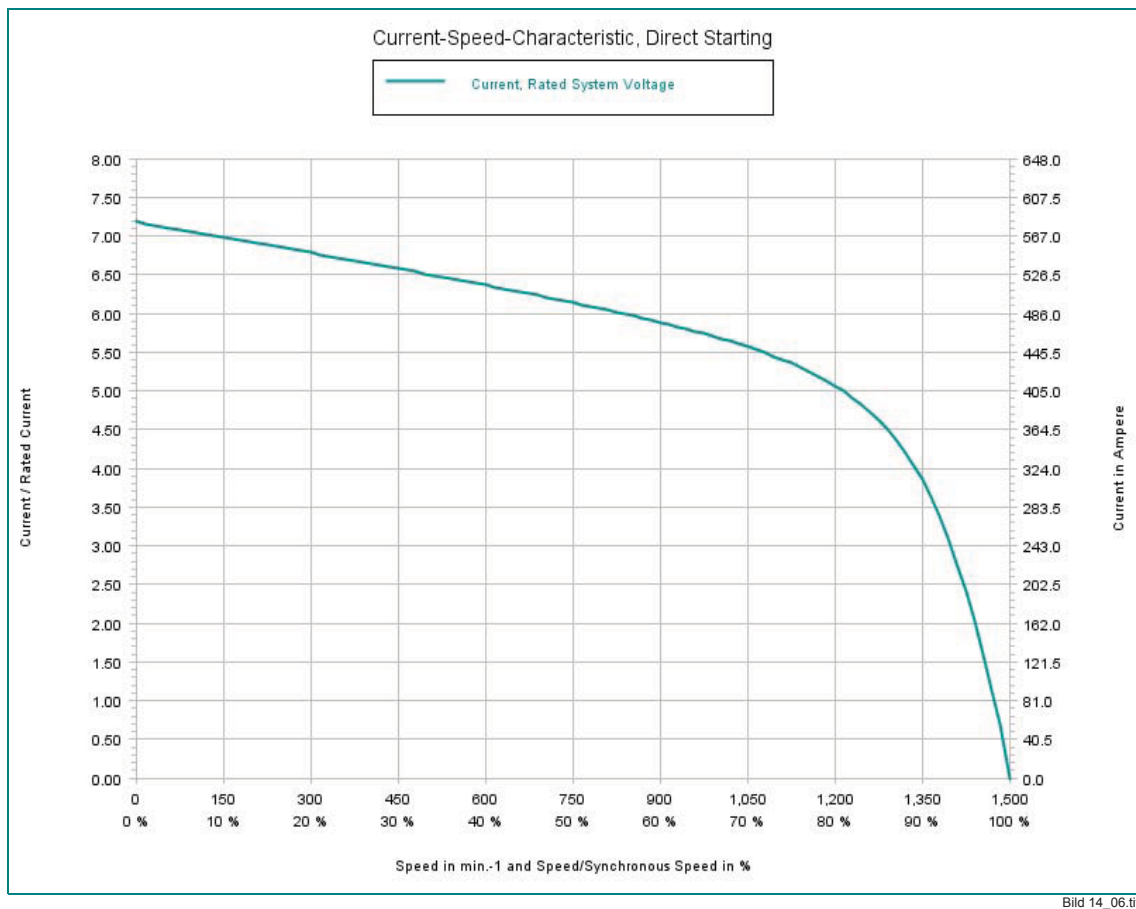


Fig. 14-6 Motor current characteristic when connecting directly to the supply system

Figure 14-6 shows the current characteristic in relation to the speed. At zero speed, a current of over 7 times the rated current is required for a short time.

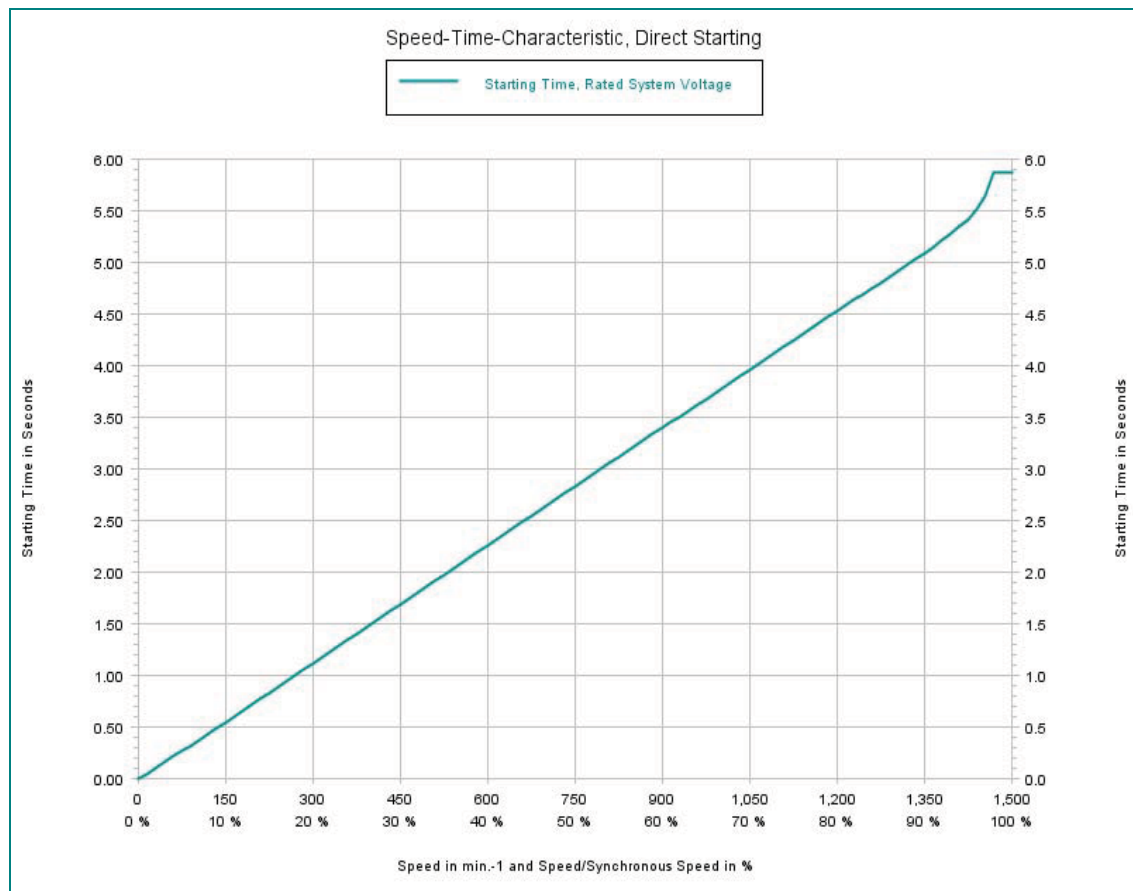


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Fig. 14-7 Acceleration time when connecting directly to the supply system

Figure 14-7 shows the speed characteristic in relation to the time.

Star/delta starting

If a motor which is suitable for star/delta starting has been selected (230/400 V at 230 V line connection, 400/690 V at 400 V line connection), you can switch from delta to star/delta operation via the operating mode.

The screenshot displays three panels of motor configuration data:

- MOTOR DATA:** System Voltage: 400 V; Variant System Voltage: 100 (Range 70 % - 110 %); Operating Mode: Delta; Y-D Starting: Yes (selected); Frequency: 50 Hz; MLFB: 1LG4223-4AA60.
- LOAD DATA:** Torque Characteristic: quadratic; Rated Speed: 1,475 min.-1; Rated Torque: 239.9 Nm; Rated Power: 37.1 kW; Moment Of Inertia: 26.57 kgm².
- STARTING TIME:** Rated System Voltage: 5.9 sec.; Variant System Voltage: 5.9 sec.; Y-D Starting: 0.00 sec.; Toggle Moment: 0.00 sec.
- STARTINGS PER HOUR:** at warm condition: 2/2; at cold condition: 4/4.

Bild 14_08.tif

Fig. 14-8 Star/delta starting of a fan on the supply system

The program is not able to determine the number of starts per hour for star/delta starting.

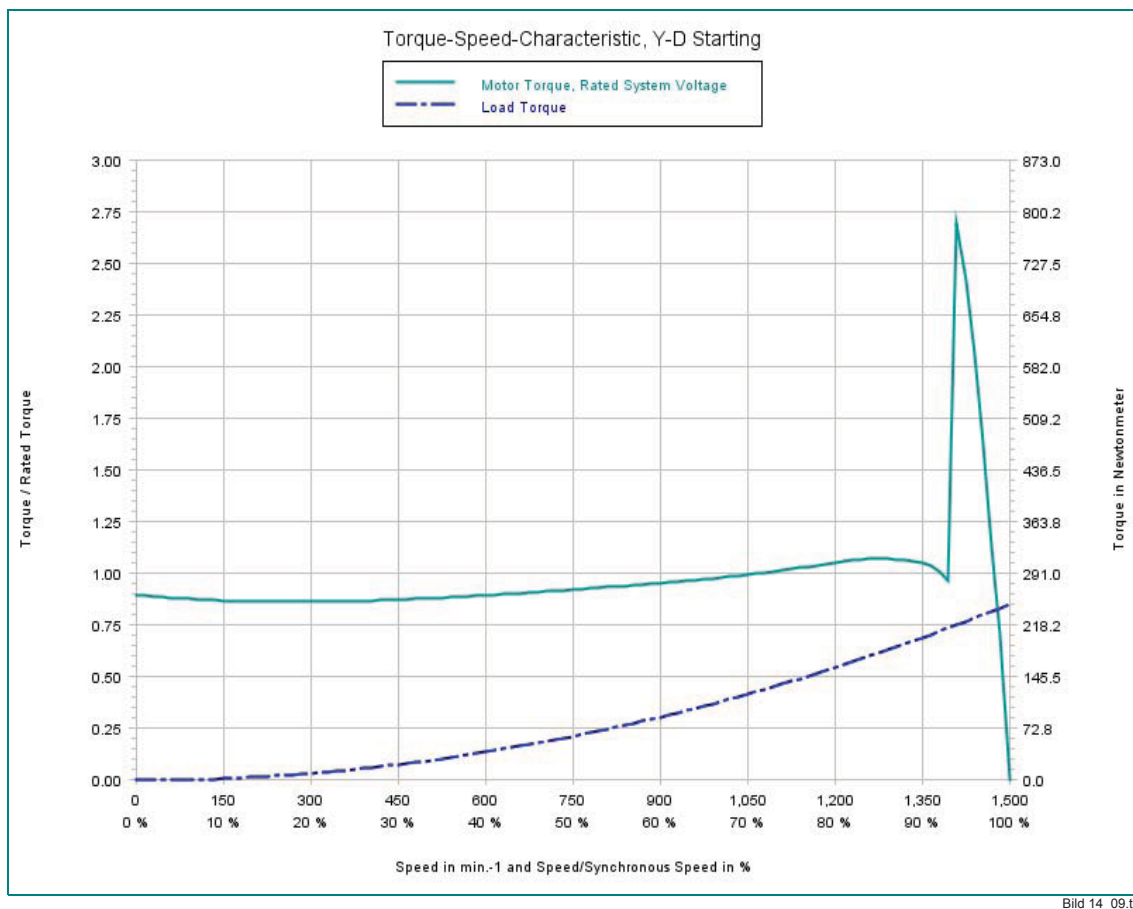


Fig. 14-9 Torque characteristic for star/delta starting

One positive effect with star/delta starting is that the starting torque of around 300 % of the rated torque drops to around 100 %; it is only on switch-over to delta operation that temporary torque peaks occur.

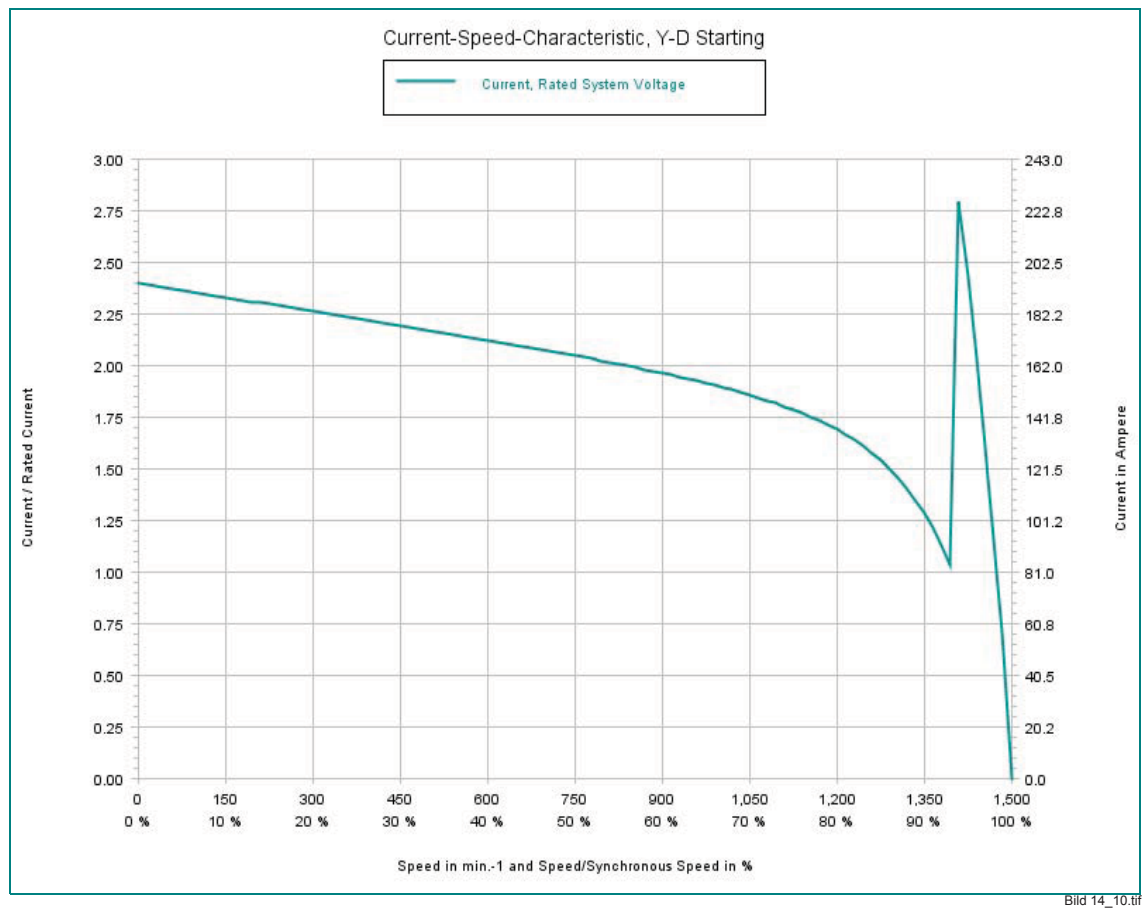


Fig. 14-10 Motor current characteristic for star/delta starting

In contrast to the 700 % starting current for direct connection, the motor current drops to around 250 % of the rated motor current.

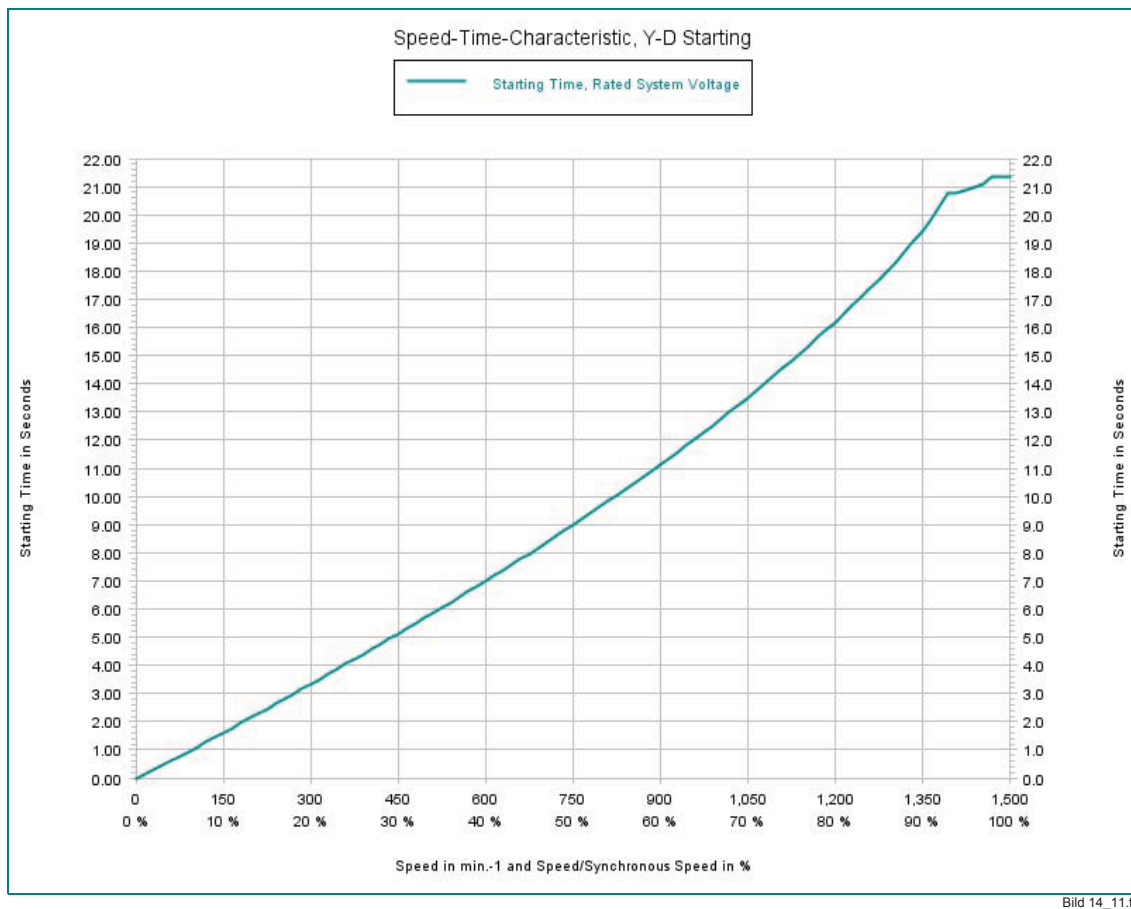


Fig. 14-11 Acceleration time for star/delta starting

The low motor torque during star operation means that the fan startup time increases to 21.39 seconds.

Direct starting at undervoltage

If you also need to ensure that the fan can start up even if the supply voltage fluctuates, you can enter the voltage deviation in order to determine the effect on the motor torque, motor current, and starting time.

MOTOR DATA		LOAD DATA		STARTING TIME	
System Voltage	400 V	Torque Characteristic	linear	Rated System Voltage	6.2 sec.
Variant System Voltage	90	Rated Speed	1,475 min.-1	Variant System Voltage	9.1 sec.
(Range 70 % - 110 %)		Rated Torque	239.9 Nm	Y-D Starting	0.00 sec.
Operating Mode	Delta	Rated Power	37.1 kW	Toggle Moment	0.00 sec.
Y-D Starting	<input checked="" type="radio"/> Yes <input type="radio"/> No	Moment Of Inertia	26.57 kgm ²		
Frequency	50 Hz				
MLFB	1LG4223-4AA60				
				STARTINGS PER HOUR	
				at warm condition	2/2
				at cold condition	4/3

Bild 14_12.tif

Fig. 14-12 Starting a fan directly at undervoltage

The undervoltage may reduce the number of permissible motor starts. In this case, at undervoltage only 3 starts per hour are permitted from a cold condition.

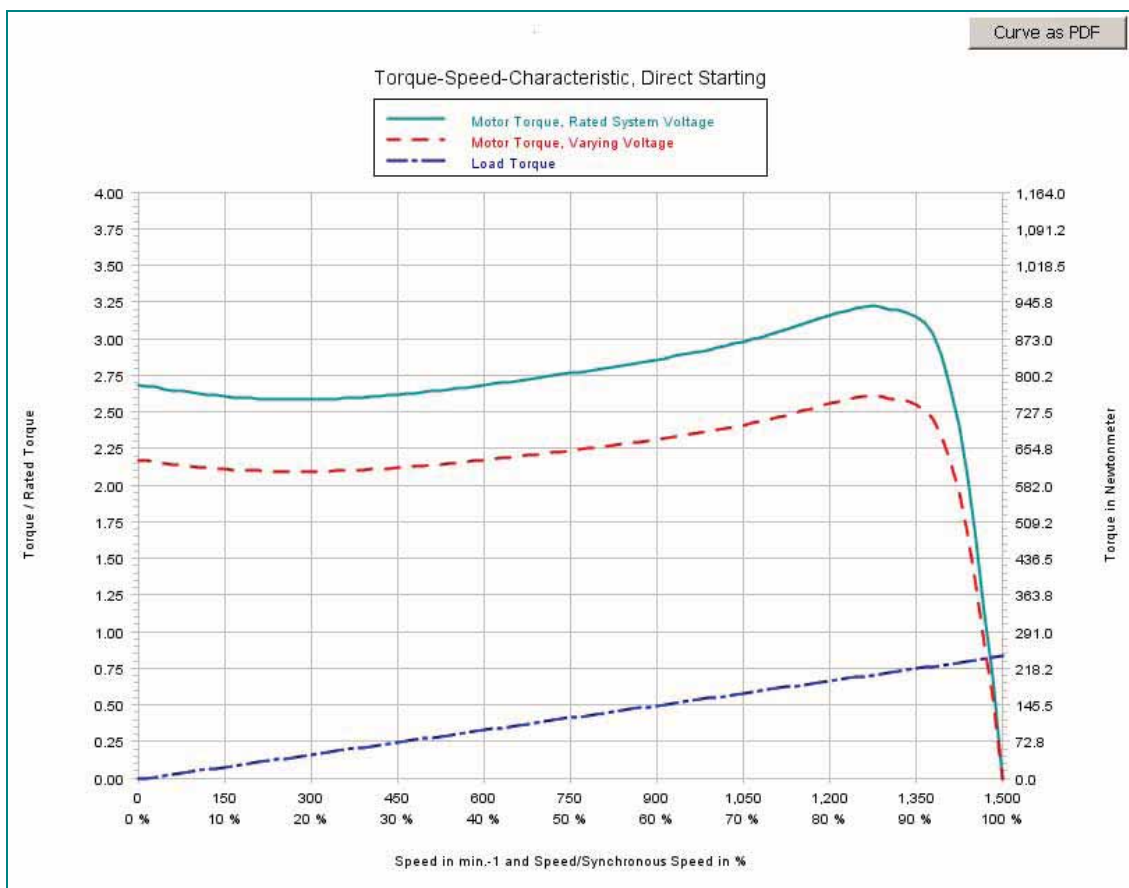


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Fig. 14-13 Torque characteristic when connecting directly and at undervoltage

Figure 14-13 now shows the motor torque at undervoltage (red) next to the motor torque at rated voltage (green). The 10 % lower supply voltage means that the motor torque is reduced to between 200 % and 260 % of the rated torque.

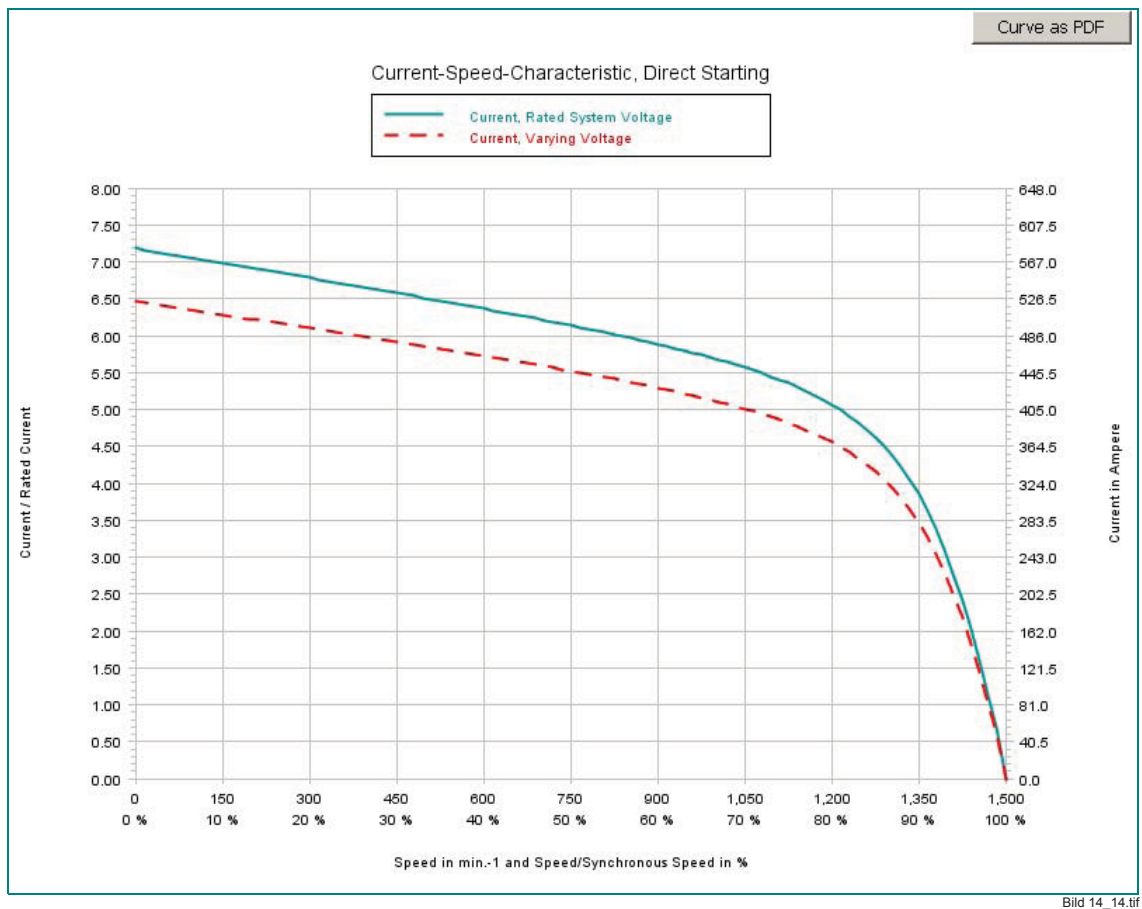


Fig. 14-14 Motor current characteristic when connecting directly and at undervoltage

The lower supply voltage results in a lower starting current; the values for rated operation (green) and undervoltage (red) are shown in figure 14-14.

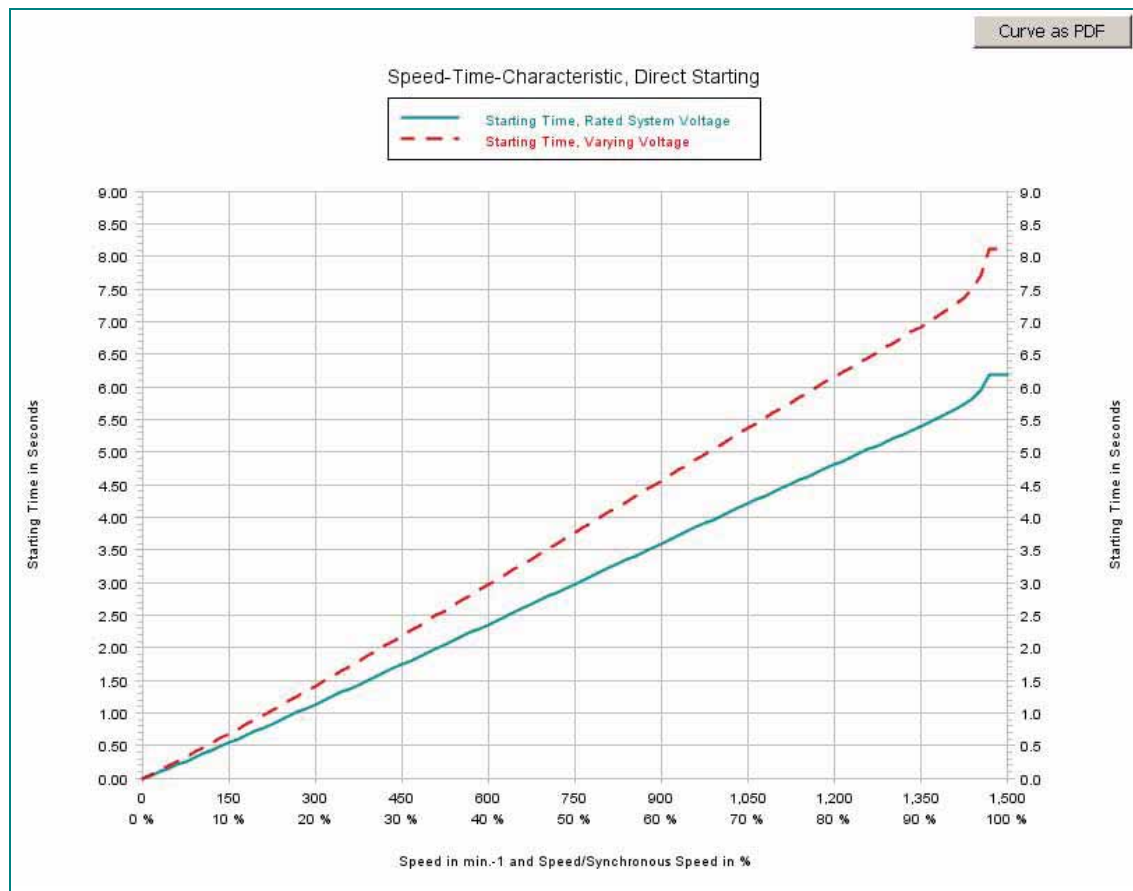


Fig. 14-15 Acceleration time when connecting directly and at undervoltage

The undervoltage lengthens the starting time by around 1.5 seconds, due to the lower motor torque.

Individual load characteristic

If the precise load-torque characteristic of the fan drive is available, you can specify it as a user-defined torque/speed diagram by means of the torque characteristic. The blue load-torque curve is replicated in accordance with the table.

Speed	1 %	2 %	3 %	5 %	7 %	10 %	15 %	20 %	25 %	30 %	40 %	50 %	60 %	70 %	80 %	90 %	100 %	110 %	120 %
Any Load Torque	22 Nm	25 Nm	30 Nm	5 Nm	10 Nm	15 Nm	22 Nm	30 Nm	38 Nm	45 Nm	60 Nm	73 Nm	87 Nm	102 Nm	118 Nm	135 Nm	154 Nm	174 Nm	195 Nm
Load/Rated Torque	0.075001	0.081546	0.094397	0.017183	0.034364	0.051546	0.075001	0.094397	0.118039	0.141681	0.206187	0.250803	0.309405	0.370511	0.435498	0.493817	0.549006	0.597938	0.642675

Bild 14_16.tif

Fig. 14-16 Entering an individual load characteristic

Characteristics

The torque/current/voltage/time diagrams shown can be displayed by pressing the *Characteristics* button and can be printed out as PDFs.

14.3 Soft starters

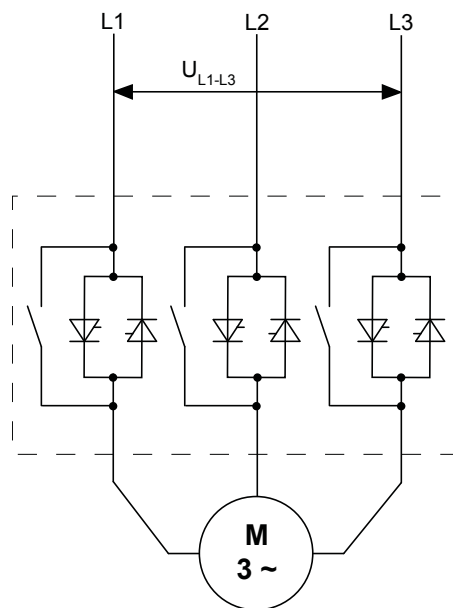
Under certain circumstances, soft starters can be used as an alternative to star-delta (wye-delta) starters and frequency converters. Their main benefits are their ability to perform soft starting, smooth ramp-downs, and uninterrupted switch overs without current peaks that put a strain on the system, as well as their small dimensions.

Soft starters are an alternative to frequency converters if the speed does not need to be controlled (open-loop or closed-loop), no high starting torques occur, and startups do not have to be performed at the rated current, or thereabouts.

14.3.1 Operating principle

Structure

For each of the phases, soft starters feature two thyristors connected in inverse parallel, with one thyristor having responsibility for the positive and one for the negative half-wave in each case.



Sanftstarter3RW4.eps

Fig. 14-17 Circuit design for the 3RW4 soft starter

The rms value of the motor voltage is increased (from an adjustable starting voltage or starting torque) to the rated motor voltage within a definable starting time by means of the leading-edge phase. This is achieved using various control methods.

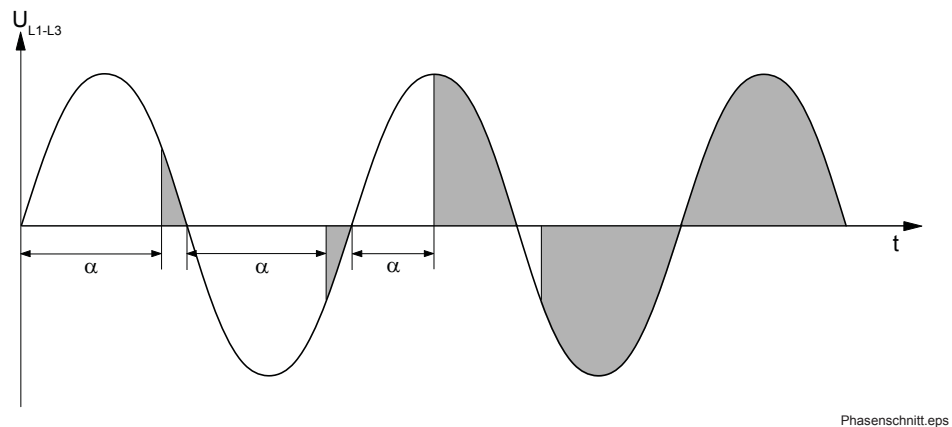


Fig. 14-18 Setting the motor terminal voltage by means of the leading-edge phase

Motor torque

The motor current changes in proportion to the voltage applied to the motor. As a result, the starting current is reduced by the factor of the voltage applied to the motor. There is a quadratic relationship between the torque and the voltage applied to the motor. As a result, the starting torque is reduced quadratically in relation to the voltage applied to the motor. This means that, since the motor voltage is controlled by the electronic soft starter during the startup procedure, the consumed starting current and the starting torque generated in the motor are also controlled.

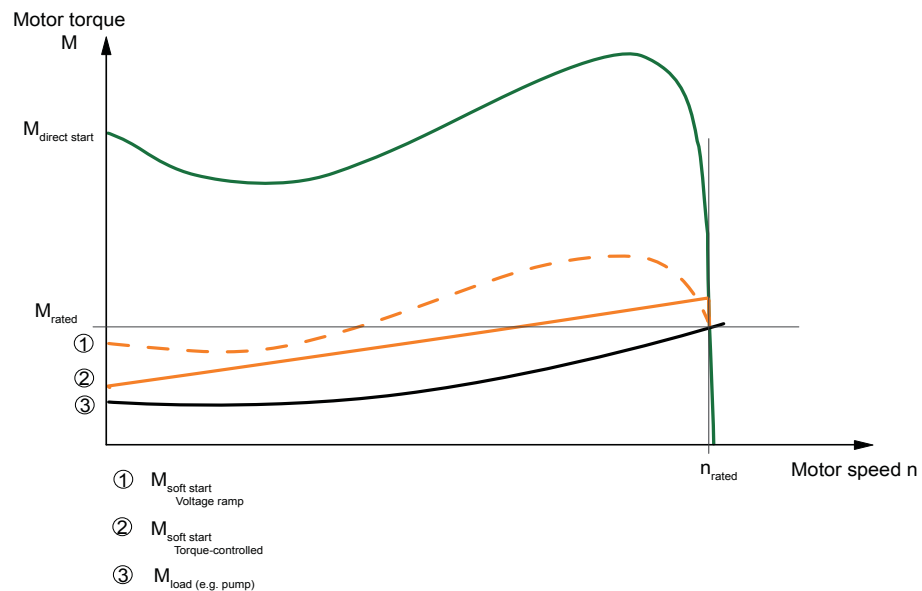


Fig. 14-19 Starting torque for voltage/torque control

The same principle can also be applied during the ramp-down procedure. This ensures that the torque generated in the motor is reduced gradually, so that the application can ramp down smoothly.

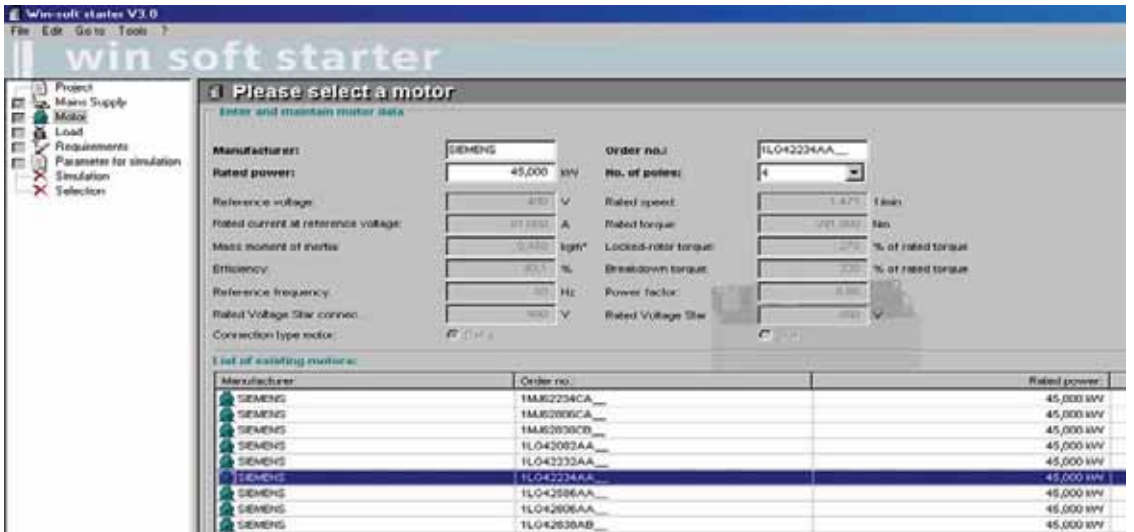
Once the motor has been accelerated correctly, the thyristors are subject to fully advanced control, meaning that the whole line voltage is applied to the motor terminals. As the motor voltage does not have to be controlled during operation, the thyristors are bridged by integrated bypass contacts. This minimizes the waste heat generated during uninterrupted duty (which is caused by the thyristor's power loss).

14.3.2 WIN-Soft Starter dimensioning program

This software can be used to simulate and select all SIEMENS soft starters, taking into account various parameters such as the supply system conditions, motor data, load data, specific application requirements, etc. It is a useful tool, which does away with the need for time-consuming and complex manual calculations in order to establish which soft starter is suitable for your particular case.

Motor selection

The first step is to select the motor for which startup is to be simulated. The list contains all the three-phase motors currently available in the CA01 selection guide. Select the basic MLFB (= order number) and the relevant motor data will be loaded to the required fields.

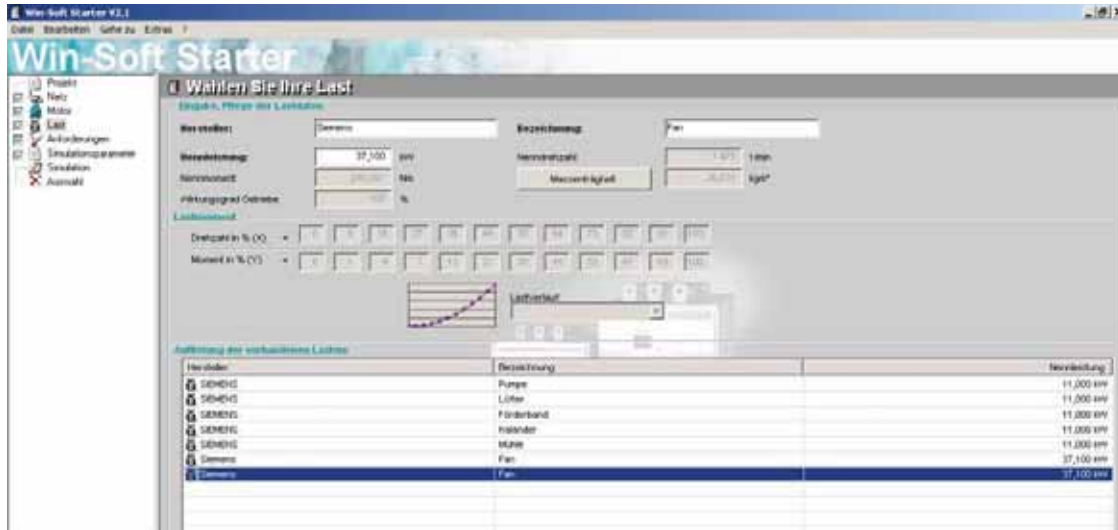


Auswahlmaske-M.tif

Fig. 14-20 Motor selection screen

Load configuration

In the next step, the load data is entered. Click the *Edit* button to enable the grayed-out fields for editing; the moment of inertia and the load curve can now be entered.

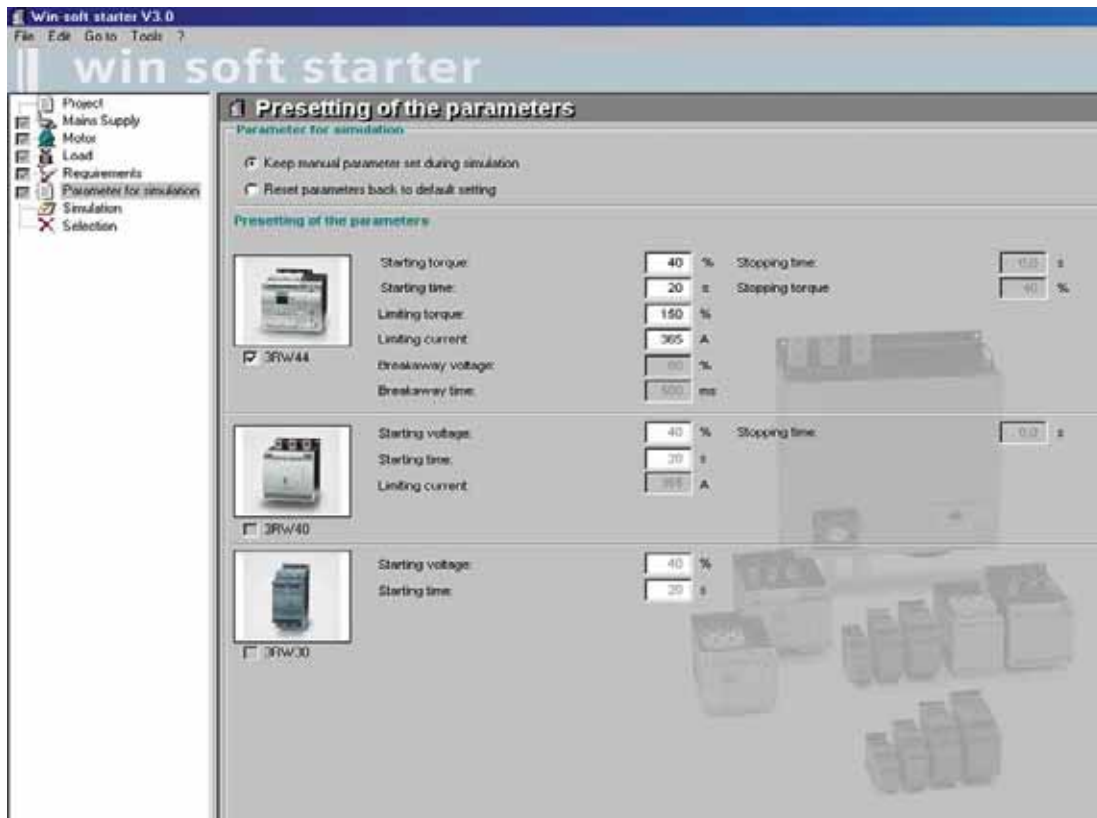


Auswahlmaske-L.tif

Fig. 14-21 Load configuration selection screen

Selecting the soft starter

Once the environmental requirements, installation conditions, device functions, and operating mode have been selected, you can select the required soft starter and its functions. The software then uses the values for the starting voltage, ramp time, and maximum current to simulate the acceleration procedure.



Auswahlmaske-S.tif

Fig. 14-22 Soft starter selection screen

Diagrams

The simulation of the drive acceleration procedure is recorded in a diagram, which shows the voltage, current, torque, and speed.

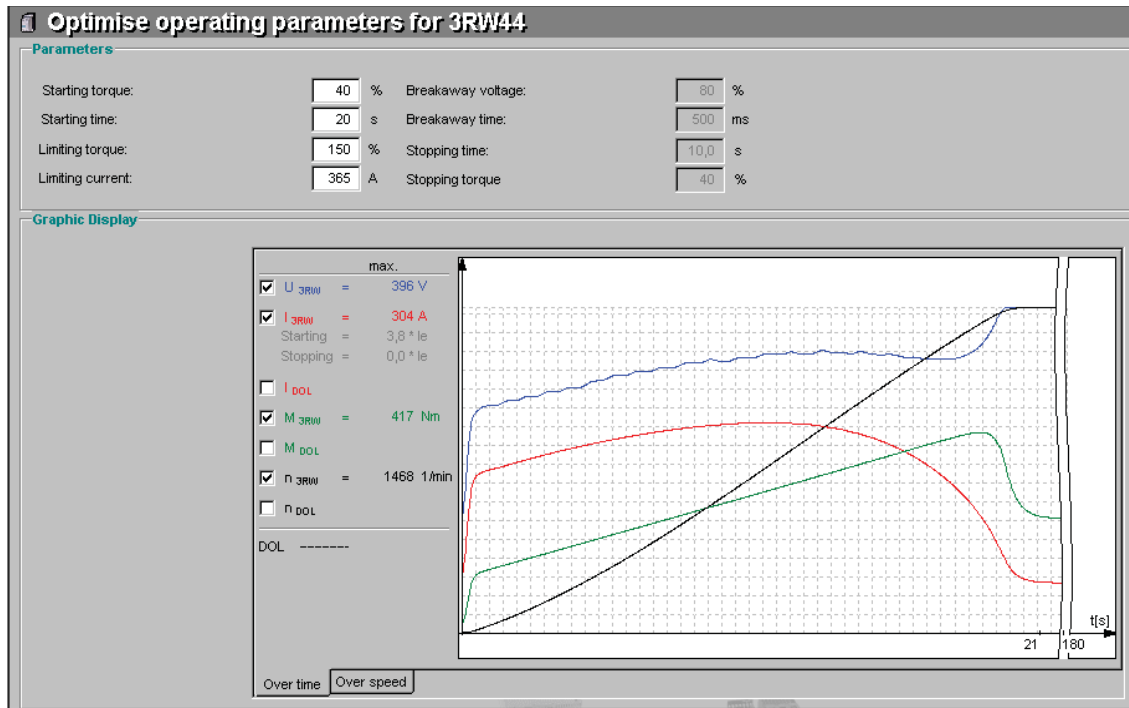


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Fig. 14-23 Voltage/current/torque diagram in relation to time

You can compare these values to direct starting on the supply system by selecting the I_{DOL} , M_{DOL} , and n_{DOL} boxes; the startup curves for direct starting on the supply system will then be shown too, as dotted lines.

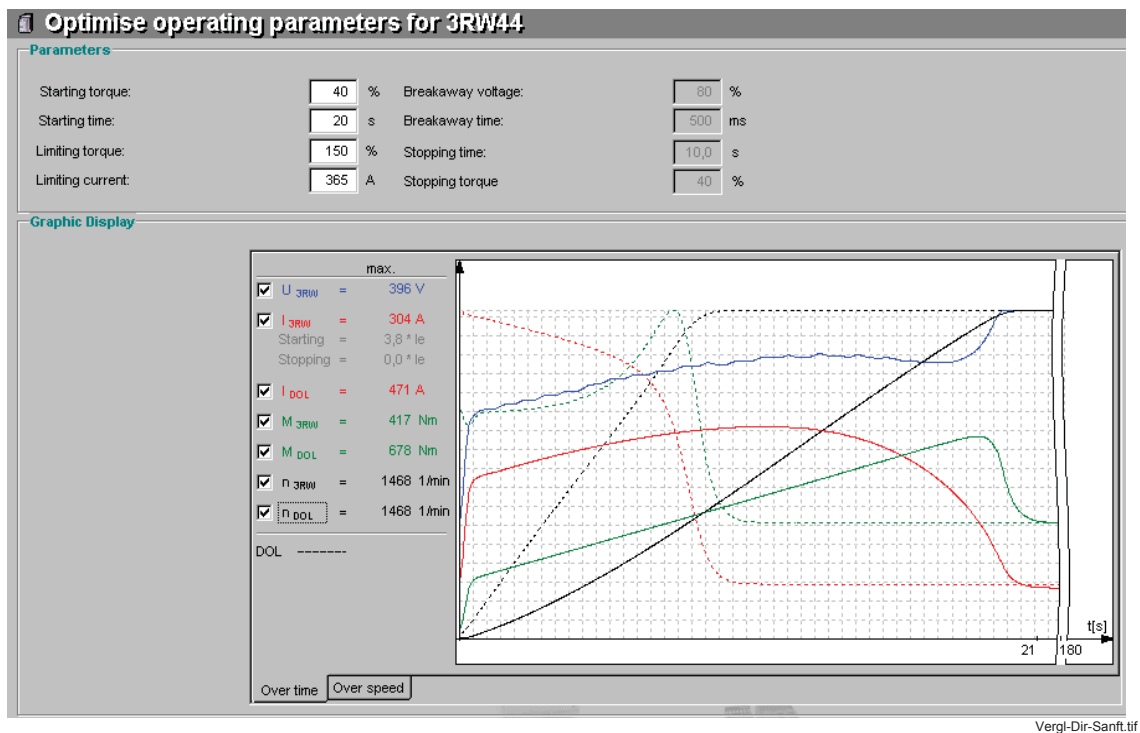


Fig. 14-24 Comparison of direct-on-line starting and starting using a soft starter

14.4 Converters

Frequency converters are used if the speed of the drive needs to be varied in addition to controlling the startup procedure for the purpose of controlling process variables.

14.4.1 Operating principle

Structure

The converters in the SINAMICS series are PWM converters with an intermediate DC circuit. At the line side, the converter consists of a rectifier (shown as a diode rectifier in the example schematic sketch), which is supplied with a constant voltage U_{line} and a constant frequency f_{line} from a three-phase system. The rectifier produces a constant DC voltage $U_{\text{DC link}}$, i.e. the DC-link voltage, which is smoothed by the DC link capacitors. The IGBT inverter at the output end (represented as ideal switches in the schematic sketch) converts the DC-link voltage to a three-phase system with a variable voltage U_{motor} and variable frequency f_{motor} . This process operates according to the principle of pulse width modulation. The speed of the connected three-phase motor can be continually changed by varying the frequency; varying the voltage in accordance with the frequency enables the torque to be kept constant over a large control range.

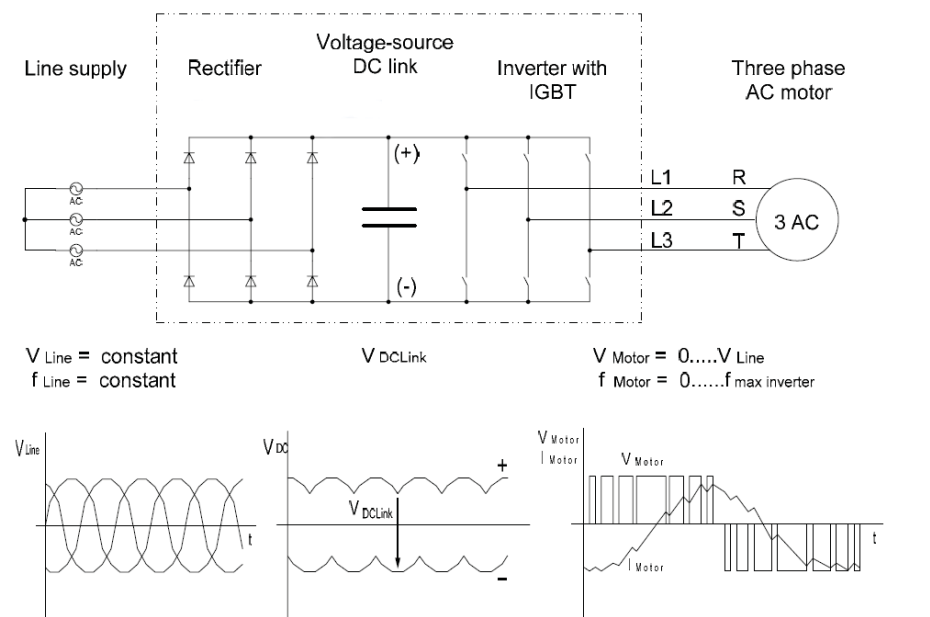
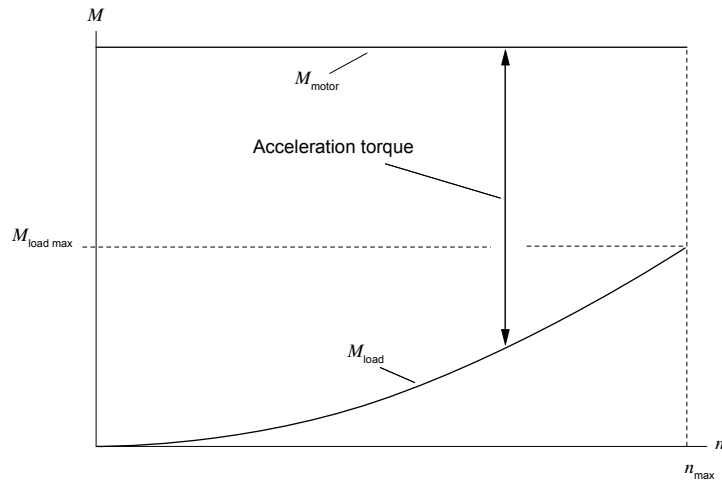


Fig. 14-25 Schematic circuit diagram of a voltage-source converter

Motor torque

When the fan is started on the frequency converter, it should be noted that, unlike the supply system, the converter can only provide a fixed maximum output current. The motor torque generated by this is constant from a frequency of zero up to the rated frequency; if the fan is to run at a higher speed, the motor torque must be reduced by a factor of $1/n$ above the rated frequency. The maximum output current can be overloaded by a factor for a certain period of time, after which the converter will shut down or the load will be reduced.



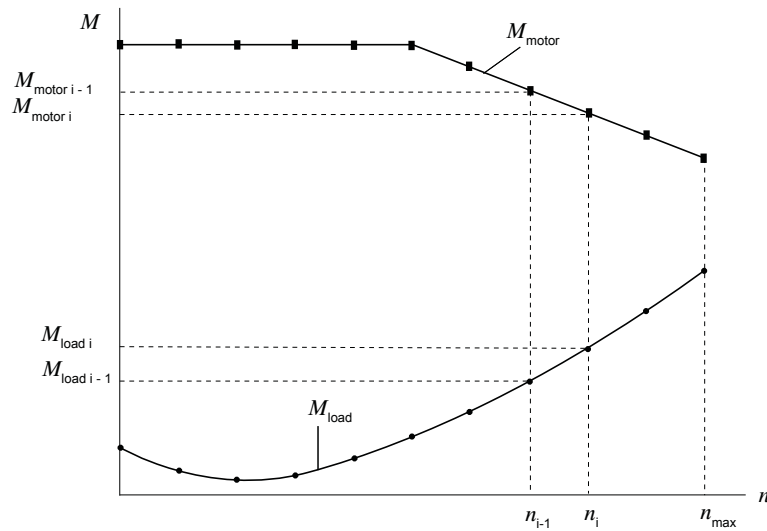
AnlaufVFU.eps

Fig. 14-26 Starting a fan on the frequency converter at a constant motor torque

If the load torque is specified for the fan in the format $M_{\text{load}} = M_{\text{load max}} \cdot \left(\frac{n}{n_{\text{max}}}\right)^2$ at a constant motor torque (e.g. acceleration at the current limit in the constant flux range), the starting time can be calculated using the formula below:

$$t_s = \frac{\pi \cdot n_{\text{max}} \cdot J}{60 \cdot \sqrt{M_{\text{motor}} \cdot M_{\text{load max}}}} \cdot \ln \frac{\sqrt{\frac{M_{\text{motor}}}{M_{\text{load max}}} + 1}}{\sqrt{\frac{M_{\text{motor}}}{M_{\text{load max}}} - 1}}$$

The difference between the motor torque and the load torque acts as the acceleration torque. Following acceleration to n_{max} , the motor torque adjusts itself to the same value as the load torque. The larger the ratio of the motor torque to the load torque and the smaller the total moment of inertia, the faster the acceleration.



AnlaufVFU_MLM.eps

Fig. 14-27 Starting a fan on the frequency converter at a user-defined motor/load torque

If the load torque is not quadratic or the motor torque is not constant during acceleration (See Fig. 14-27), the starting time has to be calculated numerically using the formula:

$$t_s = \frac{\pi \cdot J}{30} \cdot \int_0^{n_{\max}} \frac{dn}{M_{\text{motor}}(n) - M_{\text{load}}(n)}$$

Dividing the function into m sections produces the following approximation formula:

$$t_s \approx \frac{\pi \cdot J}{30} \cdot \sum_{i=1}^m \frac{1}{2} \cdot \left(\frac{1}{M_{\text{motor } i-1} - M_{\text{load } i-1}} + \frac{1}{M_{\text{motor } i} - M_{\text{load } i}} \right) \cdot (n_i - n_{i-1})$$

To evaluate this approximation formula the motor torque and the load torque must be provided in tabular format; the acceleration time is really easy to determine using a simple Excel spreadsheet.

The fan drive with the 1LG4332-4AA60 45 kW motor can once again be used as an example. An MM430 converter, 6SE6430-2AD33-7EA0, is needed for the required power of 37.1 kW at 1,470 min⁻¹. This converter provides an output current of 75 A, which can be overloaded by 10 % for 60 seconds. This means that 82.5 A are available for the acceleration phase. The motor torque thus achieved is:

$$M_{\text{motor max}} = M_{\text{motor rated}} \cdot \sqrt{\frac{I_{\text{motor max}}^2 - I_{\mu}^2}{I_{\text{motor rated}}^2 - I_{\mu}^2}}$$

At an assumed magnetizing current of 40 % of rated current, the result is a maximum motor torque of 296.4 Nm.

Table 14-1 Calculation of the starting time when operating on the converter

Step i	Speed n in rpm	M_{load} in Nm	M_{motor} in Nm	Starting time t_s in s
0	0	22	296.4	
1	74	15	296.4	0.7
2	148	10	296.4	0.7
3	221	5	296.4	0.7
4	295	10	296.4	0.7
5	369	15	296.4	0.7
6	443	22	296.4	0.7
7	516	29	296.4	0.8
8	590	38	296.4	0.8
9	664	49	296.4	0.8
10	738	60	296.4	0.9
11	811	73	296.4	0.9
12	885	87	296.4	1.0
13	959	102	296.4	1.0
14	1,033	118	296.4	1.1
15	1,106	135	296.4	1.2
16	1,180	154	296.4	1.4
17	1,254	174	296.4	1.6
18	1,328	195	296.4	1.9
19	1,401	217	296.4	2.3
20	1,475	241	296.4	3.2
Acceleration time				23.1

Therefore, the minimum acceleration time when the drive is accelerated at the current limit is 23.1 seconds, which is within the range of the permitted duty cycle.



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